A STUDY TO IMPROVE THE CRASH COMPATIBILITY BETWEEN CARS IN FRONTAL IMPACT

Final Report by

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EXECUTIVE SUMMARY

Following the introduction of the European Frontal and Side Impact Directives in October 1998, compatibility offers the next greatest potential benefit for improving car occupant safety and reducing road casualties. For frontal impact the work performed to date has focused on the structural performance of the cars, with the aim of providing a safe environment in which the restraint system can operate. Having achieved this, intelligent restraint systems could offer a way to cope with higher compartment decelerations, and give the occupant an optimised ride-down for a variety of impact severities.

The work performed in the 4th framework compatibility project has helped to understand compatibility. It concluded that an essential prerequisite for compatible cars is good structural interaction. Once this has been achieved some form of stiffness matching will be necessary to ensure that the impact energy is absorbed without exceeding the strength of the occupant compartment. The 4th framework project also outlined a number of possible test procedures to address these requirements in order to assess and control the compatibility of cars in frontal impact collisions. There are currently four candidate test procedures, which are expected to form the basis of future legislation and / or consumer testing to improve compatibility. These are a full width deformable barrier test to assess structural interaction, an ODB test to control stiffness, a high speed ODB test to control the compartment strength and a Progressive Deformable Barrier (PDB) test to assess both structural interaction and control stiffness.

The main aims of this project were:

- To further develop the crash test procedures detailed above.
- To perform an analysis to estimate the benefits of implementing compatibility measures for frontal impact.
- To perform accident analyses to further aid the understanding of compatibility and to support the cost benefit analysis.

It is expected that the 5th framework VC-COMPAT project, due to start in November 2002, should continue this work. This project was initiated to continue the development of the test procedures in the intervening period between the 4th and 5th framework projects.

The work in this project was divided into 3 work packages, namely, accident analysis, benefit analysis and crash testing.

Accident analysis

Examples of the poor structural interaction, stiffness mismatching and compartment strength compatibility problems were observed in the CCIS and Hanover accident databases for GB and Germany, respectively. For GB, poor structural interaction was found to be a major problem, with less than 2 percent of the car to car frontal impact accident cases examined, showing reasonable structural interaction. For both GB and Germany, stiffness mismatching and / or compartment strength was found to be a large problem. For GB and Germany, indications of the problem were found in 68 and 43 percent of the cases, respectively, where it was possible to identify it. For GB, structural interaction problems were also identified in some single vehicle accidents indicating that a benefit from improved compatibility could also be expected for this type of impact. It should be noted that structural interaction is the primary problem and it is not known how much it contributes to the stiffness mismatching and /or compartment strength problem.

It is recommended that further accident analysis should be performed to better quantify the magnitude of the compatibility problems for Germany. For both Germany and the UK further analysis should be performed in the future to check the conclusions of this work remain valid, as the vehicle fleet is constantly changing. Additional accident variables such as improved deformation measurements and harmonised impact severity measures would help future analyses.

Benefit analysis

Initial analyses to estimate the benefits of improved car compatibility were performed using GB and German accident data. Two different approaches were used. The first aimed to identify the number of casualties that could be expected to experience some reduction in injury risk from improved compatibility. The second aimed to predict the casualty savings resulting from the improved compatibility of cars. The second approach was only applied to the GB accident data.

The first approach, to determine the problem scope, indicated that a significant proportion of current road accident casualties would benefit from improved compatibility. In GB, for car frontal crash victims, it was predicted that approximately half (45 to 61%) of the fatalities and 2/3 (66-85%) of serious injuries would experience some reduction in injury risk as a result of improved compatibility. In Germany about half (33-67%) of current frontal crash victims would experience a reduction in injury risk.

It is expected that improved vehicle compatibility will result in far better occupant compartment integrity in frontal impact accidents. Thus, for the second approach it was assumed that improved vehicle compatibility would, pessimistically, eliminate injuries related to either contact with intruded parts of the vehicle interior, or optimistically, eliminate injuries related to contact with the vehicle interior whether it had intruded or not. It was then assumed that removal of these injuries from the existing accident data would quantify the benefits for the applicable occupant population. For GB, assuming compartment integrity is maintained for all impact severities, it was predicted that fatalities should be reduced by 40 to 60 percent and serious injuries by 11 to 29 percent, for car to car frontal impact collisions. These predictions can be regarded as an upper limit as it is unlikely that compartment integrity could be maintained for high speed impacts.

It is recommended that that an analysis to estimate the benefit of improved compatibility, in terms of the number of lives saved as opposed to the reduction in injury risk, should be performed for Germany. For GB, it is recommended that the benefit calculated for the car to car frontal impacts should be extended to cover other car accident configurations. Also, once more is known about the performance of a compatible car the assumptions made should be refined and the analysis repeated.

Crash testing

For this work package, 6 full width deformable barrier tests, 5 PDB tests, 1 car to car test and 9 EuroNCAP load cell wall (LCW) measurements were performed. This is 1 full width test and 2 EuroNCAP LCW measurements more than originally contracted.

Full width deformable barrier test to assess structural interaction

Two tests using a Mondeo car were performed to help redesign the barrier face to overcome the problem of small stiff protruding structures forming preferential load paths. The remaining tests were performed with an Astra, modified Astra, Laguna II and Rover 75. Subjective comparison of the results from the Astra and modified Astra tests showed that the modified Astra had a more homogeneous LCW force distribution which is consistent with the better structural interaction seen in the modified car to car test. However, the engine subframe to lower rail shear connection was not loaded as much in the full width tests as the car to car tests indicating that the full width test may not generate as much shear force across this type of connection as a car to car test. In the Laguna and Rover tests both lower rails and one lower rail bottomed out the barrier, respectively, to perform preferred load paths and apply large loads on the LCW, which most likely reduced the loads applied by other structures such as the subframe. Further work should be performed to ascertain whether this probable reduction in homogeneity is representative of the car's structural interaction performance in car to car collisions. Also the question of how far back a secondary load path can be positioned and still be able to contribute significantly to improving a car's structural interaction performance should be addressed. It is intended that the LCW results from the above tests will be used to help develop objective criteria to evaluate and quantify the changes observed between different vehicles in future work.

PDB test to assess structural interaction and frontal unit energy absorption

PDB tests were performed with a Mondeo, Range Rover, Astra, Smart and Volvo S80. It was concluded that the use of the load distribution measured on the LCW behind the barrier face did not give an accurate enough indication of a car's stiffness homogeneity to be used as an assessment measure. For the Mondeo test a part of the barrier remained attached to the car after the test. This would cause severe difficulties in measuring the barrier final deformation profile objectively, which the PDB approach is completely reliant upon. For this test and that of the Range Rover version 6 of PDB was used. For the Volvo S80 and the Smart tests version 7 of the barrier was used. This new version with a thicker front sheet may reduce or solve this problem. The PDB barrier was defined to represent an average car and its stiffness is such that bottoming out is unlikely, except for high mass vehicles. However, on the Range Rover test this barrier bottomed out. The implication of this should be considered in relation to future regulations and consumer testing. The test data collected in this project completes a crash test matrix, which will form a useful data set for future work to continue the development of the current assessment criteria.

Car to car test

The results of the Yaris to Clio car to car test demonstrated the poor structural performance of the Yaris. It should be noted that both of these cars had a EuroNCAP 4 star performance rating with the Yaris rated 'best in class'. It is recommended that this car could be used as a possible benchmark to help verify the full width and PDB tests and set the limit values for structural interaction performance for the proposed assessment criteria.

EuroNCAP test LCW measurements

The peak LCW forces measured were within the range measured for previous tests for vehicles of similar mass. However, the peak load cell distribution for the SUV was extremely inhomogeneous as the majority of the load was applied to a single load cell by the vehicle's lower rail. It was observed that the vertical distribution of the peak cell forces was in some cases influenced by the interaction of the engine and crossbeam with the load cell wall edge. This observation is important if it is proposed that the vertical force distribution measured in this test should be used as a criterion to control compatibility, as it may invalidate such a criterion.

CONTENTS

Page

1 INTRODUCTION

Following the introduction of the frontal and side impact Directives in October 1998, compatibility offers the next greatest potential benefit for improving car occupant safety and reducing road casualties. A Renault study (Steyer et al. 1998) has suggested that improved compatibility could reduce the number of serious injuries and fatalities by as much as a third where a car collides with one other vehicle.

Continuing the drive of the European Frontal Impact Directive and EuroNCAP the work performed to date for frontal impact has focused on the structural performance of the cars, with the aim of providing a safe environment in which the restraint system can operate. This approach is supported by the results of an accident study (Wykes et al. 1998), which show that the majority of the serious injuries received by belted occupants were contact induced as opposed to restraint system induced. Once the structure provides a safe environment within which the restraint system can operate, the next step for further improvement will be to control the compartment deceleration pulse. Following this, intelligent restraint systems could offer a way to cope with higher compartment decelerations, and give the occupant an optimised ride-down for a variety of impact severities.

The work performed in the 4th framework compatibility project has helped to understand compatibility. It concluded that for frontal impact an essential prerequisite for compatible cars is good structural interaction. Once this has been achieved some form of stiffness matching will be necessary to ensure that the impact energy is absorbed without exceeding the strength of the occupant compartment. The 4th framework project also outlined three possible test procedures to address these requirements in order to assess and control the compatibility of cars in frontal impact collisions. These are:

- A full width barrier test with a small depth of deformable barrier which uses a high resolution load cell wall to assess and control a car's local stiffness homogeneity. The aim of this test is to improve the structural interaction of cars in impacts.

- An Offset Deformable Barrier (ODB) test with a load cell wall. The aim of this test is to ensure that the global stiffnesses of cars are matched. This test is similar to the current frontal impact Directive test, except the Directive test does not require load cell wall measurements.

- An ODB test at a higher impact speed with a load cell wall to test the strength of the occupant compartment. This test would not require instrumented dummies.

Following the completion of the 4th framework project, development of the test outlines above and work to further the understanding of compatibility, has continued under government funded projects, mainly in the UK. This work has been reported at EEVC WG15 meetings and international conferences (Edwards et al. 2001 and Edwards et al. 2002).

The French, mainly Renault, have also proposed a test procedure to address compatibility issues (Delannoy and Diboine 2001 and Diboine and Delannoy 2002). This is an ODB test, which uses a recently developed Progressive Deformable Barrier (PDB). The main aim of this test is to improve the structural interaction of cars in impacts, although it does control stiffness as well.

Following further development, it is expected that these tests should form the basis of future legislation and / or consumer testing to improve compatibility. The 5^{th} framework compatibility project, which will continue this work, is not expected to start until November 2002. This project was initiated to continue the development of the tests in the intervening period. The results and recommendations from this project will be used as input for the 5^{th} framework project.

A consortium of European research institutions and a motor manufacturer was formed from members of EEVC WG15 (compatibility) to participate in this project. The partners were:

- BASt on behalf of Germany.
- Fiat on behalf of Italy.
- TRL Ltd. on behalf of the UK.
- UTAC on behalf of France.

2 OBJECTIVES

This project concentrates on the further development of the test procedures described above for frontal impact compatibility, accident analysis and a benefit analysis. The objectives of this project are:

- To further develop the crash test procedures detailed above.
- To perform an analysis to estimate the benefits of implementing compatibility measures for frontal impact.
- To perform accident analyses to further aid the understanding of compatibility and to support the cost benefit analysis.

3 THE CURRENT STATE OF COMPATIBILITY RESEARCH AND PROPOSED TEST PROCEDURES

The current state of compatibility research in Europe and world-wide is summarised as follows. In Europe, the EEVC started compatibility research in 1995. The EC made some 4th framework funding available to help with the early stages of this work. The European industry has also set up a group, ACEA/EUCAR, which has also had Commission funding for compatibility research. In the USA, NTHSA has a large compatibility project. Outside these groups, ADAC has carried out some test work, funded by DG VII. Some manufacturers, notably Renault, are also doing some internal research. The Canadians, and in particular, the Australians and Japanese are performing research work. The International Harmonisation of Research Activities (IHRA) Compatibility group acts as an international forum to bring together these international activities and interests. All of this work has lead to a fundamental understanding of compatibility. However, test procedures need to be developed so that compatibility can be implemented in the vehicle fleet, which is the ultimate aim of this work.

3.1 CURRENT COMPATIBILITY PROBLEMS

3.1.1 Structural Interaction

In rigid block crash tests, the block totally controls the way the impact deformation is distributed across the car's front. Cars designed for such tests have obtained good test performance, with limited numbers of frontal load paths having small frontal areas interacting with the block. However, when such cars impact each other, the chances of their stiff structures interacting is very limited. The Offset Deformable Frontal Impact test was intended to encourage manufacturers to increase the number of load paths being effective in car to car impacts. Unfortunately, so far, few manufacturers have taken advantage of the weight saving opportunities of this approach. Most have simply increased the stiffness of the car's main longitudinals, although some have had to weaken very stiff engine subframes. For load spreading, all cars now have substantial crossbeams between the main longitudinals but few other frontal connections have been improved. No cars currently have effective lateral connections, at the bonnet latch platform level, and few have any significant vertical connections between the lower load path and any upper load path. Consequently, when two cars collide, there is little to prevent the lateral fork effect, where the stiff members of one vehicle penetrate the soft areas of the other vehicle, due to lateral misalignment, or the over-riding of one car's structure by that of the other. With no control over the height of car structures, geometrical mismatches can give rise to over-riding from static misalignment. Even when structures are aligned statically, dynamic over-riding may occur. The result of these effects is that the impact energy is not absorbed efficiently, so the energy absorption capability of both cars is reduced, which leads to greater passenger compartment intrusion and increased injury in severe accidents.

The sensitivity of structural interaction with current cars has been demonstrated previously (Wykes et al. 1998). A 100 mm variation in ride height, in an impact between two identical cars, resulted in significant over-riding by the raised car over the lowered one. The energy absorption capability of both cars was compromised, resulting in greater intrusion for the lowered car at facia level and in the raised car at footwell level. Subsequent EUCAR simulation modelling indicated that over-riding can occur with a height difference of only 25 mm, with identical cars (EUCAR 2000). Even where structures are aligned vertically, dynamic pitch during the impact can lead to misalignment if the area of interaction is inadequate.

In order to achieve good interaction, it is important that the structures of each car meet something substantial on the other car to react against. Current views are that this is best achieved by utilising multiple load paths, with good links between them. These links may take the form of frontal inter-connections and / or shear connections set back from the front. Such structures should provide a more homogeneous front against which the other car's structure can react.

In addition to the provision of a homogeneous front, it is important that there is adequate vertical alignment. A low sports car could not interact with the front of a high off-road vehicle, even if they both had homogeneous fronts, because of their geometrical misalignment.

These aspects of compatibility are general to all impacts. They are not limited to those where there is a significant mass ratio between the cars. If impacting cars could be made to interact properly, their performance in accidents would become more predictable, in terms of energy absorption and deceleration. Apart from the resulting reduction in intrusion, this would help advanced restraint systems to perform correctly and predictably.

3.1.2 Frontal Stiffness

All current frontal impact crash tests place direct or indirect controls on energy absorption and deceleration of the car. If there is inadequate energy absorption in the frontal structure intrusion occurs which, at some level, will be detected by the instrumented dummies. Similarly, the dummies are sensitive to the car's deceleration, which is detected through such things as chest loading from the seat belt. However, there are currently no requirements controlling the frontal stiffness of the car. Indeed, the tests encourage heavier cars to be stiff, in comparison with lighter cars. As all the tests place a limit on the car's deceleration, through control of dummy loading, all cars tend to have similar stopping distances in the tests. The dummy's experience of deceleration is totally independent of the mass of the car it is travelling in. Data from EuroNCAP tests show that most cars, irrespective of size, have an overall ride-down distance of 1200 (+/- 200) mm (Figure 1). This includes the depth of the deformable barrier face of 540 mm. As most manufacturers aim to limit the length of the front structure, for a variety of reasons, crush depths tend to be kept to the minimum.

With the energy absorbed being the integral of force against distance, the only way to maintain the same crush depth, whilst at the same time absorbing the car's kinetic energy, is for the frontal stiffness to increase with vehicle mass. This means that, even without other influences, current frontal crash tests lead to a stiffness incompatibility between cars of different mass. Because stiffness is related to mass, but is not available in accident statistics, mass ratio has historically been incorrectly identified as the cause of compatibility problems.

Because, in general, heavier cars are stiffer than lighter cars in an accident between a heavy and light car, the heavy stiffer car over-crushes the light less stiff car causing greater occupant compartment intrusion and increased injury in the light car in severe accidents. To resolve this problem, vehicle stiffness matching is required. One way to control a vehicle's frontal stiffness is to limit the force imposed by the vehicle on its opponent, in the impact.

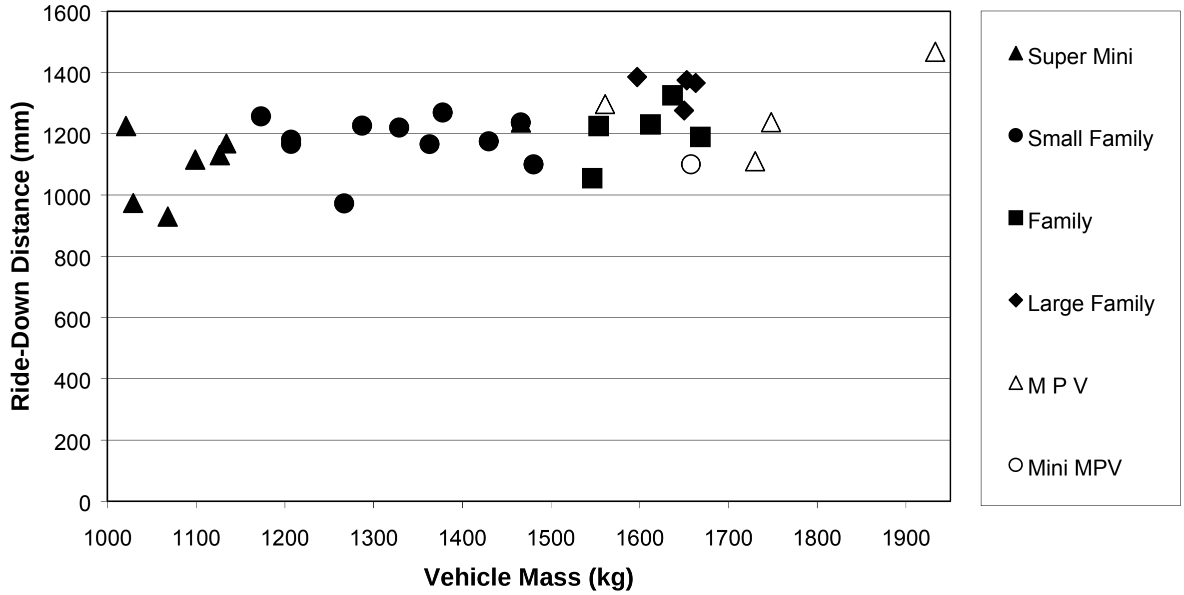

Figure 1: Ride-down distances recorded from EuroNCAP tests showing little variation with increasing mass. Note: Ride-down includes barrier depth of 540 mm.

3.1.3 Passenger Compartment Strength

Although a limit can be set for the force, which one car can impose on its opponent in an impact, this provides no guarantee that the passenger compartment can sustain the load imposed by another car. Where a car, which generated a force well below the limit, impacted one which generated a force near to the limit, there could be no confidence that its passenger compartment would survive. Furthermore, any slight variation in the impact configuration might affect the force levels. For these reasons, it will be necessary to have a requirement for the strength of the passenger compartment, ensuring that it can resist forces greater than those used to control frontal stiffness.

It is clear that the strength of the passenger compartment is dependent upon the load paths used to transmit forces to it. In a frontal impact the most important load paths are the main longitudinals, the upper longitudinals, the engine subframe, via the road wheel to the sill and via the engine to the firewall. The upper longitudinals and/or engine subframe may or may not be present. The way the loads are distributed between these load paths is dependent upon the car design, the impact configuration and the characteristics of the object hit. As the distribution of loads between the load paths varies, so the effective strength of the passenger compartment also varies. In order to ensure survival of the passenger compartment, cars should be designed to be tolerant of the distribution of the impact load. In principle this could be achieved by having a passenger compartment which is strong enough, irrespective of some variation in load path use, or by having a frontal structure that controls the way loads are distributed to the various load paths. The indications are that good structural interconnections control adjacent load paths to deform together and help to achieve this.

3.2 PROCEDURES TO ASSESS AND CONTROL FRONTAL IMPACT COMPATIBILITY

The first requirement for compatibility is to ensure good structural interaction. It helps to address problems seen in all impacts and without it any control of stiffness would have limited effect. With good structural interaction, it will then be possible to control frontal stiffness and passenger compartment strength. An inevitable consequence of these actions to reduce passenger compartment intrusion is that car deceleration will increase along with associated injuries, unless they are mitigated by improved restraint systems. Although any increase in injuries from deceleration is likely to be small compared with the decrease due to improved passenger compartment survival, there is going to be a growing need to understand the importance of and potentially control the shape of the deceleration pulse.

Two approaches using different test procedures have been proposed to assess and control frontal impact compatibility. It would be advantageous if some of the test procedures could be adaptations of current tests. The International Harmonisation of Research Activities (IHRA) Advanced Frontal Impact Working Group has recommended the universal use of two frontal tests. One, the Offset Deformable Barrier (ODB) test, as used in Europe, the other, a full width barrier impact as used in the USA.

The first approach proposed by the UK (TRL) uses a full width test to assess frontal homogeneity and hence structural interaction, a 64 km/h ODB test (such as the EuroNCAP one) for assessing frontal stiffness and a high speed ODB test to measure passenger compartment strength. All of these tests use a high definition Load Cell Wall, behind deformable barrier faces. With this approach, it is hoped that only one additional test is required for compatibility assuming that the other two tests are specified for frontal impact.

The second approach proposed by the French (Renault) uses a recently developed Progressive Deformable Barrier (PDB) face in an offset configuration test to assess the structural interaction and control the frontal unit energy absorption up to a defined Equivalent Energy Speed (EES), which is currently 50 km/h. The use of a 60 km/h (or perhaps 64 km/h) ODB test with a LCW is proposed to ensure a minimum force requirement is achieved, which should ensure self protection, i.e. a minimum passenger compartment strength and ability to absorb impact energy in the frontal unit up to an EES of about 55 km/h.

The basis of each of these tests is described in further detail below.

3.2.1 56 km/h Full Width Structural Interaction Test

As discussed earlier, cars with more homogeneous fronts offer the potential for good structural interaction with other cars. A full width impact of a car against a high definition load cell wall offers the potential to map the force deflection characteristics of the car’s front. However, there are some issues that generate problems when a rigid faced load cell wall is used:

- Localised stiff structures can hold off adjacent structures which are slightly set back

- Localised stiff structures effectively unload adjacent structures, which are slightly less stiff.

- The parts of the car that first impact the wall are decelerated instantaneously giving rise to large inertial forces, both within the structure and measured by the load cell wall. Such forces are not present in impacts with deforming structures, such as other cars.

- When the engine impacts the wall, it is brought to rest very rapidly again generating high inertial forces. In a car to car impact, the engine can rotate or move slightly out of the way of the other car's engine, so reducing its deceleration.

- No relative shear is generated in the front structure to exercise any shear connections between load paths.

In order to overcome these problems, a deformable barrier face is fitted to the front of the load cell wall. If the test is to also function as a high deceleration test for frontal impact, the overall car deceleration should not be significantly affected by the addition of the deformable face.

In summary, a full width test at 56 km/h with a deformable barrier face and a high resolution LCW is proposed to assess and control structural interaction. This will be achieved by controlling the force distribution measured on the LCW, to encourage the development of structures that behave in a more homogeneous manner.

3.2.2 64 km/h ODB Test for Frontal Stiffness

As with the full width test, a load cell wall is used to measure the forces generated by the car in an ODB test at 64 km/h. This requirement can simply be added to the current EuroNCAP test. As previously reported (Edwards et al. 2000), the load measured is a combination of the force coming from the deceleration of the passenger compartment (structural component) and the force coming from the deceleration of the mainly rigid masses ahead of the firewall (mechanical component), a large proportion of which is due to the engine and gearbox. In setting a limit for this force, it is necessary to consider the extent to which the engine force needs to be taken into account. In a car to car impact some of the engine load directly acts on the engine of the other car and has little effect on the structure. The remaining load does act on the structure, either directly or indirectly. The deformable face can attenuate the force to decelerate the engine and this may allow the maximum total force measured by the load cell wall to be used.

There may also be a need to set a minimum force level for the car front, so producing a range for the acceptable forces. This would prevent the design of small cars with excessively soft fronts, where the deceleration pulse might have to increase rapidly, when the front structure bottoms out on the strong passenger compartment. Such deceleration pulses are known to be injurious. It is unlikely that a minimum force requirement would come into play for larger cars, as there is no indication that any manufacturer has an interest in producing a long soft fronted car.

In summary, a 64 km/h ODB test with a LCW is proposed. From the load cell wall measurements, the car's frontal stiffness would be controlled by specifying that the peak force recorded should lie within a specified range.

3.2.3 80 km/h ODB Test for Passenger Compartment Strength

The frontal stiffness test described above only provides information about the car's ability to cope with loads up to that generated by the car itself. It is necessary to be able to show that its passenger compartment can survive the forces imposed by another car, which may generate a higher frontal force but still be within the requirements. This requires that an assessment be made of the passenger compartment's strength. It is proposed that this should be measured in a further ODB test carried out at an elevated speed. Currently a speed of 80 km/h is being used. It should be pointed out that there is no intention to require that cars provide a survivable performance for the occupants, at this severity. The test is simply designed to measure the strength of the passenger compartment, so would not require instrumented dummies.

If the passenger compartment becomes unstable in the impact, it will be necessary to ensure that the strength measured is prior to any major intrusion occurring. Once the passenger compartment becomes unstable, the measured load can be expected to reduce but it might again increase if subsequent structural blocking occurs. However, with conventional car designs this is unlikely.

In summary, a high speed, possibly 80 km/h, ODB test with a LCW is proposed to assess the strength of the passenger compartment by measuring the end of crash load recorded on the LCW.

3.2.4 60 km/h PDB Test for Structural Interaction and Frontal Stiffness

With a rigid wall test no relative shear is generated in the car's front structure to exercise the shear connections between load paths. As it is believed that vertical and lateral shear connections are necessary to ensure good structural interaction, it is proposed that an offset test with a deep deformable barrier that does not bottom out should be used to assess a car's structural interaction potential. It is proposed to assess the structural interaction potential from a measurement of the final deformation of the barrier; a more uniform deformation would indicate a more compatible structure. In addition the maximum barrier deformation depth would also be used to control the car's frontal energy absorption capability, i.e. stiffness, up to an Equivalent Energy Speed (EES) of 50 km/h with the currently proposed barrier design.

In summary, a 60 km/h ODB test with a recently developed progressive deformable barrier (PDB) face is proposed to control a car's structural interaction and frontal stiffness up to an EES of 50 km/h from a measurement of the barrier's final deformation profile.

3.2.5 60 km/h ODB Test for Self Protection

The 60 km/h PDB test controls the car's frontal stiffness (or energy absorption capability) up to an Equivalent Energy Speed (EES) of 50 km/h by effectively ensuring that the force level does not exceed a maximum specified value using the barrier deformation as a measure. To ensure that all cars can withstand this maximum force level a 'self protection' test is proposed in which the force level measured by a Load Cell Wall behind the barrier face would be controlled to ensure that it would be above a specified minimum value.

In summary, a 60 km/h ODB test with the EEVC barrier is proposed to control a car's self protection capability by ensuring that the peak force measured on a LCW positioned behind the barrier face exceeds a specified minimum value.

4 ACCIDENT ANALYSIS

4.1 INTRODUCTION

As mentioned previously, the 4th framework compatibility project helped to understand the compatibility problem and concluded that the major problems were structural interaction, stiffness matching and compartment strength. This accident analysis is being carried out to confirm these conclusions, further understand the problems and attempt to quantify the magnitude of the problems. In depth accident databases are required to perform this type of analysis, few of which exist for Europe. TRL and BASt will carry out the analysis for GB and Germany, respectively. TRL will use the Co-operative Crash Injury Study (CCIS) database and BASt will use the Medical University of Hanover in-depth accident databases.

4.2 TRL ANALYSIS FOR GB

4.2.1 Objectives

The main aim of this analysis was to perform detailed accident case studies in order to identify the major compatibility problems in car to car collisions, namely structural interaction, stiffness mismatching and compartment strength, and attempt to quantify the magnitude of these problems. The secondary aim was to investigate if the identified problems were present in single vehicle accidents, for example where a vehicle has collided with roadside furniture.

4.2.2 Approach

The analysis was undertaken on a case by case basis. Each accident case in the sample was assessed using the photographic evidence and detailed case information collated by the vehicle examiners, which included details of crash severity, compartment intrusion and occupant injuries. Each case was analysed individually in order to identify any compatibility problems that may have been present in the collision. Specific attention was paid to the structural performance of the vehicles, in particular structural interaction between vehicles, and any signs of a stiffness mismatch or compartment strength problem.

The compatibility problems that were identified in the analysis were illustrated with detailed case examples, which are presented in Section 4.2.4.

For each accident case a judgement was made, whether or not some or all of the major compatibility problems identified by the previous analysis were present. This allowed the magnitude of the major compatibility problems to be quantified, resulting in an estimate of the scale of compatibility problems in car to car frontal collisions.

In addition, a selection of single vehicle accidents were investigated in order to discover if compatibility issues were also present in collisions with static roadside objects such as trees and lampposts. A number of these cases were assessed in a similar way to the car to car collisions, using information and photographs from the case files. In each of the cases the behaviour of the structure of the vehicle was examined for signs of compatibility problems.

4.2.3 Data Source

The data source used for this analysis was the Co-operative Crash Injury Study (CCIS). CCIS is one of the world's largest studies investigating car occupant injury causation following real world crashes. The project started in-depth car accident investigations in 1983, building on previous UK crash-injury studies.

CCIS has the aim of investigating how car occupants are injured in crashes and helping to develop injury countermeasures, such as legislation and design solutions for improved car occupant protection. This is achieved by:

- Investigating the pattern of occupant injuries and their severity for all car crash scenarios
- Defining the population and relative frequency of different crash types
- Developing an understanding of the causes and mechanisms of injury to car occupants
- Investigating the effect of vehicle safety features on occupant injuries and providing information on the crashworthiness of vehicles
- Identifying the needs for improved vehicle safety as changes take place
- Providing biomechanical information for researchers who are developing crash test dummies.

Accidents within seven areas of England are selected for detailed investigation according to strict criteria. Detailed examinations of approximately 1500 accident-damaged cars and car-derived vans are carried out per year. CCIS vehicle investigations are typically carried out between 24 and 72 hours after the collision at a recovery garage. There is a bias towards relatively new cars. Crashes are not selected unless at least one involved car is less than seven years old at the time of the accident, contains a 'police officer' defined injured occupant and is towed from the scene. However, once a crash is selected for investigation, every effort is made to examine all car and car-derived vehicles within it. Vehicles can only be examined if they are towed to a recovery garage, or other accessible place.

For a selected crash to be investigated a stratified sampling procedure is applied. Each regional team is contracted to investigate a fixed number of vehicles per year. All of the selected crashes that are identified as having a 'police defined' fatally (killed) or seriously injured (KSI) occupant in a towed car less than seven years old at the time of the accident must be investigated. From the remainder of selected crashes, those with slightly injured casualties are randomly chosen for investigation to ensure the contractual number of vehicles is examined annually. Therefore, the CCIS database has both biases towards newer vehicles and more seriously injured casualties.

The associated occupant injuries and characteristics (gender, age, height, weight etc.) are obtained from either questionnaires, hospital records or HM coroner reports depending on the casualties' injury severity. The injuries sustained are coded using 'The Abbreviated Injury Scale (AIS) 1990 Revision' *(Association for the Advancement of Automotive Medicine).* This coding allows injuries to be coded by their type and severity or threat to life. The injuries are then correlated with the associated vehicle damage.

The data in CCIS is collected by skilled vehicle examiners and includes detailed computerised records of the vehicle damage, evidence of occupant contacts with the vehicle interior and the seat belt or airbag use or status. Although the data is not solely collected with the specific intention of identifying compatibility issues, there is information within the database that allows these effects to be determined. For example, variables such as damage to longitudinal box sections and upper rails (shotguns), detailed information about the deformation to the vehicle, measurements of changes to door aperture sizes and compartment intrusion are recorded.

Case Selection Criteria

Cases were chosen for this analysis from CCIS Phase 6, which started in June 1998 and completed in November 2002. At the time the analysis was undertaken there were 2,113 crashes available for analysis containing information on 2,776 vehicles, 4,281 occupants and 15,898 injuries.

The principal selection criteria were to identify crashes where two cars or car derivative vehicles had struck, both with frontal impact damage. The selection criteria were:

- One impact to the vehicle
- The point of initial action of the force is on the front of the vehicle
- The force direction is between 11 o'clock and 1 o'clock on the vehicle; where 12 o'clock is straight ahead

This yielded a sample of 656 crashes containing details on 842 vehicles. To fully investigate the crashes in terms of potential compatibility issues it was essential to then only select cases where both had frontal impact damage from the partner car and both vehicles had been comprehensively investigated. This further criterion reduced the cases available for detailed analysis to 162, or 324 vehicles.

4.2.4 Examples of Compatibility Problems

This section presents detailed case accident studies that have been chosen to highlight specific compatibility issues in real life accidents. These were poor structural interaction and frontal stiffness / compartment strength mismatch.

The detailed case examples in this section were selected to illustrate the major compatibility issues that were identified in the analysis. A total of eight cases are presented here, of which seven were car to car impacts and one was a single vehicle accident. In each case the vehicle damage was assessed, the intrusion measurements were analysed and the occupant injuries were recorded. However, in general these case examples were selected to illustrate primarily how the vehicles' structures performed in the collision.

The crash severity parameter used is either the Equivalent Test Speed (ETS) or the Delta-V. The ETS is calculated on the assumption that the car's deformation or crush was caused by an impact with an immovable rigid object. Delta-V is calculated as the car's change in velocity during the impact. The ETS and Delta-V cannot always be calculated due to lack of information either about the vehicle's physical properties (stiffness or mass) or vehicle damage from a subsequent impact that prevents the actual crash damage being accurately measured. In summary the crash severity, ETS or Delta-V, is calculated from knowledge of the vehicles residual crush, stiffness characteristics, mass and the direction of the impact.

Each case has an estimated value for the overlap between the vehicles at the start of the impact. The values for overlap were estimated from the amount of direct contact damage with the other vehicle or object, which were measured by the vehicle examiners.

The intrusion measurements are taken at three areas of the vehicle interior. The first is taken from the joint between the A-pillar and the top of the facia or bulkhead. The second measurement is taken from the facia, at the area where the occupant's knees are likely to contact. The last level recorded is in the footwell. These three measurements give a description of the intrusion rearwards into the compartment on both the offside and the nearside. Where possible steering wheel movement is also recorded.

Occupant injuries are coded using the AIS scale, and where possible the injury causation is assessed by examining the damage to the vehicle interior.

All of the cases in this section have been taken from the CCIS database sample described in the previous section, with the exception of one case that was taken from the On The Spot (OTS) accident investigation project. The reason for including this case was that it is a particularly good example of one type of structural interaction problem.

4.2.4.1 Structural Interaction

In rigid block crash tests, the block totally controls the way the impact deformation is distributed across the car's front. Cars designed for such tests have obtained good test performance, with limited numbers of frontal load paths having small frontal areas interacting with the block. When such cars impact each other, the chances of their stiff structures interacting is very limited. The Offset Deformable Frontal Impact test was intended to encourage manufacturers to increase the number of load paths being effective in car to car impacts. Unfortunately, so far, few manufacturers have taken advantage of the weight saving opportunities of this approach. Most have simply increased the stiffness of the car's main longitudinals, although some have had to weaken very stiff engine subframes. For load spreading, all cars now have substantial crossbeams between the main longitudinals but few other frontal connections have been improved. No cars currently have effective lateral connections, at the bonnet latch platform level, and few have any significant vertical connections between the lower load path and any upper load path. Consequently, when two cars collide, there is little to prevent the lateral fork effect, where the stiff members of one vehicle penetrate the soft areas of the other vehicle, due to lateral misalignment, or the over-riding or one car's structure by that of the other. With no control over the height of car structures, geometrical mismatches can give rise to over-riding from static misalignment. Even when structures are aligned statically, dynamic over-riding may occur.

Where there is good structural interaction in a frontal car to car collision it would be expected that the frontal structure of both vehicles would undergo homogeneous deformation over the damage width. This would result in the lower and upper rails deforming back to approximately the same crush depth. This would allow efficient energy absorption in the frontal structure, thereby reducing the amount of impact energy that has to be absorbed by the occupant compartment. This would reduce the likelihood of compartment intrusion, maintaining an occupant survival space that would allow advanced restraint systems to function efficiently.

Good structural interaction, and subsequent homogeneous deformation of the frontal structure, may be seen in cases where a vehicle has impacted a large rigid object, such as a large concrete block. Structural interaction with the block is guaranteed due to its rigidity, allowing all of the structure of the vehicle over the area of direct contact to deform, irrespective of its stiffness. This type of collision can be used to illustrate the homogeneous deformation seen as a result of good structural interaction. The example chosen is a collision between a Seat Alhambra and a bridge parapet.

Seat Alhambra v bridge parapet

A left hand drive Seat Alhambra was involved in a full frontal collision (100 percent overlap) with a bridge parapet after the vehicle left the carriageway. The impact was of high severity, the calculated value for Delta-V being 113km/h. The vehicle was fitted with a steering wheel airbag.

The front and side views of the vehicle (Figure 2) show that the frontal structure has deformed substantially over its entire width, thereby absorbing a significant amount of the impact energy. Good structural interaction with the solid bridge parapet has led to homogeneous deformation of the frontal structure. This, in combination with the width of the impact, has enabled the energy absorbing structures, such as the lower rails, to absorb a large proportion of the impact energy. This has ensured that less of the impact energy has been absorbed by the occupant compartment. This has resulted in a relatively small compartment intrusion for the high impact severity.

Figure 2: Seat Alhambra showing homogeneous deformation of frontal structure

The intrusion on the offside was similar at the three recorded facia and footwell levels, with values in the region of 340mm to 430mm (Table 1). There was less intrusion on the nearside.

The driver, a 25 year old male, sustained severe injuries but survived the collision, despite the high severity. The most severe injuries were lung contusions (AIS 4). These were caused by chest contact with the steering wheel. There were also several other serious injuries, including right lung haemothorax and left lung pneumothorax (AIS 3). These were also the result of the chest contact with the steering column. There were also upper leg and hip injuries (AIS 3), resulting from contact with the steering column and rigid bracketry behind the facia.

Table 1: Compartment intrusion measurements

	Offside (mm)	Nearside (mm)
Joint between A-pillar & facia top	340	0
Knee contact area	430	80
Footwell	360	320

The way in which the Seat Alhambra has deformed in this case shows how the car can perform in a collision if there is good structural interaction. The interaction between the structure of the car and the solid bridge parapet allowed efficient energy absorption forward of the occupant compartment. This led to a reduction in the energy absorbed by the compartment, resulting in a reduction in the levels of intrusion that would have normally been expected in a collision of this severity.

Lateral Misalignment

Lateral misalignment can occur when the longitudinal box sections of the two vehicles do not line up horizontally in a collision. It often gives rise to the so called "fork effect". The structural members of the vehicles load the less stiff areas between the longitudinals of the opposing vehicle. This can lead to passenger compartment intrusion due to poor energy absorption in the front structure.

Daewoo Espero v Mazda 626

A Daewoo Espero collided with a Mazda 626. The impact was nearside to nearside, with an overlap of approximately 65 percent. The calculated values for Delta-V were 53km/h and 54km/h for the Daewoo and Mazda respectively. The Daewoo was fitted with a steering wheel airbag, and the Mazda had a steering wheel airbag and a passenger airbag.

The greatest deformation on the Daewoo is between the longitudinals (Figure 3). The front of the Mazda has a fairly uniform deformation over the area of direct damage on the nearside (Figure 4). The deformation of the Daewoo is a result of direct contact between the nearside longitudinal of the Mazda and the less stiff structure between the longitudinals of the Daewoo.

Figure 3: Daewoo Espero

Figure 4: Mazda 626

The compartment intrusion was mainly restricted to the nearside in both vehicles. This intrusion was minor, no more that 30mm at any of the recorded levels. The only intrusion on the offside in either vehicle was the Daewoo steering wheel, which had moved 60mm towards the driver.

There were four occupants in the Daewoo. The most severe injury was a lumbar spine fracture (AIS 3), sustained by the rear nearside passenger, a six year old male. The injury was caused by the seat belt. The driver sustained broken bones in the lower leg (AIS 2) from contact with the facia panel. The other occupants in the Daewoo sustained minor injuries. These were mainly bruising to the chest and abdomen (AIS 1), caused by the seat belt webbing.

The Mazda had two occupants. Both sustained moderate injuries. The driver suffered a fracture of the left foot (AIS 2) from contact with the pedals, which had not moved towards the driver. The passenger suffered a fractured sternum (AIS 2), caused by the seat belt webbing.

This case has been included as an example of lateral misalignment resulting in the “fork effect”. A large proportion of the injuries sustained by the occupants of both vehicles were deceleration injuries from the seat belts. These injuries may have been influenced by the deceleration pulse of this collision. There may have been an initially soft pulse that stiffened towards the end of the collision. This may have been caused by the initial deformation of the less stiff sections of the Daewoo, followed by direct contact between the engine and the Mazda longitudinal (Figure 5).

Figure 5: Comparison of the damage profiles of the Daewoo and Mazda, showing deformation of less stiff structure between longitudinals on the Daewoo

Vertical Misalignment

Vertical misalignment is where the main energy absorbing structures of the two vehicles do not line up vertically in a collision. This can lead to overriding and underriding, where the main structural members of one car ride up over those of the opposing car. This misalignment can lead to loading of the less stiff areas of the frontal structure in the opposing vehicles by these stiffer members. This often results in poor energy absorption in the frontal structure of both vehicles, leading to increased occupant compartment intrusion.

There are two types of vertical misalignment problem, static and dynamic. Static misalignment is due to a geometric mismatch of the vehicle structures, where the longitudinals are located at different heights above the ground and therefore do not directly interact when the vehicles collide. This is often seen in collisions between standard cars and vehicles with high ride heights, such as “Sport Utility Vehicles” (SUVs).

Dynamic misalignment is where the longitudinals may line up geometrically before a collision, but override or underride each other during the impact. This may be due to a number of factors that occur during the impact. For example, the motion of the vehicles may have an effect on the structural alignment. The collapse mode of the vehicle structure, how it crushes or bends under direct loading, may also be different in the opposing vehicles.

Static Misalignment

Mazda RX7 v Land Rover Discovery

A collision between vehicles with significantly different structural heights, such as a sports car and an SUV, would be expected to have serious structural interaction problems. This can be illustrated by an accident case involving a Mazda RX7 and a Land Rover Discovery (Figure 6 & Figure 7).

Figure 6: Mazda RX7

Figure 7: Land Rover Discovery

The collision was an offset impact, with both vehicles impacting on the offside front. The collision can be considered to be minor, with values for Delta-V of 35km/h and 24km/h for the Mazda and Discovery respectively. The Mazda was fitted with a steering wheel airbag, and there were no airbags in the Discovery.

There was a significant difference in the damage profiles sustained by the two vehicles (Figure 8). The Mazda has the most deformation at bonnet level, and at the top of the wing. This is significantly higher than the main frontal structure, which has been largely unaffected. The Discovery has less severe deformation, with damage to the lower levels of the front of the vehicle below the main structure. The Discovery's offside road wheel has been directly loaded and displaced rearwards. The main interaction between the vehicles has been from contact of the Discovery's offside road wheel and the Mazda's bumper beam.

Figure 8: Comparison of the damage profiles of the Mazda and Discovery, showing the difference in the height where direct contact has occurred

Both compartments remain largely undeformed, the damage to both vehicles being limited to the area forward of the compartments. There was no intrusion in the Discovery, and only slight intrusion in the offside footwell of the Mazda (70mm).

There were two occupants in each vehicle. The occupants of the Mazda suffered cuts and bruises (AIS 1) from the broken glass, seat belts and contact with the facia panel. The front seat passenger in the Discovery suffered a fractured sternum (AIS 2) from the seat belt webbing. The driver sustained minor injuries.

This collision is an extreme example of static vertical misalignment. The main structures of the two vehicles involved were at significantly different heights. Subsequently, there was poor interaction between them. There were several deceleration injuries sustained by the occupants of the two vehicles. These may have been influenced by the deceleration pulse, which may have been initially soft and suddenly stiffening. Despite the low severity of the collision, the structural interaction problems in this case are easy to identify.

Renault Clio v Ford Focus

Static vertical misalignment is not restricted to collisions between vehicles of different size, such as cars and SUVs. Slight differences in the structural height of different makes and models of cars can have a similar effect. This can be seen in the case of a head-on collision between a Renault Clio and a Ford Focus (Figure 9 & Figure 10).

Figure 9: Renault Clio

Figure 10: Ford Focus

The overlap between the two vehicles was approximately 70 percent, the vehicles impacting offside to offside. The calculated values for Delta-V were 58km/h for the Clio and 43km/h for the Focus. The Focus was fitted with a steering wheel airbag, whereas the Clio did not have an airbag fitted. There was one occupant in each vehicle.

From the side it can be seen that the damage sustained by the two vehicles differs significantly (Figure 11). On the Clio the offside upper rail has been directly loaded and crushed back towards the A-pillar. It has deformed more than the main offside longitudinal. The offside road wheel has displaced rearwards and contacted the sill at the base of the A-pillar.

On the Focus the upper rail has not deformed so significantly. It has deformed, but not to the extent that can be seen in the Clio. The main longitudinals of the Focus have deformed more in this case. The front of the vehicle appears to have been pushed upwards. The offside road wheel has also displaced rearwards slightly.

Figure 11: Comparison of the damage profiles of the Clio and Focus, showing that the Clio has been overridden

The Clio sustained greater intrusion than the Focus at all levels on the offside, with a maximum intrusion of 100mm at the joint between the A-pillar and the top of the facia (Table 2). There were no measurements available for the steering wheel movement in the Clio, as it was removed by the emergency services. However, it is likely that there was some steering wheel movement in common with the movement of the facia. The Focus had a maximum intrusion of 40mm at footwell level, and no more than 10mm at the other recorded levels.

Table 2: Comparison of compartment intrusion

	Renault Clio		Ford Focus	
	Offside (mm)	Nearside (mm)	Offside (mm)	Nearside (mm)
Joint between A-pillar and top of facia	100	0	10	0
Knee contact area	70	0	10	0
Footwell	90	0	40	0

The occupant injuries in the two vehicles were significantly different. The driver of the Clio, a 65 year old male, was killed and the driver of the Focus, a 35 year old male, received minor injuries. The driver of the Clio suffered contusions to both lungs (AIS 4) and laceration of the left lung with extensive internal bleeding (AIS 4). These injuries were caused by steering wheel contact. In addition, the Clio driver suffered several broken ribs (AIS 3), also the result of steering wheel contact, and numerous minor injuries. The Focus driver sustained only one injury, a laceration with swelling to the rear of the head (AIS 1). This was the result of contact with the offside B-pillar.

The probable intrusion of the steering wheel in the Clio is likely to have had a significant effect on the chest injuries sustained by the driver. The intrusion would have reduced the distance between the driver and the steering wheel, making it more difficult for the seat belt to prevent injurious steering wheel contact. The absence of a steering wheel airbag may have also been significant. The age difference between the drivers may have had some effect on the injury severity difference, but this is unlikely to have been a large contributory factor.

The Focus has overridden the Clio in this case. The deformation profile and the intrusion measurements of both vehicles show signs of overriding by the Focus. This is due to static vertical misalignment of the structures of the two vehicles. The relatively high bumper beam and longitudinals of the Focus have overridden the structure and offside road wheel of the Clio. The structural misalignment has led to inadequate energy absorption in the frontal structure of the Clio. This has also formed load paths into the compartment, which are probably less able to absorb the impact energy and support the applied load. This has resulted in increased intrusion into the occupant compartment, which has most likely directly influenced the injury outcome.

BMW 3-Series v Volkswagen Sharan

A further example of static vertical misalignment can be seen in the collision between a BMW 3-series and a Volkswagen Sharan (Figure 12). The case was taken from a different data set, the OTS (On The Spot) accident investigation project, and does not form part of the CCIS database. The reason for including this case was that it is a particularly good example which illustrates the under/override problem caused by static vertical misalignment of the main vehicle structures.

Figure 12: BMW v Sharan Accident Scene

The accident was a frontal offset impact with an overlap of approximately 65 percent. The estimated values for Delta-V were 45km/h for the BMW and 36km/h for the Sharan. The BMW was thought to have had a less severe prior impact on the offside wing from contact with another vehicle, but it is unknown how significant it was to the accident outcome. The Sharan was fitted with a steering wheel airbag, but the BMW had no airbag fitted.

Comparison of the two vehicles shows that there was a significant difference in the damage sustained (Figure 13 & Figure 14). The longitudinals and bumper beam on the BMW were protruding from the front of the vehicle. The bumper beam was only attached to the offside longitudinal, having come away on the nearside. The longitudinals were mostly undamaged, although the offside had been bent downwards slightly. The rest of the front of the vehicle had been crushed back to a large extent.

Figure 13: BMW 3-series

Figure 14: Volkswagen Sharan

The Sharan had received damage to the bumper beam on the offside. The longitudinals had not deformed to a large extent. There was generally less deformation to the rest of the front of the Sharan when compared to the BMW.

There was no numerical data for the intrusion in either vehicle. However, both vehicles were examined to ascertain general levels of intrusion. Examination of the BMW showed that there had been minor intrusion of the facia and moderate intrusion of the footwell. The Sharan was found to have no intrusion.

The driver of the BMW was a 52 year old female. The most severe injury the BMW driver suffered was a fractured clavicle (AIS 2), most likely to have been caused by the seat belt. Other injuries sustained were bruising to the forehead and laceration to the nose (both AIS 1), caused by contact with the steering wheel, and bruising to the abdomen (AIS 1), probably caused by a combination of the seat belt and contact with the steering wheel. The driver of the Sharan suffered bruising to the arm and lower leg (AIS 1), possibly from the deployment of the airbag and contact with the lower facia respectively.

The main structures of both vehicles have not absorbed much energy in this collision (Figure 15). The longitudinals on the BMW have been overridden by those on the Sharan. This has led to direct loading of the less stiff areas of the BMW by the stiff structure of the Sharan. The longitudinals of the BMW have gone under the Sharan, where there does not appear to be any significant part of the vehicle to contact.

Figure 15: Comparison of damage to BMW and Sharan, showing lack of energy absorption in the frontal structures

The occupant injuries suggest that the crash severity may have been overestimated by the investigators. This may have been due to the large amount of crush on the BMW resulting from the lack of deformation of the main structure. It is likely that the deceleration pulse shape is back loaded, i.e. low deceleration at the beginning of the impact and high at the end of the impact, because of the poor structural interaction. This may have been a contributory factor to the fractured clavicle sustained by the BMW driver.

Static misalignment of the frontal structure has led to overriding by the Sharan. This has led to a significant reduction in crash performance for both vehicles.

Dynamic Misalignment

Two of the requirements for compatibility in car-to-car collisions are good structural interaction and stiffness matching. In a head-on collision between two identical cars it would be expected that any structural interaction or stiffness problems should be minimised. This should result in similar vehicle damage characteristics, and similar injuries for the occupants of the opposing cars. However, this is not always the case. Dynamic misalignment can lead to overriding and underriding, even in identical vehicles.

Ford Focus v Ford Focus

A Ford Focus collided head-on with another Ford Focus, the impact having an overlap of 100 percent (Figure 16 & Figure 17). As a result of the collision the first Focus spun and performed a half roll, coming to rest on its roof. The calculated values for Delta-V were 89km/h and 82km/h for the first and second Focus respectively. Both vehicles were fitted with steering wheel airbags.

Figure 16: First Ford Focus

Figure 17: Second Ford Focus

Examination of the photographic evidence reveals a difference in the damage profiles of the two vehicles (Figure 18). The bumper beam of the first Focus has been folded backwards over itself. The longitudinals appear to have been bent downwards. The bumper beam of the second Focus appears to have displaced directly rearwards, suggesting more axial deformation of the longitudinals. This difference shows the different failure mechanisms of the main structures for the two vehicles.

Figure 18: Comparison of front profiles of the two Focuses, showing that the first Focus (left) has been overridden

The first Focus has undergone larger deformation at higher levels, the front of the car having greater crush near the top of the bulkhead than at the lower levels. The second Focus has comparatively more deformation at the lower levels below the main structure of the vehicle. The recorded values for maximum crush were 1170mm for the first Focus and 900mm for the second Focus, highlighting the difference in damage between the two vehicles.

Both vehicles have sustained moderate levels of intrusion on the offside (Table 3). The first Focus had between 170mm and 240mm at the three recorded intrusion levels, and the second Focus had between 90mm and 190mm. The first Focus also had greater intrusion on the nearside. The most significant intrusion difference between the vehicles was the rearwards movement of the steering wheel. The steering wheel movement in the first Focus was double that in the second Focus.

Table 3: Comparison of compartment intrusion

	First Focus		Second Focus	
	Offside (mm)	Nearside (mm)	Offside (mm)	Nearside (mm)
Joint between A-pillar and top of facia	240	310	190	80
Knee contact area	170	230	90	70
Footwell	190	200	150	110
Steering wheel movement	200	-	100	-

Both vehicles had one occupant. The driver of the first Focus suffered extremely severe injuries. The driver of the second Focus survived with serious injuries. Both drivers were females of similar age.

The driver of the first Focus suffered complete transection of the spinal cord and separation of the skull from the spinal column (AIS 6). This injury was most likely caused by the airbag-equipped steering wheel, which had intruded towards the driver. She also suffered serious injuries to the chest and head. The most severe injury that the driver of the second Focus suffered was a compound fracture of the right knee (AIS 3). This injury was the result of contact with rigid bracketry behind the facia panel, which had intruded. The other injuries sustained included fractures to bones in the lower leg (AIS 2).

Both vehicles in this case have had inefficient energy absorption in their frontal structures. A contributory factor to this was most likely the dynamic misalignment of the main structures of the two Focuses. It is likely that the structures of the identical cars would have lined up initially, but as the collision progressed they became misaligned, leading to overriding. This may have been caused by the movement of the vehicles, or the failure mechanisms involved in deforming the frontal structure. Examination of the vehicle damage and occupant injuries show that the first Focus has been overridden by the second Focus in this case.

Low Overlap

A low overlap collision can be defined as when the opposing vehicles contact each other over a small area outside of the main structure of the vehicles. The main structural members are not involved in the collision, so there is very little energy absorption in the frontal structure due to the low stiffness of the bodywork such as the wing. It is likely that much of the collision energy would be absorbed by the passenger compartment, sometimes by loading of the A-pillar and sill by the front wheel. This can result in intrusion into the occupant compartment as the energy is absorbed.

Rover 416 v Nissan Micra

A frontal impact occurred between a Rover 416 and a Nissan Micra (Figure 19 & Figure 20). The overlap was approximately 20 percent. The collision occurred at low speed, Delta-V calculated for the Rover and Micra of 12km/h and 15km/h, respectively. The impact was severe enough for the steering wheel airbag to fire in the Rover. The Micra was not fitted with an airbag.

Figure 19: Rover 416

Figure 20: Nissan Micra

Examination of the photographic evidence shows that the main frontal structures have had no significant involvement in the collision. The main longitudinals have not deformed significantly in either vehicle, indicating that they have not interacted with each other. The upper rails can be seen clearly on both vehicles, with neither having any major deformation. This has resulted in minimal absorption of collision energy in the frontal structure. The only interaction of the two vehicles appears to have been between the front offside road wheels. These have been displaced rearwards and contacted the sill in both vehicles. The majority of the collision energy has therefore been absorbed by the occupant compartment, which was loaded via the road wheel to sill load path. Subsequently, both vehicles have significant deformation of the compartment.

In the Rover the A-pillar has been displaced rearwards at facia level, forcing the roof upwards in front of the B-pillar. The A-pillar has also buckled near the base of the windscreen. This has reduced the door aperture, with the offside front door having been forced rearwards into the B-pillar. The door was unopenable, having been split by the B-pillar.

The Micra has deformed in a different manner to the Rover, with most of the deformation occurring in the front offside sill. The sill has been forced upwards and inboard significantly at the base of the A-pillar. This is where most of the collision energy has been absorbed by the Micra, and there is only slight deformation of the roof due to loading through the A-pillar.

The occupant compartment intrusion in both vehicles was confined to the offside (Table 4). The intrusion pattern shows that both vehicles had similar intrusion at the knee contact areas, but the Rover had more substantial intrusion at the joint of the A-pillar and the top of the facia. The Micra had most intrusion at footwell level.

Table 4: Comparison of compartment intrusion

	Rover 416		Nissan Micra	
	Offside (mm)	Nearside (mm)	Offside (mm)	Nearside (mm)
Joint between A-pillar and top of facia	12	0	0	0
Knee contact area	11	0	11	0
Footwell	2	0	13	0

There was only one occupant in each vehicle, both sustaining minor injuries. The Rover driver suffered bruising to the right leg (AIS 1), caused by contact with the facia panel and lateral intrusion of the footwell. The Micra driver suffered bruising to the chest and abdomen (AIS 1), caused by the seat belt webbing. The Micra driver had little in the way of injurious contact with the interior of the vehicle.

The occupant injury levels in both vehicles were low, which would be expected from a collision at low speed. However, the damage sustained in the passenger compartments of the two vehicles mainly involved deformation of the side structure, and therefore did not affect the occupants directly. The scale of this deformation would not have been expected had there been significant interaction of the frontal structure. The main energy absorbing structures of the two vehicles have not been involved in this collision due to the low overlap. This has led to loading of the passenger compartments, and this has resulted in compartment intrusion despite the low speed of the collision. The intrusion measurements and vehicle damage suggest that there was also slight overriding in this collision, with the Micra road wheel riding up over the Rover road wheel.

4.2.4.2 Stiffness Mismatch / Compartment Strength

Even if good structural interaction were to occur in a collision, it is still possible that there would be problems involving stiffness mismatch. This is where the global stiffness of the vehicle is less stiff than that of its collision partner. This can cause the structure of the less stiff vehicle to absorb more of the collision energy than the structure of its collision partner and therefore deform more. If the passenger compartment strength is low, this in turn can lead to the less stiff vehicle being overcrushed, resulting in increased passenger compartment intrusion compared to a barrier test of equivalent severity.

All current frontal impact crash tests place direct or indirect controls on energy absorption and deceleration of the car. If there is inadequate energy absorption in the frontal structure intrusion occurs which, at some level, will be detected by the instrumented dummies. Similarly, the dummies are sensitive to the car's deceleration, which is detected through such things as chest loading from the seat belt. However, there are currently no requirements controlling the frontal stiffness of the car. Indeed, the tests encourage heavier cars to be stiff, in comparison with lighter cars. As all the tests place a limit on the car's deceleration, through control of dummy loading, all cars tend to have similar stopping distances in the tests. The dummy's experience of deceleration is totally independent of the mass of the car it is travelling in. As most manufacturers aim to limit the length of the front structure, for a variety of reasons, crush depths tend to be kept to the minimum. With the energy absorbed being the integral of force against distance, the only way to maintain the same crush depth, whilst at the same time absorbing the car's kinetic energy, is for the frontal stiffness to increase with vehicle mass. This means that, even without other influences, current frontal crash tests lead to a stiffness incompatibility between cars of different mass.

Ford Mondeo v Ford Fiesta

A Ford Mondeo had a frontal collision with a Ford Fiesta, the vehicles impacting offside to offside with an overlap of approximately 40 percent (Figure 21 & Figure 22). As a result of the collision the Fiesta performed a half roll, coming to rest on its roof. There was a significant difference in the calculated values for Delta-V, which were 45km/h and 89km/h for the Mondeo and Fiesta respectively. There was also a large mass difference between the two vehicles. The Mondeo had a mass of 1613kg and the Fiesta had a mass of 822kg, giving a mass ratio of 1.96. The Mondeo was fitted with a steering wheel airbag. There were no airbags in the Fiesta.

Figure 21: Ford Mondeo

Figure 22: Ford Fiesta

The damage sustained by the two vehicles indicates that this was a very severe impact. The frontal structure of the Mondeo has crushed significantly. There is also slight deformation of the door aperture, with buckling of the roof above the B-pillar. The driver's door is slightly buckled, and appears to be jammed in compression. The damage sustained by the Fiesta is much greater than the Mondeo. The offside structure appears to have completely crushed, and the offside sill has also deformed to a large extent. The overhead view of the car shows the full extent of the crush when compared with the nearside, which was not contacted (Figure 23). Both vehicles also exhibit slightly greater deformation between the longitudinals.

 

Figure 23: Comparison of vehicle damage for Mondeo and Fiesta, showing that the Fiesta has been overcrushed

The values for compartment intrusion differ significantly between the two vehicles (Table 5). The Mondeo had small amounts of intrusion at both recorded facia levels on the offside, and slightly more in the driver's footwell. There was no longitudinal movement of the steering wheel in the Mondeo. The Fiesta had large amounts of intrusion at all levels, over 70cm at each one. The steering wheel had also moved rearwards by over 50cm. Neither vehicle had significant intrusion on the nearside.

Table 5: Comparison of compartment intrusion

	Ford Mondeo		Ford Fiesta	
	Offside (mm)	Nearside (mm)	Offside (mm)	Nearside (mm)
Joint between A-pillar and top of facia	30	0	720	90
Knee contact area	30	0	760	0
Footwell	150	0	700	0
Steering wheel movement	0	-	520	-

The occupants of the two vehicles sustained injuries of significantly different levels. The four occupants in the Mondeo suffered cuts and bruises to various body parts (AIS 1), the result of contact with the interior of the vehicle and deceleration injuries from the seat belts. The driver sustained multiple lacerations to both legs from contact with the facia panel. The driver of the Fiesta suffered the most severe injuries in this case, with multiple rib fracture and extensive internal bleeding (AIS 5). This was the result of contact with the steering wheel, which displaced rearwards during the collision. In addition, the driver of the Fiesta received severe head injuries, also from steering wheel contact, and serious fractures to the femur and pelvis, from contact with the facia panel.

This collision shows a stiffness mismatch and/or compartment strength problem between the Ford Mondeo and the Ford Fiesta. The Fiesta has been overcrushed by the Mondeo, which was almost double the mass of the Fiesta. From the photographic evidence it appears that the frontal structure of the Mondeo has crushed up to or near its maximum crush depth in this collision. When this occurs the engine is likely to be in contact with the bulkhead, and the strength of the passenger compartment will probably have a significant effect on the outcome of the collision. Therefore in this case the reason for the overcrushing of the Fiesta is probably a combination of a stiffness mismatch between the frontal structure of the two vehicles, and the difference in the strength of the passenger compartments.

This outcome may have been influenced by the age difference of the two cars. The Mondeo was manufactured in approximately 1996, and the Fiesta was manufactured in approximately 1989, an age difference of seven years.

This case has been included as an example of stiffness mismatch. However, this is not a suggestion that there was good structural interaction between the vehicles. It is possible that there was also a lateral misalignment problem. However, this is unlikely to have had a major effect on the outcome of the collision.

4.2.5 Quantification of Compatibility Problems

This section attempts to quantify the magnitude of the compatibility problems that were identified in the case by case analysis.

The analysis involved a total of 162 frontal car to car collisions selected from the CCIS database according to the criteria described in Section 4.2.3.

4.2.5.1 Structural Interaction

The assessment of the magnitude of structural interaction problems was undertaken by analysing each case individually, looking for signs of poor structural interaction between the opposing vehicles in a collision. The sample of total cases was broken down initially into three sections (Table 6). These were low overlap, other structural interaction problems and no problem identifiable.

Table 6: Assessment of analysed cases

	No. of Cases	% of Total Cases
Low Overlap	46	28.4
Other Structural Interaction Problems	100	61.7
No Problem Identifiable	16	9.9
Total	162	100

The cases with a low overlap made up approximately 28 percent of the analysis. A low overlap was considered to be a structural interaction problem, but these cases were separated from the rest of the structural interaction problems. This was because it was considered unlikely that improved compatibility would be able to significantly improve the accident outcome, unless the vehicle's structure is extended further outboard.

Approximately 62 percent of the total number of cases had other structural interaction problems identified, and less than 10 percent of the cases had no identifiable problem.

The cases with structural interaction problems were broken down into vertical misalignment, lateral misalignment and multiple problems (Table 7). Cases were defined as having multiple problems if no single interaction problem could be identified, but a number of problems were present.

Table 7: Structural interaction problems identified

	No. of Cases	% of Total Cases
Vertical Misalignment	47	29.0
Lateral Misalignment	17	10.5
Multiple Problems	36	22.2
Total	100	61.7

Eight of the 47 vertical misalignment cases were collisions between cars and SUVs, which are vehicles with significantly different structural heights. The remainder were impacts between two cars, where the difference in structural height could not be fully determined. Therefore it was not possible to fully quantify poor structural interaction caused by static vertical misalignment or dynamic vertical misalignment in this analysis.

Of the 16 cases where there was no problem identifiable, 12 cases were minor impacts (Table 8). These were defined as cases where there was little or no structural damage, and the structure was often obscured as the bumper was still attached to the vehicle. Two cases had 'reasonable' interaction, where the structures of the opposing vehicles showed signs of good structural interaction.

Table 8: No structural interaction problem identifiable

	No. of Cases	% of Cases
Minor Impact	12	7.3
'Reasonable' Interaction	2	1.3
Unknown	2	1.3
Total	16	9.9

This shows that a high proportion (62 percent) of the cases studied exhibited signs of structural interaction problems which improved compatibility should help to address. Only two cases from the sample of 162 cases had structural interaction that could be described as ‘reasonable’.

4.2.5.2 Stiffness Mismatch / Compartment Strength

In order to assess each case for stiffness mismatching, the intrusion data collected for each case was looked at in detail. It was assumed that there would be a significant difference in the intrusion measurements between the two vehicles when a stiffness mismatch / compartment strength problem existed.

In assessing the magnitude of stiffness mismatching problems, only the cases where it is possible for a stiffness mismatch to be identified can be used (Table 9). In a large number of cases it was not possible to determine whether a stiffness mismatching / compartment strength problem was present or not because neither vehicle intruded, there were unknown intrusion measurements, or low overlap.

Table 9: Cases for which a Stiffness Mismatch Assessment cannot be made

	No. of Cases	% of Total Cases
Unknown / Incomplete Intrusion Measurements	8	4.9
No Significant Intrusion in Either Vehicle	30	18.5
Low Overlap	46	28.4
Total	84	51.9

In cases where there was unknown or incomplete intrusion data a valid assessment of the vehicles’ performance could not be made, so these cases were also removed from the analysis. Where there was no significant intrusion in either vehicle in a car to car collision it was not possible to identify a global stiffness mismatch. These cases may have been minor impacts, where a stiffness mismatch may have occurred if the collision was more severe. Therefore these cases were removed from the analysis as well.

In cases where there is a low overlap between the opposing vehicles, the frontal structure has not played a significant part in the collision. It is not possible to assess whether there is a mismatch in the global stiffness of the vehicles in these cases. Therefore cases with a low overlap were also removed from the analysis.

A total of 84 cases were excluded from this analysis for the above reasons, leaving a sample of 78 cases which were able to be assessed to determine whether a stiffness mismatching / compartment strength problem was present or not (Table 10).

Table 10: Summary of Stiffness Mismatch Analysis Case Selection

	No. of Cases	% of Total Cases
Total Cases	162	100
Unable to Assess	84	51.9
Total Cases Analysed	78	48.1

The intrusion measurements of the remaining 78 cases were analysed in detail. The measurements in each case were compared, in order to assess whether there was a significant intrusion difference between the two opposing vehicles.

The analysis showed that 53 cases had a significant intrusion difference (Table 11). This equates to almost 68 percent of the analysed cases. Similar intrusion levels were found in approximately 15 percent of the analysed cases. The intrusion measurements in 13 cases were borderline, so could not be placed in either category.

Table 11: Quantification of Compartment Intrusion for Assessed Cases

	No. of Cases	% of Analysed Cases
Significant Intrusion Difference	53	67.9
Similar Intrusion Measurements	12	15.4
Borderline cases	13	16.7
Total	78	100

A significant difference in intrusion suggests that there may have been a stiffness mismatch or compartment strength problem between the two vehicles, as the compartment of one vehicle has deformed more in the impact. Similar intrusion measurements suggest that there was no major stiffness mismatch or compartment strength problem, as both compartments have deformed to a similar extent.

The cases with a significant difference in intrusion between vehicles were split into two further categories, those where there was intrusion in only one of the opposing vehicles and those in which both vehicles had experienced intrusion (Table 12). In cases where there was intrusion in only one vehicle there was almost certainly a stiffness mismatch or compartment strength problem present. However, in the cases where there was intrusion in both vehicles it is possible that there was a stiffness mismatch or compartment strength problem, but this is not certain as there may be other issues due to the higher severity of the impact.

Table 12: Breakdown of cases with significant intrusion difference between vehicles

	No. of Cases	% of Analysed Cases
Intrusion in One Vehicle Only	24	30.7
Intrusion in Both Vehicles	29	37.2
Total	53	67.9

The analysis to quantify the stiffness mismatch / compartment strength problem shows that from 78 cases there were 24 cases in which there was a definite problem, where there was a significant intrusion difference between vehicles with only one vehicle experiencing intrusion. There were a further 29 cases in which both vehicles experienced intrusion, where it is likely that there was a problem, although it cannot be certain. Therefore the magnitude of problem determined in this analysis is somewhere in the range of 31 to 68 percent of the cases analysed.

It should be noted that in the 53 cases with significantly different intrusion measurements, and therefore a probable stiffness mismatch / compartment strength problem, the cause of the problem cannot be identified. This is because there were also structural interaction problems in 49 of these cases, and the extent to which these may have influenced the stiffness mismatching is unknown.

4.2.6 Single Vehicle Cases

Several cases involving collisions between single vehicles and roadside objects were analysed. The aim was to discover whether it was possible to identify compatibility problems, such as poor structural interaction, in these accidents.

Mercedes C-180 v bollard and signpost

A Mercedes C-180 had a frontal collision with a bollard and signpost (Figure 24). The main impact was between the longitudinals, on the nearside front of the car. There was a secondary impact to the nearside wing, but this appears to be minor. 27 percent of the vehicle width was directly damaged by the impact. The calculated value for ETS was 43km/h. The vehicle was fitted with a steering wheel airbag.

Figure 24: Mercedes C-180 showing narrow impact between longitudinals

The deformation of the front of the vehicle is mostly concentrated around the area of direct damage. Here the bumper beam has been loaded and displaced rearwards. The nearside longitudinal has been involved in this collision, and has absorbed some of the impact energy, but the maximum deformation is between the longitudinals. The engine was directly loaded and displaced rearwards into the bulkhead. The nearside wheel has displaced rearwards slightly, which may be due to the displacement of the engine.

There was little intrusion into the passenger compartment. There was no intrusion on the offside, and the only area of intrusion on the nearside was in the footwell, where there was 170mm rearwards movement. This may have been a result of the engine being displaced into the bulkhead by the object struck.

The vehicle had two occupants. The driver, a 52 year old female, suffered bruising to the arms and knees (AIS 1), caused by contact with the facia and the firing of the steering wheel airbag. The front seat passenger was known to have been injured, but there is no record of the injuries sustained. Evidence suggests that they are likely to have been minor injuries of no more than AIS 1 severity, similar to those sustained by the driver.

This accident is an example of a structural interaction problem in a single vehicle collision. The nearside longitudinal in this case has absorbed some of the impact energy, but not enough to prevent the compartment intrusion. Improved compatibility in the form of homogeneous frontal stiffness should be able to improve this type of collision by enabling the frontal structure to absorb more of the collision energy, ensuring that there is better structural engagement. This is likely to reduce intrusion levels in more severe impacts, as less energy would be absorbed by the compartment due to the increased energy absorption of the frontal structure.

Citroen Saxo v bollard and lamppost

A Citroen Saxo was involved in a frontal collision with a bollard and lamppost. The impact was outside the main longitudinal on the offside. The values for Delta-V and ETS were unable to be calculated. The vehicle sustained 13 percent direct damage from the impact.

Figure 25: Citroen Saxo showing impact between upper and lower rails

The offside lower rail appears to have been bent inboard slightly, but has absorbed little energy from the impact. The offside upper rail has slight damage to the front section and minor crumpling to the rear section, but there has been no significant deformation. The object struck has impacted between the upper and lower rails, where there is little energy absorbing structure, and has directly loaded the suspension turret.

There was little intrusion into the passenger compartment. There was only minor intrusion on the offside of no more than 40mm, and no nearside intrusion.

The driver, who was the only occupant, suffered bruising to the face and a neck strain (AIS 1). These were caused by the airbag. There was also bruising to the chest (AIS 1) from the seat belt.

The impact in this case was outside of the main structural members, the lower rails. For this reason it is considered that improved compatibility may not be able to improve collisions of this type without moving the main structure further outboard. However, improved shear connections between the upper and lower rails may be beneficial to collisions such as this.

Summary

It was possible to identify compatibility problems in some of the single vehicle cases that were analysed. In many of these cases the vehicle was involved in a collision with a narrow object that did not interact well with the vehicle's front structure. Better compatibility is likely to have a beneficial effect in these impacts.

In cases where a large, wide object has been impacted there has often been direct loading of a significant proportion of the frontal structure. In cases such as these, better compatibility may not be able to offer as much benefit because of the already significant structural interaction.

4.2.7 Conclusions

This study has shown that the major compatibility problems for car to car frontal impacts are poor structural interaction, stiffness mismatching and compartment strength.

Poor structural interaction has been shown to occur in a number of different ways. These are the fork effect caused by lateral misalignment of the car's main structural members, and under/override caused by vertical misalignment of the car's main members. Two types of the vertical misalignment problem have been identified, namely static and dynamic. Static misalignment is caused by an initial geometric mismatch of the vehicle's structures whilst dynamic misalignment occurs during the impact for structures that are approximately aligned initially.

To quantify the magnitude of the structural interaction problem it was shown that 62 percent of the 162 cases examined exhibited poor structural interaction that improved compatibility should help to address. Only two cases from a sample size of 162 had structural interaction that could be described as reasonable.

The analysis to quantify the stiffness mismatch / compartment strength problem shows that from 78 cases there were 24 cases in which there was a definite problem, where there was a significant intrusion difference between vehicles with only one vehicle experiencing intrusion. There were a further 29 cases in which both vehicles experienced intrusion, where it is likely that there was a problem, although it cannot be certain. Therefore the magnitude of problem determined in this analysis is somewhere in the range of 31 to 68 percent of the cases analysed. It should be noted that in the 53 cases with significantly different intrusion measurements, and therefore a probable stiffness mismatch / compartment strength problem, the cause of the problem cannot be identified. This is because there were also structural interaction problems in 49 of these cases, and the extent to which these may have influenced the stiffness mismatching is unknown.

It was possible to identify compatibility problems in some of the single vehicle cases that were analysed. In many of these cases the vehicle was involved in a collision with a narrow object that did not interact well with the vehicle's front structure. Better compatibility is likely to have a beneficial effect in these impacts. In cases where a large, wide object has been impacted there has often been direct loading of a significant proportion of the frontal structure. In cases such as these, better compatibility may not be able to offer as much benefit because of the already significant structural interaction.

4.3 BAST ANALYSIS FOR GERMANY

4.3.1 Introduction

In the year 2000 in Germany 2.35 million accidents were reported by the police. Among these were 1.97 million with material damage only and 382,949 accidents with personal injury. 7,503 persons were fatally injured, 4,396 of these were car occupants. 102,416 car occupants suffered severe and 401,658 minor injuries. 105,604 accidents were car to car accidents with personal injury. In 888 car to car accidents 1058 occupants were killed. In nearly half of the accidents the front of the passenger car was damaged. This indicates that the improvement of the interaction of cars in frontal impacts is quite important for road safety.

In a study commissioned by the Federal Highway Research Institute a research team consisting of physicians and technicians has been documenting traffic accidents on-scene in Greater Hanover for more than 25 years and has stored a great number of information in a database. Since 1985 this information has been documented using a statistical random sample plan and edited with the objective of obtaining statistically representative data. In 1999 a second system was established in the area of Dresden. This second team is financed by the German car industry. The aim of the expansion of the German in depth accident acquisition system is to collect a higher number of accidents per year based on the same methodology (Otte 1994).

4.3.2 Method and Database

Two research teams consisting of physicians and technicians collects information on personal injury accidents using a statistical random sample procedure. For this purpose, all traffic accidents occurring are reported continuously by the police and fire department stations active in Greater Hanover and Greater Dresden to the research teams stationed at the Surgical Accident Clinic of the Medical University of Hanover and the Technical University of Dresden, from which the teams select accidents according to a defined random procedure and documents these accidents in a comprehensive research catalogue. A detailed description of the research in Hanover is given (GIDAS project flyer). Annually, approximately 2000 traffic accidents are recorded in this way and the information stored in a database. In order to avoid distortions in the data structure of the accidents recorded by the teams, the data are weighted annually through comparison with the officially recorded accident structure. This ensures that the present accident data are regarded as representative for the investigation area of the cities and administrative districts of Hanover and Dresden. Statements on the nation-wide situation are possible only for accident characteristics which are relatively independent of regional influences. Since collision processes are generally dependent on technical background conditions and the resulting injuries often affected by these conditions, the investigations can be used for most of the aspects of passive safety.

The geographical distribution of the investigation areas correlate well to that of the Federal Republic of Germany as a whole. In both, approx. 90% of the area can be regarded as rural and 10% urban so that it can be assumed that the distribution between inside and outside built-up areas is similar. The accidents are recorded by the team daily with alternating shift times so that a uniform distribution between day and night and between the different days of the week is ensured. The technical documentation on the traffic accidents includes measurements, photographic and descriptive documentation on vehicle damage, contact points for occupants and external road users as well as indications present at the accident site such as brake and skid marks including final positions of the vehicles and persons.

The medical documentation includes a description of each individual injury according to type, locality and severity. X-rays and physicians' reports are analysed retrospectively.

Use is made of the usual classification systems for describing the severity of a patient's accidents such as AIS (Abbreviated Injury Scale - 2), ISS (Injury Severity Score) and PTS (Polytrauma Score). Documentation is accomplished under observance of the existing guidelines for medical confidentiality and the regulations for protection of data and personal rights.

Each accident is analysed in detail and the motions of the vehicles and occupants reconstructed. Collision and vehicle speed are calculated from the marks and vehicle deformation using mathematical procedures from the field of impact mechanics and scientific data given, for example EES (Equivalent Energy Speed), Delta-v (velocity change resulting from collision) and VDI (Vehicle Deformation Index).

4.3.3 Scope of the Data

For the present study, evaluations were made of accidents from the last 15 years from 1986 to 2001: During this period, a total of nearly 13,000 accidents were documented by the teams. Differentiated according to type of traffic participation, these accidents distribute as follows:

Table 13: Distribution of Accident Types in the Database.

Type of Accident	**%**
Single Car Accidents	12,3 %
Car to Truck	6,2 %
Car to Car	20,2 %
Car to Motorcycle	7,3 %
Car to Bicycle	18,7 %
Car to Pedestrian	9,8 %
Other Truck Accidents	7,5 %
Other Motorcycle Accidents	3,6 %
Multiple and Others	14,4 %
Total	**100 %**

For the question of compatibility, only accidents involving more than one vehicle are considered for evaluation. Single-vehicle accidents and accidents involving pedestrians or two-wheelers are of minor interest in this context. Accidents involving more than 2 vehicles were not evaluated due to the complex collision processes.

Multiple collisions can have additional effects on the severity of injury to the occupants. Moreover, with multiple collisions it is frequently not possible to assign beyond doubt the cause of individual injuries to a certain collision. In order to study the compatibility of passenger cars, accidents with commercial vehicles were not investigated. This means that for the question of compatibility, only collisions between two passenger cars were evaluated; all totalled these represent only 20.2 % or approx. 2,500 accidents of all personal injury accidents.

For the reason that this study engages only in frontal car to car impacts and a lot of special parameters have to be known the number of accidents considered is relatively low.

4.3.4 Case Selection Criteria

The criteria for selection were as follows:

- Only accidents between two passenger cars
- Only vehicles in single collisions
- Only front to front collisions, the force direction has to be between 11 and 1 o'clock where 12 o'clock is straight ahead
- Only fully reconstructed accidents
- Only cases with exactly investigated vehicle deformations and intrusions
- Only belted occupants.

These selection criteria lead to a sample of 135 accidents with 270 cars and 317 frontal occupants involved.

4.3.5 Example of Compatibility Problems

The main types of compatibility problems in car to car crashes are described in section 4.2.4 (TRL part of the accident analysis). Here you can find detailed descriptions of the different ways in which two cars interact in a frontal impact. You can divide these problems mainly into

- Structural interaction
- Lateral or vertical and static or dynamic misalignment
- Stiffness mismatch / compartment strength
- Low overlap.

For the different kinds of incompatibility one example is given here. Because this study intends to show the methods and the possibilities of the existing database of

BASt, this report concentrates on one single car to car accident, which illustrates most of the compatibility problems in one impact.

Porsche 911 Carrera vs. Opel Corsa

A right hand drive Porsche Carrera was involved in a frontal collision with a left hand drive Opel Corsa. This accident happened on a rural road outside of urban areas. The principle direction of force PDF was 12 o'clock for both cars. The overlap was 50 % for the Porsche and 55 % for the Corsa.

Figure 26: Opel Corsa

Figure 27: Porsche Carrera

The accident reconstruction came to the conclusion that the collision speed was 49 km/h for the Porsche and 38 km/h for the Opel Corsa. Figure 26 and Figure 27 show the final position of the vehicles after the accident.

In both cars was one occupant. The male Porsche driver was uninjured while the female Corsa driver was seriously injured and hospitalised. Injuries were located at the head (AIS 4 brain contusion / AIS 2 lower jaw injury), the thorax (AIS 4 lung injury) the abdomen (AIS 2 vertebral fracture) and the pelvis (AIS 1 contusion). All these injuries were caused by intrusion into the passenger compartment.

The resulting deformation of the front ends are shown in Figure 28 and Figure 29. The percentage of deformation according to the Vehicle Deformation Index VDI are 7% for the Porsche and 34% for the Corsa.

Figure 28: **Front Deformation Opel**

Figure 29: **Front Deformation Porsche**

This is not only caused by the different masses of the vehicles (the Porsche had an kerb weight of 1395 kg the Corsa of 850 kg) but also by an obvious stiffness mismatch of the front ends and the passenger compartments.

Beside this other kinds of incompatibilities can be observed in this accident. Figure 5 illustrates the deformation of the passenger compartment of the Corsa. There was a footwell rupture as well as a folding of the left door sill. This is caused by vertical misalignment of the longitudinals, which leads to some overriding by the Corsa.

Figure 30: **Deformation of the Passenger Compartment of the Corsa**

The deceleration of the passenger compartment is directly addressed to its stiffness and to the overall stiffness of the front end. The accident reconstruction results in a deceleration of 21 g for the Porsche and 14 g for the Corsa although the relation of the delta v's are the other way round. The matching delta-v's were 28 km/h for the Porsche and 48 km/h for the Opel. The structural interaction of these two cars has to be assessed as very poor.

This collision is an extreme example of compatibility problems. The structures of the two vehicles involved are significantly different so that nearly no structural interaction between these two cars is possible. The stiffness of the front ends and passenger compartments is so different that you have in one car no intrusion and in the other very extensive intrusion. This mismatch results directly in extremely different levels of injuries for the occupants.

4.3.6 Quantification of Compatibility Problems

This section attempts to assess generally the relevance of compatibility problems in real world crashes. It tries to find out the main points in which compatibility problems appear. Therefore the GIDAS database of the accident investigation in Hanover (Medical University Hanover) and Dresden (Technical University Dresden) was analysed. The exact selection criteria are described above in this investigation.

As mentioned above 135 frontal accidents out of the GIDAS database were analysed. The complete table of the selected accidents is given in the annex.

The assessment of the quantification of the possible compatibility problem in the data set was undertaken by different attempts. The following tables show the relevance of different factors which may have an influence on car to car compatibility or which may have been influenced by compatibility problems.

The distribution of the mass ratio of the 135 selected accidents is given in Table 14. In compatibility investigations this ratio is often determined as responsible for all compatibility problems. This might be true for the 11% of the cases over 1.6 but even in these cases there are other factors influencing car to car compatibility. The influence of these other factors described below might be not as strong as in cases with a mass ratio of nearly 1 but they are still present.

Table 14: Mass Ratio between both Cars involved

Mass Ratio	**No**. of Cases	**%** of Total Cases
1	3	2,2
1,1	38	28,1
1,2	36	26,7
1,3	15	11,1
1,4	11	8,1
1,5	8	5,9
1,6	6	4,4
> 1,6	15	11,1
unknown	3	2,2
Total	**135**	**100,0**

Table 15 shows that in 10% of the 135 cases the maximum front end deformation in the one car was between 20 and 30 cm higher than in the opposite car. In 5% this difference is higher than 30 cm. Among other things the front end deformation depends on the design of the car front structure and a great difference in front end deformation between the both vehicles involved in an accident indicates possibly an stiffness mismatch. A deeper look on these 21 cases shows that the mass ratio is nearly evenly distributed between 1 and 1.5. Only 3 cases have a higher mass ratio.

Table 15: Difference in max. Front End Deformation between both Cars involved

Difference in **max. Front End Deformation** up to	**No**. of Cases	**%** of Total Cases
10 cm	77	57,0
20 cm	31	23,0
30 cm	14	10,4
>30 cm	7	5,2
unknown	6	4,4
Total	**135**	**100,0**

A similar result follows if the Vehicle Deformation Index VDI is taken into consideration. Table 16 shows that 19.3 % of the cases have significant differences in this item. As a significant difference here a variation of more than 10 % of the respective car length is supposed.

Table 16: Significant Difference in Vehicle Deformation Index VDI between both Cars involved

Significant Difference in **Vehicle Deformation Index VDI**	**No**. of Cases	**%** of Total Cases
yes	26	19,3
no	108	80,0
unknown	1	0,7
Total	**135**	**100,0**

Another possible indicator for compatibility problems is displayed in Table 16 and Table 17. In Table 17 the direct distribution of the delta v of all 270 cars is shown. Delta v is the change of the velocity of a car, which occurs as direct consequence of an impact. According to Table 17 approx. half of the 270 cars have a delta v between 11 km/h and 30 km/h. This high portion of less severe impacts makes it difficult to conclude problems of car to car interaction.

Table 17: Distribution of Delta v for all Cars involved

Delta v up to		**No**. of Cars	**%** of Total Cars
10	km/h	26	9,6
20	km/h	81	30,0
30	km/h	61	22,6
40	km/h	49	18,1
50	km/h	26	9,6
60	km/h	19	7,0
>60	km/h	6	2,2
unknown		2	0,7
Total		**270**	**100,0**

A bad interaction between two cars might lead to higher intrusion or deformation in one car which influence the deceleration of the car. So consequently the difference in delta v between two cars in one impact could indicate compatibility problems.

89 of the 135 cases have just the small difference up to 5 km/h in delta v (Table 18). A delta v difference of more than 10 km/h as observed in 18 cases could depend from incompatibilities. Another factor to be taken into account here is the mass ratio. Even here there is no direct connection between delta v difference and mass ratio detectable just like in the deformation analysis above.

Table 18: Difference in Delta v between both Cars involved

Difference in **delta v** up to		**No**. of Cases	**%** of Total Cases
5	km/h	89	65,9
10	km/h	28	20,7
15	km/h	13	9,6
20	km/h	3	2,2
25	km/h	1	0,7
30	km/h	0	0,0
>30	km/h	1	0,7
unknown		0	0,0
Total		**135**	**100,0**

In front to front collisions between 11 o'clock and 1 o'clock the overlap between the both cars differs in a very small range, which depends on the single car width. For analysing this point it is sufficient to consider only the mean overlap between both cars involved.

Table 19 shows the distribution of this mean overlap for the 135 cases. It is an even distribution over all values. It will be very difficult to develop solutions for compatibility problems, which will be able to address the 17% of the cases having an overlap of less than 20 %. In such cases the main structures of the car body cannot interact in the way they are designed for.

Table 19: Distribution of Mean Overlap from both Cars involved

mean **Overlap**	**No**. of Cases	**%** of Total Cases
20 %	23	17,0
40 %	23	17,0
60 %	34	25,2
80 %	24	17,8
100 %	20	14,8
unknown	11	8,1
Total	**135**	**100**

The most important point which indicates car to car interaction problems is the compartment intrusion. It seems to be the only aspect, which directly reflects the different degrees of occupant injuries. In the data sample there were 76 cases in which at least one vehicle experienced compartment intrusion. Table 20 shows that 33 cases were identified out of these 76 in which the one car had no intrusion and the other one had relevant intrusion into the passenger compartment.

Table 20: Significant Difference in Compartment Intrusion between both Cars involved in the 76 cases where at least one vehicle experienced intrusion

Significant Difference in **Compartment Intrusion**	**No**. of Cases	**%** of Cases
yes	33	43,4
no	43	56,6
Total	**76**	**100,0**

Relevant intrusion means that the intrusion is evaluated as responsible for the injuries of the driver or the front seat passenger. In these 33 cases it seems reasonable to suppose that stiffness mismatch or compartment strength problems are identifiable.

Table 21 clarifies that in total 119 of the 270 considered cars have significant intrusions in the passenger compartment. Serious injuries are directly linked to compartment intrusion. In all 14 cars with seriously injured occupants there is intrusion at the same time.

Table 21: MAIS Distribution depending on Compartment Intrusion

MAIS	Intrusion		Total
	Yes	No	
0	25	59	84
1 - 2	76	90	166
3 +	14	0	14
unknown	4	2	6
Total	119	151	**270**

Big differences in the MAIS value between the two cars may be another indicator for a non optimal car to car interaction. If one assumes that a difference of 2 in MAIS is significant, then Table 22 makes once more clear that approx. 10 - 15 % of the selected cases seem to have compatibility problems.

Table 22: Difference in MAIS between both Cars involved

Difference in **MAIS**	**No**. of Cases	**%** of Total Cases
1	114	84,4
2	14	10,4
3	0	0,0
4	2	1,5
5	1	0,7
unknown	4	3,0
Total	**135**	**100,0**

Figure 31 and Table 23 serve as example to make clear that it is intrusion and not vehicle deformation which is responsible for occupant injuries. In Figure 31 the definition of the Vehicle Deformation Index VDI for frontal impacts is given. The VDI has the categories from 1 to 9. 1 up to 5 means that the deformation lies only in the front area from the bumper to the lower part of the wind screen frame and so on.

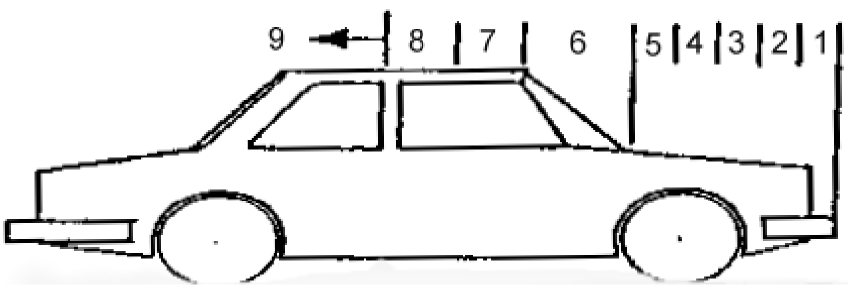

Figure 31: Definition of Vehicle Deformation Index VDI 6 for Frontal Impact

Table 23: MAIS Distribution depending on Vehicle Deformation Index VDI 6

MAIS	Vehicle Deformation Index VDI 6				Total
	1 - 2	3 - 5	> 5	unknown	
0	66	15	1	2	84
1 - 2	92	67	5	2	166
3 +	5	7	2		14
unknown	1	3	2		6
Total	164	92	10	4	**270**

Of course one can see in Table 23 that there is a relation between the amount of deformation and the severity of injuries. The higher the deformation is the higher the corresponding MAIS value is. But the clearness of this tendency is broken if you concentrate on the cases with MAIS 3+ injuries. Here one detects even in relatively low deformed cars with a VDI of 1 or 2, 5 occupants with severe injuries.

4.3.7 Conclusions

This study has shown that major compatibility problems, namely structural interaction, stiffness mismatching and compartment strength can be observed in the accident data from the in depth accident investigation GIDAS.

The GIDAS database was examined to attempt to quantify the size of the major compatibility problems. From the 135 accident cases examined it was found that the 14 MAIS 3+ injuries correlated well with compartment intrusion, i.e. no MAIS 3+ injuries occurred unless the compartment had intruded. This confirms the results of previous studies that show that intrusion is the major cause of deaths and serious injuries (Wykes 1998). However, there was no correlation of MAIS 3+ injuries with the Vehicle Deformation Index (VDI). The stiffness mismatch / compartment strength problem magnitude was quantified by identifying the cases where one vehicle had injury causing intrusion and the other no intrusion. In the data sample there were 76 cases where at least one of the vehicles had intruded. From these 76 cases, 33 (43%) had no intrusion in one vehicle and injury causing indicating that stiffness mismatch / compartment strength is a large problem. The degree to which poor structural interaction may have contributed to this problem is unknown. The measures currently envisaged for improved compatibility are not expected to address accidents having a low overlap because the main car structures, such as the lower rails, are usually not involved in these impacts. 23 (17%) of the cases investigated had a low overlap of less than 20 percent. No variables were found that appeared to have the potential to identify poor structural interaction. This problem can probably only be quantified using detailed case studies.

This study was mainly based on the electronic version GIDAS database. It is intended that a deeper analysis of the single accidents should follow in the research activities of the VC-COMPAT project. Then every single accident folder of the 135 accidents will be carefully examined to better understand the interaction of the vehicles in view of the major compatibility problems. During the running period of the VC-COMPAT project further accidents will be recorded (and reconstructed) in Hanover and especially Dresden. Obviously these will also be taken into consideration.

4.4 SUMMARY OF CONCLUSIONS AND RECOMMENDATIONS

Conclusions

For GB and Germany, it was confirmed that the compatibility problems for car to car frontal impacts are structural interaction, stiffness matching and compartment strength.

Poor structural interaction was seen to occur in a number of different ways, namely the fork effect caused by lateral misalignment and under/override caused by vertical misalignment. Two types of the vertical misalignment problem have been identified, static and dynamic. Static misalignment is caused by an initial geometric mismatch of the vehicle's structures. Dynamic misalignment occurs for structures, initially approximately aligned, deforming to become misaligned during the impact.

For GB, poor structural interaction was found to be a major problem. Of the 162 cases examined only 2 had structural interaction that could be described as reasonable. However, some of the cases had poor structural interaction caused by low overlap, which improved compatibility is not expected to address. 100 (62%) cases had structural interaction problems that improved compatibility is likely to address. For Germany, it was found that structural interaction problems could probably only be quantified using detailed case studies. Unfortunately, unlike the UJK, detailed case studies were not performed for all the selected cases. However, it is intended that this should be done in the VC-COMPAT project.

For GB and Germany stiffness mismatch / compartment strength was found to be a large problem. For GB, the problem magnitude was quantified by identifying the cases where there was a significant intrusion difference between the colliding vehicles. In the data sample there were 78 cases where at least one of the vehicles had intruded and therefore it was possible to identify an intrusion difference. A significant intrusion difference was identified in 68 percent of these cases indicating that stiffness mismatch / compartment strength is a large problem. For Germany, the problem magnitude was quantified by identifying the cases where one vehicle had injury causing intrusion and the other no intrusion. In the data sample there were 76 cases where at least one of the vehicles had intruded. From these 76 cases, 33 (43%) had no intrusion in one vehicle and injury causing indicating that stiffness mismatch / compartment strength is a large problem. It should be noted that the extent to which poor structural interaction contributed to this problem is unknown.

For Germany, from the 135 accident cases examined it was found that the 14 MAIS 3+ injuries correlated well with compartment intrusion, i.e. no MAIS 3+ injuries occurred unless the compartment had intruded. This confirms the results of previous studies that show that intrusion is the major cause of deaths and serious injuries (Wykes 1998). However, there was no correlation of MAIS 3+ injuries with the Vehicle Deformation Index (VDI).

For GB, structural interaction problems were also identified in some single vehicle accidents indicating that a benefit from improved compatibility could also be expected in this type of impact.

Recommendations

It is recommended that this type of study should be repeated in the future to check that the conclusions are still valid, as the vehicle fleet is constantly changing. In particular, for Germany, it is recommended that detailed case studies should be performed for all selected cases so that the structural interaction problem can be quantified.

5 BENEFIT ANALYSIS

The aim of this work is to provide an initial estimate of the benefits of implementing compatibility measures for frontal impact for Germany and Great Britain. BASt and TRL will perform this work for Germany and the UK, respectively.

In order to perform this work assumptions have to be made to how improved compatibility will effect the crash performance of future cars, because this is not known, as yet. It is difficult to make reasonable assumptions that allow this work to be done. Hence, when considering the conclusions that this work draws the underlying assumptions should also be considered. Future work should be able to refine these assumptions and hence the benefit estimates, once more is known about the performance of compatible cars.

5.1 TRL ANALYSIS FOR GB

5.1.1 Introduction

Great Britain's national road accident statistics are compiled from police reports of all accidents involving a personal injury. For the year 2000, these show that two-thirds of the road accident casualties were in cars or light good vehicles. Occupants of these vehicles accounted for just under half of the fatalities and just under half of the seriously injured, which is typical for recent years (Table 24, Road Accidents GB, 2000). Using costs estimated by the UK Department of Transport to identify the average value of preventing a road accident casualty; the cost to society of these casualties was about £6.3 billion in 2000.

Table 24: Distribution of road accident casualties in Great Britain in the year 2000.

	Fatalities		Seriously injured		Slightly injured	
	No	(%)	No	(%)	No	(%)
Car occupants	1665	(49)	18054	(47)	187080	(67)
Other road users	1744	(51)	20101	(53)	91639	(33)
Total	3409	(100)	38155	(100)	278719	(100)

Although the improved structural interaction aspects of compatibility are relevant for virtually all car frontal impacts, the main benefits from stiffness matching are expected in car impacts with another vehicle. The national data contain only basic information about the accident configuration, but include the "First point of impact" on the car. Over sixty percent of fatal and serious occupants in 2000 were in frontal impacts (Table 25) and approximately two-thirds of these occurred in an impact with at least one other vehicle (Table 26).

Table 25: Distribution of car occupant casualties by first point of impact for Great Britain in the year 2000

First point of impact	Fatalities		Seriously injured	
	No	(%)	No	(%)
Front	1002	(60)	11931	(66)
Side	560	(34)	4123	(23)
Rear	65	(4)	1527	(8)
Other	38	(2)	473	(3)
Total	1665	(100)	18054	(100)

Table 26: Distribution of car occupant casualties in frontal impacts by impact partner for Great Britain in the year 2000.

Impact partner	Fatalities		Seriously injured	
	No	(%)	No	(%)
Frontal impacts with one other vehicle	454	(45)	6483	(54)
Frontal impacts with more than one other vehicle	208	(21)	2077	(18)
Frontal impacts where no other vehicle was involved (e.g. roadside obstacle etc.)	340	(34)	3371	(28)
Total	1002	(100)	11931	(100)

The aim of this work is to provide initial estimates of the benefits of implementing compatibility measures for frontal impact for GB. At present, a ‘compatible’ car does not exist so the accident statistics contain no data for crashes with such vehicles. In order to proceed with any benefit analysis, assumptions about the performance benefits of such a car have to be made, which the current accident statistics can identify and quantify. It is difficult to make reasonable assumptions of what these benefits may be and the results of any analysis are totally reliant upon the validity of the assumptions made. For these reasons, two different approaches for the analysis were taken to give a range of the possible benefits of frontal impact compatibility. These analyses could be refined at a later date when more is known about the crash performance of a ‘compatible’ car.

The first approach aimed to identify the number of car occupant casualties that could be expected to experience some reduction in injury risk from improved frontal impact compatibility. This approach is referred to as the ‘potential benefit’ approach. The reason for this is that it only identifies the number of casualties that are likely to experience an injury risk reduction, it does not quantify the injury risk reduction.

The second approach aimed to estimate the reduction in the number of car occupant fatalities and serious injuries for a specific category of accidents that could be expected as a result of improved compatibility. The accident category chosen was car frontal impacts with another car or van because this is the type of impact that improved compatibility should benefit most. This approach is referred to as the 'benefit for car to car/van impacts' approach.

As both of these approaches were based on a number of key approximations relating to the improved crash performance of a 'compatible' car, optimistic and pessimistic estimates of these approximations were used so that a range for the potential benefit and benefit of car to car/van impacts of improved compatibility was made. When more is known about the likely crash performance of a 'compatible' car these approximations could be revised and the accuracy of the analysis improved.

The accident data sources used for these analyses were the Great Britain STATS19 national accident database and the Co-operative Crash Injury Study (CCIS) detailed accident database. The STATS19 accident statistics are compiled from police reports of all road accidents involving a personal injury. Unfortunately this database does not contain enough details about the accident to be able to assess whether or not improved compatibility would have reduced the injury risk potential or number of casualties. For this type of information the CCIS database was used, which has been described previously in section 4.2.3.

The methodology used, the assumptions made and the results of the analyses performed for both approaches are described in the sections below.

5.1.2 Estimate of The Potential Benefit of Improved Frontal Impact Compatibility

The aim of this work was to identify the number of car occupant casualties that could be expected to experience some reduction in injury risk from improved frontal impact compatibility.

The accident data used for this analysis were:

- The national accident data for Great Britain (STATS19) for car occupant fatal and seriously injured casualties for the years 1996 to 2000, inclusively.

- Various subsets from the CCIS phase VI data, as detailed in the methodology below.

5.1.2.1 Methodology

The methodology used for this analysis was as follows:

1. The car occupant casualty STATS19 data for the years 1996 to 2000, inclusively, were adjusted to remove the effect of cars that were greater than 7 years old. An average of the adjusted data for the years 1996 to 2000 inclusively was made. This was deemed necessary as it was thought that inclusion of older cars would over estimate the potential benefit of improved compatibility as these, generally poor performing vehicles, will be replaced by newer better performing vehicles in

the vehicle fleet regardless of the introduction of compatibility improvements. This has the additional advantage that the CCIS data sample will be a more representative subset of these adjusted data as the CCIS is biased to newer cars. This is because one of the accident selection criteria is that one of the cars involved must be less than 7 years old.

2. The car occupant casualty average adjusted data were broken into categories by impact partner and first point of impact. The categories by impact partner are:

    - Car collides with object off carriageway, subdivided into narrow object, wide object and other object.

    - Car collides with one other vehicle, subdivided into car and van; public service vehicle and heavy goods vehicle; and other.

    - Car collides with more than one other vehicle.

    The categories by first point of impact are front, rear, side and other. It was assumed that there would only be potential benefit for casualties involved in frontal impact collisions, i.e. no potential benefit for casualties in side, rear and other impact collisions.

    The reason for breaking down the data in this manner was that it was thought that the relative potential benefit of improved compatibility for each of these groups would be quite different, therefore they needed to be treated separately. For example, improved compatibility is expected to deliver its greatest benefit for the accident configuration of a frontal impact with another car.

3. For each of the frontal impact categories defined above an equivalent data subset was derived from the CCIS data sample. The CCIS subset was then used to estimate the proportion of fatal and seriously injured casualties[1] that were likely to experience a potential benefit from improved compatibility. This was achieved by considering parameters such as overlap, impact severity and the impact principle direction of force (pdf). For each of these parameters a lower (pessimistic) and upper (optimistic) estimate was made for which a potential benefit would be expected as a result of implementing improved compatibility. The results were combined to give a somewhat optimistic and a somewhat pessimistic estimate of the accident subset in which the casualties could expect to experience a potential benefit for improved compatibility. The number of casualties in each of these accident subsets was determined. These were compared to the number of casualties in the originally derived equivalent data subset to determine a lower (somewhat pessimistic) and upper (somewhat optimistic) bound for the proportion of fatalities and seriously injured that would be expected to see a potential benefit from improved compatibility.

4. An upper and lower estimate of the number of fatalities and seriously injured casualties that would be expected to see a potential benefit for improved compatibility, annually in GB, was determined by scaling the results obtained from the analysis using the CCIS data sub-set to the STATS19 national accident data, for each of the defined frontal impact categories.

[1] Fatal and seriously injured are defined according to the Police's injury severity rating.

Step 3 of the methodology described above is fully illustrated below for the impact partner category, car frontal impact with another car or van. Details of the other impact partner categories are shown in the Tables in Appendix 1.

The accident parameters considered for the category, car frontal impact with another car or van, were; impact direction, overlap, multiple impacts, rollover and accident severity. The lower (somewhat pessimistic) and upper (somewhat optimistic) limits, shown in Table 27 below, were chosen and used as selection criteria to determine the proportion of fatalities and seriously injured in the CCIS equivalent data sample sub-set that would be expected to experience a potential benefit. Some reasoning as to why the particular limits used were chosen is given below.

It is expected that improved compatibility should offer some potential benefit for frontal collisions with nearly all impact directions, except possibly those with a substantial side component. Therefore, an upper limit to include impacts with 10, 11 12, 1 and 2 o'clock principle direction of force (pdf) and a lower limit of 11, 12 and 1 o'clock pdf were chosen. In considering the limits for overlap, it is not expected that improved compatibility will offer significant benefits for side-swipe or low overlap accidents where the main structure of the car, such as the lower rails, is not involved. This is because it would be difficult to obtain good structural interaction in these cases. So to exclude these accidents the upper and lower limits were set at 20 and 30 percent, respectively. The accident data sub-set will include some multiple impact accidents where a car has impacted a roadside obstacle following a frontal impact. In some cases this secondary impact may be a side impact and more injurious than the frontal impact. Improved compatibility will probably not benefit these types of cases. To take this into account an upper limit to exclude all cases in which a significant[2] side impact occurred and a lower limit to exclude cases in which a significant side impact occurred and cases in which the other impact was judged to be more injurious than the frontal impact[3] were used. The accident data subset will also include some cases where the car has rolled over following the frontal impact. In some cases it is possible that the rollover was more injurious than the frontal impact. To take this into account an upper limit to include all accident cases in which rollover occurred and a lower limit to include only rollover cases where the rollover was judged to be less injurious were used. Finally, impact severity was considered. Some potential benefit will be expected at almost all impact severities, but obviously this will be very small or zero in accidents of very high severity. To attempt to take this into account an impact severity limit was used, up to which all occupants are expected to experience potential benefit, but above which only half the occupants are expected to experience potential benefit. The upper value chosen for this limit was 56 km/h ETS as this is widely believed to be a good approximation of the severity of the 64 km/h ODB test, the severity up to which a 'compatible' car is expected to offer 'good compatible' performance. However, recent work has estimated the average ETS for a number of EuroNCAP tested cars (a 64 km/h ODB test) to be 48 km/h. Hence, this value was used as the lower limit.

[2] "Significant side impact" is defined as having a CDC extent code of at least 2.

[3] "Less injurious" assessment is based on the vehicle examiners' judgement of the relative likelihood of a particular part of the accident causing the serious injuries.

Table 27:Upper and lower limits for accident parameters used to identify proportion of fatalities and seriously injured expected to experience a potential benefit.

Accident Parameter	Upper (somewhat Optimistic) Limit	Lower (somewhat Pessimistic) Limit	Basic Sample for all frontal impacts with another car / van
Principle direction of force (pdf) (o'clock)	10,11,12,1,2	11,12,1	10,11,12,1,2
Overlap	Include ≥ 20%	Include ≥ 30%	All
Multiple impacts	Exclude cases in which a significant[4] side impact occurred.	Exclude cases in which a significant side impact occurred and cases in which the other impact was judged to be more injurious than the frontal impact.[5]	All multiple impacts with frontal as the initial point of contact.
Rollover	Include all, with or without rollover	Include only those where rollover was less injurious than the frontal impact.	All, with or without rollover
Impact severity	Include all with ETS up to 56km/h and 50% of those more severe.	Include all with ETS up to 48km/h and 50% of those more severe.	All with known ETS.

This methodology was repeated for cars that suffered an impact with a PSV or HGV, with a wide object and with a narrow object. Similar accident case selection parameters and limit values were used with a number of exceptions that are described below. Tables showing these parameters and the limit values used are shown in Appendix 1.

The analysis for impacts with a Public Service Vehicle (PSV) or Heavy Goods Vehicle (HGV) included an extra parameter regarding underrun and impact severity. The lower (pessimistic) limit for the underrun parameter excluded all underrun cases but for the upper limit 20 percent of underrun cases were included. The reason for this is that there is the possibility that a more compatible car's front structure should

[4] "Significant side impact" is defined as having a CDC extent code of at least 2.

[5] "Less injurious" assessment is based on the vehicle examiners' judgement of the relative likelihood of a particular part of the accident causing the serious injuries.

engage more effectively with either an underrun protection system or with the wheels of the other vehicle. Accordingly, there should be some benefit in this scenario. The additional parameter for the impact severity was the Δv of the car. For the lower limit the change in velocity (Δv) of the car must not have been greater than 56km/h. For the upper limit, all Δvs were included. This parameter was added as an attempt to account for the likelihood of there being little or no energy absorbing structure on the front of HGVs and hence the car will have to absorb all of the impact energy. This assumption would need to be reviewed if energy absorbing front underrun guards were introduced on HGVs.

The analysis of impacts with narrow objects was difficult to perform accurately. In this type of impact the damage pattern seen on the car consists of direct damage (caused by contact of the car with the object) and induced damage (caused as a consequence of the crush of the vehicle structure). The CDC records the direct damage and thus identifies a “narrow” damage pattern, while the overall damage pattern is quantified by measuring the direct and induced damage as one area of crush. Overlap is not a meaningful parameter in this context. Therefore another parameter was devised to replace overlap. All patterns of damage more than 750mm in width were included in the case selection for both the upper and lower limits because an impact with this much damage should allow interaction with a significant proportion of the car’s front structure. In addition, for the upper limit, cases with a damage width of less than 750 mm were included as long as the damage midpoint was no more than 700 mm from the midpoint of the car. The reason for this was as follows. From a survey of car structures (INSIA, 1997), the centreline of a typical longitudinal stiff member is located approximately 500mm from the centreline of the car. Narrow-object damage patterns with a mid-point offset of up to 700mm were considered as falling within a central “catchment” area for the main structure, while those beyond this were considered as tending towards sideswipe scenarios.

5.1.2.2 Results

These accident selection parameters with the upper and lower limits were applied to the appropriate CCIS data subsets to estimate an upper (optimistic) and lower (pessimistic) bound to the number and proportion of fatalities and seriously injured casualties that are likely to experience a potential benefit as a consequence of improved compatibility. The results are shown in Table 28 below. The proportion of fatalities estimated to experience a potential benefit for the car in frontal impact with car / van category was between 41 and 60 percent and for seriously injured between 71 and 87 percent. The CCIS sample for this category contained a total of 1833 occupants of which 539 were seriously injured and 63 killed. The CCIS sample for the PSV/HGV category, wide objects and narrow objects contained 125, 253, and 139 occupants, respectively.

Table 28: Estimate of potential benefit of improved compatibility for car frontal impacts.

Casualty type	Upper (somewhat optimistic) limit		Lower (somewhat pessimistic) limit		Basic sample for all frontal impacts in each category	
	No	(%)	No	(%)	No	(%)
Car frontal impact with another car / van						
Fatal occupants[6]	37.5	(60)	26	(41)	63	(100)
Serious occupants	469	(87)	381	(71)	539	(100)
Car frontal impact with PSV or HGV						
Fatal occupants	8.2	(75)	4.5	(41)	11	(100)
Serious occupants	32.2	(75)	25	(58)	43	(100)
Car frontal impact with Wide Object						
Fatal occupants	6.5	(36)	5	(28)	18	(100)
Serious occupants	71.5	(82)	56.5	(65)	87	(100)
Car frontal impact with Narrow Object						
Fatal occupants	1.5	(75)	1.5	(75)	2	(100)
Serious occupants	33	(80)	21	(51)	41	(100)

Next, the final step in the methodology outlined previously was taken, namely to scale the results obtained from the analysis using the CCIS data sub-set to the STATS19 national accident data, for each of the frontal impact categories defined. As explained previously, the data set used was adjusted to remove the effect of cars greater than seven years old and was the average of the five years, 1996 to 2000 (Table 29).

[6] Fractional estimates of the numbers of occupants reflect that the estimate is calculated from elements that are weighted in accordance with the criteria, such as "Fifty percent of occupants in severe impacts".

Table 29: Average of car occupant casualties in frontal impacts annually (Great Britain, 1996-2000), adjusted to remove effect of cars over 7 years old, by struck-object category.

Accident Category	Killed casualties	Serious casualties	Slight casualties	All casualties
Car frontal impact with one other vehicle				
Car or Van	254	5557	48695	54506
HGV or PSV	113	644	3081	3838
Other vehicle	43	643	5591	6277
Car frontal impact with object off carriageway				
Wide object	142	1522	7884	9549
Narrow object	162	1366	5733	7261
Other object	28	564	4102	4694
Car frontal impact with more than one other vehicle				
All	189	2089	14929	17207
TOTAL (all car frontal impacts)	931	12385	90015	103331

Note: Frontal impact was defined as first point of impact.

The proportions of casualties estimated to expect a potential benefit calculated from the CCIS data sub sets as described above, were applied to the appropriate impact category in Table 29 to calculate the potential benefit expected nationally for killed and seriously injured casualties shown in Table 30 and Table 31, respectively. It should be noted that:

- For the ‘car frontal impact with one other vehicle’ category, ‘other vehicle’ sub-category, the proportion used for scaling was estimated from the proportions for the car or van and HGV or PSV sub-categories, weighted to reflect the annual casualty ratio of those groups.

- For the ‘car frontal impact with object off carriageway’ category, ‘other’ sub-category, the proportion used for scaling was taken to be equal to that for ‘wide object’ sub-category, due to low confidence in the small sample of narrow object data.

- For the ‘car frontal impact with more than one other vehicle’ category, the proportion used for scaling was estimated from the proportions of the car or van and HGV or PSV, ‘car frontal impact with one other vehicle’ sub-categories, weighted to reflect the annual casualty ratio of those groups.

Table 30: Number of Fatal Casualties estimated to experience a potential benefit for improved compatibility annually in Great Britain.

Accident category	Potential benefit proportion		Average annual number of fatalities	Number of fatalities expected to experience potential benefit	
	Lower (somewhat pessimistic) estimate (%)	Upper (somewhat optimistic) estimate (%)		Lower (somewhat pessimistic) estimate	Upper (somewhat optimistic) estimate
Car frontal impact with one other vehicle					
Car or Van	41	60	254	105	151
HGV or PSV	41	75	113	46	84
Other vehicle	41	64	43	18	28
Car frontal impact with object off carriageway					
Wide object	28	36	142	39	51
Narrow object	75	75	162	122	122
Other Object	28	36	28	8	10
Car frontal impact with more than one other vehicle					
All	41	64	189	77	121
Total (all car frontal impacts)			**931**	**415**	**567**

Table 31: Number of Seriously Injured Casualties estimated to experience a potential benefit for improved compatibility annually in Great Britain.

Accident category	Potential benefit proportion		Average annual number of seriously injured casualties	Number of seriously injured expected to experience potential benefit	
	Lower (somewhat pessimistic) estimate (%)	Upper (somewhat optimistic) estimate (%)		Lower (somewhat pessimistic) estimate	Upper (somewhat optimistic) estimate
Car frontal impact with one other vehicle					
Car or Van	71	87	5557	3928	4835
HGV or PSV	58	75	644	374	482
Other vehicle	69	86	643	446	551
Car frontal impact with object off carriageway					
Wide object	65	82	1522	988	1251
Narrow object	51	80	1366	700	1099
Other	58	81	564	330	459
Car frontal impact with more than one other vehicle					
All	69	86	2089	1450	1792
Total (all car frontal impacts)			**12385**	**8216**	**10470**

5.1.2.3 Conclusions

In summary, the above Tables show that the potential benefit of improved frontal impact compatibility for car occupant casualties involved in frontal impact collisions in Great Britain is estimated to be:

- *some* reduction in injury risk for between 415 and 567 fatalities per year, (currently out of 931 frontal impact car occupant fatalities per year on average[7]).
- *some* reduction in injury risk for between 8216 and 10470 seriously injured casualties per year, (currently out of 12385 frontal impact seriously injured car occupant casualties per year on average).

[7] This figure is adjusted to remove the effect of cars greater than 7 years old.

It should be recognised that potential benefits can be expected in accidents excluded from this analysis such as side impacts.

5.1.3 Estimate of Benefit for Car to Car Impacts of Improved Frontal Impact Compatibility

The second approach aimed to identify the reduction in the number of car occupant fatalities and seriously injured casualties that could be expected as a result of improved compatibility for accidents in which the car was involved in a frontal impact with another car or van. The main benefits of improved compatibility are expected in these types of accidents, i.e. where a car impacts another vehicle.

Currently, accident analysis shows that the majority of fatalities and serious injuries are caused by the occupant impacting the interior of the car, the remainder being caused by the restraint system loading (Wykes, 1998). Many of these injuries are also in the presence of occupant intrusion. Indeed, accident analysis reported by Renault states that 75 percent of car occupant fatal injuries are caused by contact with an intruding part of the vehicle (IHRA Compatibility Workshop, 2002). The main aim of compatibility is to improve the structural performance of vehicles so that in collisions they interact in a predictable manner to absorb the impact energy with little or no occupant compartment intrusion. This will provide a safe environment in which the restraint system can operate. The next step would be to develop better restraint systems that are able to cope with higher compartment decelerations and give the occupant an optimised ridedown for a variety of impact severities.

Hence, a likely benefit of improved compatibility is that it will largely eliminate injuries caused by contact with intruding parts of the vehicle (because it will prevent the occupant compartment intruding) and reduce the injuries caused by contact with the vehicle interior (because of the better restraint systems) up to at least the impact severity that the vehicles will be tested at. However, it may increase the injuries caused by restraint system loading, because cars may become stiffer and hence have a reduced ridedown in an accident. Hopefully, better restraint systems should largely compensate for this.

Following these arguments, this analysis was based on a somewhat pessimistic and somewhat optimistic assumption. The pessimistic assumption was that improved compatibility would prevent injuries caused by contact with parts of the vehicle that had intruded (contact with intrusion) up to a defined impact severity. The optimistic assumption was that improved compatibility would prevent all injuries caused by contact with the vehicle interior up to a defined impact severity. The UK CCIS database codes each individual occupant injury to what caused it, and if this was contact with a part of the vehicle interior, it also uniquely records whether or not this part had intruded. This provided sufficient information to perform an analysis to predict the likely accident outcome, in terms of occupant injury, if contact related to intrusion or all contact caused injuries could be prevented. This was then compared to the known accident outcome to provide an estimate of the benefit, in terms of reduced occupant injury, of improved compatibility. The data set, analysis methodology, results and conclusions are described in detail below.

5.1.3.1 Data Source

The data source used for this analysis was the Co-operative Injury Study (CCIS) Phase 6, which started in June 1998 and will be completed in November 2002. At the time the analysis was undertaken there were 5803 occupants in the database, 3560 of which were restrained front seat occupants. The manner in which CCIS cases are sampled, as described previously in section 4.2.3, is based on injury severity and hence the sample includes a greater proportion of unrestrained occupants than the general population. Various selection criteria were used to obtain a sample of frontal car to car impacts containing 1098 occupants. These included criteria to remove unrestrained occupants, impacts with vehicles other than cars and impacts in which rollovers occurred.

5.1.3.2 Methodology

The methodology used for this analysis was as follows:

1. The MAIS[8] injury severity for each occupant was identified for the CCIS car to car impact data subset described above. For the pessimistic assumption that improved compatibility would prevent injuries caused by contact with intruding parts (intrusion caused) of the vehicle the following analysis was performed. The MAIS for each occupant was identified assuming that contact induced injuries associated with an intruded part of the car, up to a defined impact severity did not occur. The changes in the distribution of the number of occupants at each MAIS level were compared to give an estimate of the benefit of improved compatibility. For the optimistic assumption that improved compatibility would prevent all injuries caused by contact with the vehicle interior the analysis was repeated and the MAIS for each occupant identified assuming that all contact induced injuries did not occur.

2. The CCIS database contains both the occupant injuries, described by the AIS and the police assessed occupant severity outcome, described as fatal, seriously injured, slightly injured and non-injured. A relationship was derived between the injury scale and the outcome assessment using data from all accidents in CCIS phase 6 in order to ensure that the relationship was statistically significant. This relationship was then used to calculate the equivalent change in fatalities and seriously injured casualties for the changes in the occupant MAIS level distribution calculated in step 1 above.

3. The proportional changes in the number of fatalities and seriously injured casualties were scaled to the national statistics to estimate a possible range for the benefit of improved compatibility for car occupants casualties involved in car to one other car / van impacts annually in GB.

5.1.3.3 Results

The change in the MAIS distribution for the occupants in the CCIS data subset for all impacts and up to an impact severity of 56 km/h and 48 km/h ETS is shown, for the pessimistic assumption that improved compatibility would prevent injuries caused by contact with intruding parts of the vehicle[9] (Table 32).

An impact severity of 56 km/h ETS was used as this is widely believed to be a good approximation of the severity of the 64 km/h ODB test, the severity up to which it is expected to offer 'good compatible' performance. However, recent work has estimated the average ETS for a number of EuroNCAP tested (a 64 km/h ODB test)

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[8] MAIS Maximum Abbreviated Injury Scale.
[9] Estimated Test Speed.

cars to be 48 km/h. Hence, this value was also used. It should be noted that using an impact severity equivalent to the ODB test as an upper limit will result in a low estimate because some casualties in more severe impacts will experience benefits which will be ignored.

Table 32: Predicted changes in occupant MAIS distribution using improved compatibility pessimistic assumption.

Occupant MAIS Level	Occu-pants	Occupants, assuming prevention of intrusion-caused injuries up to 48km/h ETS		Occupants, assuming prevention of intrusion-caused injuries up to 56km/h ETS		Occupants, assuming prevention of intrusion-caused injuries in all impacts	
	No	No	Change	No	Change	No	Change
Unknown	34	34	0	34	0	34	0
6	11	11	0	11	0	4	-7
5	9	8	-1	6	-3	5	-4
4	23	22	-1	23	0	17	-6
3	80	70	-10	59	-21	45	-35
2	185	177	-8	171	-14	168	-17
1	611	621	+10	633	+22	652	+41
Uninjured	145	155	+10	161	+16	173	+28
Total	1098	1098	0	1098	0	1098	0

It is seen that removing the intrusion-caused injuries from the MAIS calculation for occupants of cars in the sample generates a new MAIS distribution in which the number of occupants at the higher MAIS levels has decreased with a corresponding increase in the number at lower levels. Overall, the average MAIS level for the occupant group has reduced, which gives an indication of the benefit of improved compatibility.

The change in the MAIS distribution for the occupants in the CCIS data subset up to an impact severity of infinity (i.e. all impacts), 56 km/h and 48 km/h ETS is shown, for the optimistic assumption that improved compatibility would prevent injuries caused by contact with the vehicle interior (Table 33).

Table 33: Predicted changes in occupant MAIS distribution using improved compatibility optimistic assumption.

Occupant MAIS Level	Occu-pants	Occupants, assuming prevention of contact-caused injuries up to 48km/h ETS		Occupants, assuming prevention of contact-caused injuries up to 56km/h ETS		Occupants, assuming prevention of contact-caused injuries in all impacts	
	No	No	Change	No	Change	No	Change
Unknown	34	6	-28	6	-28	2	-32
6	11	11	0	11	0	2	-9
5	9	8	-1	3	-6	1	-8
4	23	20	-3	23	0	13	-10
3	80	57	-23	41	-39	20	-60
2	185	143	-42	134	-51	124	-61
1	611	590	-21	597	-14	603	-8
Uninjured	145	263	+118	283	+138	333	+188
Total	1098	1098	0	1098	0	1098	0

The predicted benefits of improved compatibility above, expressed in terms of the MAIS redistribution for the sample of occupants, were converted into the less detailed police-assessed “Fatal, Serious, Slight” categories of injury severity used in the national statistics and are tabulated below (Table 34 and Table 35).

Table 34: Predicted changes in police assessed occupant severity distribution using improved compatibility pessimistic assumption.

Occupant severity Level	Occu-pants	Occupants, assuming prevention of intrusion-caused injuries up to 48km/h ETS		Occupants, assuming prevention of intrusion-caused injuries up to 56km/h ETS		Occupants, assuming prevention of intrusion-caused injuries in all impacts	
	No	No	Change	No	Change	No	Change
Unknown	23	23	0	24	+1	24	+1
Fatal	40	38	-2	36	-4	24	-16
Serious	322	311	-11	300	-22	288	-34
Slight	611	619	+8	628	+17	645	+34
Uninjured	102	107	+5	110	+8	117	+15
Total	1098	1098	0	1098	0	1098	0

Table 35: Predicted changes in police-assessed occupant severity distribution using improved compatibility optimistic assumption.

Occupant severity Level	Occu-pants	Occupants, assuming prevention of contact-caused injuries up to 48km/h ETS		Occupants, assuming prevention of contact-caused injuries up to 56km/h ETS		Occupants, assuming prevention of contact-caused injuries in all impacts	
	No	No	Change	No	Change	No	Change
Unknown	23	22	-1	22	-1	23	0
Fatal	40	37	-3	33	-7	16	-24
Serious	322	271	-51	255	-67	229	-93
Slight	611	613	+2	623	+12	641	+30
Uninjured	102	155	+53	165	+63	189	+87
Total	1098	1098	0	1098	0	1098	0

From the Tables above, the predicted benefit of improved compatibility for car occupants involved in a single frontal impact with one other car or van can be determined in terms of the proportion of fatalities and seriously injured casualties prevented (Table 36).

Table 36: Predicted benefit of improved compatibility for car occupants involved in a single frontal impact with one other car / van expressed in terms of proportion of fatalities and seriously injured casualties prevented.

Assumption on which prediction was based.	Proportion of fatalities prevented (%)	Proportion of seriously injured casualties prevented (%)
Somewhat pessimistic assumption		
Improved compatibility prevents intrusion-caused injuries up to 48km/h ETS	5	3
Improved compatibility prevents intrusion-caused injuries up to 56km/h ETS	10	7
Improved compatibility prevents intrusion-caused injuries for all impact severities	40	11
Somewhat optimistic assumption		
Improved compatibility prevents contact-caused injuries up to 48km/h ETS	8	16
Improved compatibility prevents contact-caused injuries up to 56km/h ETS	18	21
Improved compatibility prevents contact-caused injuries for all impact severities	60	29

The average annual number of fatally and seriously injured car occupants in frontal impacts adjusted to remove the effect of cars over 7 years old, for impacts where the car is involved in a single impact with one other car or van, is 254 and 5557, respectively (Table 29). If it is assumed that improved compatibility offers increased protection up to an impact severity of 56 km/h ETS, it is predicted that between 25 (10%) and 46 (18%) fatalities and between 389 (7%) and 1167 (21%) seriously injured casualties would be prevented. However, if it is assumed that improved compatibility offers increased protection for all impact severities, it is predicted that between 102 (40%) and 152 (60%) fatalities and between 587 (11%) and 1605 (29%) seriously injured casualties would be prevented. Obviously, the latter assumption is overly optimistic, but the former assumption could be viewed as overly pessimistic. So the actual benefit probably lies somewhere between these estimates.

It should be noted that this benefit has been predicted for car occupants involved in a single frontal impact with one other car or van, only. Improved compatibility should also reduce the number of fatalities and serious injuries for car occupants involved in other categories of frontal impacts such as multiple impacts where the main impact is a frontal one, and frontal impacts where the car collides with an object off the carriageway. The scope for potential benefit in these impacts was discussed in the previous section.

5.1.3.4 Conclusions

For the accident type where a car was involved in a frontal impact with one other car / van there were on average 254 fatalities and 5557 serious injuries annually in recent years in GB. For this group, from the analysis performed, using the assumptions that optimistically 'compatible' cars should prevent contact related injuries and pessimistically 'compatible' cars should prevent injuries caused by intrusion up to a given impact severity, the following predictions were made:

- If it is assumed that improved compatibility offers increased protection for all impact severities, it is predicted that between 102 (40%) and 152 (60%) fatalities and between 587 (11%) and 1605 (29%) serious casualties would be prevented.

- If it is assumed that improved compatibility offers increased protection up to an impact severity of 56 km/h ETS, it is predicted that between 25 (10%) and 46 (18%) fatalities and between 389 (7%) and 1167 (21%) serious casualties would be prevented. It should be noted that compatibility is expected to offer some benefit above an impact severity of 56 km/h ETS, so these predictions are most likely low.

Improved compatibility will also offer casualty savings in impact categories other than car frontal impacts with one other vehicle. As discussed in the previous section it is expected that compatibility will offer some benefit for most car frontal impacts and possibly for side impacts as well. For example, in multiple car impacts the frontal impact is often the most injurious, so for these impacts a similar benefit to that predicted for car to one other vehicle impacts may be expected.

The seriously injured casualty category defined to the Police's injury severity rating covers a wide range of injury severities. It should be noted that the benefit from, for example, reducing a MAIS 4 serious injury to a MAIS 2 serious injury is not accounted for in the analysis performed.

It is recommended that this analysis should be repeated for other car frontal impact accident types to obtain a better estimate of the benefit for GB.

The benefits predicted above are largely dependent on the assumptions made for how 'compatible' cars will perform. Hence, it is recommended that once more about a 'compatible' car's performance is known, the assumptions made should be refined and the analysis repeated.

5.2 BAST ANALYSIS FOR GERMANY

5.3 INTRODUCTION

This study is based on two different accident databases. On the one hand there are the collected police accident reports which are combined in the official German Road Accident Statistics on the other hand there is the GIDAS database of the in depth accident investigations in Hanover and Dresden. The GIDAS database was built up in the year 1999 in combination with the accidents recorded in Hanover in the years from 1990 up to 1999. Out of the official statistics it is possible to determine the road accident casualties depending on the kind of traffic participation but it is impossible to come to reliable statements concerning the actual accident details. Therefore the official German figures are scaled with the different distributions concluded out of the in depth data (see below). Due to the fact that the GIDAS data are collected using a statistical sampling plan in two areas, which nearly reflect the German traffic situation, they can be called representative and scaling should lead to trustworthy results.

The distribution of the road accident casualties in Germany is given in Table 37. It shows that nearly two thirds of the road accident casualties were in cars. 59% of the killed people in the German traffic of the year 2000 were car occupants. For the seriously injured this percentage is 52. Using average costs estimated by the Federal Highway Research Institute (BASt) of Germany all casualties of car occupants in the year 2000 have produced costs of 10.4 billion euro. (€ 1.15 million per fatality; € 83412 for every seriously injured person; € 3737 for every slightly injured person)

Table 37: Distribution of road accident casualties in Germany in the year 2000.

Germany 2000	Fatalities		Seriously injured		Slightly injured	
	No	(%)	No	(%)	No	(%)
Car occupants	4,396	59%	52,759	52%	256,737	64%
Other road users	3,107	41%	49,657	48%	144,921	36%
Total	7,503	100%	102,416	100%	401,658	100%

The official German road accident statistics do not reliably record if the involved vehicles had a side or frontal or rear accident so scaling from a more detailed accident database is necessary. Table 38 shows the impact directions of car accidents for the last 10 years out of the Hanover / GIDAS database. Here one can see that half of all accidents and also half of those with fatal or serious injuries were frontal collisions. In 19% of the cases is the impact direction sideways. These 19% lead to 35% of the fatalities and to 30% of the serious injuries. Even in that part of the side collision in which they collide with another car the occupants will benefit from improved or more compatible car front structures.

Table 38: Distribution of Car Occupant Casualties by Impact Direction in Hanover Database from 1991 to 2000.

Hanover Database Years 1991 – 2000	All Accidents		Accidents with Fatalities		Accidents with Serious Injuries	
Impact Direction	No	(%)	No	(%)	No	(%)
Front	3,945	53%	47	47%	185	53%
Side	1,400	19%	35	35%	106	30%
Rear	1,225	16%	7	7%	23	7%
unknown / others	876	12%	11	11%	35	10%
Total	7,446	100%	100	100%	349	100%

The scaled numbers of the official German statistics are given in Table 39. This scaling was done by multiplying the original German numbers for the year 2000 with the percentages from Table 38.

Table 39: Distribution of Car Occupant Casualties by Impact Direction Extrapolated for Germany for the Year 2000.

Germany 2000 **Extrapolated**	Accidents with Fatalities		Accidents with Serious Injuries	
Impact Direction	No	(%)	No	(%)
Front	2,066	47%	27,967	53%
Side	1,539	35%	16,024	30%
Rear	308	7%	3,477	7%
unknown / others	483	11%	5,291	10%
Total	4,396	100%	52,759	100%

Even the collision partners can not be detected exactly within the official statistics. Therefore the same method of scaling was used to come to the results in Table 40 and Table 41. These Tables deal just with the frontal impacts of killed or seriously injured car occupants. These were all accidents with an impact direction between 10 o'clock and 2 o'clock. Nearly two thirds of the fatal and serious casualties of car occupants were occurred in collisions with another vehicle or in multiple collisions with vehicles and/or fixed obstacles.

Table 40: Distribution of Car Occupant Casualties in Frontal Impacts by Impact Partner in Hanover Database from 1991 to 2000.

Hanover Database Years 1991 – 2000	Accidents with Fatalities		Accidents with Serious Injuries	
Impact Partner	No	(%)	No	(%)
Other Vehicle	28	28%	109	31%
Fixed Obstacle	39	39%	106	30%
Multiple	33	33%	134	38%
Total	100	100%	349	100%

The distribution of impact partners is nearly even. One can roughly conclude here that improved compatibility without worsening self protection should have positive consequences for about two thirds of the car occupants. For a more detailed estimation the next chapter deals with a specific methodology.

Table 41: Distribution of Car Occupant Casualties in Frontal Impacts by Impact Partner, Extrapolated for Germany for the Year 2000.

Germany 2000 **Extrapolated**	Accidents with Fatalities		Accidents with Serious Injuries	
Impact Partner	No	(%)	No	(%)
Other Vehicle	578	28%	8,735	31%
Fixed Obstacle	806	39%	8,494	30%
Multiple	682	33%	10,738	38%
Total	2,066	100%	27,967	100%

The aim of the work is to provide initial estimates of the benefits of implementing compatibility measures for frontal impacts for Germany. For the following analysis a similar methodology to that described in section 5.1.2.1 of the TRL part of the analysis was used.

5.4 ESTIMATE OF THE POTENTIAL BENEFIT OF IMPROVED FRONTAL IMPACT COMPATIBILITY

The objective of the work was to identify the number of car occupants which could probably be less injured in frontal impacts then they actually were because of improved frontal compatibility. To get this type of information it was necessary to analyse the following accident data:

- The Official German Accident Statistics, here especially extrapolated values for fatal and seriously injured car occupants in frontal impacts
- The GIDAS database of the in depth accident investigation carried out in Hanover and Dresden (description in the accident analysis part of BASt) in combination with accidents collected in Hanover during the last 10 years. The start of the second investigation team in Dresden was Summer 1999.

5.4.1 Methodolgy

For this analysis a similar methodology to that used by TRL (section 5.1.2.1) was followed. Because of some differences in the accident selection criteria between the English CCIS database and the German GIDAS database the total number and the percentages of fatal and serious casualties in the GIDAS database is by far lower. The car impact partner categories used were difference to those used by TRL. They were just subdivided into single collisions with other vehicles or fixed obstacles and multiple collisions with other vehicles and /or fixed obstacles.

Table 42: **Upper and Lower Limits.**

Accident parameter	Upper (somewhat optimistic) limit	Lower (somewhat pessimistic) limit	Basic sample for all frontal impacts
Principle direction of force (pdf) (o'clock)	10,11,12,1,2	11,12,1	10,11,12,1,2
Overlap	Include ≥ 20%	Include ≥ 30%	All
Multiple impacts	Exclude cases in which a significant side impact occurred.	Exclude cases in which a significant side impact occurred and cases in which the other impact was judged to be more injurious than the frontal impact.	All multiple impacts with frontal as the initial point of contact.
Rollover	Include all, with or without rollover	Include only those where rollover was less injurious than the frontal impact.	All, with or without rollover
Impact severity	Include all with EES up to 56km/h	Include all with EES up to 48km/h	All with known EES.

Table 42 shows the upper and lower limits for accident parameters used to identify proportion of fatalities and seriously injured expected to experience a potential benefit. The limits are the same as those used by TRL with the exception of the impact severity. The TRL impact severity optimistic limit included half of the cases with an ETS higher than 56 km/h and the pessimistic limit half of the cases with an ETS higher than 48 km/h. The limits used for this analysis do not include these cases. The analysis of the different EES (here the EES, Energy Equivalent Speed, instead of the ETS is used) and delta v values have shown that nearly all of the cases with very high changes in velocity during the impact would not benefit from improved compatibility.

5.4.2 Results

The accident selection parameters were applied to the GIDAS data subset to estimate the upper and lower bound of the potential benefit for car occupants in frontal collisions. Table 43 shows the result of this application.

Table 43: **Estimate of potential benefit of improved compatibility for car frontal impacts out of the GIDAS database.**

Casualty type	Upper (somewhat Optimistic) Limit		Lower (somewhat Pessimistic) Limit		Basic Sample for all frontal impacts in each category	
	No	(%)	No	(%)	No	(%)
Car Frontal Impact with Other Vehicle						
Fatal occupants	3	25%	0	0%	12	100%
Serious occupants	69	82%	42	50%	84	100%
Car Frontal Impact with Fixed Obstacle						
Fatal occupants	3	25%	2	17%	12	100%
Serious occupants	21	49%	6	14%	43	100%
Car Frontal Impacts with multiple Vehicles or Obstacles						
Fatal occupants	3	33%	2	22%	9	100%
Serious occupants	37	69%	19	35%	54	100%
Total						
Fatal occupants	9	27%	4	12%	33	100%
Serious occupants	127	70%	67	37%	181	100%

The proportion of fatalities, which could potentially benefit from improved compatibility, lies in the range of 25% to 33% for the optimistic estimate depending on the impact configuration and similarly between 0% and 22% for the pessimistic estimate. It is possible that this result is not statistically significant because the GIDAS database, which this analysis used, contained only 33 killed occupants in frontal impacts. As already mentioned above, the criteria for including an accident in the GIDAS database are different to that of CCIS. Nevertheless the figures are quite comparable with these out of the CCIS database if only the seriously injured casualties are taken into account. Here the optimistic percentages of the addressable cases are 49% in collisions with fixed obstacles and 82% (CCIS = 87%) in car to other vehicle accidents. The same applies for the multiple collisions with 69% for the optimistic approach (CCIS = 64%).

The last step was to scale the official German statistics with the estimated percentages. The results of doing this are shown in Table 44 for fatal and seriously injured car occupants.

Table 44: Estimate of potential benefit of improved compatibility for car occupants in car frontal impacts for Germany.

Extrapolated for Germany for the year 2000	Potential benefit proportion		Number of **fatalities** for the year 2000	Number of **fatalities** expected to experience potential benefit	
	Lower (Somewhat Pessimistic) estimate (%)	Upper (Somewhat Optimistic) estimate (%)		Lower (Somewhat Pessimistic) estimate	Upper (Somewhat Optimistic) estimate
Car Frontal Impact with					
Other Vehicle	0%	25%	578	0	145
Fixed Obstacle	17%	25%	806	137	202
Multiple Impact	22%	33%	682	150	225
Total	**14%**	**28%**	**2,066**	**287**	**572**

Extrapolated for Germany for the year 2000	Potential benefit proportion		Number of **seriously injured** casualties for the year 2000	Number of **seriously injured** expected to experience potential benefit	
	Lower (Somewhat Pessimistic) estimate (%)	Upper (Somewhat Optimistic) estimate (%)		Lower (Somewhat Pessimistic) estimate	Upper (Somewhat Optimistic) estimate
Car Frontal Impact with					
Other Vehicle	50%	82%	8,735	4,368	7,163
Fixed Obstacle	14%	49%	8,494	1,190	4,163
Multiple Impact	35%	69%	10,738	3,759	7,410
Total	**33%**	**67%**	**27,967**	**9,317**	**18,736**

5.4.3 Conclusions

The potential benefit of improved compatibility for car occupant casualties involved in frontal impact collisions in Germany based on accident data for the year 2000 was estimated to be:

- some reduction in injury risk or injury outcome for between 9,317 (33%) and 18.736 (67%) seriously injured car occupants per year, (there were 27,967 frontal impact car occupant seriously injured casualties in the year 2000).

An estimate was also made for fatalities. However, it is possible that this result was not statistically significant as the GIDAS database, on which the analysis was based, contained only 33 fatalities for this impact configuration. Noting this caveat, the estimate was:

- some reduction in injury risk or injury outcome for between 287 (14%) and 572 (28%) fatalities per year, (there were 2,066 frontal impact car occupant fatalities in the year 2000).

It should be noted that potential benefit means that the casualties could be expected to experience some reduction in injury risk or outcome, the injury will not necessarily be prevented.

Further analyses are required in order to obtain a more accurate estimate of the benefit of improved frontal impact compatibility. It is intended that these analyses will be performed as part of the research activities of the VC-COMPAT project within the 5th framework programme.

5.5 SUMMARY OF CONCLUSIONS AND RECOMMENDATIONS

Conclusions

For GB the potential benefit of improved frontal impact compatibility for car occupant casualties involved in frontal impact collisions in Great Britain is estimated to be:

- *some* reduction in injury risk for between 415 (45%) and 567 (61%) fatalities per year, (currently out of 931 frontal impact car occupant fatalities per year on average[10]).

- *some* reduction in injury risk for between 8216 (66%) and 10470 (85%) seriously injured casualties per year, (currently out of 12385 frontal impact seriously injured car occupant casualties per year on average).

For GB the benefit has been estimated for one particular type of accident only, namely a car frontal impact with one other car / van. For this accident type there were on average 254 fatalities and 5557 serious injuries annually in recent years in GB. From the analysis performed, using the assumptions that optimistically ‘compatible’ cars should prevent contact related injuries and pessimistically ‘compatible’ cars should prevent injuries caused by intrusion up to a given impact severity, the following predictions were made:

[10] This figure is adjusted to remove the effect of cars greater than 7 years old.

- If it is assumed that improved compatibility offers increased protection for all impact severities, it is predicted that between 102 (40%) and 152 (60%) fatalities and between 587 (11%) and 1605 (29%) serious casualties would be prevented.

- If it is assumed that improved compatibility offers increased protection up to an impact severity of 56 km/h ETS, it is predicted that between 25 (10%) and 46 (18%) fatalities and between 389 (7%) and 1167 (21%) serious casualties would be prevented. It should be noted that compatibility is expected to offer some benefit above an impact severity of 56 km/h ETS, so these predictions are most likely low.

It should be recognised that much further benefit can be expected for other accident types, especially car to vehicle frontal impacts, most likely car frontal collisions with roadside obstacles and possibly for side impacts as well. The seriously injured casualty category defined to the Police's injury severity rating covers a wide range of injury severities. It should be noted that the benefit from, for example, reducing a MAIS 4 serious injury to a MAIS 2 serious injury is not accounted for in the analysis performed.

For Germany, the potential benefit of improved compatibility for car occupant casualties involved in frontal impact collisions based on accident data for the year 2000 has been estimated to be:

- some reduction in injury risk for between 9,317 (33%) and 18.736 (67%) seriously injured car occupants per year, (there were 27,967 frontal impact car occupant seriously injured casualties in the year 2000).

An estimate was also made for fatalities. However, it is possible that this result was not statistically significant as the GIDAS database, on which the analysis was based, contained only 33 fatalities for this impact configuration. Noting this caveat, the estimate was:

- some reduction in injury risk or for between 287 (14%) and 572 (28%) fatalities per year, (there were 2,066 frontal impact car occupant fatalities in the year 2000).

Recommendations

In order to obtain a more complete benefit estimate for GB, it is recommended that a similar benefit analysis to that performed for the car frontal impact with one other car or van type of accident should be conducted for other car frontal impact accident types. For Germany, it is recommended that an analysis to estimate the benefit of improved compatibility, in terms of the number of lives saved as opposed to the reduction in injury risk, should be performed.

The benefits predicted are largely dependent on the assumptions made for how 'compatible' cars will perform. Hence, it is recommended that once more about a 'compatible' car's performance is known, the assumptions made should be refined and the analysis repeated.

6 CRASH TESTING

6.1 INTRODUCTION

The aim of this work package was to perform crash tests and associated analysis to continue the development of the full width deformable barrier and Progressive Deformable Barrier (PDB) test procedures, described in section 3.2. In addition, this work package provided funding for additional instrumentation for 7 EuroNCAP tests, namely a Load Cell Wall, and analysis of the results.

The table below lists the tests performed with the car(s) used, the organisation that performed the test and the aim of test. In total 6 full width deformable barrier tests, 5 PDB tests, 1 car to car test, and 10 EuroNCAP LCW measurements were performed. TRL were able to perform 1 additional full width test to that quoted for in the original bid, because of cost savings made, such as Land Rover supplying the Range Rover free of charge to the project. In addition, an extra 2 LCW measurements are reported to that quoted for in the original bid, 1 by BASt and 1 by TRL.

Table 45: Full Width Deformable Barrier Tests

	Vehicle	Organisation	Aim of test
1.	Mondeo	TRL	To redesign the full width deformable element depth and stiffness.
2.	Mondeo	TRL	To redesign the full width deformable element depth and stiffness.
3.	Renault Laguna II	TRL	To provide benchmark data for the development of the assessment protocol. Note: The Laguna II is the best performing car tested in EuroNCAP to date.
4.	Astra	UTAC	To provide test data for the development of the assessment protocol and for future comparison of the proposed test procedures.
5.	Astra modified	UTAC	To provide test data for the development of the assessment protocol and for future comparison of the proposed test procedures.
6.	Rover 75	UTAC	To investigate the performance of a ‘stiff’ vehicle with multiple load paths and provide test data for the development of the assessment protocol.

Table 46: Progressive Deformable Barrier (PDB) Tests

	Vehicle	Organisation	Aim of test
1.	Smart	BASt	To investigate the effect of a low mass vehicle on PDB compatibility assessment.
2.	Range Rover	TRL	To investigate the effect of a high mass and body on frame vehicle on PDB compatibility assessment.
3.	Mondeo	TRL	To provide test data for the development of the assessment protocol and for future comparison of the proposed test procedures.
4.	Astra (approx 200 kg mass decrease)	BASt	To investigate the influence of mass on the PDB assessment of the structural interaction aspect of compatibility.
5.	Volvo S80	BASt	To complete PREDIT/EUCAR/Renault test matrix to form complete data set for development of PDB evaluation criteria.

Table 47: Car to Car Test

	Vehicles	Organisation	Aim of test
1.	Toyota Yaris Renault Clio	Fiat	To investigate the structural performance of the Yaris. Note: The Yaris has only one major load path, i.e. lower rails.

Table 48: Offset Deformable Barrier (ODB) Tests

	Vehicles	Organisation	Aim of tests
1.	Renault Laguna	TRL	These tests are EuroNCAP assessments and as such are funded by EuroNCAP. This project provided funding for additional instrumentation for these tests, namely a Load Cell Wall. This project analysed and interpreted the Load Cell Wall results. The eventual aim of recording Load Cell Wall results is to ascertain whether this measure is adequate to assess and control a car's global stiffness.
2.	Ford Mondeo	TRL	
3.	Volvo S60	TRL	
4.	Fiat Multipla	TRL	
5.	Peugeot 307	TRL	
6.	Honda Civic Stream	TRL	
7.	Vauxhall Frontera	TRL	
8.	Mercedes SLK	BASt	
9.	Mazda MX5	BASt	

6.2 FULL WIDTH DEFORMABLE BARRIER TESTS

As mentioned in section 3, to improve compatibility an essential prerequisite is good structural interaction. The hypothesis on which the full width test is based states that cars with more homogeneous fronts offer the potential for good structural interaction with other cars. A full width impact of a car against a high definition load cell wall offers the potential to map the force deflection characteristics of the car's front. However, there are some issues that generate problems when a rigid faced load cell wall is used:

- Localised stiff structures can hold off adjacent structures which are slightly set back.

- Localised stiff structures effectively unload adjacent structures, which are slightly less stiff.

- The parts of the car that first impact the wall are decelerated instantaneously giving rise to large inertial forces, both within the structure and measured by the load cell wall. Such forces are not present in impacts with deforming structures, such as other cars.

- When the engine impacts the wall, it is brought to rest very rapidly again generating high inertial forces. In a car to car impact, the engine can rotate or move slightly out of the way of the other car's engine, so reducing its deceleration.

- No relative shear is generated in the front structure to exercise any shear connections between load paths.

In order to overcome these problems, a deformable barrier face is fitted to the front of the load cell wall. If the test is to also function as a high deceleration test for frontal impact, the overall car deceleration should not be significantly affected by the addition of the deformable face.

The depth and stiffness of the barrier face were initially set at 150mm and 0.34MPa, respectively, primarily for three reasons. The first was so that, compared to a rigid wall test, the initial high decelerations at the front of the car were attenuated to make the test more representative of a vehicle to vehicle impact. The second was to reduce the magnitude of the engine deceleration loading on the wall to avoid high engine loads masking the loads from the car structure. The third was to minimise the effect that the face had on the occupant compartment deceleration pulse so that the test could also be used as a high deceleration frontal impact test similar to US FMVSS 208.

Unfortunately, initial tests with this barrier face have shown that localised stiff structures on the car can form preferential load paths, which dramatically reduce loading from adjacent structures, indicating that the barrier depth and / or stiffness may need to be altered (Edwards et al. 2002). An example of this effect is seen with a family sized car, which has several such structures, namely a towing eye and radiator mount brackets located on the engine subframe (Figure 32).

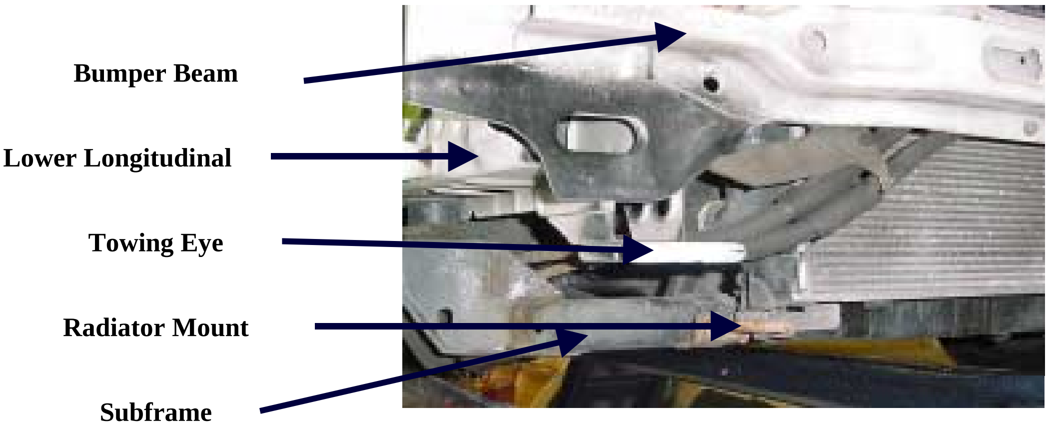


Figure 32: View of family sized car structure (right hand side) showing towing eye and radiator mount bracket protruding structures.

Examination of the deformed car and barrier face showed that the front crossbeam of the engine subframe applied load to the load cell wall (LCW) with over 50 percent of this load being applied to two cells. This was caused by the radiator mount brackets penetrating the deformable barrier face to make direct contact with the LCW to form preferential load paths. These unloaded the adjacent crossbeam structure. Unfortunately, this load distribution is not representative of the stiffness homogeneity of the crossbeam structure.

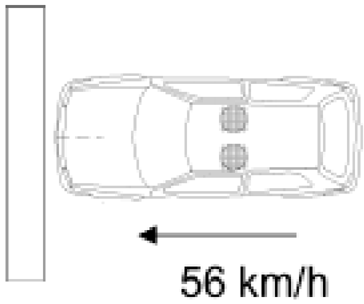
To attempt to resolve this problem the test was repeated using a stiffer deformable barrier face (1.71 MPa) of the same depth. The radiator mount brackets penetrated the barrier face but did not contact the wall, which allowed loading from the rest of the crossbeam to load the wall. However, even though this stiffer barrier face appeared to solve the preferential load path problem, it was not a viable solution as it dramatically increased the engine deceleration load.

The first two tests performed for this project were used to help redesign the barrier face to overcome the problems with both the 0.34MPa barrier and the 1.71MPa barrier faces detailed above. Following this, two tests were conducted in order to evaluate the ability of this revised barrier face to assess the difference between an Astra and a modified Astra with improved structural interaction. Two further tests were performed with the Renault Laguna II and the Rover 75 to provide data for the future development of the ‘compatibility’ assessment protocol for the full width test.

6.2.1 TRL tests

Ford Mondeo against development full width barrier

Date	24/9/2001
Location	TRL
Topic Group	Development FWT
Mass Ratio	-
Test Number	11MF



Vehicle 1:	Ford Mondeo 93	Barrier:	Development Full Width Impact Barrier
Impact Side:	Front		
Test Speed:	56km/h		
Overlap:	100%		
Test Mass:	1455kg		
Dummies:	RHS – HybridIII LHS – HybridIII* *not instrumented		

Test Objective

This test was one of a series for the DTLR/EC compatibility project, the objective being to redesign the full width test barrier face. Previous full width impact tests using barriers of 150mm depth, with a stiffness of 0.34MPa, revealed problems with localised stiff structures, such as towing eyes, that are found on many cars. When the barrier bottomed out, these localised stiff structures formed preferential load paths between the vehicle and the load cell wall, dramatically reducing loading from adjacent structures. Increasing the barrier stiffness to 1.71MPa prevented direct contact of these structures with the load cell wall, but was not a viable solution as it increased the engine deceleration load. This test combined these two previous barriers with the objective of both preventing the engine deceleration problems and the formation of preferential load paths.

Test Details

The Ford Mondeo was impacted into an aluminium honeycomb barrier of size 2m x 0.75m x 0.30m and a crush strength of 0.34MPa for the first 150mm and 1.71MPa thereafter, mounted onto a high resolution load cell wall (Figure 33).

Figure 33: Development full width test barrier mounted onto load cell wall.

The load cell wall was formed of 128 load cells of 125mm x 125mm arranged in a 16 x 8 matrix and mounted 50mm from the ground. The deformable barrier covered the lower 6 rows of the load cell wall.

The Ford Mondeo was instrumented with accelerometers at various locations around the car, predominantly to allow the interface force to be calculated. One instrumented HybridIII dummy was placed in the driver's seat and one uninstrumented HybridIII dummy was placed in the passenger's seat. They were positioned according to the EuroNCAP Testing Protocol Version 3 dated April 2001. The driver and passenger dummies were restrained by three point seat belts.

Results and Discussion

The test speed and overlap were within the specified tolerances. Pre and post test images of the test vehicle are shown (Figure 34 and Figure 35), along with the resultant barrier deformation (Figure 36).

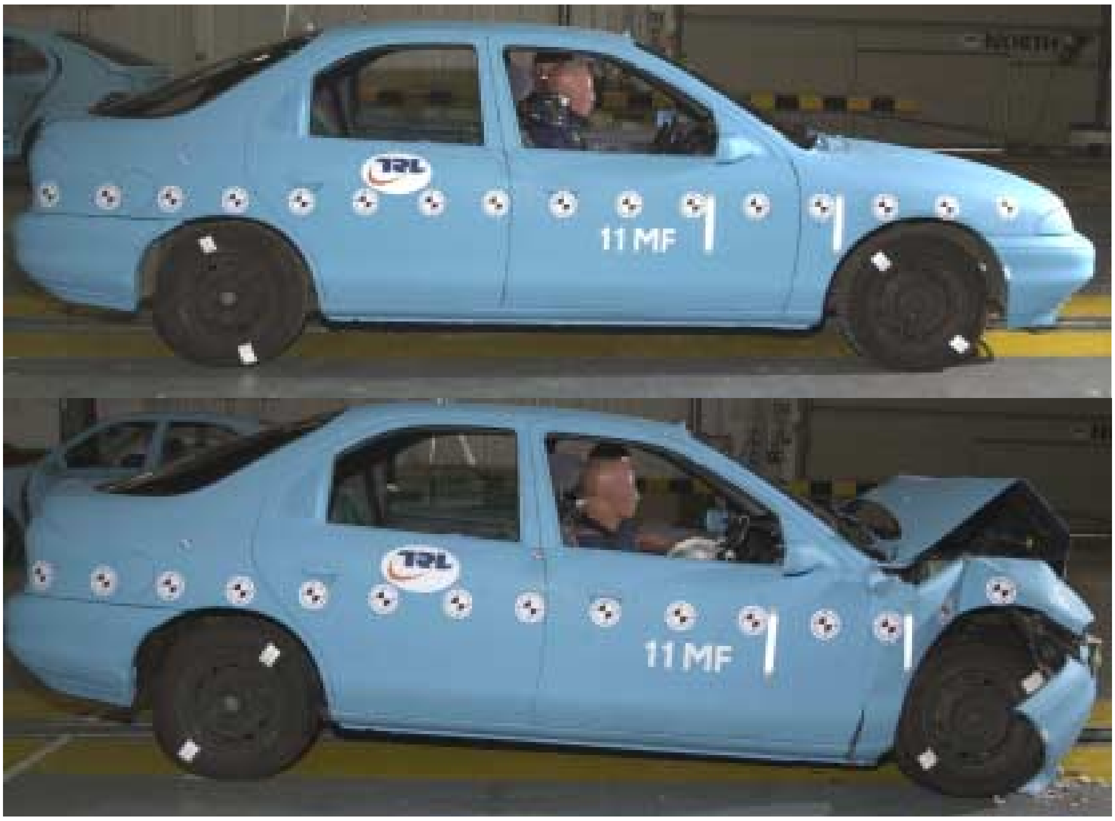


Figure 34: Pre and post test driver's side view of test vehicle.

Figure 35: Pre and post test front view of test vehicle.

Figure 36: View of the resultant barrier deformation.

Comparison of the load cell wall force, with that recorded from a rigid wall impact showed that the initial 150mm layer of 0.34MPa aluminium honeycomb reduced the engine deceleration loading (Figure 37).

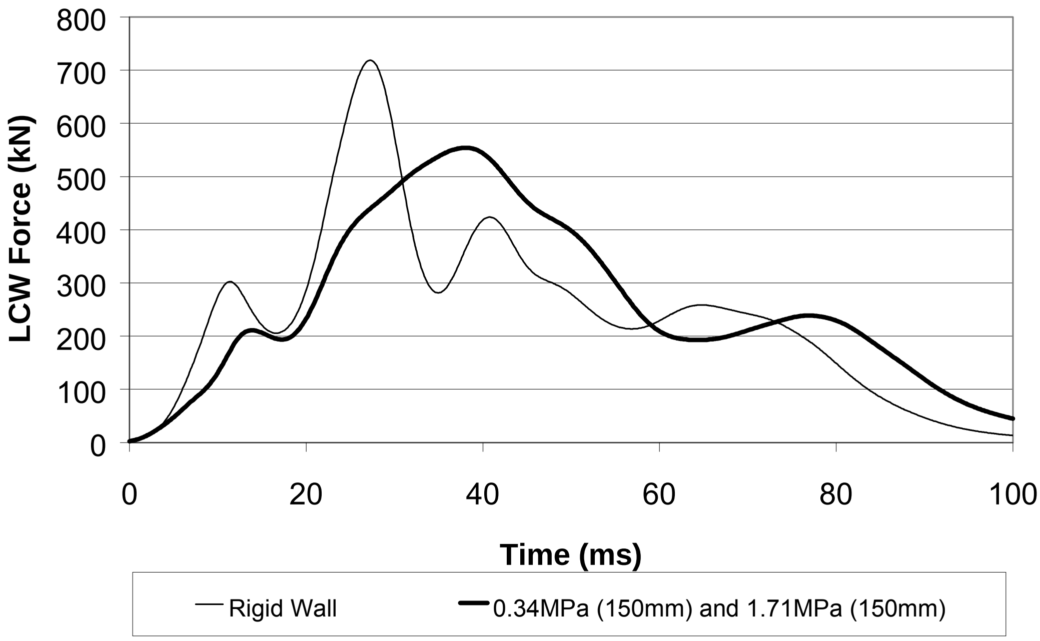


Figure 37: Load cell wall forces showing the reduction in engine deceleration loading peak force of 700kN for the test with the development barrier face.

Whilst the 150mm layer of 0.34MPa aluminium honeycomb resulted in an attenuation of the engine loading, the effect of the adding the additional 1.71MPa layer was marginal when considering the occupant compartment deceleration (Figure 38). Due to the stiffness of the additional layer, the vehicle behaved in a similar manner to having a rigid wall behind the initial 0.34MPa layer.

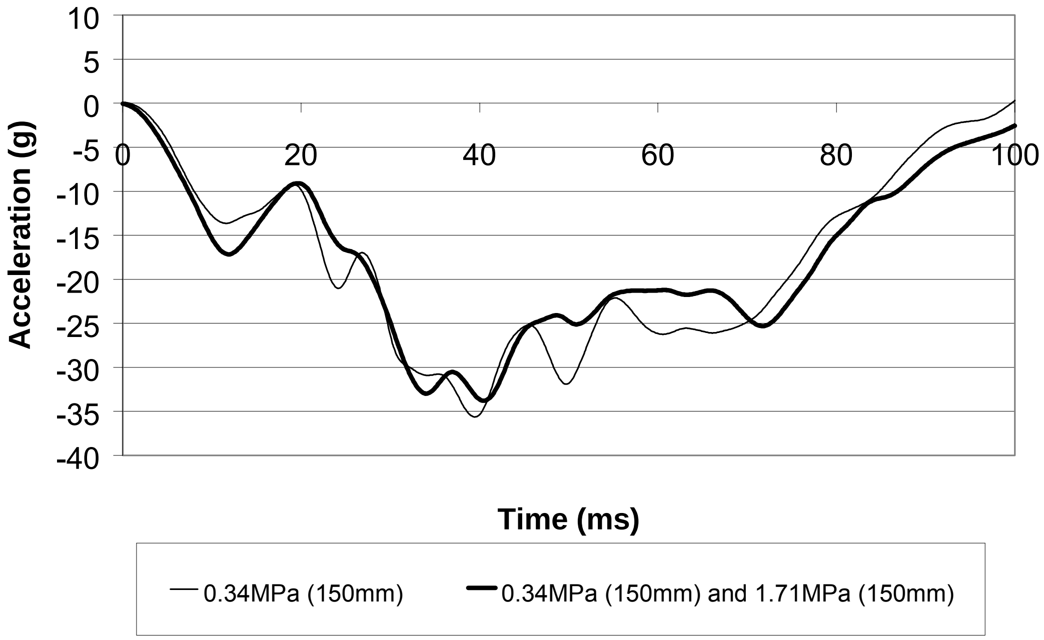


Figure 38: Occupant compartment acceleration showing similarity between tests with single 0.34MPa layer and additional 1.71MPa layer.

Comparison of the effect on the driver chest acceleration confirmed that the effect of adding the two layer barrier to the rigid wall test had no significant effect (Figure 39).

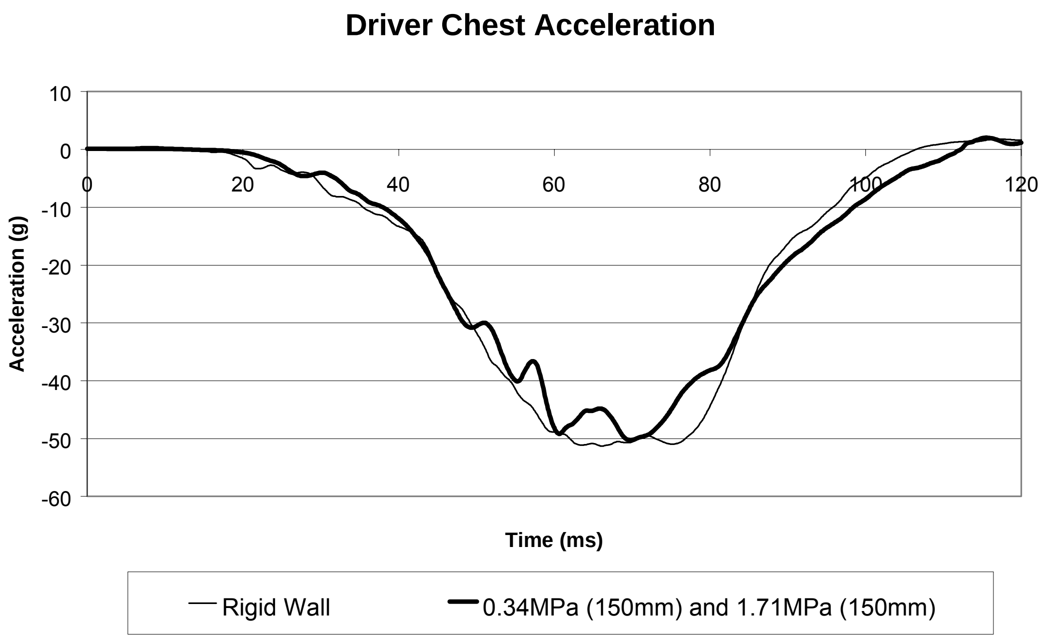


Figure 39: Driver chest acceleration showing the similarity between the rigid wall impact and the double layer barrier impact tests.

The difference between having a rigid wall and the 1.71MPa layer is that the localised stiff structures, due to their small frontal area, could penetrate the 1.71MPa layer (Figure 40). In the rigid wall impact these structures deformed to an extent where they became rigid and effectively held the remaining structure away from the wall. It was the penetration of these localised stiff structures that allowed the stiffer layer of the barrier to load adjacent structures and therefore prevented the formation of preferential load paths with the load cell wall.

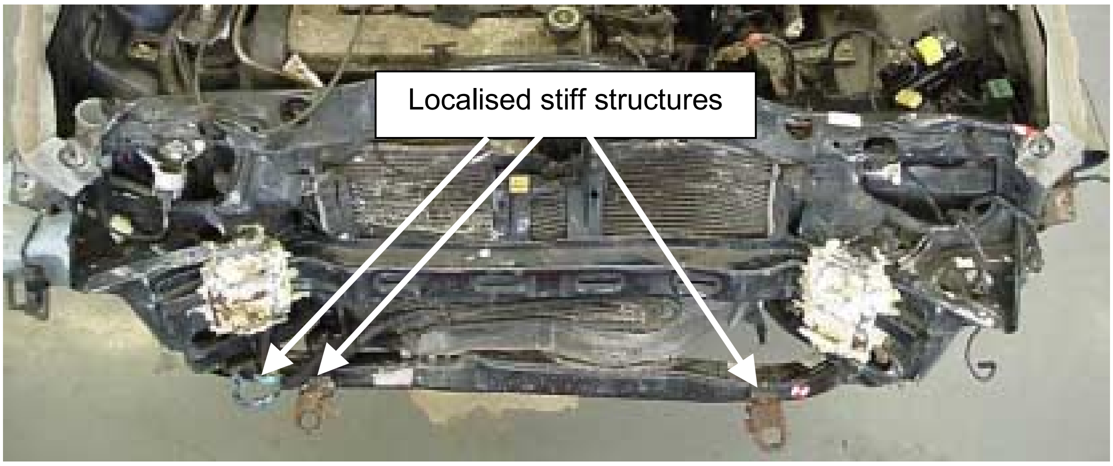


Figure 40: Vehicle deformation showing the localised stiff structures such as the radiator mounting brackets and the vehicle towing eye.

Evidence that this barrier prevented the formation of these preferential load paths is seen by examining the LCW force distribution (Figure 41). This shows the two lower rails, the engine exhaust manifold and centre section of the subframe crossbeam generating high loads. The timing of these loads can be assessed by examining the force time history for each of the individual load cells (Figure 42).

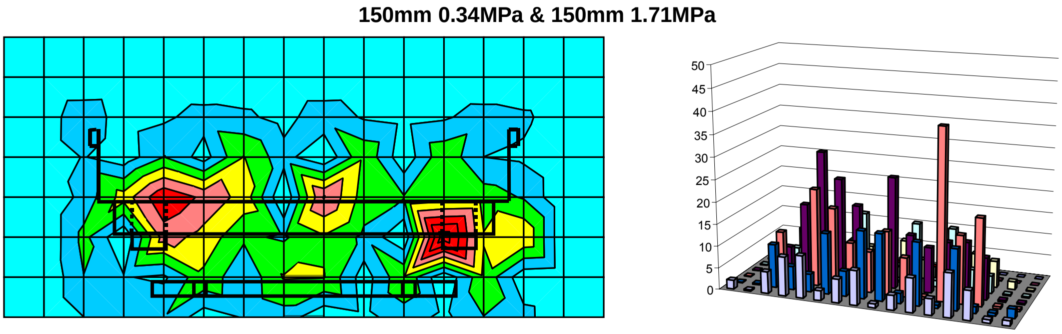


Figure 41: Force distribution based on peak cell loads showing the high loading in front of the lower rails, subframe and engine exhaust manifold. Note: the grid for the 2D plot represents the centre point of each load cell.

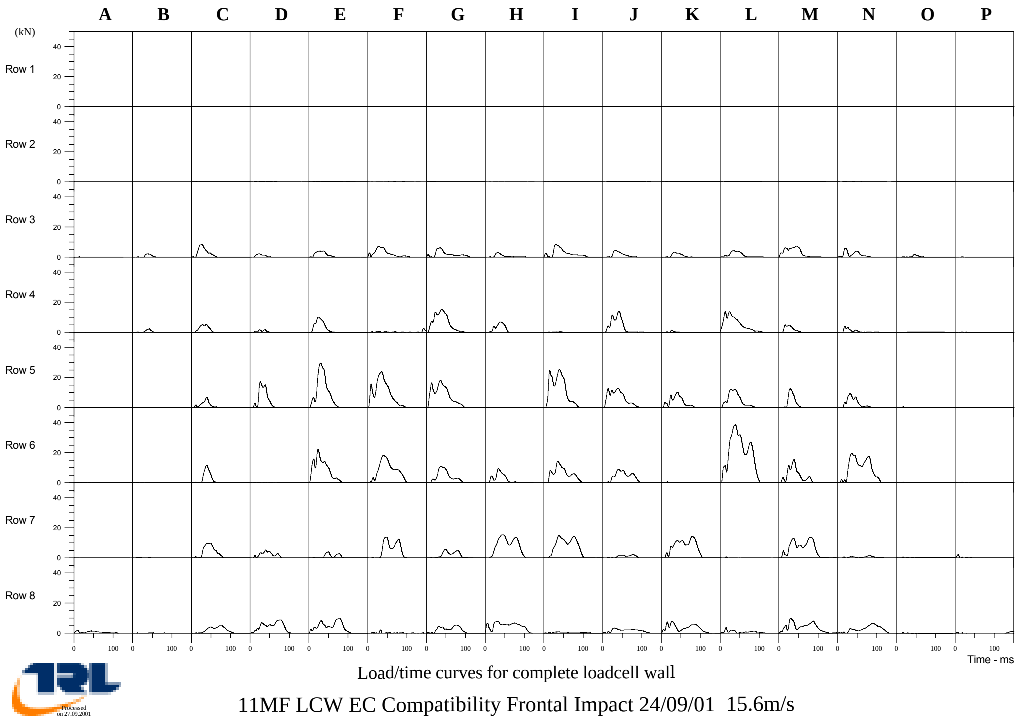


Figure 42: Load cell wall force time history showing the high forces generated in front of the lower rails, subframe and the engine exhaust manifold.

A comparison of the post test barrier deformation with the location of the LCW peak forces showed that the barrier deformation in the location of the engine manifold was not consistent with the measured force. Further investigation of these test results, and other development full width test results, using barriers of similar stiffness, revealed a problem that some load cells were bridged. This was caused by the shear stiffness of the barrier and slight variations in the height of individual load cells. The bridging effect unloaded some load cells and increased the load on others, which would result in an incorrect representation of the homogeneity of the vehicle frontal structure. The barrier face needs to be redesigned to remove this problem.

Conclusions

- The barrier configuration of a 150mm of 0.34MPa followed by 150mm of 1.71MPa solved the problem of localised stiff structures on the test vehicle forming preferential load paths

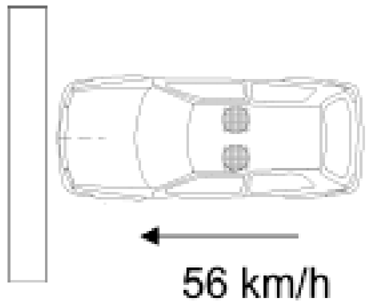
- The initial 150mm 0.34MPa achieved the aim of attenuating the engine loading.

- The additional 1.71MPa layer had little effect on the deceleration measured within the occupant compartment, which is an important factor for using this proposed test to fulfil a requirement for an additional frontal impact test with a high passenger compartment deceleration pulse.

- The shear strength of the barrier face stiff rear honeycomb layer caused bridging of load cells and possible load spreading. This resulted in a measured force distribution, which was not representative of the stiffness homogeneity of the vehicle tested.

Ford Mondeo against development full width barrier

Date	8/2/2002
Location	TRL
Topic Group	Development FWT
Mass Ratio	-
Test Number	27MF



Vehicle 1:	Ford Mondeo 93	Barrier:	Development Full Width Impact Barrier
Impact Side:	Front		
Test Speed:	56km/h		
Overlap:	100%		
Test Mass:	1455kg		
Dummies:	RHS – HybridIII LHS – HybridIII* *not instrumented		

Test Objective

This test was one of a series for the DTLR/EC compatibility project, the objective being to redesign the full width test barrier face. From a previous test (11MF), it was determined that a desirable barrier configuration was a two layer barrier, with the first layer used to limit the engine deceleration load and the second layer to prevent any localised stiff structures forming preferential load paths. However, the increased shear stiffness of the second layer created problems with both load cell bridging and load spreading. The objective of this test was therefore to assess a barrier with a similar configuration, but with a reduction in the shear stiffness of the second layer. Segmenting the stiffer rear layer into individual blocks achieved this reduction.

Test Details

The Ford Mondeo was impacted into an aluminium honeycomb barrier of size 2m x 0.75m x 0.235m. The front layer had a depth of 150mm and a crush strength of 0.34MPa. The rear layer had a depth of 85mm and a crush strength of 1.71MPa. The stiffer rear layer of the barrier was segmented into individual blocks. The size of these blocks was 125mm by 125mm, which represented the frontal area of each load cell. The segmenting resulted in 96 individual blocks that lined up with the load cells on the wall. Due to technical difficulties with the manufacture of the barrier, the depth of the second layer was limited to 85mm, as opposed to the previously tested 150mm (11MF). In addition, the front cladding of the barrier was omitted in order to minimise any barrier deformation caused by bending.

The barrier was mounted onto a high resolution load cell wall (Figure 43). The load cell wall was formed of 128 load cells of 125mm x 125mm arranged in a 16 x 8 matrix. The deformable barrier covered the lower 6 rows of the load cell wall.

Figure 43: Development full width test barrier mounted onto load cell wall.

The Ford Mondeo was instrumented with accelerometers at various locations around the car, predominantly to allow the interface force to be calculated. One instrumented HybridIII dummy was placed in the driver's seat and one uninstrumented HybridIII dummy was placed in the passenger's seat. They were positioned according to the EuroNCAP Testing Protocol Version 3 dated April 2001. The driver and passenger dummies were restrained by three point seat belts.

Results and Discussion

The test speed and overlap were within the specified tolerances. Pre and post test images of the test vehicle are shown (Figure 44 and Figure 45), along with the resultant barrier deformation (Figure 46).

Figure 44: Pre and post test driver's side view of test vehicle.

Figure 45: Pre and post test front view of test vehicle.

Figure 46: View of the resultant barrier deformation.

The change in the shear stiffness and depth of the second layer had no significant effect on either the total load cell wall force (Figure 47) or the occupant compartment deceleration pulse (Figure 48).

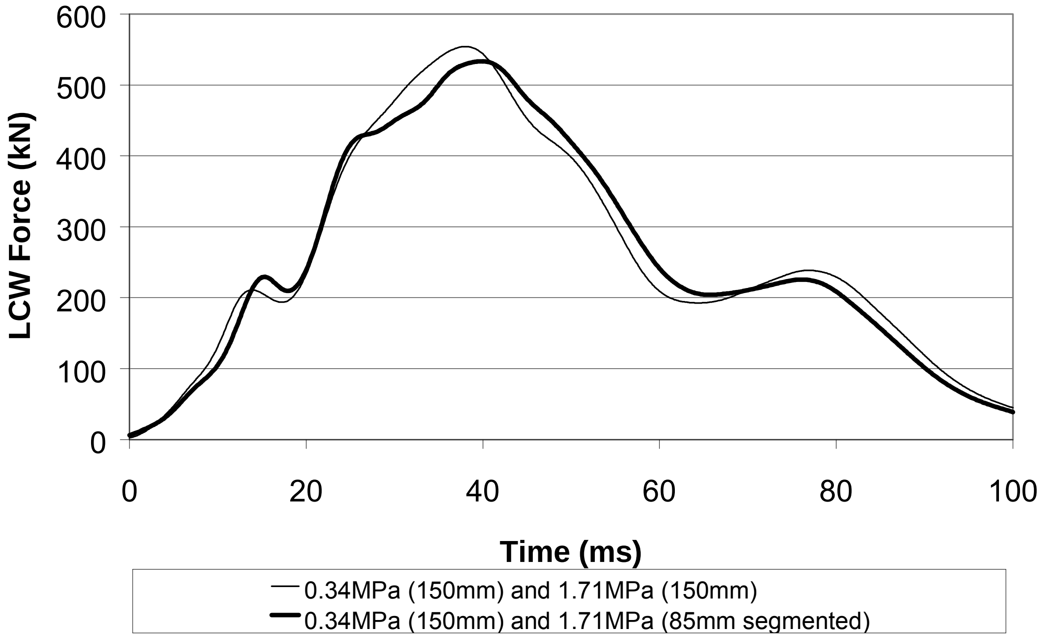


Figure 47: Load cell wall total force against time showing the similarity between the segmented and non-segmented barrier face.

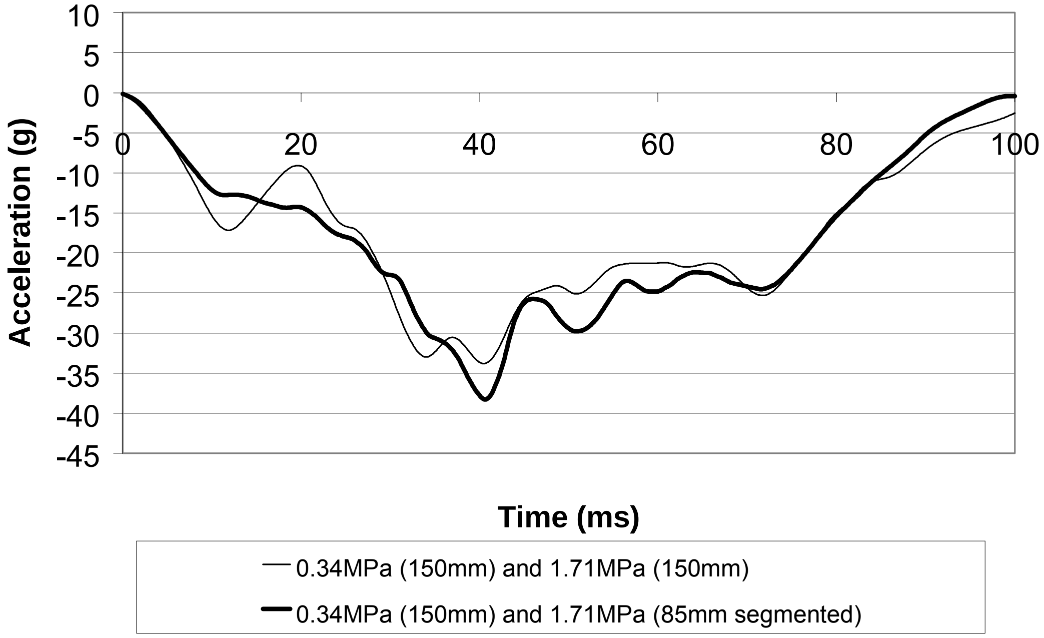


Figure 48: Occupant compartment acceleration showing the similarity between the segmented and non-segmented barrier face.

However, the reduction in shear stiffness of the rear layer of the barrier resulted in a visible alteration of the force distribution (Figure 49). For the test with the segmented barrier, the highest loads are those located in front of the lower rails and the subframe crossbeam. The timing of these loads can be assessed by looking at the force time history for each of the individual load cells (Figure 50).

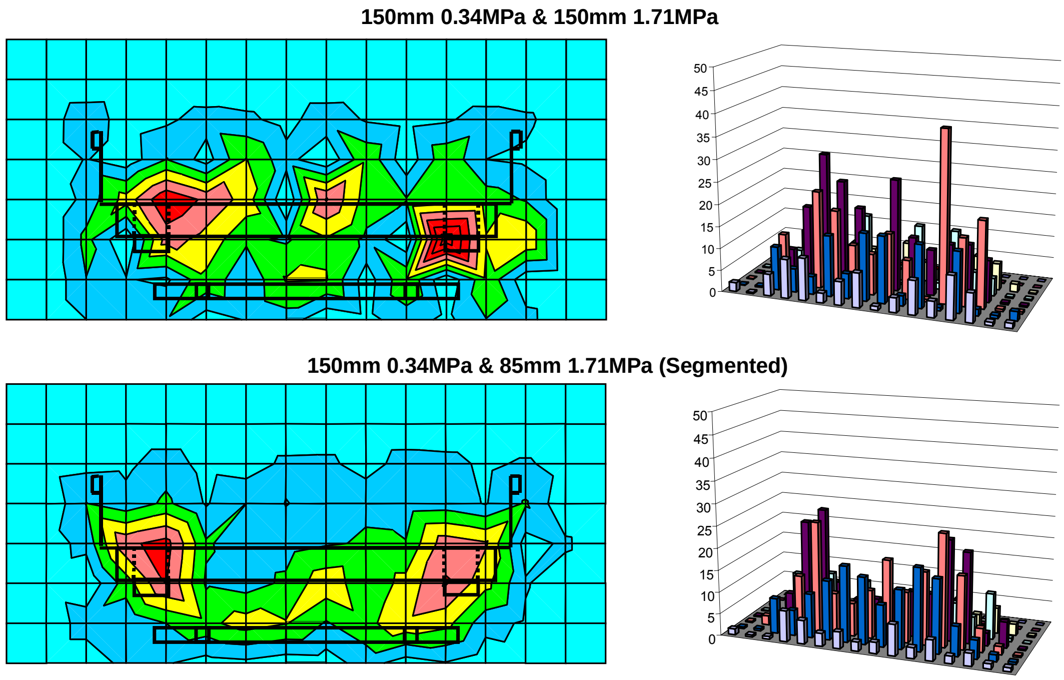


Figure 49: Comparison of the force distributions based on peak cell loads for the segmented and non-segmented barriers. The segmenting of the barrier provided a more realistic load distribution over the centre section of the load cell wall.

Figure 50: Load cell wall force time history showing the high loads applied by the lower rails and subframe crossbeam.

The barrier deformation in this test was consistent with the force distribution recorded by the load cell wall. This indicates that the segmenting of the stiffer layer into individual blocks to reduce its shear stiffness removed the problem of load cell bridging. The segmenting of the stiffer layer should also remove any effect of load spreading caused by the shear stiffness of the barrier, which would have limited the possible depth of this layer.

Due to the limited 85mm depth of the stiffer layer, the barrier bottomed out in front of both lower rails (Figure 51). The bending of the leading edge of the radiator mountings (Figure 52), along with localised deformation of the barrier indicated that they also contacted the load cell wall. However, in the case of the radiator mountings, the applied load was still less than that required by the adjacent structure to deform the surrounding honeycomb.

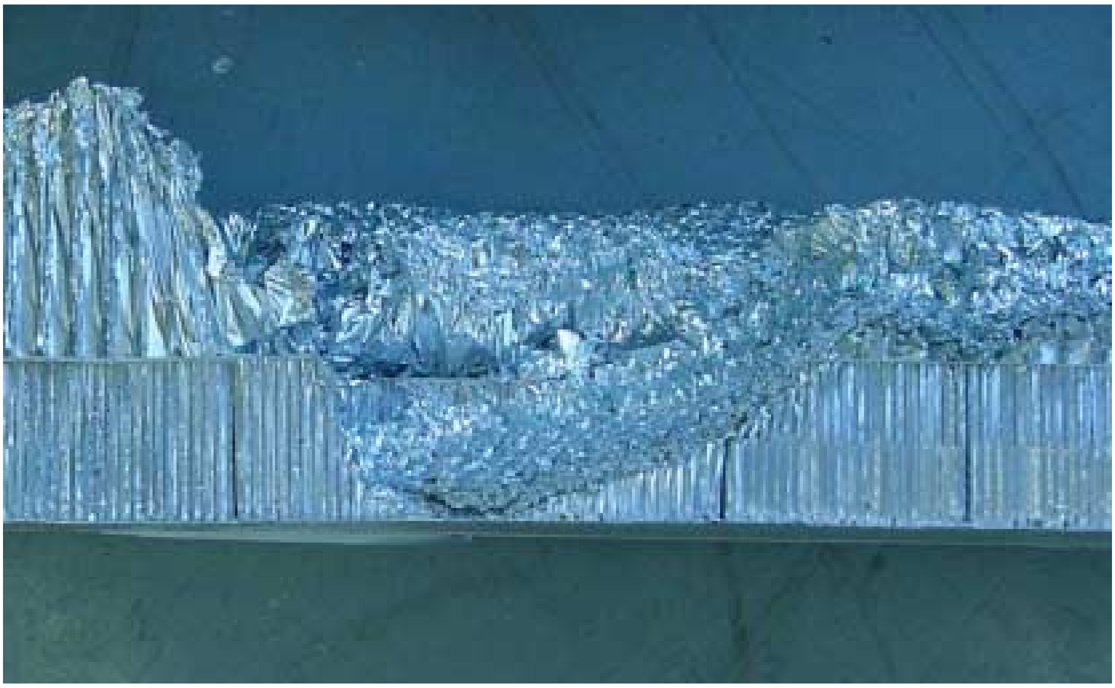


Figure 51: Deformation of barrier showing the bottoming out at the location of the left hand side lower rail impact.

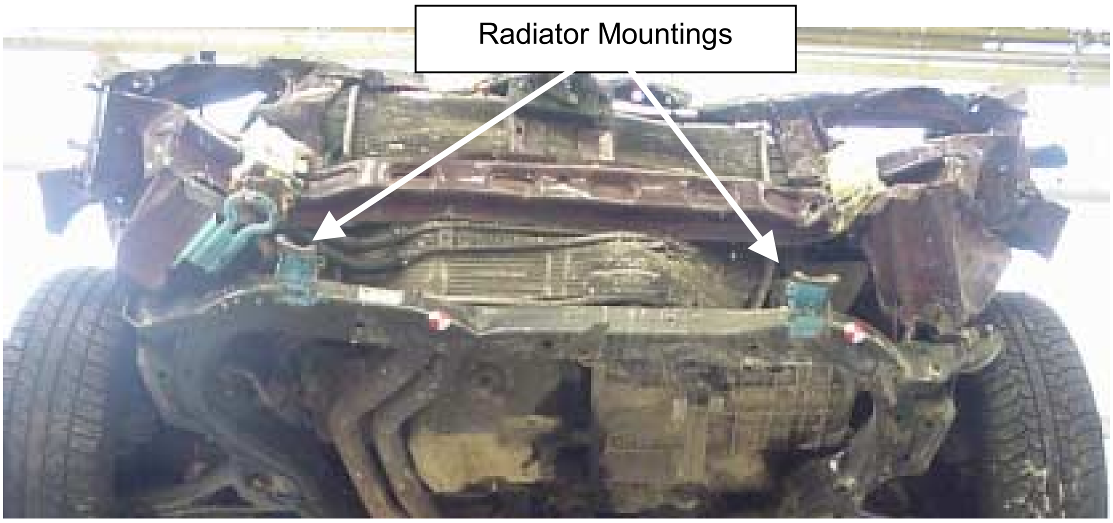


Figure 52: Vehicle deformation showing the resultant deformation of the radiator mountings from contact with the load cell wall.

Conclusions

- The segmenting of the rear stiffer layer of the honeycomb face effectively reduced its shear strength and prevented the problem of the load cell bridging to give a measured load cell wall force distribution that was more representative of the car’s stiffness homogeneity than the previous test result.

- The vehicle bottomed out the barrier. However, this did not significantly alter the total load cell wall force or the occupant compartment deceleration compared with the previous test which used a non-segmented barrier with a deeper rear layer.

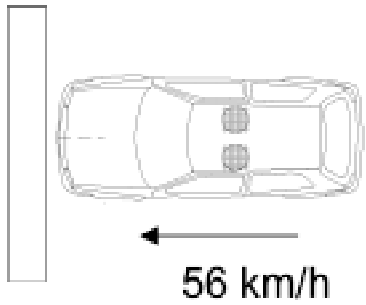
- The barrier face achieved the aim of attenuating the engine loading.

- The barrier face achieved the aim of maintaining a similar occupant compartment deceleration to that measured in a full width rigid wall impact.

To prevent bottoming out of the barrier, it is recommended that the depth of the second layer should be 150mm, the same as used for the non-segmented barrier.

Renault Laguna II against development full width barrier

Date	7/6/2002
Location	TRL
Topic Group	Development FWT
Mass Ratio	-
Test Number	07NF



Vehicle 1:	Renault Laguna II	Barrier:	Development Full Width Impact Barrier
Impact Side:	Front		
Test Speed:	56km/h		
Overlap:	100%		
Test Mass:	1635.5kg		
Dummies:	RHS – HybridIII* LHS – HybridIII *not instrumented		

Test Objective

The Renault Laguna II is the highest scoring EuroNCAP car ever tested to date. Visual inspection shows that it possesses desirable features for improved compatibility, such as strong vertical shear connections. The aim of this test is to assess the performance of this benchmark car, in particular its stiffness homogeneity. This assessment can be used for comparison with its performance in car to car tests to be conducted at a later date.

Test Details

The left hand drive Renault Laguna II was impacted into an aluminium honeycomb barrier of size 2m x 0.75m x 0.3m. The front layer had a depth of 150mm and a crush strength of 0.34MPa. The rear layer had a depth of 150mm and a crush strength of 1.71MPa. The stiffer rear layer of the barrier was segmented into individual blocks. The size of these blocks was 125mm by 125mm, which represented the frontal area of each load cell. The segmenting resulted in 96 individual blocks that lined up with the load cells on the wall. In addition, the front cladding of the barrier was omitted in order to minimise any barrier deformation caused by bending.

The barrier was mounted onto a high resolution load cell wall (Figure 54). The load cell wall was formed of 128 load cells of 125mm x 125mm arranged in a 16 x 8 matrix. The deformable barrier covered the lower 6 rows of the load cell wall.

Figure 53: Full width test barrier mounted onto load cell wall.

The Renault Laguna II was instrumented with accelerometers at various locations around the car, predominantly to allow the interface force to be calculated. One instrumented HybridIII dummy was placed in the driver's seat and one uninstrumented HybridIII dummy was placed in the passenger's seat. They were positioned according to the EuroNCAP Testing Protocol Version 3 dated April 2001. The driver and passenger dummies were restrained by three point seat belts.

Results and Discussion

The test speed and overlap were within the specified tolerances. Pre and post test images of the test vehicle are shown (Figure 44 and Figure 45), along with the resultant barrier deformation (Figure 46).

Figure 54: Pre and post test driver's side view of test vehicle.

Figure 55: Pre and post test front view of test vehicle.

Figure 56: View of the resultant barrier deformation.

The total load cell wall force against B-pillar displacement indicates a peak force of approximately 543 kN at about 590 mm (Figure 57).

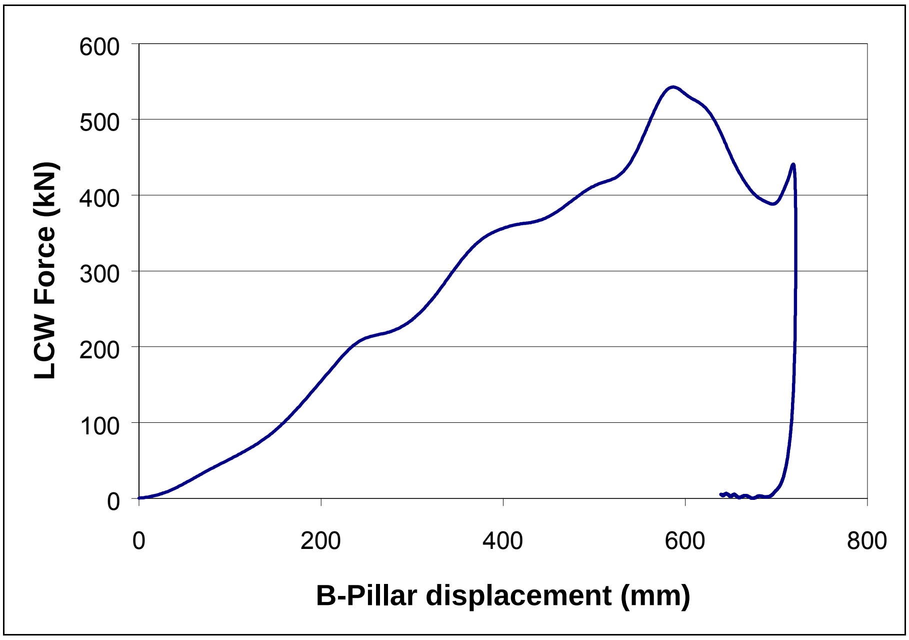


Figure 57: LCW Force against B-Pillar displacement.

The force distribution based on the peak loads recorded by each load cell is shown (Figure 492). The timing of these peak forces can be assessed by viewing the force time history of each load cell (Figure 59).

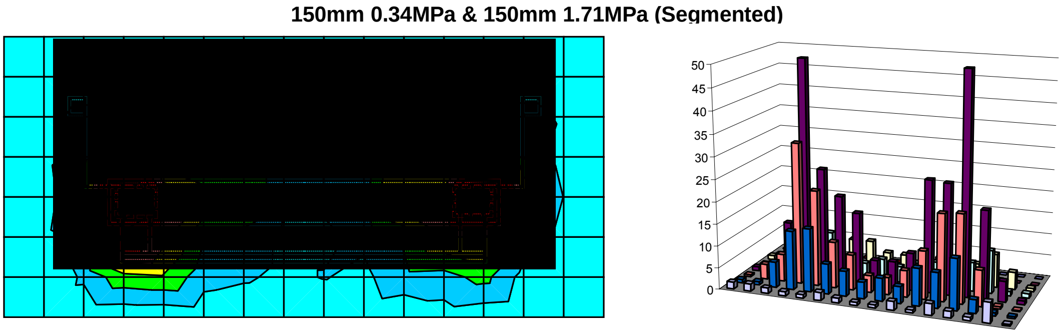


Figure 58: The force distribution based on peak cell loads for the Renault Laguna II, showing high load concentrations in front of the lower rails.

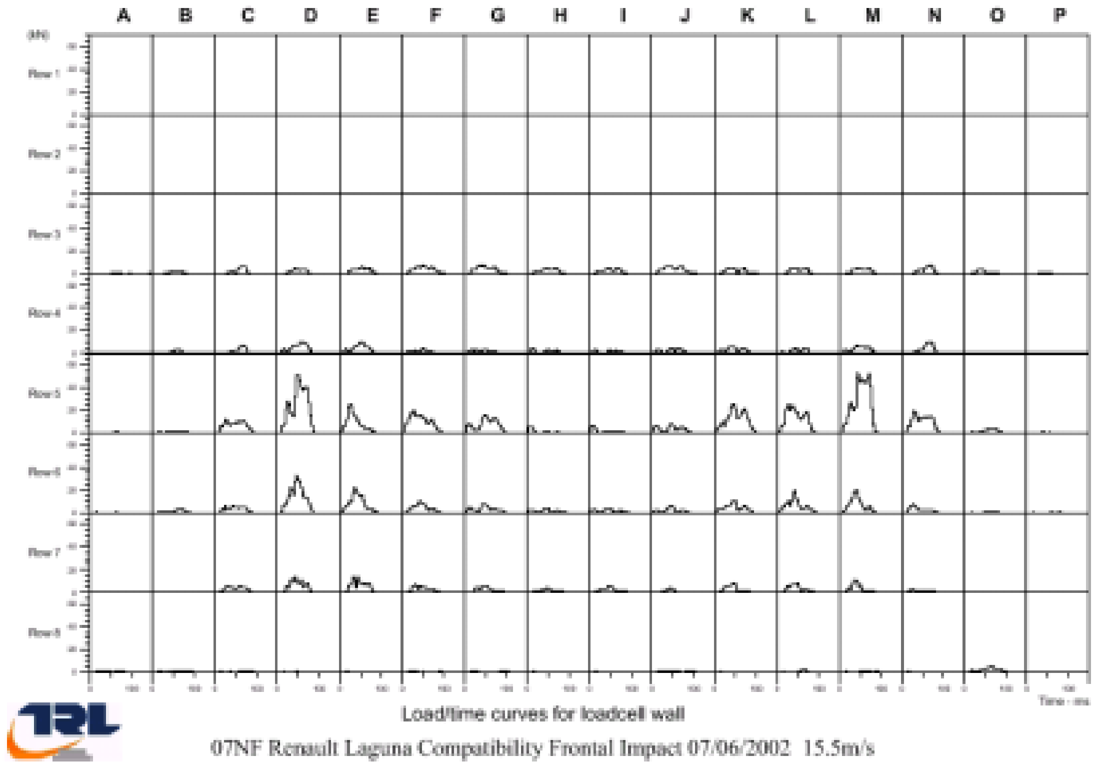


Figure 59: Load cell wall force time history showing the high loads applied by the lower rails.

The car applied high loads (~50kN) to two cells on the LCW (Row 5, columns D and M), which relate to the pre-impact location of the lower rails (Figure 59). The reason for these high loads was the bottoming out of the barrier face in front of the lower rails (Figure 60).

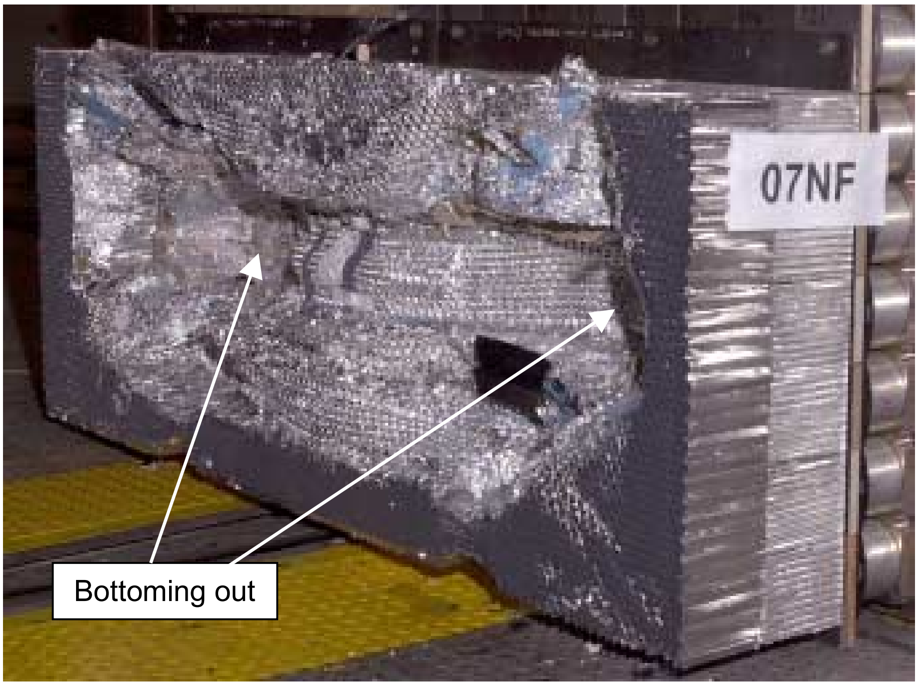


Figure 60: View of barrier deformation, showing bottoming out of barrier in front of lower rails and less deformation in front of the bumper beam.

A possible effect of the barrier bottoming out is that the lower rails formed a preferential load path, which may have prevented other load paths such as the subframe applying any significant load. The likelihood of this effect is increased when it is noted that the subframe was set back approximately 135mm from the front of the lower rails. However, the subframe still managed to apply a peak load of approximately 14kN to the cells in row 7 columns D, E, L and M, which corresponds approximately to where it was supported.

There was little loading of the middle of the wall (Figure 59, columns G, H, I, J). This is consistent with the failure of the subframe crossbeam and the bumper beam (Figure 61 and Figure 62).

Figure 61: Comparison of bumper beam, pre and post-test, showing failure between lower rails.

Figure 62: Centre of bumper beam and subframe crossbeam, showing bending failures.

The LCW force distribution for the maximum loads measured on each cell was not ‘homogeneous’ in a subjective assessment. This was because of the high loads in front of the lower rails, caused by the bottoming out of the barrier face, and the low loads in the middle of the load cell wall, caused by the failure of the bumper and subframe crossbeams.

Conclusions

- The lower rails of the Renault Laguna II bottomed out the barrier face and applied load directly to the load cell wall. This resulted in high peak forces of approximately 50kN being applied to 2 load cells.

- The subframe applied a peak load of 14kN over each load cell at its support locations. The centre section applied low load due to its failure in bending.

- The bumper beam C-section between the lower rails failed during the impact. This type of failure was also observed in the EuroNCAP test.

- Overall, the Renault Laguna II did not exhibit good stiffness homogeneity. This was due to the lower rails bottoming out the barrier and applying large loads directly on the load cell wall and the low loading applied by the centre of the bumper and subframe crossbeams due to their failure. The bottoming out of the lower rails formed preferential load paths, which may have reduced the load applied by other structures. The stability of the lower rails was most likely helped by the good vertical connections.

6.2.2 UTAC tests

Test Details

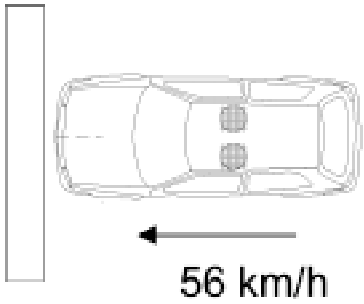
The cars were impacted into an aluminium honeycomb barrier of size 2m x 0.75m x 0.30m and a crush strength of 0.34MPa for the first 150mm and 1.71MPa thereafter, mounted onto a high resolution load cell wall (Figure 63). The stiffer rear layer of the barrier was segmented into individual blocks. The size of these blocks was 125mm by 125mm, which represented the frontal area of each load cell. The segmenting resulted in 96 individual blocks that lined up with the load cells on the wall.

Figure 63: Full Width Test Barrier and Load Cell Wall

The load cell wall was formed of 128 load cells of 125mm x 125mm arranged in a 16 x 8 matrix and mounted 165mm from the ground (TRL tests 50mm). The deformable barrier covered the lower 6 rows of the load cell wall.

The cars were instrumented with accelerometers at various locations around the car, predominantly to allow the interface force to be calculated. Instrumented HybridIII dummies were placed in the driver's seat and in the passenger's seat. They were positioned according to the EuroNCAP Testing Protocol Version 3. The driver and passenger dummies were restrained by three point seat belts.

Opel Astra against development full width barrier

Date	12/03/2002				
Location	UTAC				
Topic Group	FWT				
Mass Ratio	-				
Test Number	01/07497				
		Vehicle 1:	Opel Astra 98	Barrier:	Development Full Width Impact Barrier
		Impact Side:	Front		
		Test Speed:	56km/h		
		Overlap:	100%		
		Test Mass:	1364 kg		
		Dummies:	RHS – HybridIII LHS – HybridIII		

Test Aim

To provide test data for the development of the assessment protocol and for future comparison of the proposed test procedures.

Results and Discussion

The test speed and overlap were within the specified tolerances. Pre and post-test images of the test vehicle (Figure 64 and Figure 65), and the resultant barrier deformation (Figure 66) are shown.

Figure 64: Pre and post test driver's side view

During the test, the bottom of the barrier became detached from the wall (Figure 64). Analysis of the video showed that it occurred during the back up of the car after the crash so there was no influence on the results.

Figure 65: Post test front view

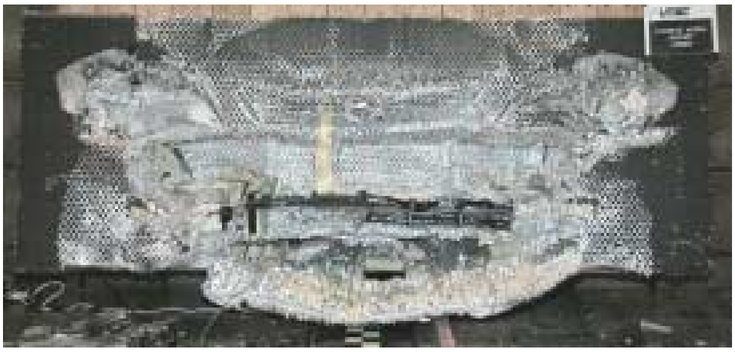


Figure 66: Resultant barrier deformation

The load cell wall force against B-Pillar displacement, indicates a peak force of approximately 447 kN at about 430 mm (Figure 67).

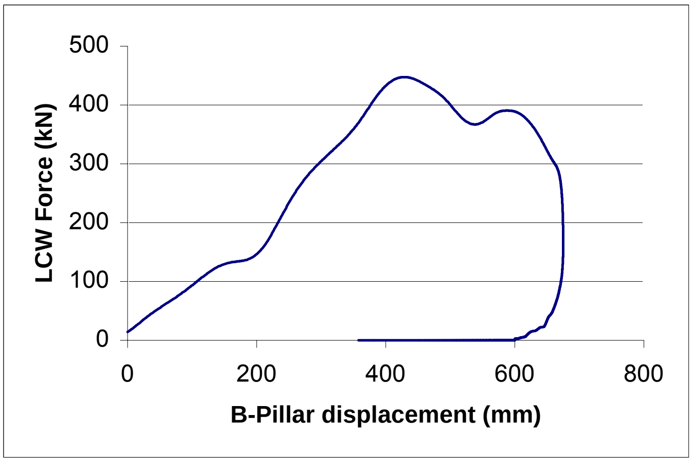


Figure 67: LCW force against B-Pillar displacement

The force distribution, based on the peak loads recorded by each load cell is shown (Figure 68).

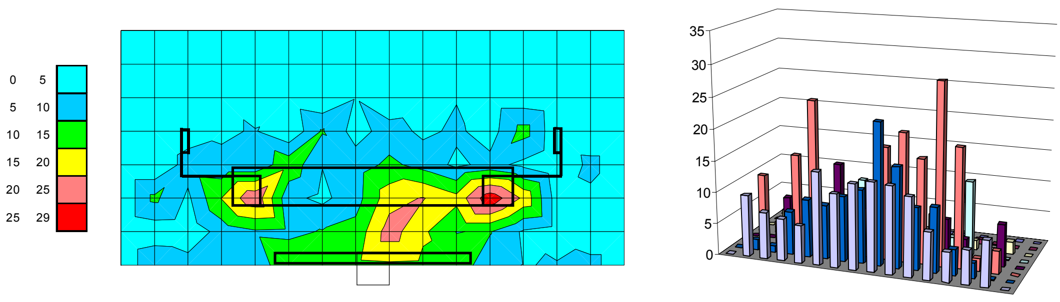


Figure 68: Force distribution using peak load cell forces

The highest loads were located in front of the lower rails and the subframe crossbeam. It was noticed also that the engine and alternator on the RHS pushed on the upper crossbeam, which generated a high load concentration.

The Load Cell Wall force distribution and the deformation of the barrier matched with the car stiffness homogeneity.

When comparing the deformation obtained in the car to car test, it was noticed that on the full width test the connection between the lower rails and the subframe was not destroyed. This indicates that the shear force generated in this test was not as great as that generated in the car to car test

Conclusions

- From the load cell wall force distribution it was possible to identify the multiple load paths of the Opel Astra.

- A load path between the engine and crossbeam was created, most likely by the alternator, which produced a significant load on the load cell wall.

- The engine subframe to lower rail shear connection was not broken in this test unlike the car to car test. This was due to the fact that this test did not generate as much shear force across this connection as experienced in the car to car test.

Modified Opel Astra against development full width barrier

<table>
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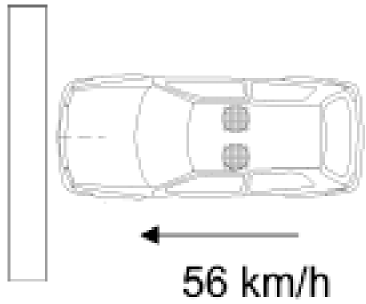
Location: UTAC
Topic Group: FWT
Mass Ratio: -
Test Number: 01/07559</td>
<td colspan="2">
</td>
</tr>
<tr>
<td>Vehicle 1: Opel Astra 98
Impact Side: Front
Test Speed: 56km/h
Overlap: 100%
Test Mass: 1364 kg
Dummies: RHS – HybridIII
LHS – HybridIII</td>
<td>Barrier: Development Full Width Impact Barrier</td>
</tr>
</table>

Test Aim

To provide test data for the development of the assessment protocol and for future comparison of the proposed test procedures.

Modification of the Opel Astra

The Opel Astra has multiple load paths, which can be considered as a compatible feature. To maximise this feature, modifications were made to reinforce the cross member and the connections between load paths, and to increase the pushing surface. The purpose is to check if the test method is able to differentiate such a modification.

Figure 69: Structure of the modification

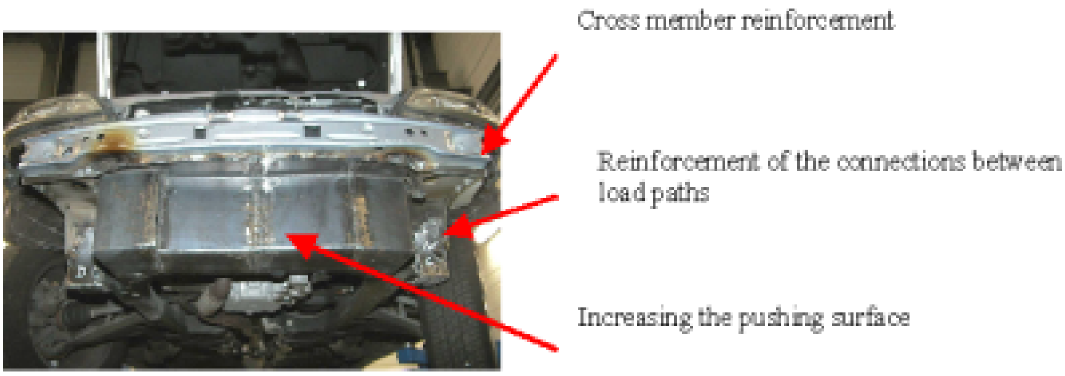


Figure 70: Modifications of the Opel Astra

Results and Discussion

The test speed and overlap were within the specified tolerances. Pre and post-test images of the test vehicle (Figure 71 and Figure 72), and the resultant barrier deformation (Figure 73) are shown.

Figure 71: Pre and post test driver's side view

Figure 72: Post test front view

Figure 73: Resultant barrier deformation

The load cell wall force against B-Pillar displacement, indicates a peak force of approximately 474 kN at about 430 mm (Figure 74).

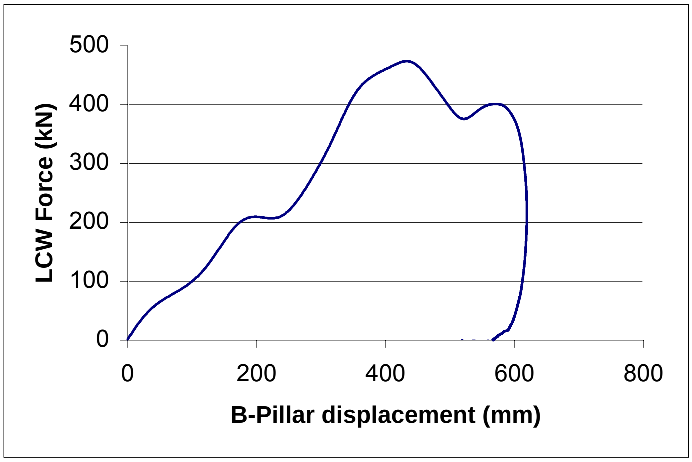


Figure 74: LCW force against B-Pillar displacement

The force distribution, based on the peak loads recorded by each load cell is shown (Figure 75).

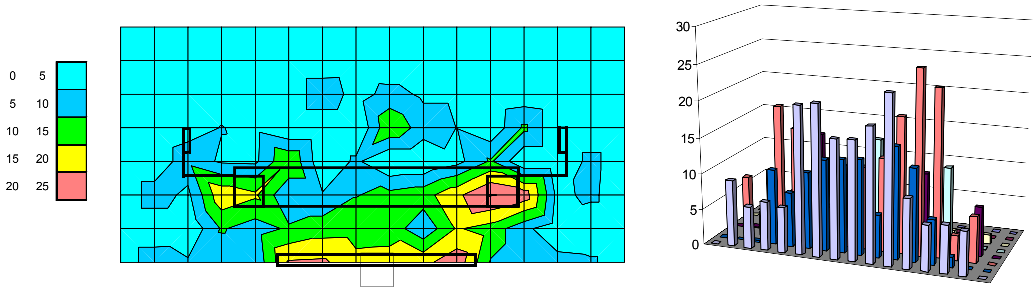


Figure 75: Force distribution using peak load cell forces

The highest loads were located in front of the lower rails, particularly on the RHS and on the subframe crossbeam.

The Load Cell Wall force distribution and the deformation of the barrier matched with the car stiffness homogeneity.

When comparing the deformation obtained in the car to car test, it was noticed that on the full width test the connection between the lower rails and subframe was not exercised as much because the shear force generated is not so great as on a car to car test.

Comparison Opel Astra / Modified Opel Astra tests results

The purpose of the modification was to reinforce the connections between load paths and to increase the pushing surface. The aim was to demonstrate that the full width test was able to indicate a difference from the load distribution recorded by the Load Cell Wall.

The comparison between the total load cell wall force against time history (Figure 76), shows that the maximum total load cell force was different for the two cars: 447 kN for the Opel Astra and 474 kN for the modified Opel Astra.

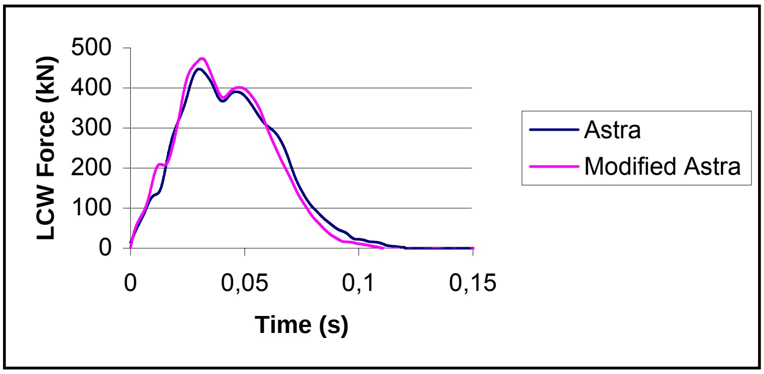
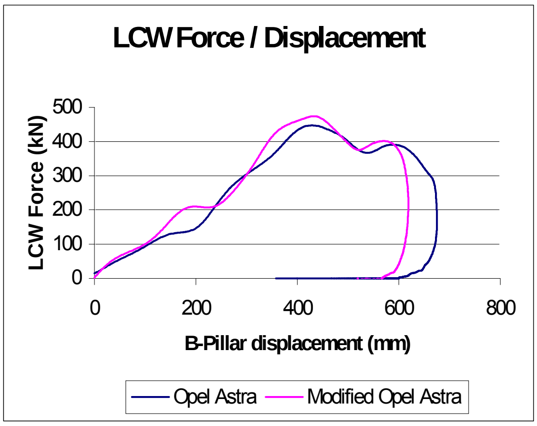


Figure 76: LCW Force / Time history

With the load cell wall force against B-Pillar displacement graph (Figure 77), the total energy absorbed by each car can be calculated. It was noticed that the absorbed energy was lower for the modified car (173 kJ) than for the standard one (181 kJ).

Although a slight increase in the vehicle stiffness was observed, the change in the energy absorption was mainly due to the change in the B-Pillar displacement. One probable cause for this would have been the increased interaction area due to the increased pushing surface. This increase would have resulted in less penetration into the barrier and better energy absorption by the barrier.



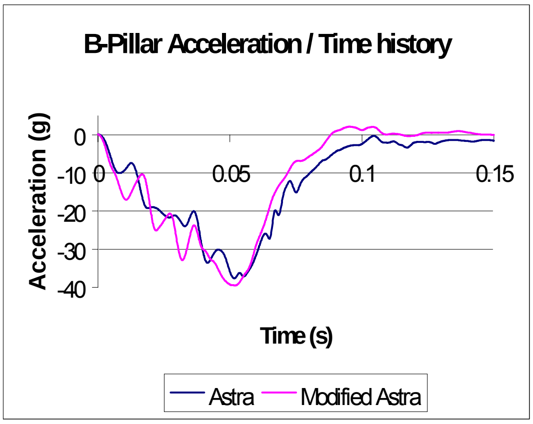
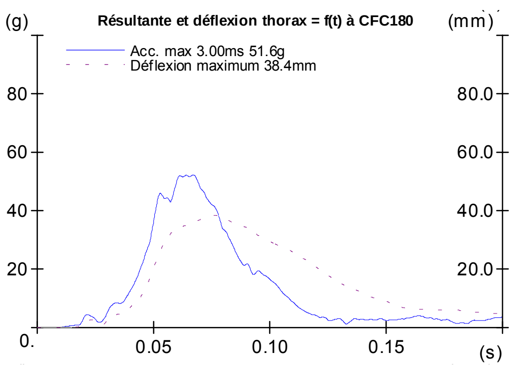


Figure 77: LCW force against displacement and B-Pillar acceleration against time

The compartment deceleration pulse (Figure 77), was also more severe for the modified Opel Astra. The modification seems to have increased the global stiffness of the frontal unit, this may have been due to the reinforcement of the connections between load paths resulting in slightly different failure modes of the main structure. As the deceleration pulse is more severe for the modified Opel and the energy absorbed is less, the dummy criteria are more important. The chest deflection for the driver dummy was 38.4 mm for the Astra and 45.5 mm for the modified Astra (Figure 78).

Opel Astra



Modified Opel Astra

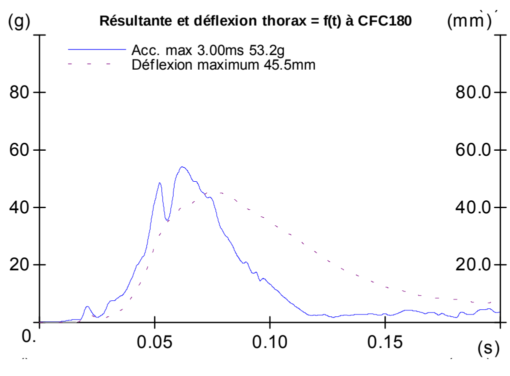


Figure 78: Dummy criteria

The force distribution, based on the peak loads recorded by each load cell is shown (Figure 79).

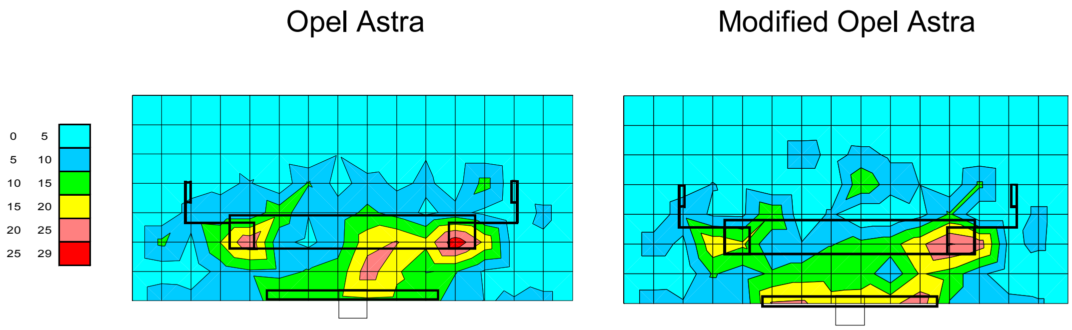


Figure 79: Comparison Opel Astra / Modified Opel Astra

The force cartography of the two cars shows that the load level and distribution is different. On the Opel Astra the highest loads were located in front of the lower rails and the subframe crossbeam, the engine is also pushing on the crossbeam. For the modified Opel Astra, the highest loads were located in front of the lower rails. The loads caused by the engine are much more distributed, probably because of the plate that increased the pushing surface. Furthermore the loads due to the engine subframe are very high, but when we analyse the frontal unit of the two cars (Figure 80), we noticed that the subframe is less deformed. By the modification we reinforced the load paths and made it stiffer causing a high load concentration on this area.

On these two tests, the connections between the lower rails and the subframe were not destroyed. The shear connection in the unmodified car was not used as on a car to car test. This indicates that this test did not generate the same shear forces as seen in the car to car test.

Figure 80: Subframe deformation

Conclusions

- The test method with the full width barrier was able to show differences between two cars. The modifications to the Opel Astra appear to have changed the load path behaviour leading to additional load generated through the subframe.

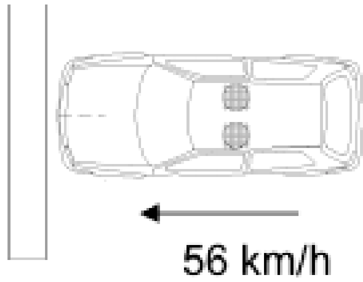
- The Load Cell Wall force distribution appeared more homogenous for the modified car based on a subjective assessment of the load cell wall force distribution.

- From interpretation of the load cell wall total force / displacement plots, the modifications to the Opel Astra appear to have increased the global stiffness of the frontal structure.

- At this stage of the test method development, differences in the performance of the two vehicles are based on subjective analysis of the load cell wall force distribution, criteria need to be developed to evaluate and quantify the difference between the Opel Astra and the Modified Opel Astra.

Rover 75 against development full width barrier

Date	19/03/2002
Location	UTAC
Topic Group	FWT
Mass Ratio	-
Test Number	01/07560



Vehicle 1:	Rover 75 (1998)	Barrier:	Development Full Width Impact Barrier
Impact Side:	Front		
Test Speed:	56km/h		
Overlap:	100%		
Test Mass:	1596 kg		
Dummies:	RHS – HybridIII LHS – HybridIII		

Test Aim

To investigate the performance of a ‘stiff’ vehicle with multiple load paths and provide test data for the development of the assessment protocol.

Results and Discussion

The test speed and overlap were within the specified tolerances. Pre and post test images of the test vehicle (Figure 81 and Figure 82), and the resultant barrier deformation (Figure 83) are shown.

Figure 81: Pre and post test driver’s side view

Figure 82: Post test front view

Figure 83: Resultant barrier deformation

The load cell wall force against B-Pillar displacement, indicated a peak force of approximately 569 kN at about 545 mm (Figure 84).

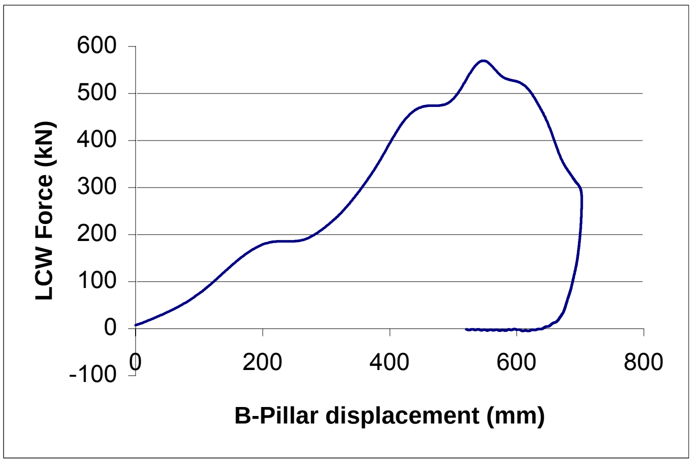


Figure 84: LCW force against B-Pillar displacement

The force distribution, based on the peak loads recorded by each load cell is shown (Figure 85).

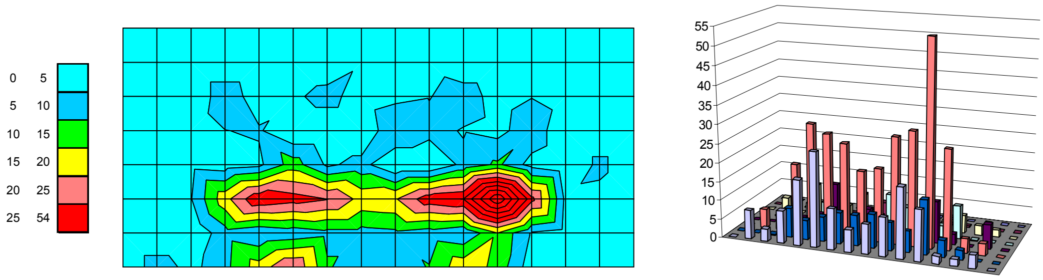


Figure 85: Force distribution using peak load cell forces

The highest loads are located in front of the lower rails and the cross member. The load level was very high, approximately 54kN, and focused on one point which corresponds to the RHS lower rail position. At this location the barrier bottomed out and the backplate of the barrier was torn up. The difference in the force applied by the LHS and RHS lower rails was probably due to the different failure modes experienced. The support provided in the lateral plane for the RHS lower rail by the location of the alternator, positioned near the lower rail and between the crossbeam and the engine, prevented any significant failure in bending of that rail. In comparison, the LHS lower rail lacked this lateral support and suffered a significant failure in bending (Figure 90). The different failure modes meant a significant difference in the applied force, with the result that the RHS lower rail bottomed out the barrier and applied load directly to the LCW. The difference in stiffness of the two lower rails, due to the distinct failure modes can be seen through analysis of the force contour plot over time. During the impact, the force in front of both lower rails is about 30kN, which is approximately the crush force of the rear layer of the barrier per load cell. Towards the end of the impact, the force suddenly increases over one load cell to a peak of 54kN. This sudden increase is representative of the lower rail on that side of the car penetrating the barrier late on in the impact and applying load directly to the load cell wall.

Figure 86: Bottom view

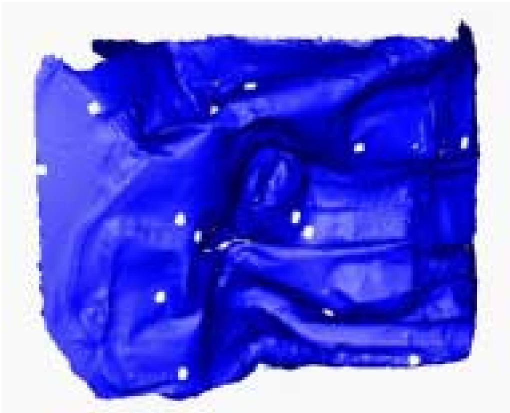
Regarding the force distribution graph, it was noticed that there was a difference between the force measured for the lower rail and cross member, in comparison to the subframe. The first conclusion may be that the stiffness of these two parts seems to be relatively different.

Excluding the RHS lower rails, that bottomed out the barrier and resulted in a force that was in excess of the barrier resistance, this difference in the force applied by the lower rails and the subframe was approximately 5-10kN. This difference may be accounted for by the relative frontal area of the two components, the subframe being smaller, and therefore generating less resistance over the frontal area of one load cell. Evaluation criteria are under development for assessing differences such as this.

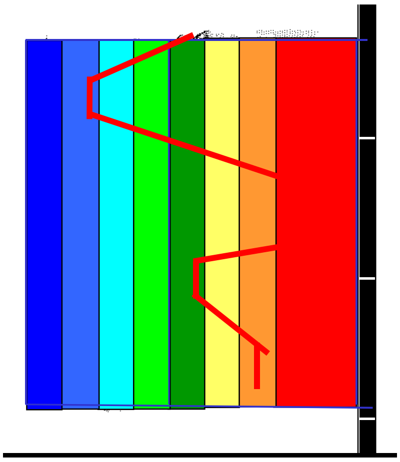
Tests conducted on the Rover 75 against a PDB have demonstrated that the engine subframe is an important load path of the car and contributed to the non-overriding car to car test. The deformation of the barrier due to the subframe is nearly the same as the deformation observed for the lower rails (Figure 87).

The engine subframe is positioned behind the front of the lower rails and the cross member (according to the longitudinal axis of the car), so to deform the subframe and to check the loads on the load cell wall, the car needs to be more deformed than the 700 mm we have on the full width test.

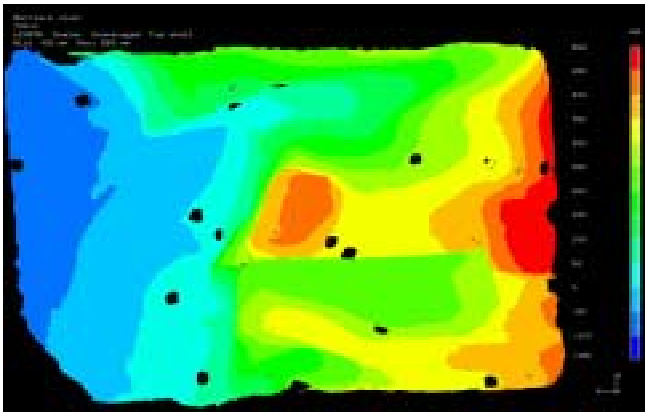
Barrier deformation



Digitisation



Side view



Deformation map

Figure 87: PDB resultant deformation

Conclusions

- The Rover 75 RHS lower rail bottomed out the barrier to form a preferential load path, which may have reduced the load, applied by the subframe to the wall. This resulted in a high load of 54 kN on one load cell, which reduced the homogeneity of the loads on the wall.

- Ideally, a test method to evaluate compatibility needs to be able to deform a car as much as it is deformed in accidents so that all the possible load paths and the shear connections between these load paths are exercised. This test has shown that the frontal unit deformation achieved in the full width test was perhaps not sufficient to check all these load paths, especially if they are positioned some distance behind other paths, for example a subframe positioned behind the front of the lower rails.

6.2.3 Summary of Conclusions and Recommendations

Conclusions

- Two tests using a Mondeo car were performed to help in the redesign of the barrier face in order to overcome the problem of small stiff protruding structures forming preferential load paths. The second test demonstrated that the redesigned face overcame this problem, whilst still achieving the aims of the initial barrier face which were:
  - To prevent unrealistic decelerations at the front of the car.
  - To attenuate the engine inertial loading
  - To have a similar compartment deceleration to an equivalent rigid wall test.

- The multiple loads of the Opel Astra and modified Astra could be identified from the homogeneity of the load cell wall (LCW) force distribution recorded in the full width tests. A difference was distinguished between the Astra and modified Astra, the modified Astra showing better homogeneity for the LCW force distribution, which is consistent with the better structural interaction seen in the modified car to car crash test. However, the engine subframe to lower rail shear connection was not loaded as much in either of these tests compared to the car to car tests. This indicates that the full width test may not generate as much shear force across this type of connection as in a car to car impact.

- The LCW results from the Renault Laguna II test showed that the Laguna II did not exhibit good stiffness homogeneity. This was due to the lower rails bottoming out the barrier and applying large loads directly on the load cell wall and the low loading applied by the centre of the bumper and subframe crossbeams due to their failure. The bottoming out of the lower rails formed preferential load paths, which most likely reduced the load applied by other structures, such as the subframe. The stability of the lower rails was most likely helped by the good vertical connections. The formation of a preferential load path was also seen in the Rover 75 test, in which one lower rail bottomed out the barrier.

- Ideally, a test method to evaluate compatibility needs to be able to deform a car as much as it is deformed in accidents so that all the possible load paths and the shear connections between these load paths are exercised. The tests performed in this project have shown that the frontal unit deformation achieved may not be sufficient to adequately check all these load paths and the shear connections, especially if they are positioned some distance behind other paths, for example, a subframe positioned more than about 150 mm behind the front of the lower rails.

Recommendations

- At this stage of the test method development, differences in the performance of the vehicles are based on subjective analysis of the load cell wall force distribution. Criteria should be developed to evaluate and quantify the changes observed between different vehicles. This will require additional crash test data to be generated from a larger range of vehicle designs.

- In the Laguna and Rover 75 tests preferential load paths were formed because the lower rails bottomed out the barrier. This most likely reduced the load carried by other structures set further back in the car, such as the subframe, resulting in the reduction of the homogeneity of the load recorded on the wall. A study should be performed to address the following questions:

- Is the probable reduction in homogeneity representative of the car's structural interaction performance in car to car collisions?

- Approximately, how far back can a secondary load path be positioned from the front of the main load path and still be able to contribute significantly to improving a car's compatibility?

6.3 PROGRESSIVE DEFORMABLE BARRIER (PDB) TEST

The offset progressive deformable barrier test is one of the proposals for assessing compatibility, the objective being to quantify the capacity of the frontal unit to absorb energy. To achieve this, a calculated speed of 60km/h is sufficient to take account of the absorption capacities of the barrier as well as the vehicle stiffness. This results in an equivalent energy speed (EES) of 50km/h. The overlap width of 750mm, means that the barrier will always generate the same load.

Three criteria were originally proposed, the first of these being the deformation of the barrier as it is clear that a large local force would provide increased local deformation. The second was the maximum load measured behind the barrier. The third was the height of the resultant force assessed from the force distribution recorded by a high resolution load cell wall located behind the barrier. These three criteria have been used in the assessment of the tests performed for this report.

A new assessment method being put forward for this test requires a more detailed deformation measurement process than used for the analysis performed in this report. This process involves a wax skin being applied to even out any small local deformation and tearing of the barrier. From this, the deformation of the entire barrier surface can then be measured. This deformation measurement process has been used on all the PDB tests conducted as part of this report. The assessment of this data will be reported on at a later date when the assessment criteria have been clearly defined.

The PDB barrier used in the first two tests performed by TRL was version 6. This barrier has two sections separated by an aluminium sheet, with additional aluminium sheets on the front and rear faces. A diagram indicating the crush strengths of the barrier is included within this report (Figure 88). The PDB barrier used in the final three tests performed at BASt used version 7 of the PDB barrier. Version 7 differed from version 6 in that an additional aluminium sheet has been included covering both the upper, front and lower surface of the barrier. The attachment of this additional sheet was to the existing front face of the version 6 barrier.

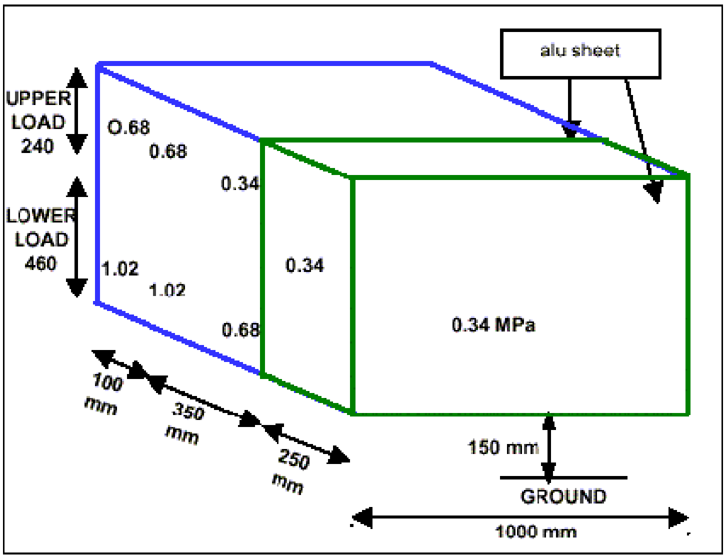


Figure 88: Progressive Deformable Barrier crush strengths

6.3.1 TRL tests

Ford Mondeo against progressive deformable barrier

<table>
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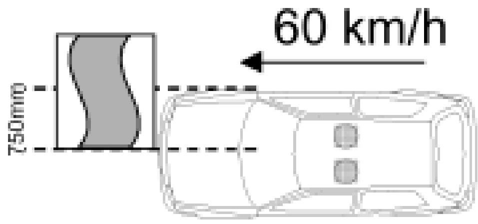
Location TRL

Topic Group PDB Impacts

Mass Ratio -

Test Number 10MF</td><td colspan="2">
</td></tr>
<tr><td>Vehicle 1: Ford Mondeo 93
Impact Side: Front
Test Speed: 60km/h
Overlap: 750mm
Test Mass: 1455kg
Dummies: RHS – HybridIII
LHS – HybridIII*
*not instrumented</td><td>Barrier: PDB, version 6</td></tr>
</table>

Test Objective

The aim of the test was to investigate the performance of the proposed Progressive Deformable Barrier (PDB) test in assessing the compatibility of the Ford Mondeo. This car is a medium sized vehicle with two levels of load path, namely the lower rails and the engine subframe. The Ford Mondeo was also used as the vehicle in the development of the full width barrier test. Therefore, at a later date when the assessment protocols have been developed, the test data will be available to make a comparison between the two methods for this car

Test Details

The PDB was mounted on a high resolution load cell wall formed of 64 cells of 125mm by 125mm, arranged in an 8 x 8 matrix (Figure 90). The position of the load cell wall was such that the lower four rows were below the split line of the upper load and lower load areas of the barrier, whilst the remaining four rows were above the split line. Therefore, the barrier did not cover the two highest rows of the load cell wall, whilst the lowest and highest rows that were in contact with the barrier had only 64% and 92% coverage respectively.

Figure 89: Progressive deformable barrier version 6 mounted on load cell wall.

The Ford Mondeo was instrumented with accelerometers at various locations around the car, predominantly to allow the interface force to be calculated. One instrumented HybridIII dummy was placed in the driver's seat and one uninstrumented HybridIII dummy was placed in the passenger's seat. They were positioned according to the EuroNCAP Testing Protocol Version 3 dated April 2001. The driver and passenger dummies were restrained by three point seat belts.

Results and Discussion

The test speed and overlap were within the specified tolerances. Pre and post test images of the test vehicle are shown (Figure 90). During the impact the PDB became attached to the vehicle structure and remained attached post impact.

Figure 90: Vehicle pre and post test views showing the attachment of the barrier to the vehicle post impact.

The removal of the PDB from the vehicle structure required the use of tools. Furthermore, part of the barrier became detached from the main body and remained attached to the vehicle structure (Figure 91)

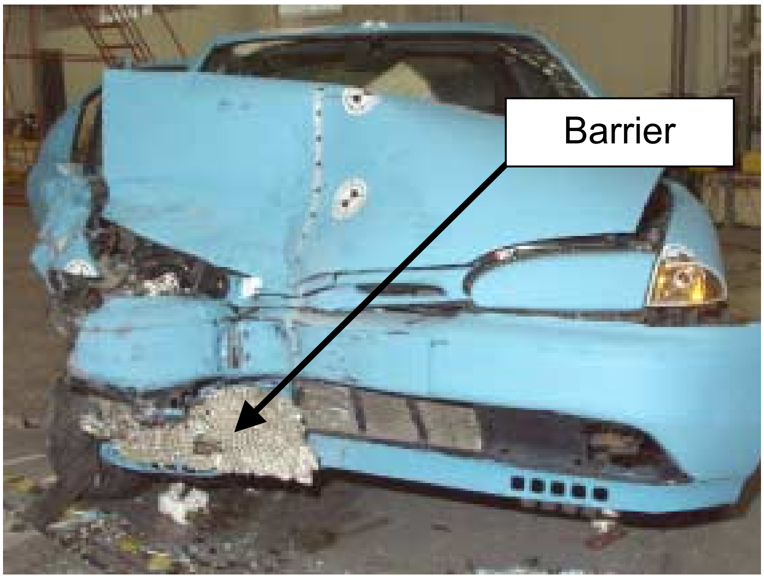


Figure 91: Post test front view of vehicle showing part of barrier that remained attached to the test vehicle following removal of the main body of the barrier.

After removal of the separated part of the barrier from the vehicle, the two sections of the barrier were joined together (Figure 92 and Figure 93).

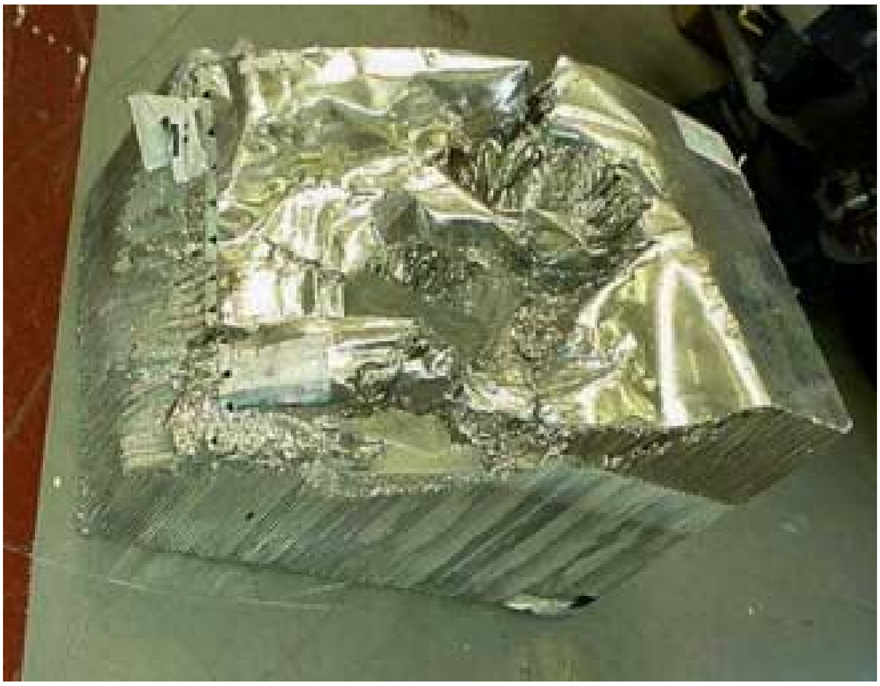


Figure 92: Progressive deformable barrier post impact after removal of all sections from the test vehicle.

Figure 93: Progressive deformable barrier post impact after removal of all sections from the test vehicle.

The load cell wall force against B-Pillar displacement, indicates a peak force of approximately 290kN at about 700mm (Figure 94). This value is below the proposed 300kN limit that indicates a compatible car (Steyer et al. 1998). After 700mm displacement, the force remained approximately constant until the maximum displacement of 1000mm was reached.

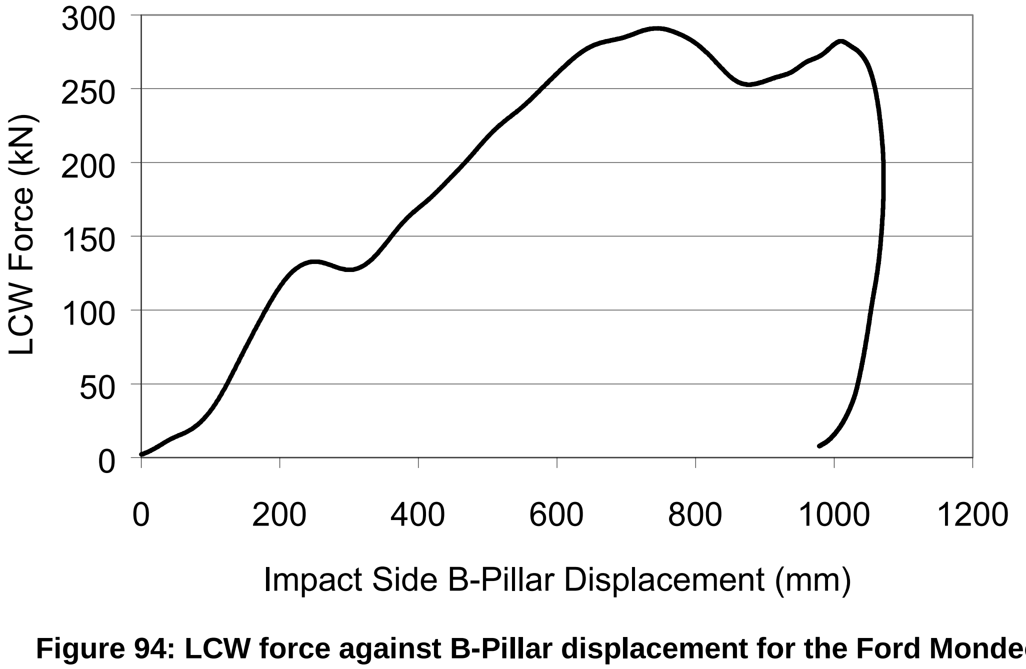


Figure 94: LCW force against B-Pillar displacement for the Ford Mondeo

The force distribution, based on the peak loads recorded by each load cell is shown (Figure 95). Analysis of this force distribution and the post test inspection of the vehicle indicated that the peak force shown in red at the lower LHS of the wall was directly in front of the vehicle's towing eye and its supporting structure. Viewing the force distribution over time highlighted the impact of the main crossbeam at 40ms, and the effect that the vehicle rotation had in generating higher than expected loads on the outer column of load cells late in the impact. These high loads can distort the true force distribution observed during the impact.

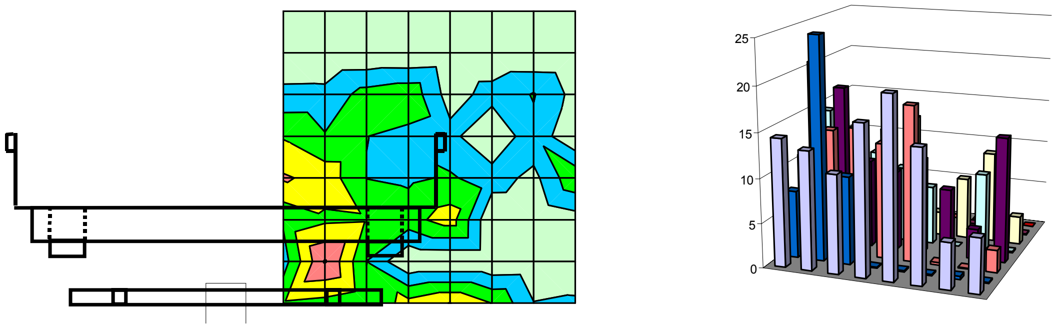


Figure 95: Force distribution using peak load cell forces.
Note: the grid for the 2D plot represents the centre point of each load cell.

Post test, the barrier crush depth at the centre point of each load cell was measured (Figure 96). The proposed assessment looks at the shape of the barrier post impact. The shape refers to both the maximum penetration of the barrier and the overall deformation profile.

	A	B	C	D	E	F	G	H
1								
2								
3	292	185	119	63	92	0	0	0
4	413	296	234	222	243	18	0	0
5	466	381	323	175	112	72	0	0
6	505	391	369	401	232	27	0	0
7	465	423	382	424	107	69	0	0
8	261	428	441	378	135	90	0	0

Figure 96: Barrier Crush Depth (mm) as measured at the centre point of each load cell.

As the outer column of load cells would be excluded due to the excessive crush caused by the vehicle rotation, the maximum penetration was determined to be 441mm located in front of the vehicle subframe. Along with the subframe, there was also increased penetration in front of the lower rail and the crossbeam between the lower rails, however this was not large enough to indicate substantial large local forces. The deformation profile indicates that this car possesses compatible features such as two levels of load paths. Based on the original assessment proposal, this vehicle would be assessed between non-aggressive and potentially aggressive (Figure 97).

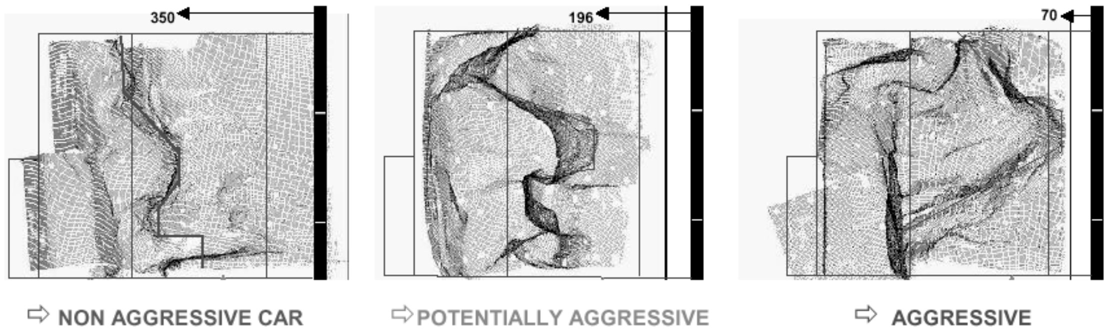


Figure 97: Indications of vehicle aggressivity based on barrier deformation profile. (Courtesy of EUCAR)

The deformation is also shown as a two dimensional and a three dimensional plot (Figure 98 and Figure 99). These plots interpolate the deformation profile between the centre point of each load cell.

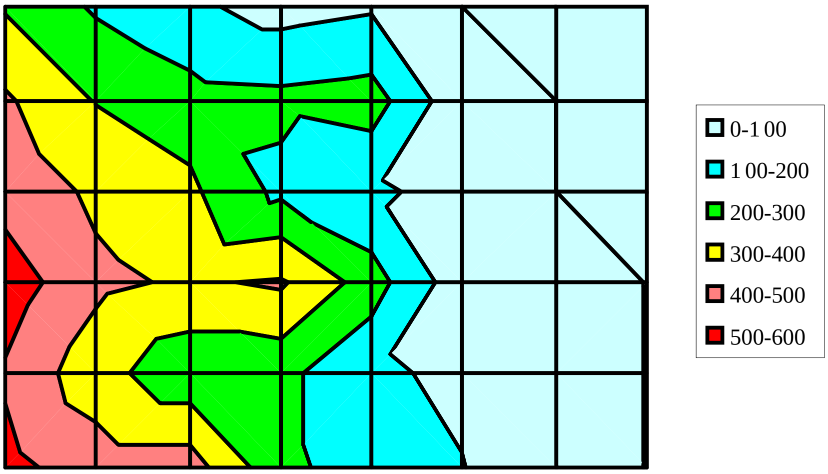


Figure 98: Barrier Deformation Plot – 2D.
Note: the grid for the 2D plot represents the centre point of each load cell.

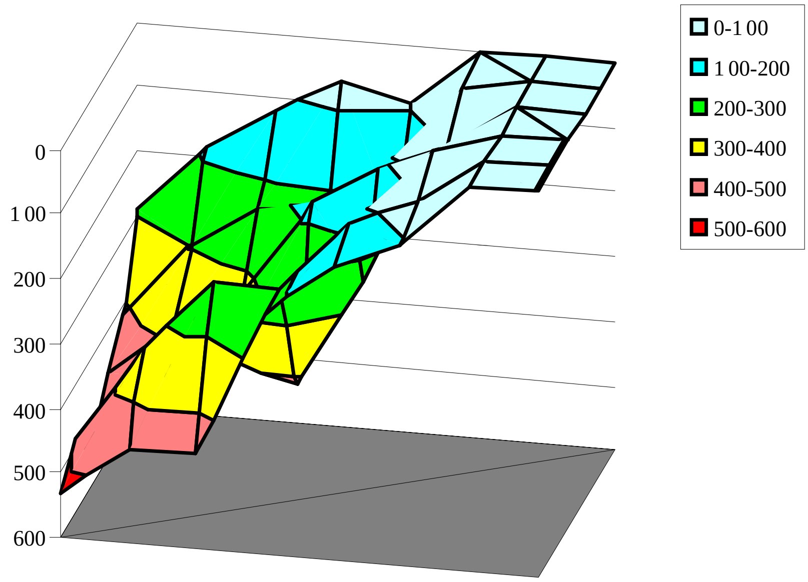


Figure 99: Barrier Deformation Plot – 3D.
Note: the grid for the 3D plot represents the centre point of each load cell.

Analysis of the results indicated that the peak load cell forces were higher than the static crush strength of the aluminium honeycomb used in the barrier. This is possibly due to the fact that when subject to a dynamic impact, the honeycomb stiffness increases. Tests at TRL have shown that for 1.71MPa honeycomb subject to an impact of 16m/s, the increase was approximately 25%. Also, as honeycomb will only crush to a fraction of its original volume, the measured deformation of the barrier will not represent the actual crush depth of the barrier. The volume of deformed honeycomb will create an incompressible section between the surface of the barrier and the actual point of crush. It has been determined from post test measurements of the barrier used in this test that the depth of this incompressible section would be equivalent to 15% of the measured deformation for the 0.34MPa layer and 25% for the progressive layers. This is assuming that all crush happened axially and there was no movement of the honeycomb in any other axis.

Using the deformation measurements, and taking into account both the dynamic stiffness and the fact that the honeycomb will only crush to a fraction of its original volume, the peak force applied to each cell was predicted. These predicted forces, in comparison to the actual forces recorded by the load cell wall are shown (Figure 100 and Figure 101).

Predicted Force
(Barrier Crush)

0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
8.5	6.1	6.1	6.1	6.1	0.0	0.0	0.0
13.1	9.4	7.4	7.0	7.7	6.6	0.0	0.0
21.4	18.7	16.9	6.6	6.6	6.6	0.0	0.0
24.0	20.0	18.8	19.8	14.4	6.6	6.6	0.0
22.1	17.3	13.8	13.8	6.6	6.6	6.6	0.0
15.0	13.4	13.4	8.8	4.3	4.3	4.3	0.0

Total Force = 390kN

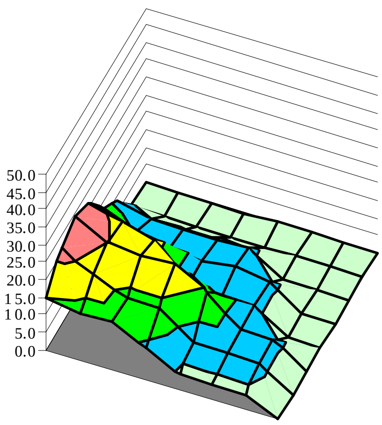
Load Cell Peak
Forces

0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
6.5	6.7	12.0	13.7	2.6	7.0	10.3	3.2
14.5	10.4	10.1	8.5	6.6	0.0	8.7	0.2
20.7	18.0	9.9	9.0	10.6	7.6	3.4	14.0
10.8	14.0	14.5	13.0	17.5	0.3	0.1	2.5
7.6	25.3	9.9	0.1	0.1	0.1	0.4	0.1
14.4	13.3	11.1	16.8	20.1	14.9	5.2	6.1

Total Force = 371kN

Figure 100: Comparison of the predicted forces based on the barrier deformation with the actual load cell peak forces.

Predicted Force
(Barrier Crush)



Load Cell Peak
Forces

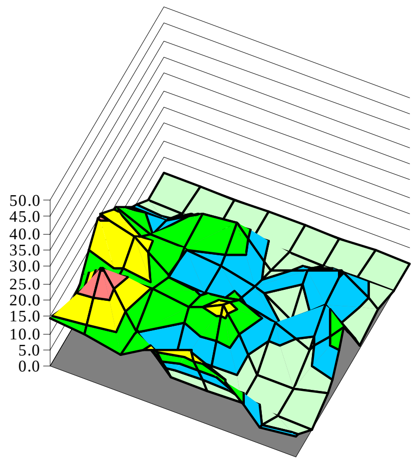


Figure 101: Predicted Forces and Load Cell Peak Forces – 3D
Note: the grid for the 3D plot represents the centre point of each load cell.

In general, the actual and predicted force levels compare reasonably well although there are some slight variations. A number of reasons exist to explain these variations.

- The shear stiffness of the barrier coupled with slight variation in the height of individual load cells, may have led to load concentrations, or bridging of load cells, before the backing sheet of the barrier deformed enough to enable even load spreading.

- It was observed that part of the stiffer progressive layer the barrier failed prior to the softer front layer (Figure 102). The deformation of the progressive layer may have occurred due to the transfer of shear through an aluminium plate between the two layers. This transfer of force would of resulted in variation between the measured force and the measured deformation.

Figure 102: Barrier deformation showing the failure of the stiffer progressive layer prior to the softer front layer.

- The predicted force distribution would not match that actual force distribution if failure of the aluminium honeycomb occurred in bending as opposed to axial crush. This is due to the fact that the force required to fail the aluminium honeycomb in bending would be far less than that required for failure in axial crush.

- Features such as the towing eye may have caused increased local deformation that was greater than that measured at the centre point of the load cell The greater the deformation of a progressive barrier, the higher the force recorded. It must be noted that any local deformation may alone not cause the higher force, but will contribute to it by deforming the stiffer parts of the progressive barrier. However, estimates based on the Mondeo towing eye have put this contribution to as little as 1/4kN due to the low frontal area.

Conclusions

- The deformation of the barrier indicates that the Ford Mondeo tested possessed two levels of load path. Multiple load paths are considered to be beneficial for compatibility.

- Based upon the three deformation profile examples reported by EUCAR, this vehicle would be assessed between non-aggressive and potentially aggressive. Further analysis based on a detailed digitisation process of the barrier will be conducted and reported on at a later date.

- The peak total load cell wall force was approximately 290kN. This value was below the proposed 300kN limit, a limit proposed by Renault (Steyer et al. 1998).

- Some agreement was found between the measured LCW force distribution and the force predicted from the barrier deformation. However, due to the problems encountered with load cell bridging that have been reported previously in the full width impact test development, it was decided not to investigate further.

- During the impact, the barrier became attached to the vehicle and remained attached post impact. Removal required tools and part of the barrier became detached from the main body. For post test analysis of the barrier deformation, the separated piece was replaced approximately.

- Part of the stiffer progressive layer of the barrier failed prior to the softer front layer. This may have been due to the interaction of the aluminium plate between the two layers transferring shear forces, or the failure in bending of the aluminium honeycomb. The failure of the aluminium honeycomb in bending would require less force then for failure in axial crush and would have therefore resulted in different levels of barrier deformation.

Range Rover against progressive deformable barrier

<table>
<tr><td>Date

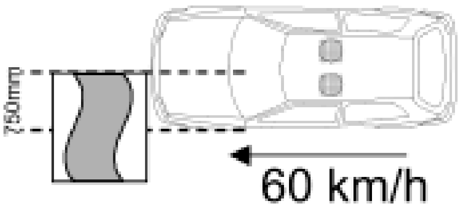
Location

Topic Group

Mass Ratio</td><td>3/10/2001

TRL

PDB Impacts

-</td><td colspan="4">
</td></tr>
<tr><td>Test Number</td><td>12MF</td><td>Vehicle 1:
Impact Side:
Test Speed:
Overlap:
Test Mass:
Dummies:</td><td>Range Rover 94
Front
60km/h
750mm
2377kg
RHS – HybridIII*
LHS – HybridIII
*not instrumented</td><td>Barrier:</td><td>PDB, version 6</td></tr>
</table>

Test Objective

The aim of this test was to investigate the effect of a high mass body on frame vehicle on the Progressive Deformable Barrier (PDB) compatibility evaluation assessment. The test vehicle was a Range Rover, as it met both the conditions for high mass and vehicle construction.

Test Details

The test was an offset frontal impact of a Range Rover with a test mass of 2377kg into a progressive deformable barrier. The barrier was as in the previous test 10MF. The Range Rover was instrumented with accelerometers at various locations around the car, predominantly to allow the interface force to be calculated. One instrumented HybridIII dummy was placed in the driver’s seat and one uninstrumented HybridIII dummy was placed in the passenger’s seat. They were positioned according to the EuroNCAP Testing Protocol Version 3 dated April 2001. The driver and passenger dummies were restrained by three point seat belts.

Results and Discussion

The test speed and overlap were within the specified tolerances. Pre and post test images of the test vehicle are shown (Figure 103 and Figure 104).

Figure 103: Vehicle pre and post test driver's side view of Range Rover.

Figure 104: Vehicle pre and post test front view of Range Rover.

The post test images of the barrier are shown (Figure 105 and Figure 106). These images show that the test vehicle bottomed out the barrier over a significant area in front of the main chassis member, the latch platform, the LHS body structure along with the LHS front wheel. During the impact, the barrier became attached to the vehicle and began to pull away from the load cell wall. Although the attachment of the barrier to the load cell wall held, the action of the vehicle pulling against the load cell wall attachment resulted in additional deformation of the barrier and its mounting plate.

Figure 105: Progressive deformable barrier post impact showing bottoming out of barrier.

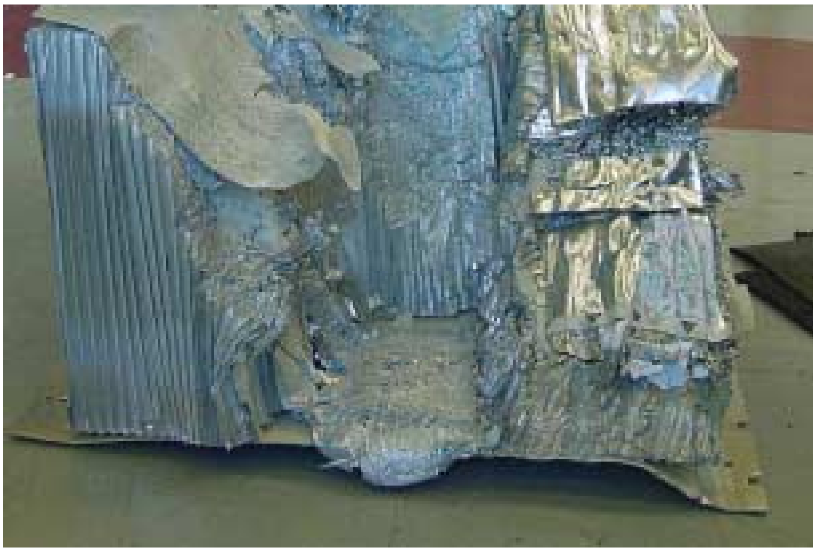


Figure 106: Progressive deformable barrier post impact showing bottoming out of barrier post impact.

The load cell wall force against B-Pillar displacement, indicates a peak force of approximately 480kN at about 1100mm (Figure 107). This value is above the proposed 350kN limit that indicates a compatible car. The point at which the vehicle bottomed out the barrier and impacted the load cell wall is indicated by the sharp increase in force at approximately 950mm.

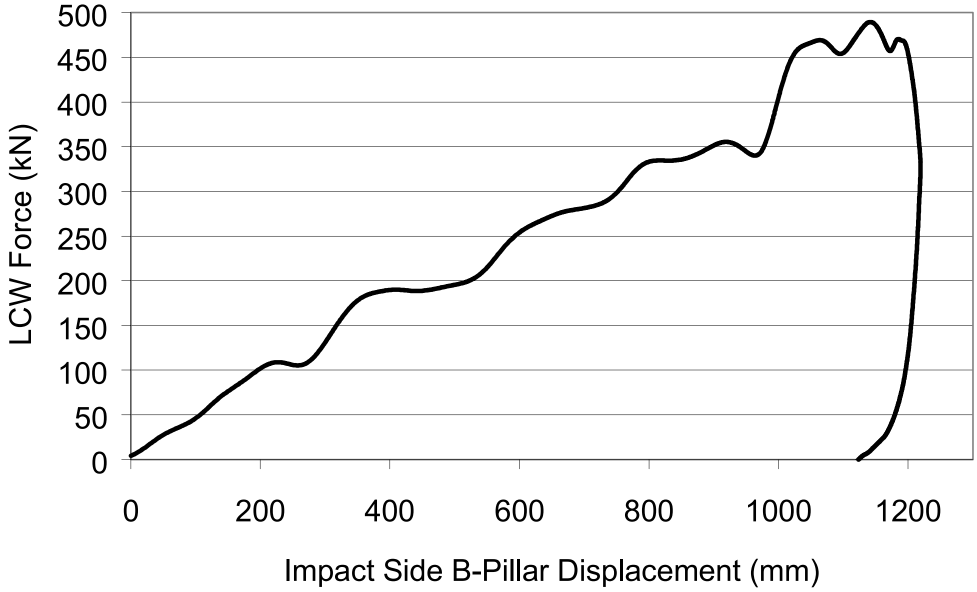


Figure 107: LCW force against B-Pillar displacement showing the sudden increase in force at approximately 950mm when the vehicle bottomed out the barrier.

Examination of the distribution of the peak load cell forces clearly identifies the stiff parts of the vehicle (Figure 108). The peak force on any one load cell during this test was 66kN, which was well in excess of the peak crush force for the barrier over one load cell of 16kN. Even when taking into account that the dynamic stiffness of the honeycomb is greater than its static stiffness, a peak force of only 20kN should be possible. It is therefore evident that the test vehicle bottomed out the barrier over a large number of load cells.

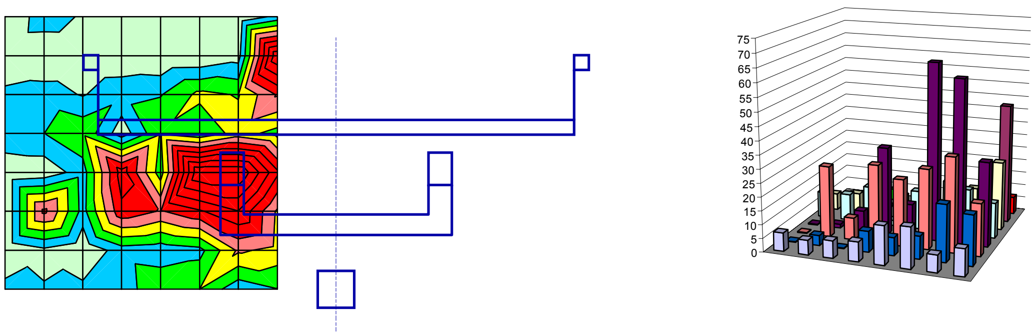


Figure 108: Force distribution using peak load cell forces.
Note: the grid for the 2D plot represents the centre point of each load cell.

To investigate the extent of the bottoming out of the barrier, the crush depth of the barrier at the centre point of each load cell has been measured and the results are shown (Figure 109). These measurements are also shown as 2D and 3D contour plots (Figure 110 and Figure 111). Based on these measurements, the vehicle would be considered aggressive when assessed against the maximum 450mm of barrier crush specified as one of the assessment criteria.

	A	B	C	D	E	F	G	H
1								
2								
3	249	1	343	221	247	386	541	648
4	107	1	142	210	246	484	583	656
5	99	1	185	225	370	701	675	668
6	116	1	200	224	208	670	664	672
7	152	1	182	211	197	41	267	341
8	251	51	169	211	140	34	1	16

Figure 109: Barrier crush depth (mm) as measured at the centre point of each load cell.

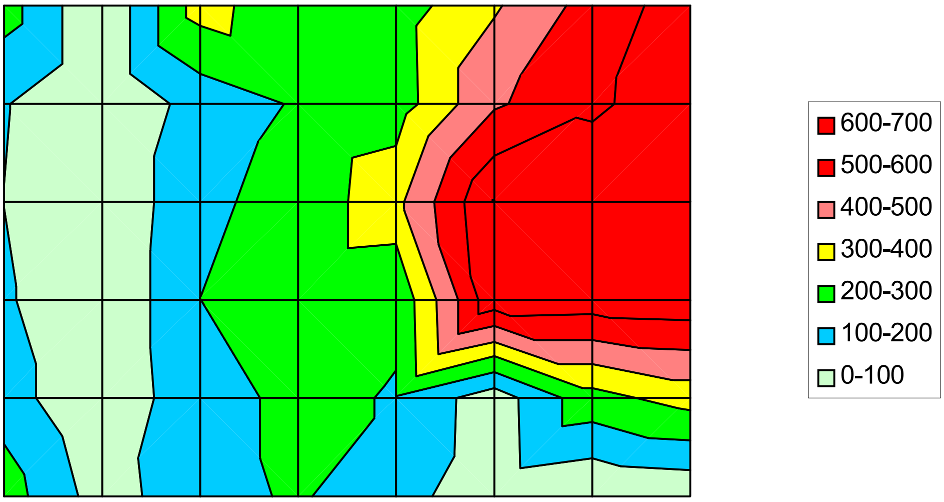


Figure 110: Barrier deformation Plot – 2D.
Note: the grid for the 2D plot represents the centre point of each load cell.

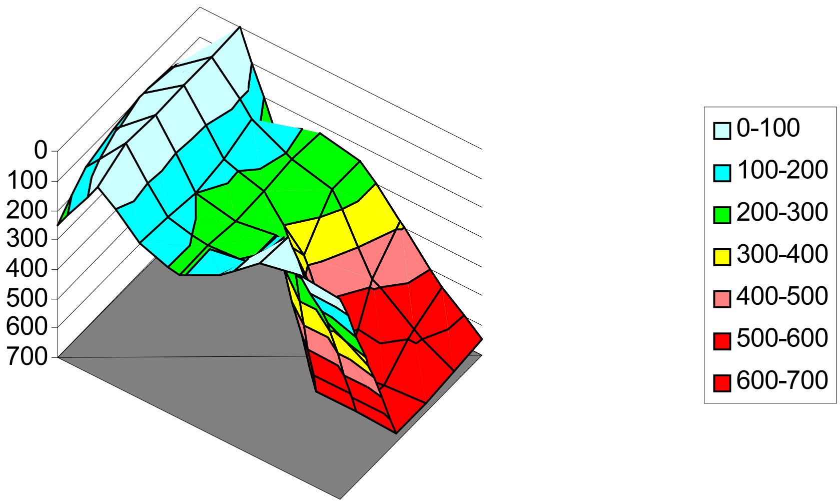


Figure 111: Barrier Deformation Plot – 3D.
Note: the grid for the 3D plot represents the centre point of each load cell.

As for the previous test, 10MF, these deformation measurements can then be used to calculate a predicted force for comparison with the actual forces recorded by the load cells. The predicted forces are shown (Figure 112) and the barrier had effectively bottomed out against the load cell wall over 12 of the 48 load cells.

Predicted Force

	A	B	C	D	E	F	G	H
1								
2								
3	7.3	6.1	10.0	6.4	7.2	BO	BO	BO
4	6.6	6.6	6.6	6.6	7.8	BO	BO	BO
5	6.6	6.6	6.6	13.8	18.3	BO	BO	BO
6	6.6	6.6	6.6	13.7	6.6	BO	BO	BO
7	0.0	0.0	0.0	0.0	0.0	0.0	15.1	17.4
8	0.0	0.0	0.0	0.0	0.0	0.0	4.3	4.3

Actual Force

	A	B	C	D	E	F	G	H
1	0.1	8.0	4.2	0.2	0.3	0.1	0.5	6.4
2	0.7	0.2	0.4	0.7	0.0	0.2	2.7	44.4
3	6.3	7.4	4.6	8.5	9.7	16.3	14.7	25.5
4	9.7	10.0	13.9	2.0	14.3	10.8	17.2	13.2
5	0.2	1.4	7.6	32.5	12.4	64.7	59.8	31.0
6	0.3	26.3	8.1	28.8	24.5	29.3	34.7	19.2
7	0.2	3.7	0.2	7.7	6.6	8.4	20.9	18.3
8	6.8	5.3	6.4	7.2	14.3	15.0	6.4	9.8

BO = Barrier Bottomed Out

Figure 112: Predicted forces based on barrier deformation measurements indicating the extent of the bottoming out of the barrier. Note: The PDB covered only the lower six rows of the load cell wall.

The large amount of crush at the edge of the progressive layer also caused a bending failure in bending of the honeycomb due to the action of the shear forces (Figure 113). The presence of the aluminium sheet at the front of the progressive layer would have increased the value of this shear force.

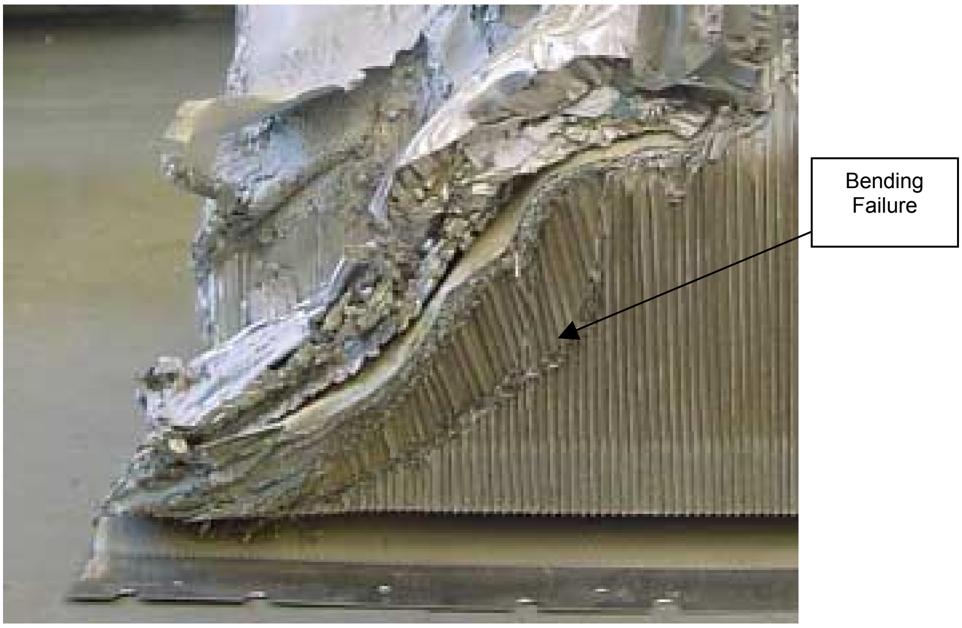


Figure 113: PDB showing the bending failure of the progressive rear layer.

Conclusions

- The vehicle's main chassis rail, LHS body structure and latch platform bottomed out the barrier over a large area. The extensive contact induced significant damage to the honeycomb not directly impacted, mainly through the transfer of shear forces.

- The load cell wall total peak force was approximately 480kN. Both the force time history and the force displacement history showed a sudden increase in force when the vehicle bottomed out the barrier.

- No useful comparison between the predicted LCW force and the actual peak load cell forces could be made due to the fact that barrier had bottomed out over such a large area.

- The attachment of the vehicle to the barrier during the impact, and the subsequent action of the vehicle in pulling the barrier away from the wall, resulted in additional deformation of the barrier and the backplate.

6.3.2 BASt tests

BASt carried out three tests against the progressive deformable barrier (PDB). As TRL, BASt measured the impact forces behind the PDB. For the tests the load cell wall of BASt was modified. The measuring area for each force transducer was reduced to 125 x 125 mm. The measuring elements have an extremely high overload capacity, are compensated to temperature and oblique forces so that only longitudinal forces are measured. The gap between the measuring fields is 1 mm. Each measuring field is covered by 124.5 x 124.5 mm plywood, 20 mm thick. Figure 114 below shows the arrangement of the available 44 load cells (at the shaded positions no forces could be measured) and the mounting of the PDB to the load cell wall.

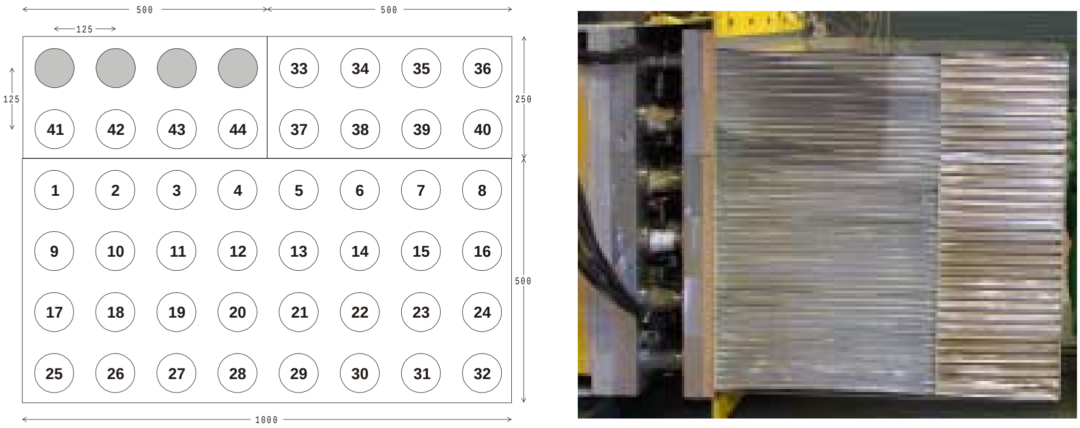
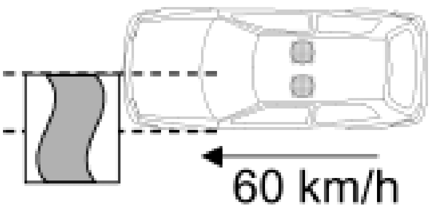


Figure 114: Progressive Deformable Barrier and Load Cell Wall

The PDB elements were supplied by AFL in France. In the test with the modified ASTRA a PDB version 6 and in the tests with the SMART and the Volvo S80 a PDB version 7 were used. From PDB version 6 to version 7 the thickness of the aluminium front cover was increased by adding a second aluminium front sheet. The other cover sheets remained unchanged. So the thickness of the aluminium front sheet is in version 6 1 mm and in version 7 2 mm. In all tests the position of the load cell wall was such that the lower four rows were below the split line of the upper load and lower load areas of the barrier, whilst the remaining two rows were above the split line. Therefore the PDB element did not fully cover the lowest and highest rows of the force measuring wall. As illustrated above the four outer (upper left) measuring areas of the highest row could not be equipped with load cells, but as in the three BASt tests the impacting vehicles did not load these areas, the force measurement was not affected by this deficiency.

Smart against progressive deformable barrier

Date	06.06.2002
Location	BASt
Topic Group	PDB Impacts
Mass Ratio	
Test Number	**02CO16FO**



Vehicle:	Smart	Barrier:	PDB
Impact Side:	Front		Version 7
Test Speed:	60km/h		
Overlap:	750mm		
Test Mass:	841 kg		
Dummies:	LHS – Hybrid III		

Test Objective

The aim of this test is to investigate the performance of the PDB version 7 (with thicker aluminium cover sheet) in an impact test with a very small and light vehicle with an extreme frontal stiffness.

Results and Discussion

The test was an offset frontal impact of a SMART with a test mass of 845 kg into a PDB-element version 7 as described above. The SMART was instrumented with accelerometers at the bottom of the B-pillars. Although the engine is mounted at the rear of the vehicle three accelerometers were applied on the top and the bottom of the engine. One instrumented Hybrid III dummy was placed on the driver's seat and one uninstrumented Hybrid III dummy was placed on the passenger's seat. They were positioned according to the Euro NCAP Testing Protocol Version 3 dated April 2001. Driver and passenger dummies were restrained by three point seat belts.

The test speed and overlap were within the specified tolerances. Pre and post test images of the test vehicle are shown in Figure 115 and Figure 116.

Figure 115: Vehicle pre and post Test Driver's Side View, Test 02CO16FO

Figure 116: Vehicle pre and post Test Front View, Test 02CO16FO

The post test images of the deformed barrier are shown in Figure 117. These images show that the SMART deformed the PDB relatively homogeneously over the full contact area between SMART front and the PDB face. Only the extreme stiff left longitudinal penetrated locally the PDB element version 7 with the thicker aluminium cover sheet. The overlap of 750 mm covers about two thirds of the SMART front, additionally the longitudinals are located closer to the middle of this car in comparison to other car models; these two facts are the reason for very little lateral deformation of the PDB caused by rotation of the vehicle during impact.

Figure 117: Progressive Deformable Barrier Post Impact, Test 02CO16FO

According to the low weight of the SMART, the deformation depth of the PDB and the local penetration of the longitudinal are relatively low as the deformation plots in Figure 118 below show.

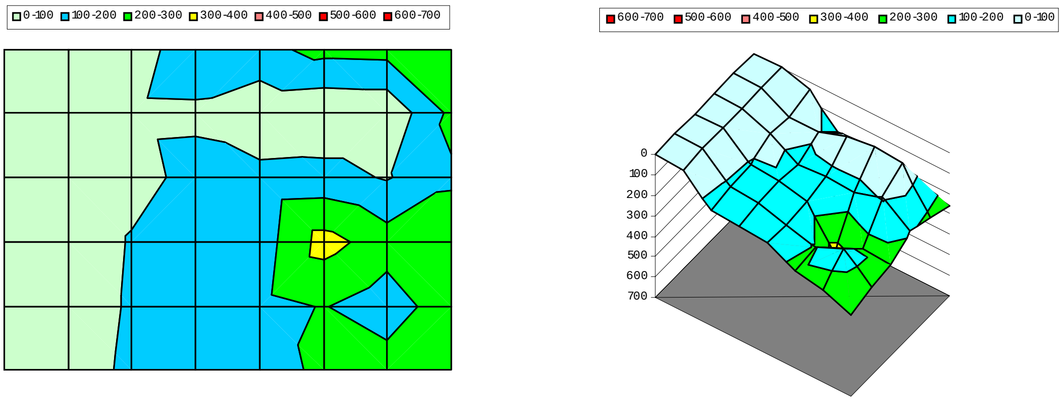


Figure 118: Barrier Deformation Plot – 2D and Plot – 3D, Test 02CO16FO

The maximum deformation amounts to 338 mm as Figure 119 of the deformation measurement shows. This means that the penetration into the stiffer second layer is only about 80 mm.

	A	B	C	D	E	F	G	H
1	0	0	37	177	167	232	237	262
2	0	0	75	79	30	14	21	223
3	0	0	56	138	126	135	92	187
4	0	0	110	117	147	338	246	252
5	0	0	121	126	170	204	145	260
6	0	12	136	142	156	220	246	299

Figure 119: Barrier Crush Depth (mm),Test 02CO16FO

The load cell wall force against B-Pillar displacement, indicates a peak force of approximately 250 kN at about 400 mm vehicle displacement, see Figure 120. The small drop in the impact force may be caused by the bending of the front end of the longitudinal when it had reached the stiffer deeper layer.

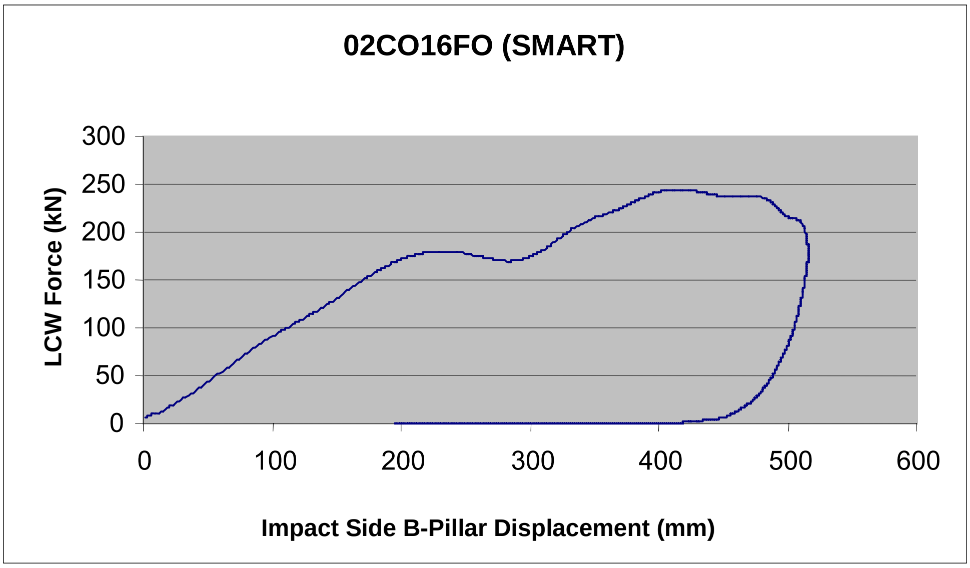


Figure 120: LCW Force against B-Pillar Displacement (mm), Test 02CO16FO

In relation to the B-Pillar displacement the PDB deformation is relatively high. This means that the front of the SMART is very stiff and is only slightly deformed after the test. The high stiffness of this small and light car also illustrates Figure 121. The peak deceleration value reaches nearly 50 g, the impact pulse duration is about 80 msec.

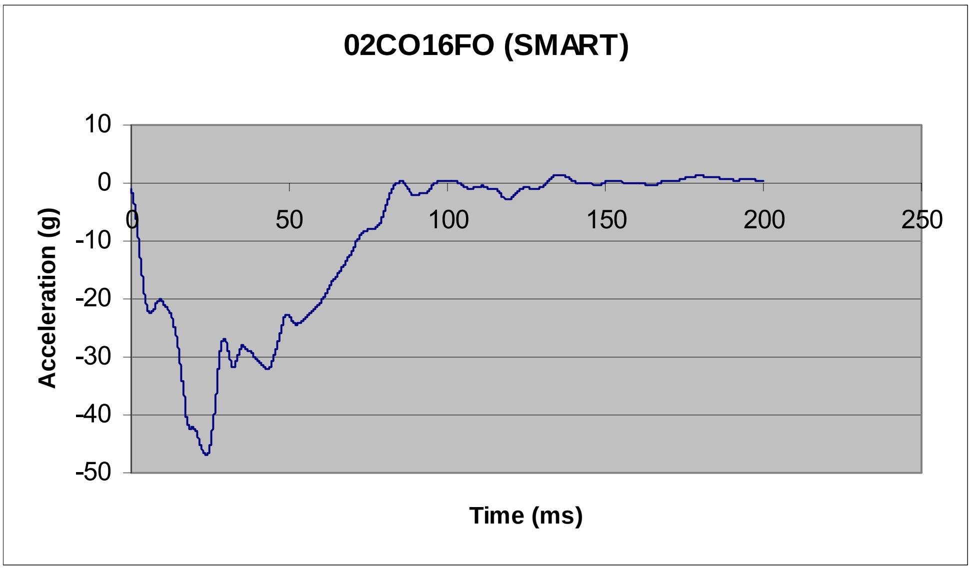


Figure 121: Vehicle Acceleration against Time, Test 02CO16FO

The force distribution, based on the peak loads recorded by each load cell is shown in Figure 122. Analysis of this force distribution and the post test inspection of the vehicle indicates that the front stiffness is relatively well distributed. The deeper local penetration of the PDP by the longitudinal does not lead to local higher peak load cell forces because the penetration to the deeper layer is more wedge shaped and the aluminium honeycombs are pressed sideways.

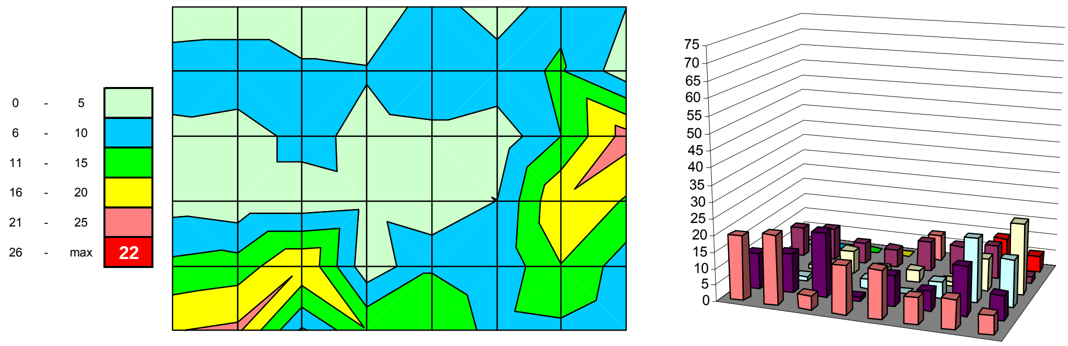


Figure 122: Force Distribution using Peak Load Cell Forces, Test 02CO16FO

Conclusions

Based on the assessment measures applied, the test vehicle SMART would be considered as a non aggressive vehicle although the vehicle itself is extremely stiff. The deformation shape of the progressive deformable barrier is in principle homogenous over the contact area between vehicle and PDB. The thickness of the cover of the PDB version 7 avoids a large tear of the cover sheet. Only under the localised loading of the bended longitudinal was a small area torn. This unusual narrow vehicle type was loaded by the PDB test procedure in a different way compared with other wider vehicle types. The overlap degree is much higher for this car than for the most other vehicle types. This leads to lower vehicle rotation during the impact phase with correspondingly less lateral force on the PDB.

Opel Astra (200kg mass decrease) against progressive deformable barrier

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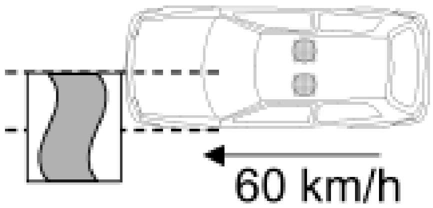
<tr><td>Date</td><td>23.04.2002</td><td colspan="4" rowspan="4">
</td></tr>
<tr><td>Location</td><td>BASt</td></tr>
<tr><td>Topic Group</td><td>PDB Impacts</td></tr>
<tr><td>Mass Ratio</td><td></td></tr>
<tr><td>Test Number</td><td>02CO14FO</td><td>Vehicle
Impact Side:
Test Speed:
Overlap:
Test Mass:
Dummies:</td><td>Opel Astra 2001
Front
60km/h
685 mm / 40%
975 kg</td><td>Barrier:</td><td>PDB
Version 6</td></tr>
</table>

Test Objective

In this test the effect of different mass of the same vehicle model in impacts to the PDB should be studied. The mass of the ASTRA was lowered by 200 kg through removing heavier car parts which are not involved in the structural deformation, e.g. seats, spare wheel, the glass of the windows.

Test Details

By error the overlap of the car and the ground clearance of the PDB were not adjusted correctly in this test. The overlap was 685 mm instead of 750 mm of the proposed test procedure. The ground clearance was 200 mm instead of 150 mm. Particularly for this vehicle model the ground clearance of the PDB is relevant because the ASTRA has a stiff subframe, which underrode the PDB in the test. This did not occur in a similar test at UTAC. So the tests at UTAC (ASTRA with normal mass) and BASt (ASTRA with reduced mass) can not be fully compared. Therefore the test has to be repeated in the next phase of the compatibility study. However, the behaviour of the PDB-element can still be assessed. The lack of the possibility to compare the tests at UTAC and BASt is compensated by the presentation of high resolution force measurements of an additional Euro NCAP test.

The test was an offset frontal impact of an ASTRA of a test mass of 975 kg into a PDB-element version 6 (only one layer of 1 mm aluminium front cover sheet) as described above. The ASTRA was instrumented with accelerometers at the bottom of the B-pillars and on the top and the bottom of the engine. To keep the mass of the impacting vehicle low no dummies were used in this test.

The test speed was within the specified tolerances. Pre and post test images of the test vehicle are shown in Figure 123 and Figure 124. The PDB-element was trapped by the front structure of the ASTRA and torn off from the back plate during the vehicle rebound. Figure 125 shows the deformed car front after removal of the PDB and destroyed plastic front covers.

Results and Discussion

Figure 123: Vehicle pre and post Test Driver's Side View, Test 02CO14FO

Figure 124: Vehicle pre and post Test Front View, Test 02CO14FO

The significant deformation characteristic of the vehicle front is the deep deformation of the front lateral beam close to the left longitudinal. This shows that the PDB clearly generates shear when weaker structures are contacted and loaded. The stiffer second layer apparently does not reduce this effect as a rigid block would do.

Figure 125: Vehicle Post Test Front View, Test 02CO14FO

The post test images of the deformed barrier are shown in Figure 126. These images illustrate that the ASTRA deeply deformed the PDB at different locations of the contact area. The stiff left longitudinal penetrated locally the PDB element. Apparently when it reached the second stiffer honeycomb layer the foremost end of the longitudinal was bent nearly 90° upwards and as a consequence of this the lateral beam was twisted to the same degree. The support of the cooler was also bent upwards so that a sharp wedge was generated. This wedge easily cut deeply into the PDB-element. A stiffer front cover would probably have avoided this effect caused by a part of the front structure, which does not significantly contribute to the vehicle's frontal stiffness distribution.

Figure 126: Progressive Deformable Barrier Post Impact, Test 02CO14FO

Although the ASTRA is only about 140 kg heavier than the SMART it deeply deforms the PDB. The different structures of the ASTRA tear up the softer front honeycombs so that a reasonable measurement of the PDB deformation profile seems impossible. Figure 127 below only by way of suggestion show this result.

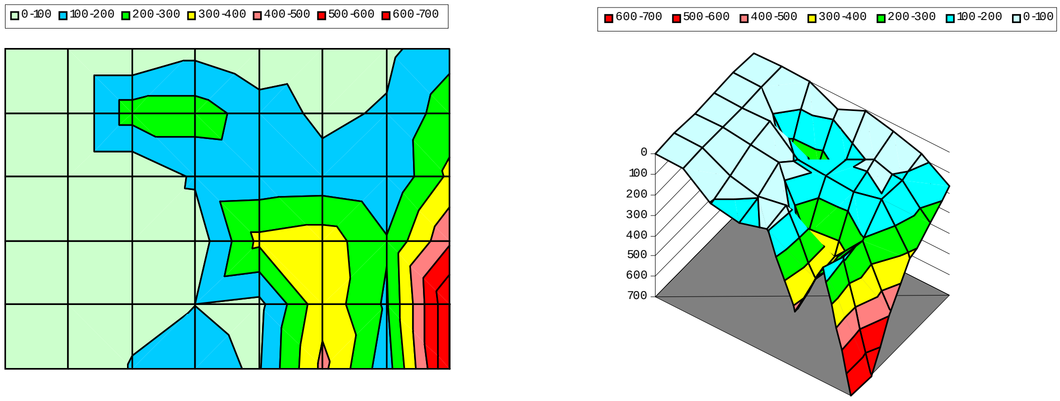


Figure 127: Barrier Deformation Plot – 2D and Plot – 3D, Test 02CO14FO

The maximum deformation amounts to 425 mm to 484 mm (the measurements in the fields H5 and H6 are not considered because here the honeycomb materials was pressed side wards) as Figure 128 of the deformation measurement shows. This means that the penetration into the stiffer second layer is up to 235 mm.

	A	B	C	D	E	F	G	H
1	0	0	0	67	11	39	76	167
2	0	0	248	248	150	78	111	251
3	0	0	2	118	130	135	104	322
4	0	0	17	31	327	357	209	484
5	0	0	70	99	72	367	186	700
6	0	0	107	131	89	425	137	700

Figure 128: Barrier Crush Depth (mm),Test 02CO14FO

The load cell wall force against B-Pillar displacement indicates a peak force of approximately 250 kN at about 620 mm to 850 mm vehicle displacement, see Figure 129. The peak force is very similar to the peak force measured in the SMART test but the B-Pillar displacement (PDB intrusion plus vehicle deformation) is significantly higher.

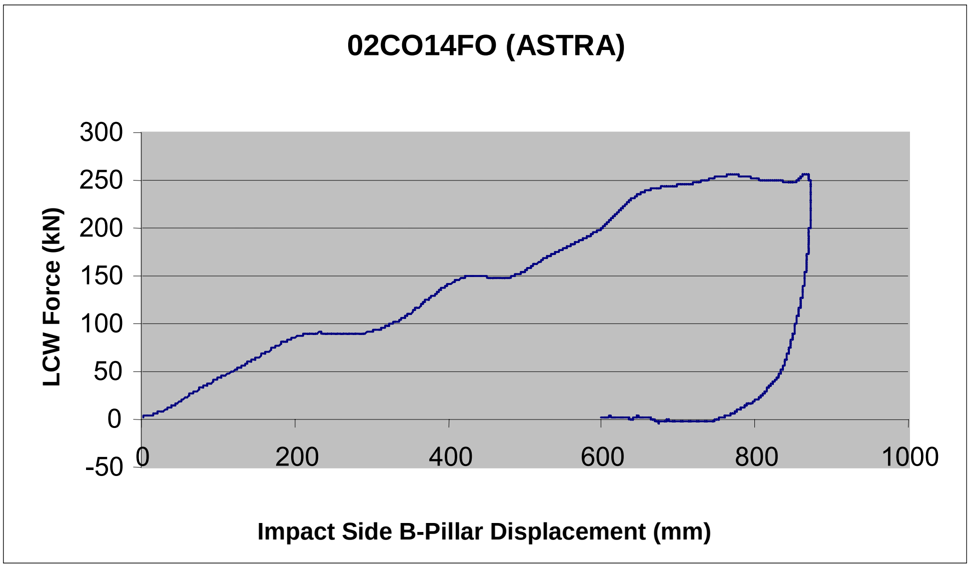


Figure 129: LCW Force against B-Pillar Displacement (mm), Test 02CO14FO

The vehicle deceleration, particularly at the end of the pulse when the longitudinal reaches the stiffer layer of the PDB, is relatively high also caused by the reduced weight of the test vehicle and the structure stiffness which is developed for higher masses in the self protection tests.

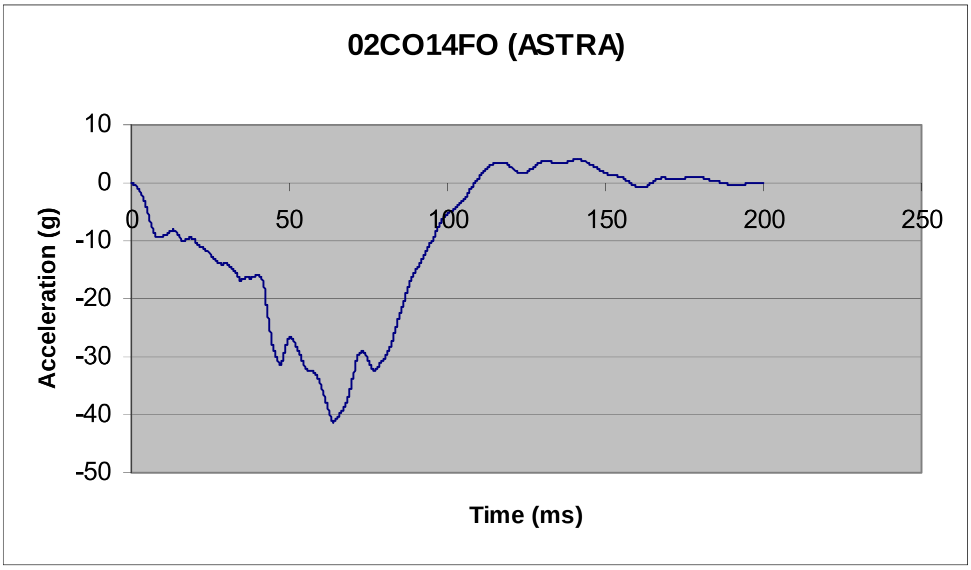


Figure 130: Vehicle Acceleration against Time, Test 02CO14FO

The force distribution, based on the peak loads recorded by each load cell is shown in Figure 131. Analysis of this force distribution and the post test inspection of the vehicle indicates that the front stiffness was not very well distributed. The deeper local penetration of the PDB by the longitudinal led to local higher peak load cell forces. The second near the cooler fan is caused by the force transmission through the fan motor to the engine.

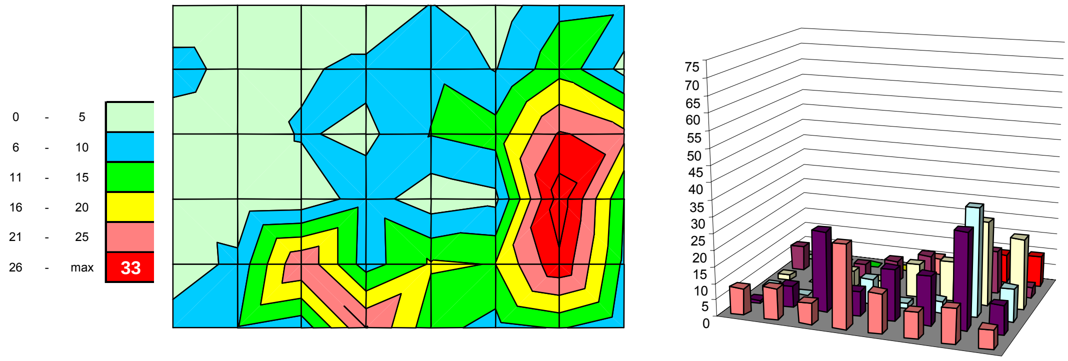


Figure 131: Force Distribution using Peak Load Cell Forces, Test 02CO14FO

Conclusions

Based on the assessment measures applied, the ASTRA would be considered as a moderately aggressive vehicle. The front structure stiffness distribution is not very homogeneous. The bumper crossbeam undergoes a large deformation in the region adjacent to the longitudinal. The deformation shape of the progressive deformable barrier is extremely inhomogeneous over the contact area between vehicle and PDB. The thickness of the cover of the PDB version 6 was not great enough to avoid a massive tear of the cover sheet. It appears impossible to develop an objective procedure to make reasonable measurements of the barrier deformation for the degree of deformation seen in this test. The PDB test method showed good potential to detect shear weaknesses in the car front structure.

Volvo S80 against progressive deformable barrier

<table>
<tr><td>Date

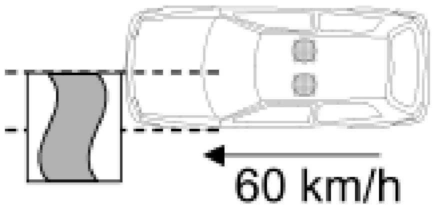
Location

Topic Group

Mass Ratio</td><td>25.04.2002

BASt

PDB Impacts

</td><td colspan="4">
</td></tr>
<tr><td>Test Number</td><td>02CO15FO</td><td>Vehicle:
Impact Side:
Test Speed:
Overlap:
Test Mass:
Dummies:</td><td>Volvo S80 1998
Front
60km/h
733mm / 40%
1810 Kg
LHS – HybridIII
RHS – HybridIII*
Rear LHS – P3
Rear RHS – P1,5</td><td>Barrier:</td><td>PDB
Version 7</td></tr>
</table>

Test Objective

The aim of this test is to investigate the performance of the PDB version 7 (with thicker aluminium cover sheet) in an impact test with a large and heavy vehicle. The tested vehicle represents a modern large sized vehicle with more than one level of load paths.

Test Details

By error the overlap of the car and the ground clearance of the PDB were not adjusted correctly in this test. The overlap was 733 mm instead of 750 mm of the proposed test procedure. The ground clearance was 200 mm instead of 150 mm. The test was an offset frontal impact of a Volvo S 80 with a test mass of 1810 kg into a PDB-element version 7 (two layers of 1 mm aluminium front cover sheet each) as described above. The Volvo was instrumented with accelerometers at the bottom of the B-pillars and on the top and the bottom of the engine. One instrumented Hybrid III dummy was placed on the driver's seat and one uninstrumented Hybrid III dummy was placed on the passenger's seat. They were positioned according to the Euro NCAP Testing Protocol Version 3 dated April 2001. Driver and passenger dummies were restrained by three point seat belts.

Results and Discussion

The test speed was within the specified tolerance. Pre and post test images of the test vehicle are shown in Figure 132 and Figure 133.

Figure 132: Vehicle pre and post Test Driver's Side Views, Test 02CO15FO

Figure 133: Vehicle pre and post Test Front View, Test 02CO15FO

The post test images of the deformed barrier are shown in Figure 134. These images show that the Volvo S 80 deformed the PDB very homogeneously over the full contact area. There was no local penetration found on the PDB element version 7 with the thicker aluminium cover sheet. The deep deformation of the lower row of the PDB shows that the main stiff load paths of this vehicle are placed also relatively deep. In contrast to this the upper load paths of the Volvo seem to be relatively weak. This might result in a susceptibility to being overridden by other vehicles. As the Volvo is a very heavy vehicle the deformation depth of the PDB is substantial. Due to the rotation of the test vehicle at the end of the impact the inner honeycomb material is pressed sideways.

Figure 134: Progressive Deformable Barrier Post Impact, Test 02CO15FO

The PDB deformation measurements shown in Figure 135 clearly show the very uniform deformation with a slope towards the right end.

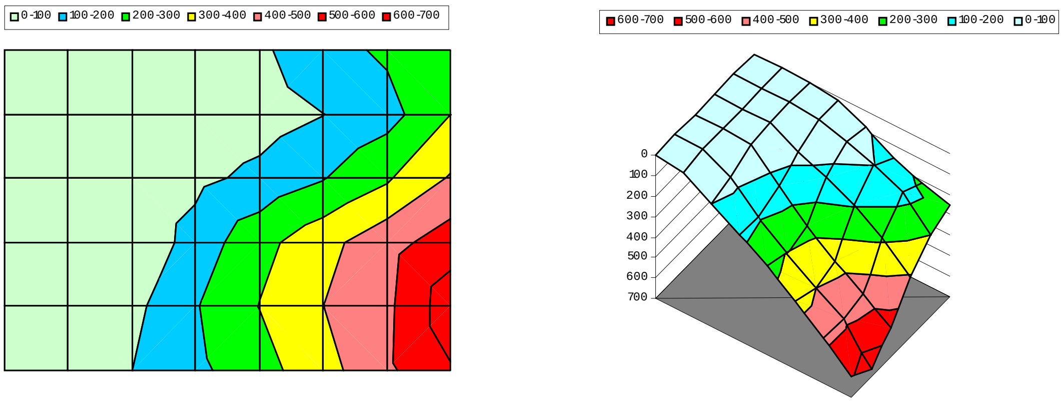


Figure 135: Barrier Deformation Plot – 2D and Plot – 3D, Test 02CO15FO

Neglecting the part of the PDB with lateral bending the maximum deformation amounts to about 500 mm as Figure 136 of the deformation measurements shows.

	A	B	C	D	E	F	G	H
1	0	0	0	19	83	162	217	252
2	0	0	0	17	51	98	161	300
3	0	0	0	72	127	195	287	407
4	0	0	10	142	268	367	462	552
5	0	0	70	194	303	398	479	659
6	0	15	100	177	263	363	480	592

Figure 136: Barrier Crush Depth (mm), Test 02CO15FO

The load cell wall force against B-Pillar displacement indicates a peak force of approximately 350 kN at about 1100 mm vehicle displacement, see Figure 137.

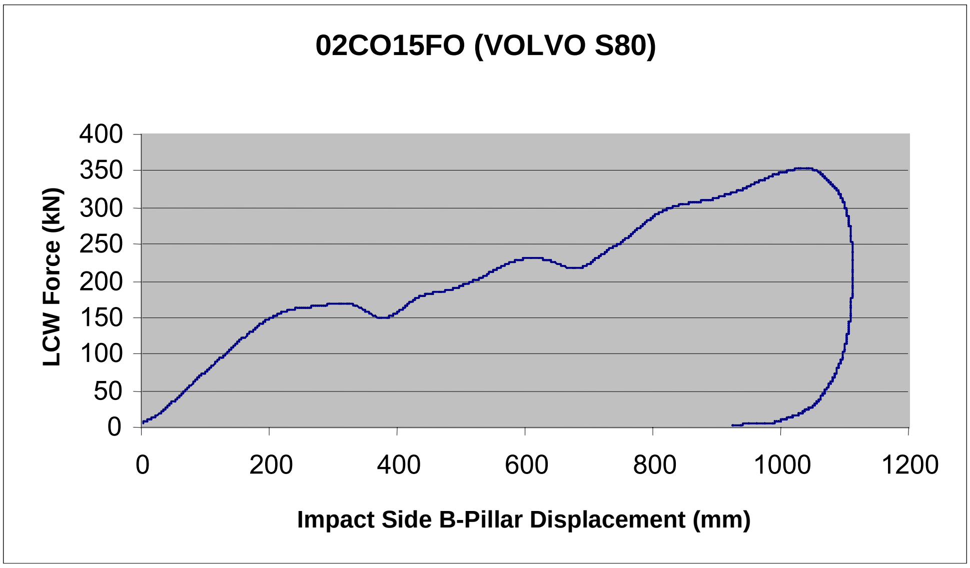


Figure 137: LCW Force against B-Pillar Displacement (mm), Test 02CO15FO

The Volvo occupant compartment undergoes a peak deceleration of 28 g in this test (Figure 138).

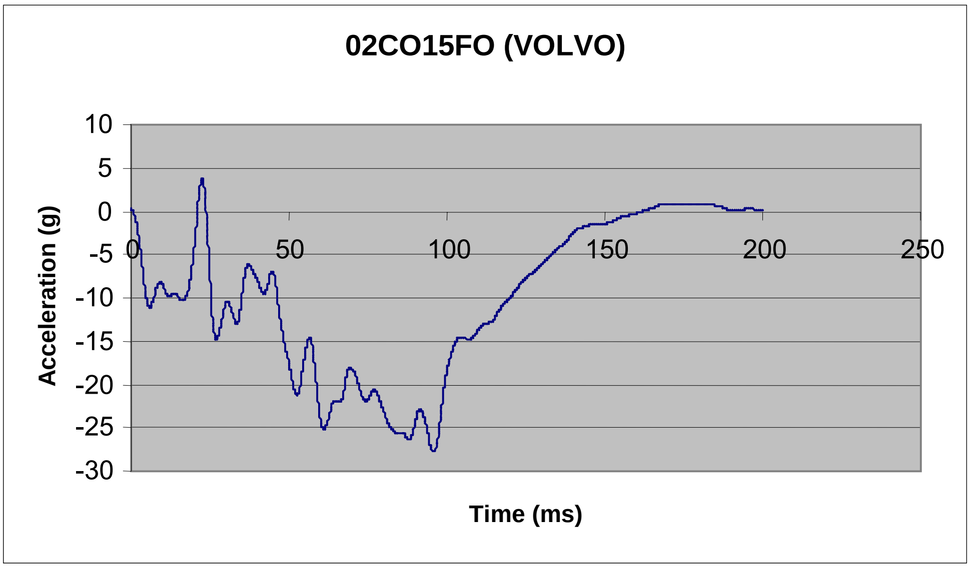
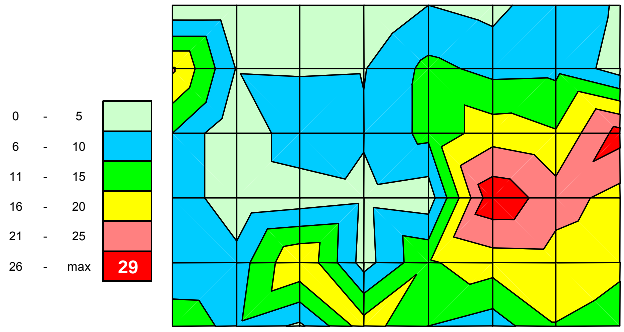


Figure 138: Vehicle Acceleration against Time, Test 02CO15FO

Figure 139 based on the peak load recorded by each load cell demonstrates that the front stiffness of the Volvo is well distributed.



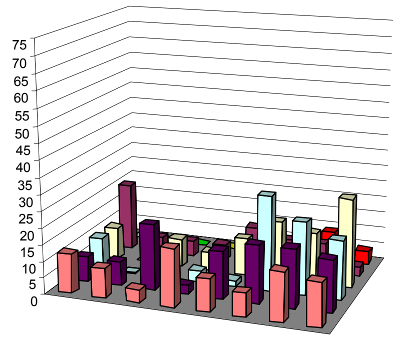


Figure 139: Force Distribution using Peak Load Cell Forces, Test 02CO15FO

Conclusions

The Volvo S 80 with a mass of 1810 kg is a heavy vehicle. Based on the assessment measures applied, the test vehicle would be considered as a non aggressive vehicle although the vehicle itself is heavy. The deformation shape of the progressive deformable barrier is very homogenous over the contact area between vehicle and PDB. The thickness of the cover sheets of the PDB version 7 helps to avoid any tearing. Comparing the front face cover of the PDBs after the SMART and Volvo tests the two layer thicker cover of PDB version 7 appears to be a good compromise to detect extremely aggressive local front structures.

6.3.3 Summary of Conclusions and Recommendations

It should be noted that the purpose of the PDB test is to assess structural interaction and the frontal unit energy absorption up to an Equivalent Energy Speed (EES) of 50 km/h. An impact speed of 60 km/h was calculated to give a vehicle EES of 50 km/h, which takes into account the energy absorption of the barrier and the vehicle stiffness. A fixed overlap width of 750 mm is used to ensure that the barrier generates the same load for cars of different widths.

Conclusions

- The use of the load distribution on the LCW behind the PDB does not appear to give an accurate enough measure of a car's stiffness homogeneity and hence is not worth pursuing further as an assessment method. This is because problems similar to those encountered with the full width tests, such load cell bridging caused by the shear strength of the honeycomb, occur to some degree with this test. This conclusion is supported by a separate French study, which found an uneven load distribution was recorded on the load cell wall for an impact against the PDB using a trolley with a flat rigid face.

- In the Mondeo test a part of the barrier remained attached to the car after the test. This would cause severe difficulties in measuring the barrier final deformation profile objectively, which the PDB approach is completely reliant upon. For this test the version 6 of PDB was used. Version 7 of the barrier has a thicker front sheet, which may reduce or solve this problem. The lack of penetration of the barrier front sheet in the test with the Volvo S80 indicates the improved performance of version 7 of the barrier in this respect.

- In the Range Rover test, the barrier bottomed out. The PDB barrier was defined to represent an average car and its stiffness is such that bottoming out is unlikely, except for high mass vehicles. The test results show that the Range Rover's mass and stiffness exceed the capabilities of the current barrier face.

- The Smart is a very light car. It was judged to be a non aggressive car based on the shape of the barrier deformation after the impact, even though its stiffness is very high.

- The Volvo S80 is a heavy car with a mass of 1810 kg. Even though this car is heavy it was judged to be non aggressive, based on a subjective assessment of the relatively homogeneous barrier deformation.

Recommendations

The current assessment criteria require further development. At present these criteria are based on the shape of the barrier face final deformation. A formula has been developed to assess the barrier deformation in terms of its height and depth, but limit values for these parameters still need to be defined. The test data collected in this project completes a crash test matrix, which will form a useful data set for future work to continue the development of the current assessment criteria. However, further data will be required to validate the procedure and set definitive limit values.

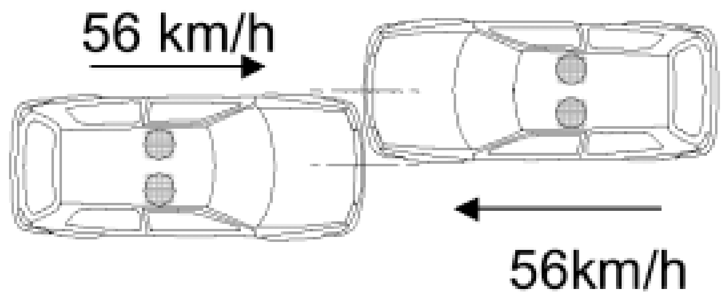
The implications of the vehicle mass and stiffness limits that the current barrier face can be used to test should be considered. The suitability of the current barrier face design should be considered in terms of its likely effect on the future vehicle fleet design. An example of a parameter that should be investigated is the stiffness distribution between its upper and lower sections.

6.4 CAR TO CAR

6.4.1 Fiat test

Renault Clio against Toyota Yaris

Date	
Location	FIAT
Topic Group	Car to Car
Mass Ratio	1.05
Test Number	13599



Vehicle 1:	Renault Clio 98	Vehicle 2:	Toyota Yaris
Impact Side:	Front	Impact Side:	Front
Test Speed:	56km/h	Test Speed:	56km/h
Overlap:	50%	Overlap:	50%
Test Mass:	1 047kg	Test Mass:	1 100kg
Dummies:	RHS – HybridIII LHS – HybridIII	Dummies:	RHS – HybridIII LHS – HybridIII

Test Objective

The aim is to compare the behaviour of cars involved in car-to-car crashes and the results that each car had in the test performed against a barrier. The chosen cars were the Toyota Yaris and the Renault Clio II, two cars of the same mass. The Toyota Yaris was the best in class performer of the small cars in EuroNCAP rating (self-protection) and the Renault Clio II represented a small size vehicle with more than one level of load path.

Test Details

The test was planned at 112 kph closing speed with an overlap of 50% for each car on the left. Both cars were left-hand drive.

Each car was instrumented with accelerometers at various locations around the car, predominantly to allow the interface force (both mechanical and structural) to be calculated. Two instrumented HybridIII dummies were placed in the front seats of each car. The driver and the passenger dummies were restrained by three point seat belts with pretensioners and protected by airbags.

From an analysis of the frontal structure of both cars, a good lower beam interaction was expected. The vertical height of RENAULT CLIO longitudinals was from 400 to 480 mm from the ground and the TOYOTA YARIS longitudinals was from 380 mm to 500 mm from the ground so, theoretically, a vertical overlap between them was guaranteed. A horizontal overlap was expected too, because the distance of the longitudinal from the longitudinal axis was approximately the same (only 10 mm of difference).

Results and Discussion

The test was performed at 118 kph (59 kph for each car) due to a problem at the logical control of external engine of the laboratory. After the crash, both cars had a rotation of about 90° from the initial position (Figure 140).

Figure 140: Initial and final position of crashed cars

The overall shapes of the compartment pulses were similar, although there were large differences for short time periods during some oscillations (Figure 141).

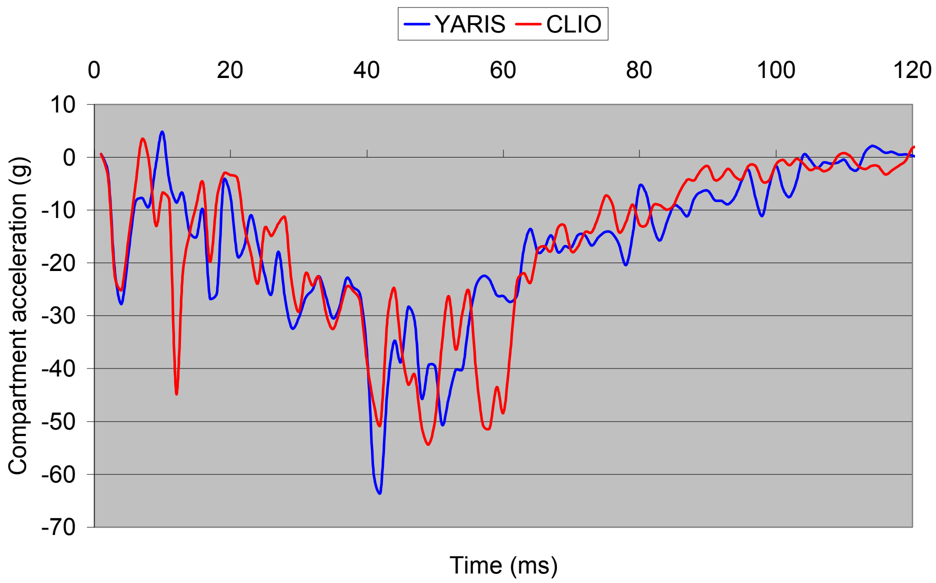


Figure 141: Compartment accelerations vs time

Using a previously used methodology for force calculation (Steyer et al. 1998 and Wykes et al. 1998), in case of car to car crash, the forces, both mechanical and structural, were calculated (Figure 142 and Figure 143).

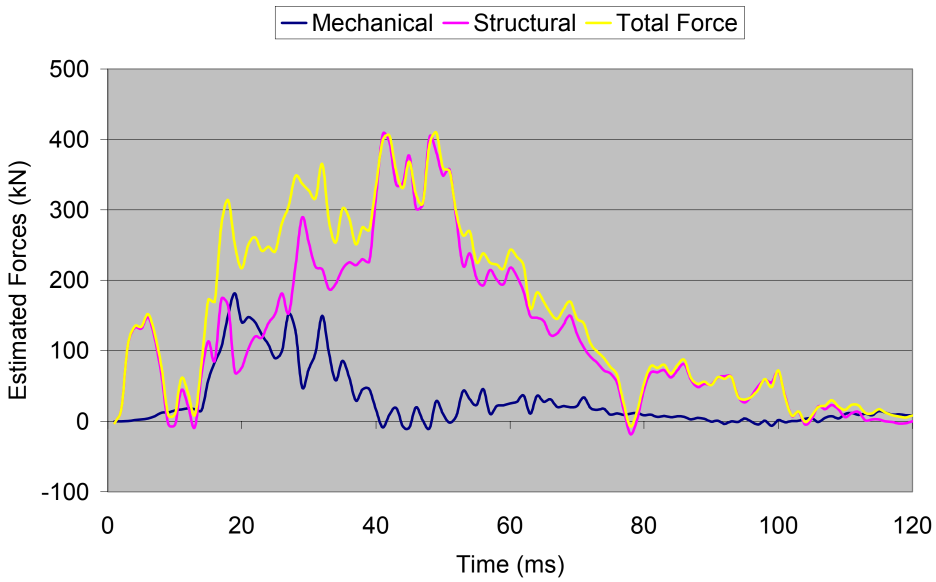


Figure 142: Toyota Yaris: calculated forces vs time

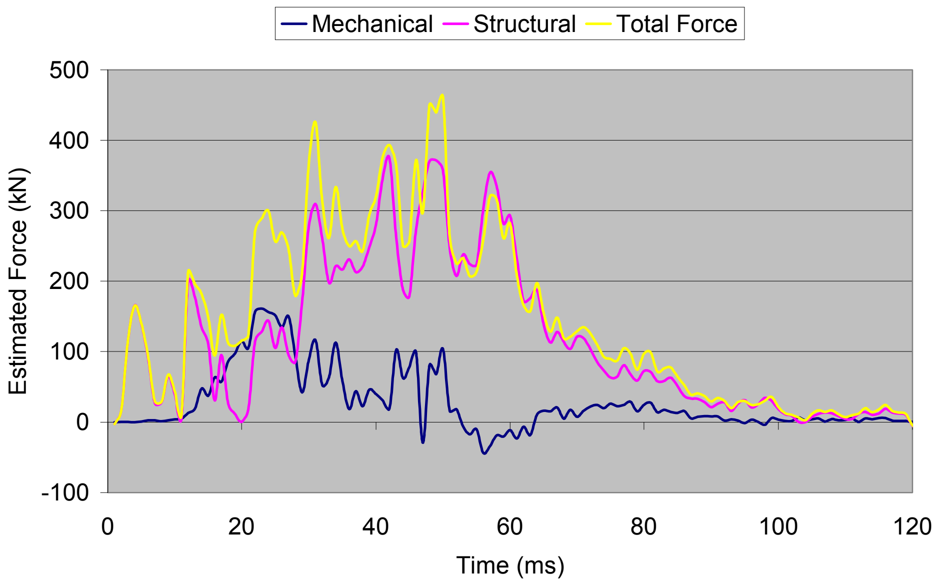


Figure 143: Renault Clio: calculated forces vs time

Comparison of the calculated forces vs. time and vs. B-pillars closing distance; in the second graph it can be seen that the maintenance load of the two compartments is about 360 kN (Figure 144 and Figure 145).

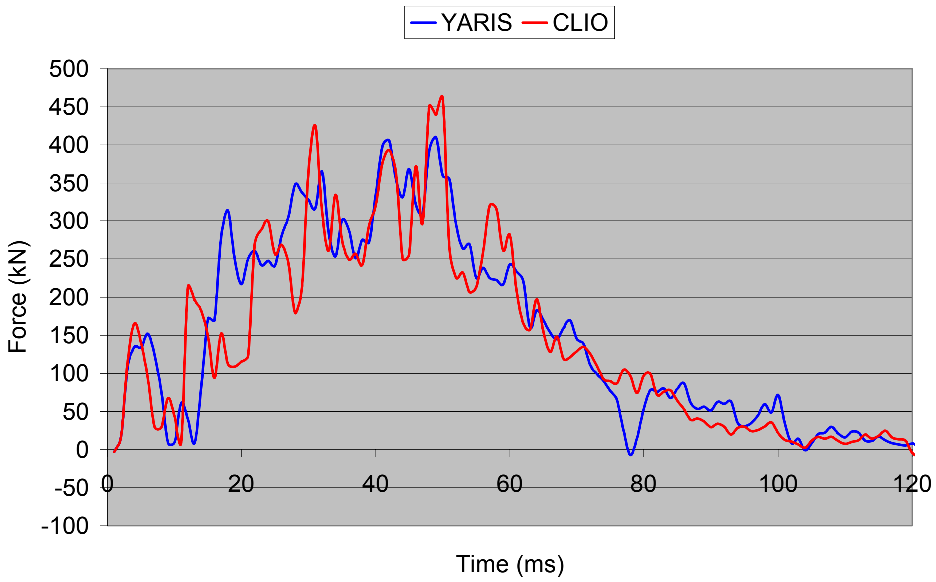


Figure 144: Yaris and Clio calculated forces vs time

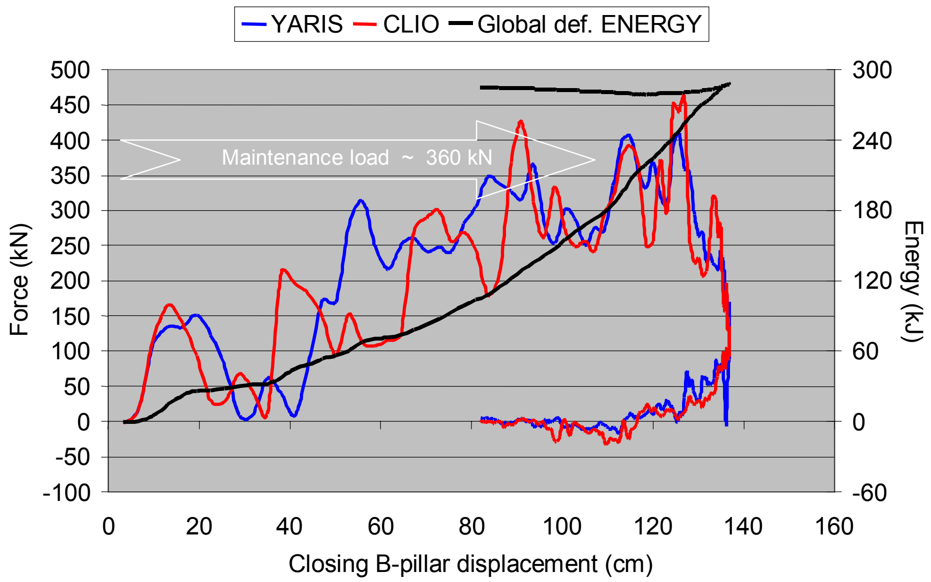


Figure 145: Yaris and Clio calculated forces vs closing B-pillar displacement and energy

Pre and post test images of the Toyota Yaris are shown (Figure 146 and Figure 147). In these photographs there is evidence of insufficient stability of the occupant compartment in a crash of this severity.

Figure 146: Toyota Yaris: pre and post test driver's side view

Figure 147 : Toyota Yaris: pre and post test driver's angled view

Pre and post test images of the Renault Clio are shown (Figure 148 and Figure 149). The images show the good stability of the Clio occupant compartment that received very low intrusion in the front part (it is possible to observe a deformation below the rear left wheel).

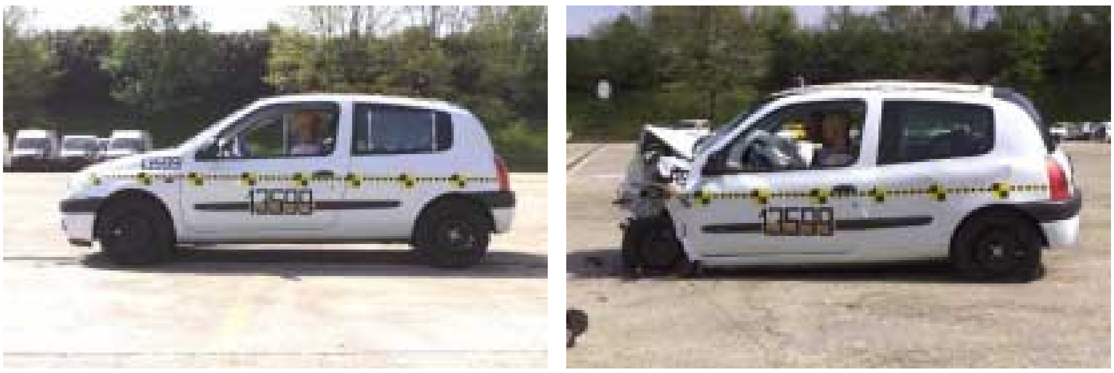


Figure 148: Renault Clio: pre and post test driver's side view

Figure 149: Renault Clio: pre and post test driver's angled view

As it said above, good interaction was expected between the longitudinals of both cars, due to good horizontal and vertical overlap.

Analysing the TOYOTA YARIS front-end, it is evident that the longitudinal did not collapse as designed; in fact, it moved upwards and this caused a fold of the rear part of the rail. The height of the longitudinal extremity was measured after the crash and it was verified an upwards movement of 90 mm; this measure was not the whole movement because, after the crash, the front tyre was deflated while, before the crash, it was at the correct pressure.

Figure 150: Toyota Yaris: left longitudinal after crash

It is possible to identify the impacts of some parts of the Renault Clio (Figure 151). The wheel impacted on the lower beam and interacted with the radiator behind it; whilst the longitudinal impacted against the lower part of the Toyota Yaris longitudinal and produced more rotation of its extremity.

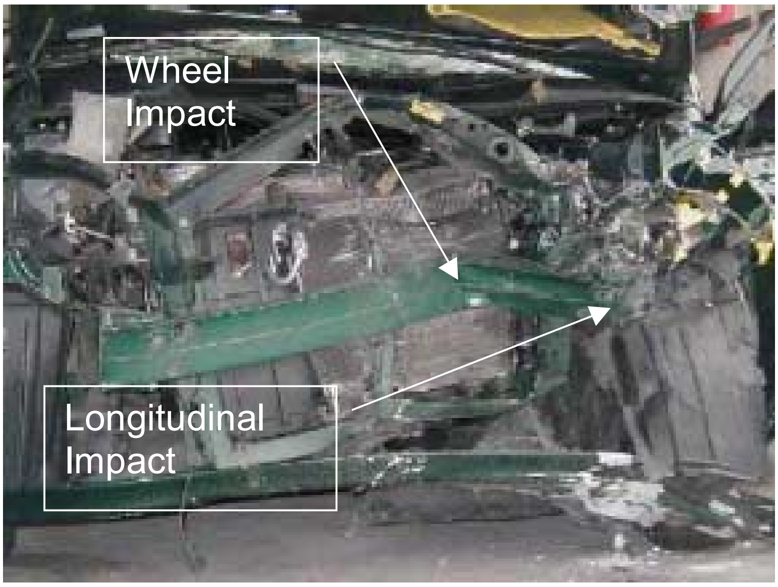


Figure 151: Toyota Yaris: front-end frontal view

The frontal view of the Renault Clio allowed the observation that the Clio suffered a more homogeneous deformation in comparison with the Yaris (Figure 152). Furthermore, it is possible to identify some parts of the Toyota Yaris that impacted against it. The Yaris longitudinal impacted just above the Clio longitudinal (good horizontal position, but bad vertical one) while the Yaris wheel collided against the lower beam and the sub-frame, cutting it (Figure 153).

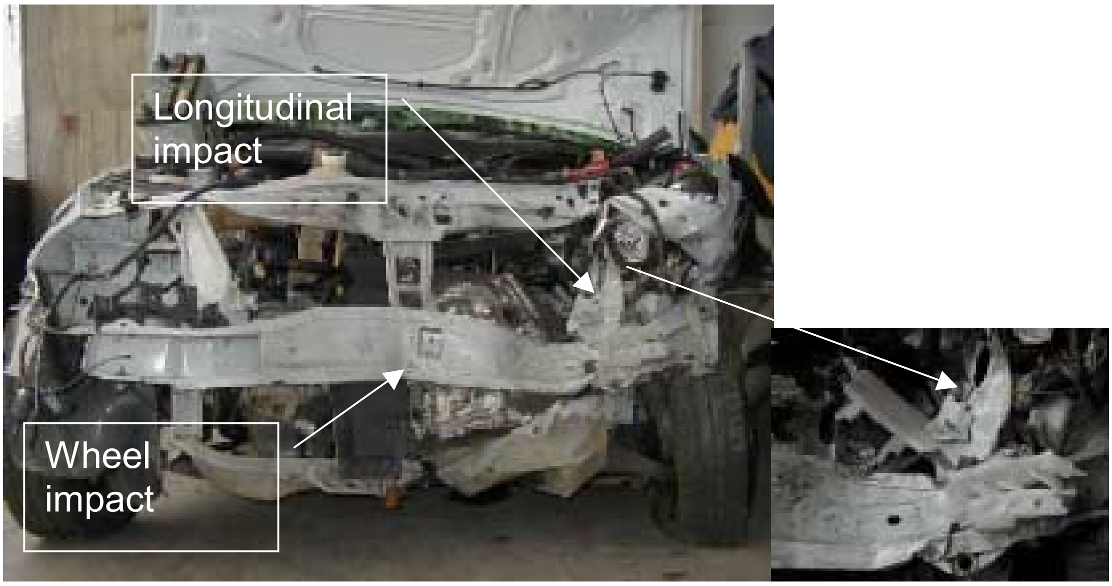


Figure 152: Renault Clio, frontal view showing Yaris longitudinal impact

Figure 153: Renault Clio: sub-frame cut from Yaris wheel

The deformation of the floor pan for both cars is shown (Figure 154 and Figure 155). It is possible to observe that the Toyota Yaris floor pan was totally deformed, not only in the left front part but also in the rear and the right part. Instead, the Renault Clio floor pan was not considerably deformed, considering the violence of crash.

Figure 154: Toyota Yaris, bottom view

Figure 155: Renault Clio, bottom view

The static intrusions measured in both cars are reported (Figure 156). The Toyota Yaris suffered intrusions approximately twice that for the Renault Clio. The average value of intrusions was calculated excluding the brake pedal movement.

Static intrusions	YARIS	CLIO
Dashboard	105	52
A-Pillar high level	144	57
A-Pillar low level	110	10
Footwell	215	128
Brake pedal	*179*	*91*
Average value	**144**	**62**

Figure 156: Measurements of static intrusions of compartment

There is the evidence of greater low level intrusions in the Toyota Yaris occupant compartment compared to the Clio, due to the crush of the lateral sill at the height of the front door (Figure 157 and Figure 158).

Figure 157: Toyota Yaris: Intrusions at low level of occupant compartment

Figure 158: Renault Clio: Intrusions at low level of occupant compartment

The difference in the driver dummy responses is generally consistent with the difference in the deformation/intrusion of the corresponding cars, i.e. the Yaris intrudes more than the Clio and generally has higher dummy injury criteria values (Table 49). The main inconsistency is that the chest deflection of the Renault Clio dummy is bigger than that of the Toyota Yaris dummy. Probably, this is the consequence of the large deformation of the Yaris floor pan, which caused the driver seat to rotate during the crash, resulting in the dummy chest not impacting the steering wheel correctly (Figure 159). Most dummy injury criteria values are less than regulatory limits, but there are some that exceed the limits.

Figure 159: Toyota Yaris: driver seat rotation

Table 49: Driver dummy injury criteria.

Injury Criterion		Regulation Limit	Toyota Yaris	Renault Clio
HEAD				
Peak acceleration	g	N/a	79.8	76.0
HIC_{36}		1000	820	397
3 ms exceedence	g	80	77	48
NECK				
Shear level max	kN	Varies with duration	0.25	0.44
Tension level max	kN	Varies with duration	2.78	1.93
Neck extension	Nm	-57	-29.2	-18.6
CHEST				
Compression	mm	50	37	47
Viscous criterion	m/s	1	0.43	0.36
3 ms exceedence	g	N/a	63	46
KNEE, FEMUR and PELVIS				
Left knee slide	mm	15	1.0	0.5
Left femur compression level max	kN	7.58 @≥10msec	6.1	0.9
Right knee slide	mm	15	1.3	0.6
Right femur compression level max	kN	7.58 @≥10msec	2.8	3.8
Pelvis acceleration	g	N/a	72	62
LOWER LEG				
Left tibia compression	kN	8	1.0	3.2
Left upper Tibia Index		1.3	1.2	1.2
Left lower Tibia Index		1.3	1.8	0.8
Right tibia compression	kN	8	1.8	3.6
Right upper Tibia Index		1.3	0.3	0.6
Right lower Tibia Index		1.3	0.8	0.7

Comparison with EuroNCAP tests

The Renault Clio and Toyota Yaris had a similar performance rating for the frontal impact tests for the EuroNCAP program: the Clio achieved a score of 11 points, the Yaris achieved a slightly higher score of 13 points.

Generally, an ODB test at 64 km/h is considered comparable with a car-to-car test at a closing speed of 112 km/h. The car-to-car test, between the Clio and Yaris, was performed with a closing speed of 118 km/h; consequently the total energy of the crash was increased by about 28 kJ. This amount of energy could be considered negligible.

The difference in the performance of the cars between the EuroNCAP tests and the car to car test is described below.

Unfortunately, for the EuroNCAP tests only a few static intrusion measurements are recorded (steering wheel, brake pedal and door aperture). In addition, the steering wheel movement is not usable in the comparison because, during the car-to-car crash, the Clio driver impacted against the steering wheel producing a reduction of the intrusion into the compartment.

Table 50: Comparison of Yaris and Clio intrusion measurements in EuroNCAP and Car to Car tests.

	EuroNCAP		**Car to Car**	
Static Intrusions	**YARIS**	**CLIO**	**YARIS**	**CLIO**
A-Pillar waist level	72	42	144	57
A-Pillar sill level	39	20	110	10
Brake pedal	-5	36	179	91

A comparison of the intrusion measurements in Table 50 shows that:

- Both in EuroNCAP and in the car-to-car test, the Yaris intrusions were larger than corresponding Clio values.
- Although both cars suffered higher intrusions in the car-to-car test compared to the EuroNCAP test, the increase of the Yaris intrusions was significantly higher than Clio values (from 1.8 to 5.5 times more).

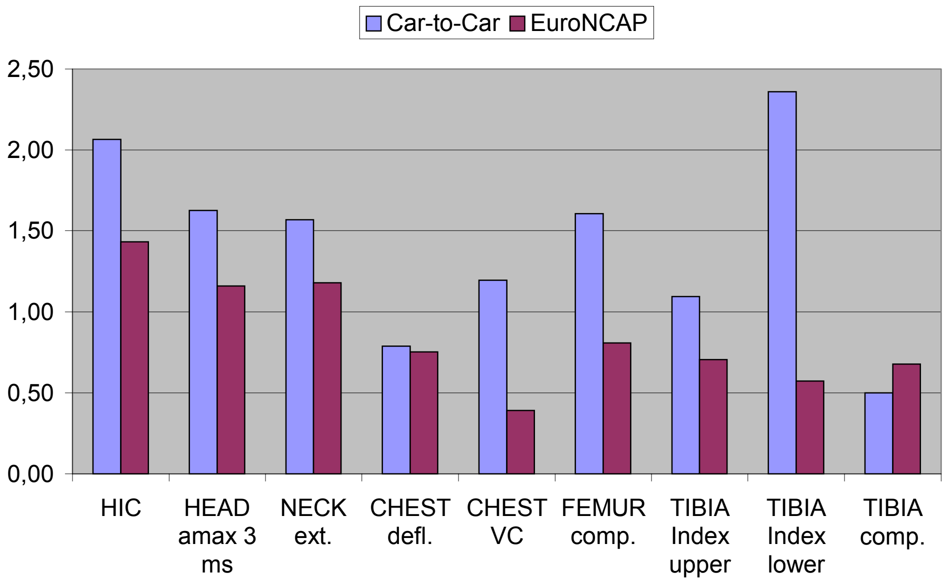


Figure 160: Comparison between Yaris and Clio driver dummy responses.

Figure 160 shows the Yaris driver dummy responses divided for correspondent Clio values (for lower leg parameters, the higher value of the right and left responses was used).

- For the EuroNCAP tests, most of the Clio dummy responses were bigger than those for the Yaris (except for Head and Neck ratios that are higher than 1).

- In contrast, for the car-to-car test, only the Chest deflection and Tibia compression were higher for the Clio driver dummy (Yaris/Clio ratios were less than 1).

The relative increase in the driver dummy responses between the EuroNCAP test and the car to car test was higher for the Yaris than the Clio (except for Tibia compression value). The maximum increase (4 times more) was registered for the lower Tibia Index.

Conclusions

- The Toyota Yaris has a design consisting of one main load path, the lower rails. The Renault Clio has a multi-level load path design. Examination of the cars prior to the test showed that there was good structural alignment between them, which indicated that good structural interaction between the lower rails might be expected. However, poor structural interaction was seen in the test caused by dynamic effects. When the Toyota Yaris lower rail impacted against the Renault Clio one, the Yaris lower rail bent upwards resulting in poor interaction with the Clio structure. As a result of this the Yaris occupant compartment intruded significantly and became unstable. The Clio compartment performed well without significant intrusion.

- A comparison of the relative performance of the Yaris and Clio in the car to car test and the EuroNCAP tests based on the intrusion measurements showed that the Clio performance was slightly worse in the car to car test compared to the

EuroNCAP test. In contrast the Yaris performance was significantly worse in the car to car test. It is believed that the main reason for this difference was the change in the structural performance of the Yaris, namely the longitudinal, caused by poor structural interaction, which in turn was a result of the Yaris having a design based on a single main load path.

6.5 EURONCAP 64 KM/H ODB IMPACT TESTS

6.5.1 Introduction

As mentioned in section 3, in order to overcome the stiffness aspect of compatibility, it is necessary to control frontal stiffness by limiting the force imposed by the car on its opponent, in an impact. A proposal is to measure the force imposed on a load cell wall behind the barrier face in a 64 km/h ODB test, such as the current EuroNCAP one. Limits would be placed upon this force measurement to ensure that the frontal stiffnesses of cars are matched. To help determine where to set these force limits and to assess whether the proposal is viable, test data for current cars is needed.

6.5.2 Test Objectives

The objective was to collect load cell wall data from a high resolution load cell wall located behind the offset deformable barrier for a range of cars used in the EuroNCAP frontal impact assessment. The load cell wall data was then analysed and interpreted as part of this project.

6.5.3 Test Details

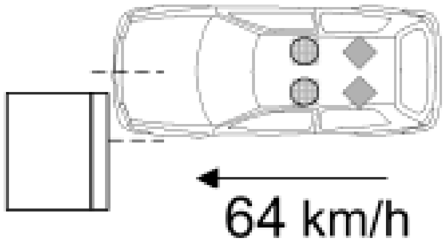
The respective vehicles where impacted into an EEVC frontal impact barrier with a 40% overlap. The barrier was mounted onto a high resolution load cell wall formed of 64 load cells of 125mm x 125mm arranged in an 8x8 matrix. The position of the load cell wall was such that 5 of the eight rows of load cells are centred behind the deformable element. Of the three remaining rows, two were above the barrier and one below (Figure 161).

Figure 161: Offset deformable barrier (ODB) mounted onto high resolution load cell wall.

Each of the test vehicles were instrumented with accelerometers at the base of the LHS and RHS B-Pillars. The acceleration measured by these accelerometers when multiplied by the mass of the occupant compartment provides an indication of the force seen by the occupant compartment during the impact, which has been described previously as the structural force. The contribution of the mechanical components, or mechanical force, is then assumed to be the difference between the measured load cell wall force and the calculated structural force. This method has limitations when analysing the various forces and therefore in later EuroNCAP tests, additional accelerometers have been added to the major mechanical components such as the engine and the gearbox. 09MF (Honda Civic Stream) and 35MF (Body on Frame SUV) have these additional accelerometers located on the engine top, engine sump and gearbox. The advantage of the additional accelerometers is that the contribution of the mechanical components to the force measured by the load cell wall can be calculated as opposed to derived. The masses used in the calculation of the structural and mechanical forces are generic masses estimated from the component mass proportions measured from other vehicles in its class. Each vehicle contained two instrumented HybridIII dummies in the front seats, and a P3 and P1.5 in the rear seats.

6.5.4 Renault Laguna II against Offset Deformable Barrier (ODB)

Date	20/10/00
Location	TRL
Topic Group	ODB Impacts
Mass Ratio	-
Test Number	27LF



Vehicle 1:	Renault Laguna II	Barrier:	EEVC Frontal
Impact Side:	Front		
Test Speed:	64.2km/h		
Overlap:	40%		
Test Mass:	1612 kg		
Dummies:	2 x HIII, P3 and P1.5		

Results and Discussion

The test speed and overlap were within the specified tolerances. The force recorded by the load cell wall, along with the mechanical and structural components of that force, is shown (Figure 162). The peak force was approximately 440kN at 1150mm displacement, which is around the average of the end of crash forces for this class of car.

At the peak load cell wall force of 440kN, the structural force was approximately 270kN, the difference between the two being the contribution of the mechanical force. The peak structural force of 320kN occurred later in the impact. Therefore the peak load cell wall force measured in this impact would not accurately represent the peak force seen by the occupant compartment. The load cell wall peak force would represent a combination of both the force seen by the occupant compartment and that being applied to the wall by the mechanical components of the vehicle, mainly the engine and transmission. The structural component contributed approximately 65% to the peak load cell wall force.

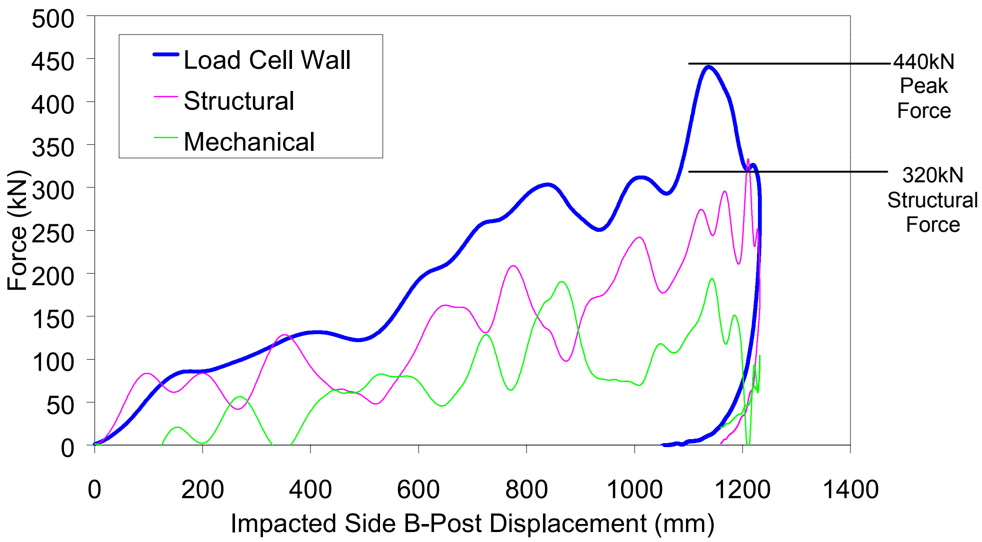


Figure 162: Total load cell wall force and its structural and mechanical components showing a peak force of 440kN at 1150mm displacement and an estimated end of crash structural force of 320kN.

The force distribution measured by the load cell wall was examined using the peak cell forces recorded during the impact (Figure 163). The red area towards the centre of the force distribution plot and along the RHS edge represents the high loads applied by the lower rail and the rotation of the car around the edge of the load cell wall towards the end of the impact. The peak force for one load cell in front of the lower rail was approximately 120kN whilst the rotation of the vehicle around the outer edge of the load cell wall produced a peak load of 41kN on one other load cell. The force applied by other structures was low in comparison and could not be clearly identified from the force distribution plot.

In comparison to vehicles of the same class tested previously at TRL, the impact of the lower rail produced a higher peak load, indicating a stiffer lower rail than in comparable vehicles. However, the force distribution was typical of many cars in that there were two main peaks, one generated by the lower rail impact and the other by the rotation of the car around the outer edge of the load cell wall. Post test observation of the car showed failure of the main cross beam, which indicated that there was little load transfer across the car via this load path. In addition, the subframe extension on the impacted side exhibited one failure in bending, which would have limited the energy absorption capability of this structure.

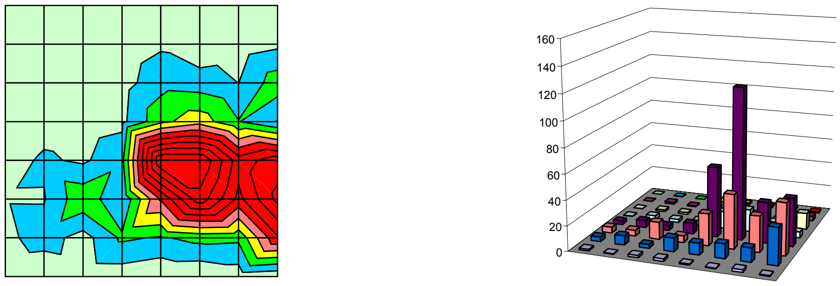
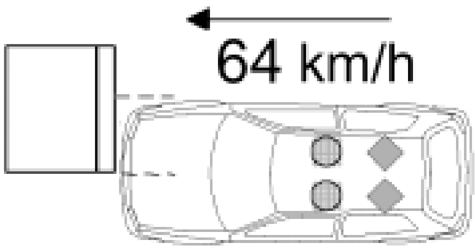


Figure 163: Force distribution based on peak cell forces showing the impact of the lower rails and the rotation of the car around the outer edge of the load cell wall as the red area.

6.5.5 Ford Mondeo against Offset Deformable Barrier (ODB)

Date	22/02/01
Location	TRL
Topic Group	ODB Impacts
Mass Ratio	-
Test Number	48LF



Vehicle 1:	Ford Mondeo	Barrier:	EEVC Frontal
Impact Side:	Front		
Test Speed:	64.2km/h		
Overlap:	40%		
Test Mass:	1612 kg		
Dummies:	2 x HIII, P3 and P1.5		

Results and Discussion

The test speed and overlap were within the specified tolerances. The force recorded by the load cell wall, along with the mechanical and structural components of that force, is shown (Figure 164). The peak force was approximately 400kN at 1150mm displacement, which is at the lower end of the range of end of crash forces for this class of car.

At the peak load cell wall force of 440kN, the structural force was approximately 300kN, the difference between the two being the contribution of the mechanical force. The peak structural force was approximately 300kN. Therefore the peak load cell wall force measured in this impact would not accurately represent the peak force seen by the occupant compartment. The load cell wall peak force would represent a combination of both the force seen by the occupant compartment and that being applied to the wall by the mechanical components of the vehicle, mainly the engine and transmission. The structural component contributed approximately 75% to the peak load cell wall force.

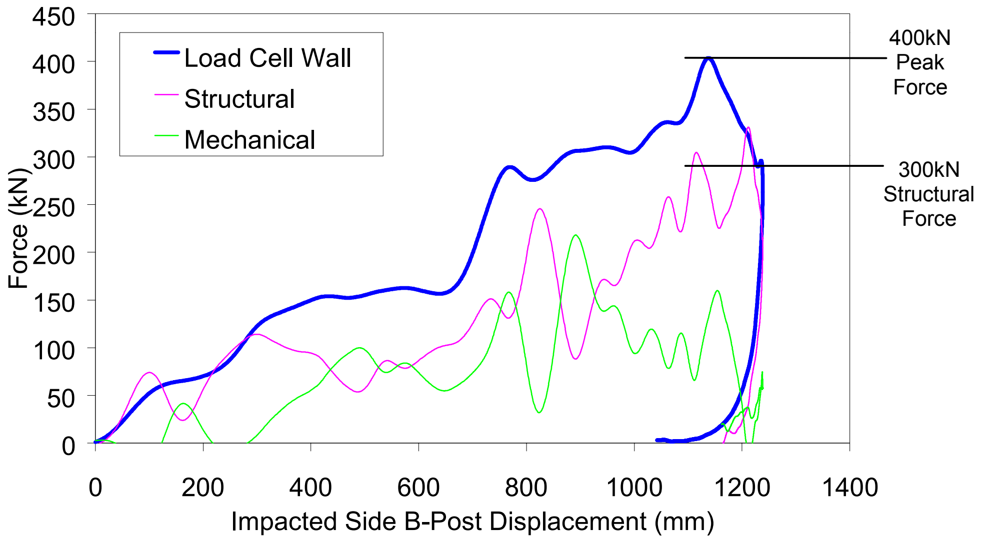


Figure 164: Total load cell wall force and its structural and mechanical components showing a peak force of 400kN at 1150mm displacement and an estimated end of crash structural force of 300kN.

The force distribution measured by the load cell wall was examined by using the peak cell forces recorded during the impact (Figure 165). The red area towards the lower LHS of the force distribution plot represents the high loads applied by the lower crossbeam and the rotation of the car around the edge of the load cell wall towards the end of the impact. The peak force on the load cell wall in front of the lower rail was approximately 50kN whilst the rotation of the vehicle around the outer edge of the load cell wall produced a peak load of 90kN on one load cell.

In comparison to vehicles of the same class tested previously at TRL, the impact of the lower rails was closer to the inside edge of the load cell. This was due to the lower rail bending inwards, prior to contacting the load cell wall. In addition, the force applied by the vehicle as it rotated around the edge of the load cell wall lacked the typical vertical distribution seen with other vehicles. This was due to the vertical profile of the engine at the point where it lined up with the load cell wall, along with its subsequent rotation during the impact. This meant that the load applied by the engine was focused at the same height as the vehicle crossbeam. Therefore, the crossbeam would have been the main point of contact between the car and the load cell wall edge at that point in the impact.

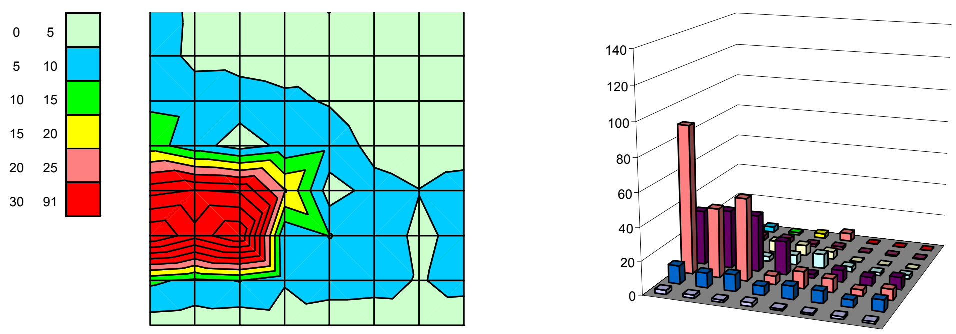
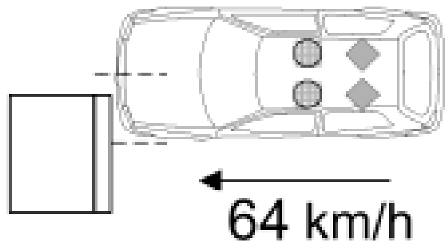


Figure 165: Force distribution based on peak cell forces showing the impact of the lower rails and the rotation of the car around the outer edge of the load cell wall as the red area.

6.5.6 Volvo S60 against Offset Deformable Barrier (ODB)

Date	01/03/01
Location	TRL
Topic Group	ODB Impacts
Mass Ratio	-
Test Number	49LF



Vehicle 1:	Volvo S60	Barrier:	EEVC Frontal
Impact Side:	Front		
Test Speed:	64.3km/h		
Overlap:	40%		
Test Mass:	1669 kg		
Dummies:	2 x HIII, P3 and P1.5		

Results and Discussion

The test speed and overlap were within the specified tolerances. The force recorded by the load cell wall, along with the mechanical and structural components of that force, is shown (Figure 166). The peak force was approximately 475kN at 1100mm displacement, which is typical for this class of car. The structural force at the peak load cell wall force was approximately 400kN. As in the previous test (48LF) the use of the peak force would represent contribution from both the structural and mechanical components of the vehicle. In this test, the structural force was approximately 80% of the peak force.

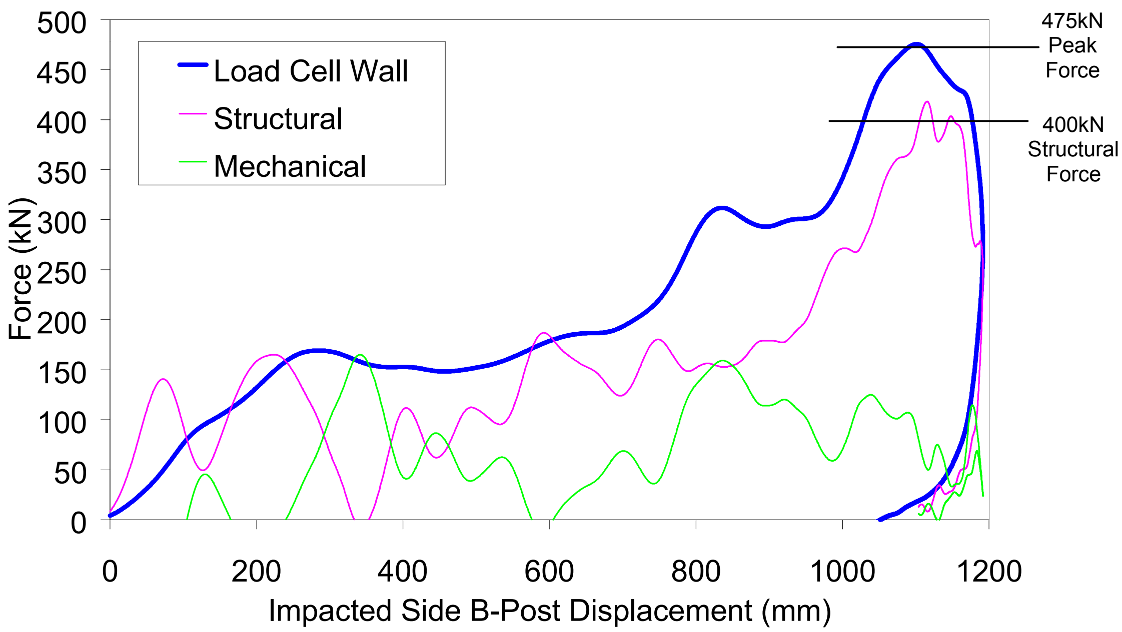


Figure 166: Total load cell wall force and its structural and mechanical components showing a peak force of 475kN at 1100mm displacement and an estimated end of crash structural force of 400kN.

The force distribution measured by the load cell wall was examined using the peak cell forces recorded during the impact (Figure 167). The force distribution shows the impact of the lower rail as the red area towards the centre of the plot. The rotation of the vehicle around the outer edge of the load cell wall can also clearly be seen as the red area along the right hand side of the plot. The vertical spread of the force along the right hand side edge of the wall indicates that the contact between the load cell wall and the vehicle during the rotation would have been through the engine. This can be compared with test 48LF (Figure 165), in which the point of contact between the vehicle and the load cell wall during rotation was the front crossbeam. The peak individual load cell force recorded in front of the lower rail was 41kN.

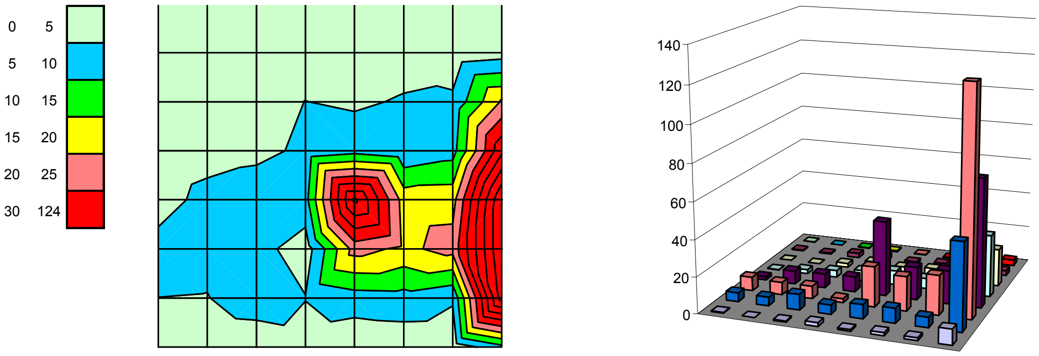
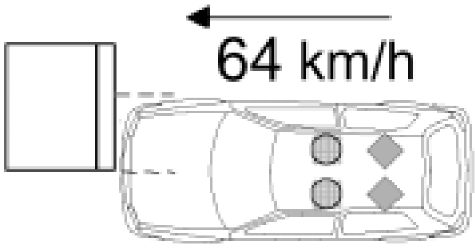


Figure 167: Force distribution based on peak cell forces showing the impact of the lower rail as the red area towards the centre of the plot and the rotation of the vehicle around the outer edge of the load cell wall as the red area along the RHS edge.

6.5.7 Fiat Multipla against Offset Deformable Barrier (ODB)

<table>
<tr>
<td rowspan="2">Date 30/03/01
Location TRL
Topic Group ODB Impacts
Mass Ratio -
Test Number 50LF</td>
<td colspan="2">
</td>
</tr>
<tr>
<td>Vehicle 1: Fiat Multipla
Impact Side: Front
Test Speed: 64.3km/h
Overlap: 40%
Test Mass: 1730 kg
Dummies: 2 x HIII, P3 and P1.5</td>
<td>Barrier: EEVC Frontal</td>
</tr>
</table>

Results and Discussion

The test speed and overlap were within the specified tolerances. The force recorded by the load cell wall, along with the mechanical and structural components of that force, is shown (Figure 168). The peak force was approximately 475kN at 910mm displacement, which is typical for this class of car.

The structural force towards the end of the impact was approximately 330kN. In this test, the occupant compartment of the vehicle was judged to have become unstable and the structural force at the end of the impact, from 900mm displacement, could be considered to be an accurate indication of the maximum occupant compartment strength when loaded in this manner. The force applied by the mechanical components, mainly the engine and transmission, has, allowing for test tolerances, reached zero at 1000mm. In order to achieve this, the engine and transmission would have had to bottom out the barrier and impacted the load cell wall and this can be seen by the large peak in the mechanical force at approximately 810mm. It is this bottoming out of the engine and transmission on the load cell wall that has resulted in the high peak load cell wall force in relation to the structural force. The structural force at the time of the peak force was approximately 55% of the global force. However, the actual occupant compartment strength was 70% of the peak global force.

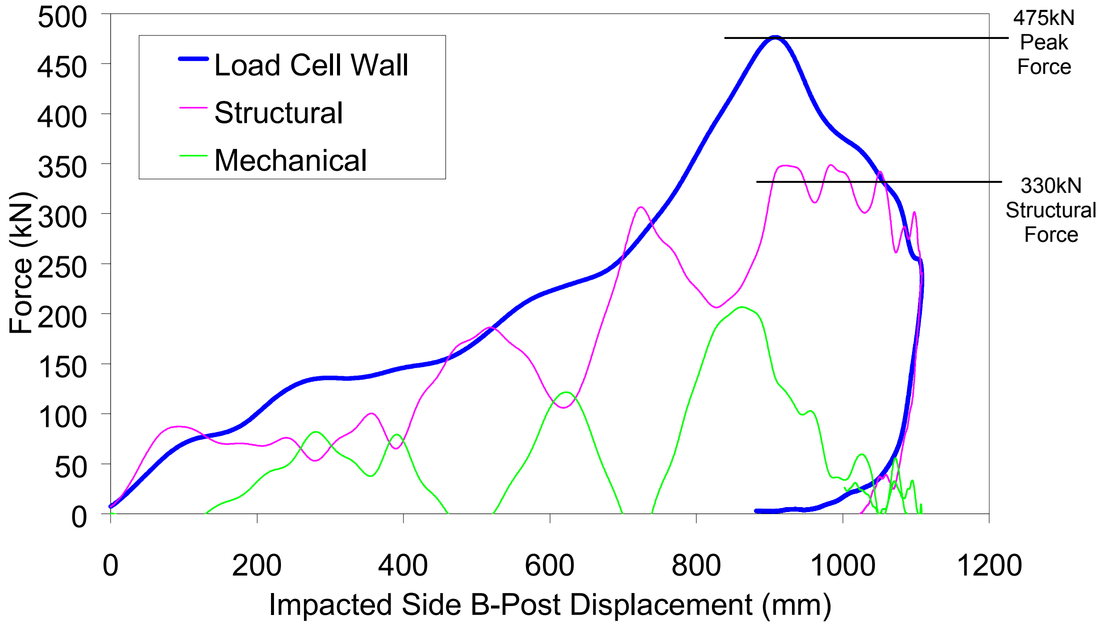
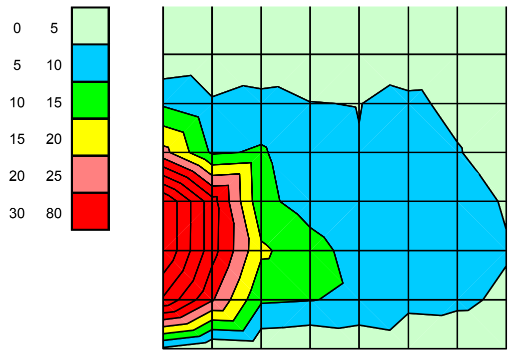


Figure 168: Total load cell wall force and its structural and mechanical components showing a peak force of 475kN at 910mm displacement and an estimated end of crash structural force of 330kN. The large increase in the mechanical force at 810mm indicates the impact of the engine onto the load cell wall.

The force distribution based on the peak cell forces recorded during the impact is shown (Figure 169). The high forces, shown in the plot as the red area, are all concentrated towards the edge of the load cell wall. The vertical spread of these high forces indicates that they are primarily due to the impact of the engine and the subsequent rotation of the vehicle around the outer edge of the wall. There was indication of loading by the lower rail early on in the impact. However the force applied by the lower rail was low in comparison to that applied by the engine. This low force could indicate a less stiff lower rail in comparison to the previous three tests, 27LF, 48LF and 49LF.



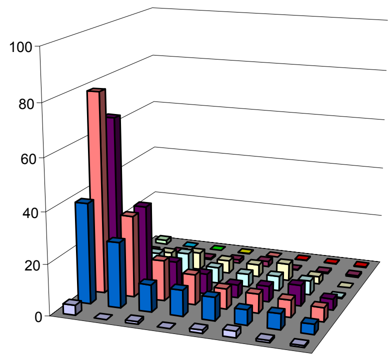
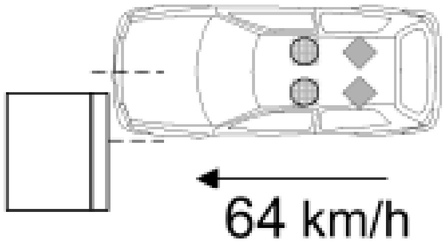


Figure 169: Force distribution based on peak cell forces showing the impact of the engine and the subsequent rotation of the vehicle around the edge of the load cell wall as the red area along the LHS side.

6.5.8 Peugeot 307 against Offset Deformable Barrier (ODB)



Date	26/07/01	Vehicle 1:	Peugeot 307	Barrier:	EEVC Frontal
Location	TRL	Impact Side:	Front		
Topic Group	ODB Impacts	Test Speed:	64.2km/h		
Mass Ratio	-	Overlap:	40%		
Test Number	07MF	Test Mass:	1481 kg		
		Dummies:	2 x HIII, P3 and P1.5		

Results and Discussion

The test speed and overlap were within the specified tolerances. The force recorded by the load cell wall, along with the mechanical and structural components of that force, is shown (Figure 170). The peak force was approximately 420kN at 1185mm displacement, which is typical for this class of car. However, this force was only applied for a very short displacement, due to a large increase in the mechanical force. In this test, it can be observed that the structural force contributed to a high proportion of the overall global force for much of the impact. However, due to the sharp peak force, the structural component of the global peak force was only 75%.

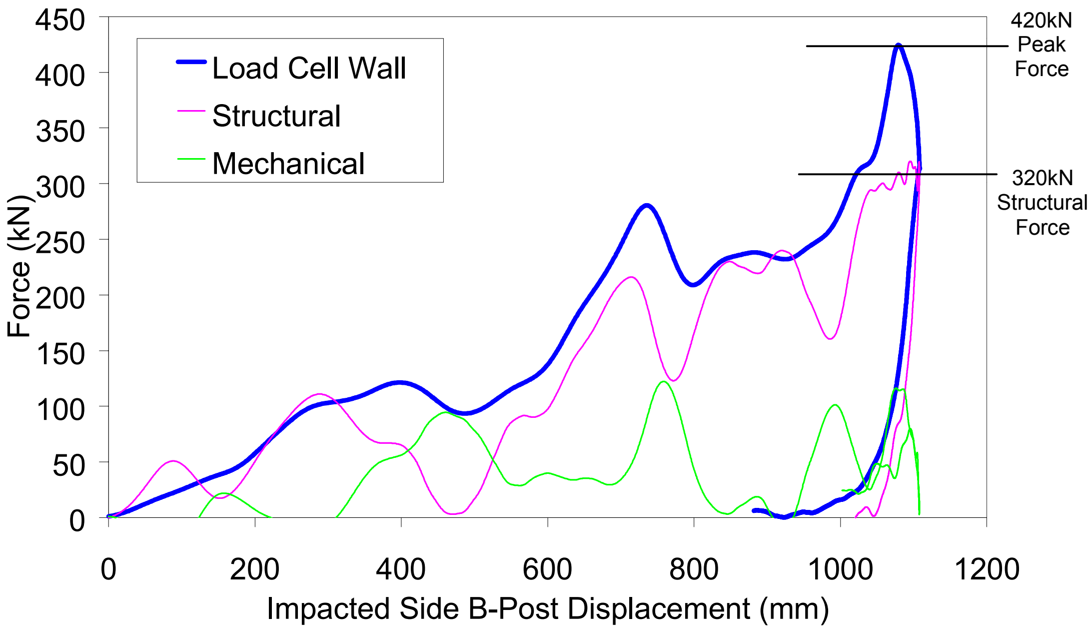


Figure 170: Total load cell wall force and its structural and mechanical components showing a peak force of 420kN at 1185mm displacement and an estimated end of crash structural force of 320kN.

The force distribution, based on the peak load cell forces is shown (Figure 171). This figure shows two significant areas of high force. The red area towards the centre of the plot was the impact of the lower rail, whilst the red area along the RHS edge was due to the rotation of the vehicle around the outer edge of the load cell wall. The point of contact as the vehicle rotated was the engine block, hence the significant vertical distribution in comparison to 48LF in which the point of contact was the vehicle's crossbeam. It is this force, applied by the engine at the end of the impact, that resulted in a peak load cell wall force that was significantly higher than the structural force. Observation of the force distribution through time shows the continual rise and fall of the force applied by the lower rail as it fails. This failure mode of the lower rails can clearly be seen by the three dips at 390mm, 780mm and 990mm when looking at the structural force. Another observation was the height of the lower rail impact, which was significantly higher than the lower rail impacts observed in the previous tests.

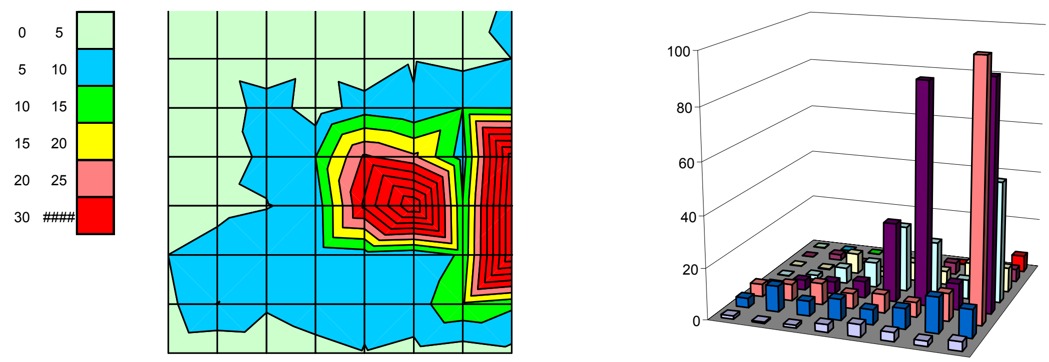
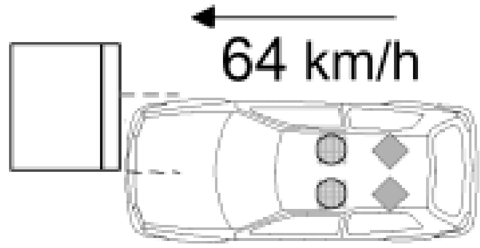


Figure 171: Force distribution based on peak cell forces showing the impact of the lower rail and the high force applied by the engine as the vehicle rotated around the edge of the load cell wall.

6.5.9 Honda Civic Stream against Offset Deformable Barrier (ODB)

Date	19/09/01
Location	TRL
Topic Group	ODB Impacts
Mass Ratio	-
Test Number	09MF



Vehicle 1:	Honda Civic Stream	Barrier:	EEVC Frontal
Impact Side:	Front		
Test Speed:	64.1km/h		
Overlap:	40%		
Test Mass:	1662 kg		
Dummies:	2 x HIII, P3 and P1.5		

Results and Discussion

The test speed and overlap were within the specified tolerances. The force recorded by the load cell wall, along with the mechanical and structural components of that force, is shown (Figure 172). The peak force was approximately 460kN at 1160mm displacement, which is typical for this class of car. The peak structural force was 429kN and occurred earlier in the impact at 980mm displacement. However, the average structural force at the end of the impact was only 360kN. Therefore, the structural force at the end of the crash represented only 80% of the peak global force.

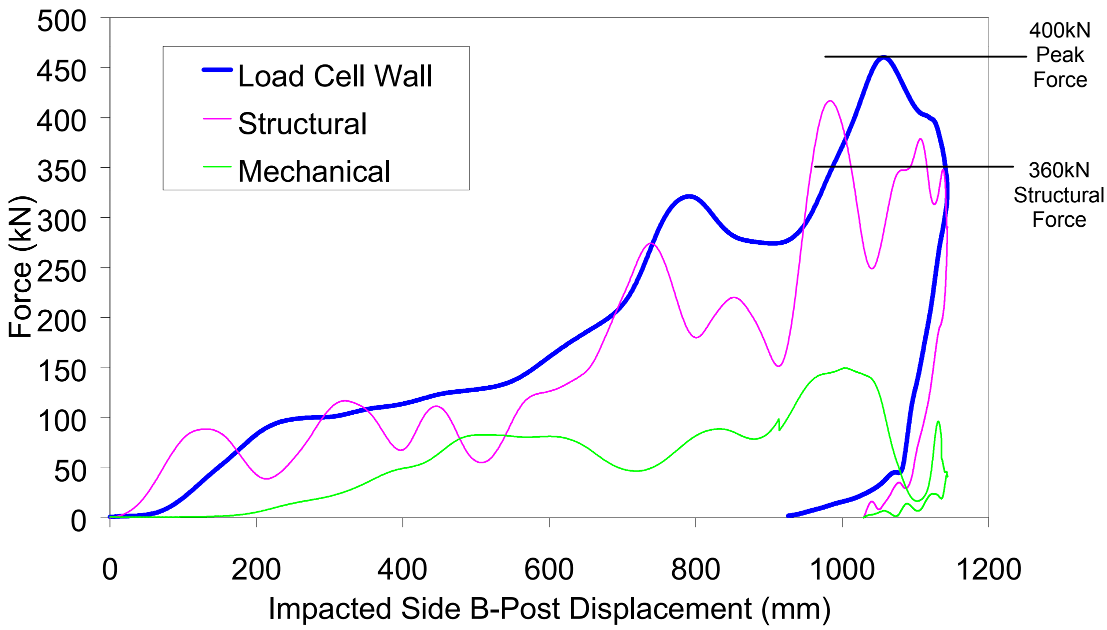


Figure 172: Total load cell wall force and its structural and mechanical components showing a peak force of 460kN at 1160mm displacement and an estimated end of crash structural force of 360kN.

The force distribution based on the peak individual load cell forces is shown (Figure 173). The red area towards the centre of the plot shows the impact of the lower rail, whilst the red area along the LHS edge is due to the rotation of the vehicle around that outer edge of the load cell wall towards the end of the impact. The individual cell load in front of the lower rail impact was 55kN, whilst the rotation of the car around the outer edge of the load cell wall produced a maximum cell force of 56kN.

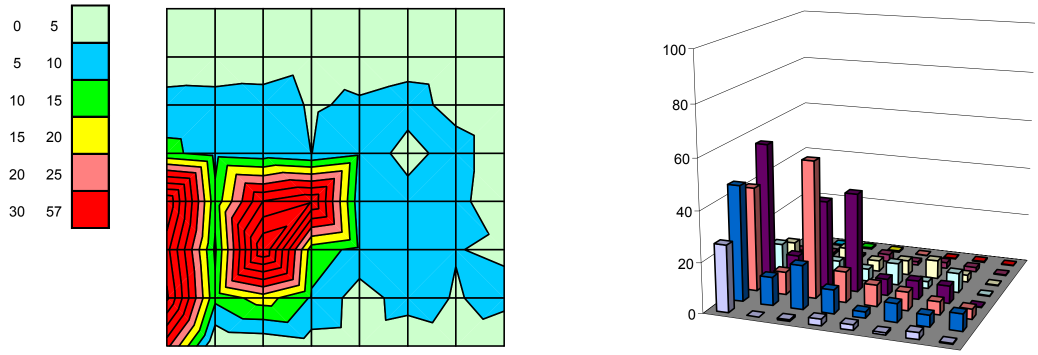
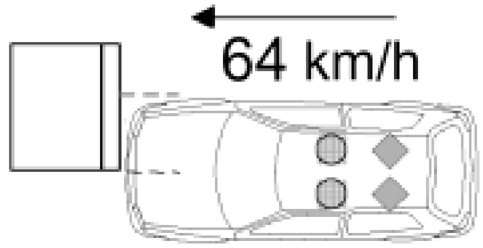


Figure 173: Force distribution based on peak cell forces

6.5.10 Body on Frame SUV against Offset Deformable Barrier (ODB)

Date	28/02/02
Location	TRL
Topic Group	ODB Impacts
Mass Ratio	-
Test Number	35MF



Vehicle 1:	SUV (Body on Frame)	Barrier:	EEVC Frontal
Impact Side:	Front		
Test Speed:	64.2km/h		
Overlap:	40%		
Test Mass:	2060 kg		
Dummies:	2 x HIII, P3 and P1.5		

Results and Discussion

The test speed and overlap were within the specified tolerances. The force recorded by the load cell wall, along with the mechanical and structural components of that force, are shown (Figure 174). The peak force was approximately 490kN at 680mm displacement. This is a relatively low displacement in comparison to typical cars, although the peak force was similar to that seen for a large family car.

The force displacement plot shows a large increase in the global force at about 600mm displacement. This was due to a large increase in the structural force, with the mechanical loading not occurring till later in the impact at about 800mm displacement. The reason for this is the body on frame construction of the test vehicle. The longitudinal rails of the frame, which run the entire length of the vehicle, require a far higher loading to initiate failure than the lower rails found in a typical car. These lower rails penetrated the barrier and impacted the load cell wall behind. This bottoming out was responsible for the large rise in the load cell wall force, beginning at approximately 600mm displacement and the peak force of 490kN. The load applied by the mass of the occupant compartment was initially not high enough to initiate failure in the longitudinal rails of the frame. This resulted in less crush forward of the occupant compartment and the resultant increase in the structural force at a relatively low displacement of 680mm. This sudden loading of the occupant compartment may have been responsible for the observed oscillation of the structural force between 200kN and 600kN from 700mm and 950mm. The load cell wall force does not replicate this oscillation. The occupant compartment force assessed from the load cell wall force was determined to be 300kN and therefore approximately 61% of the peak global force.

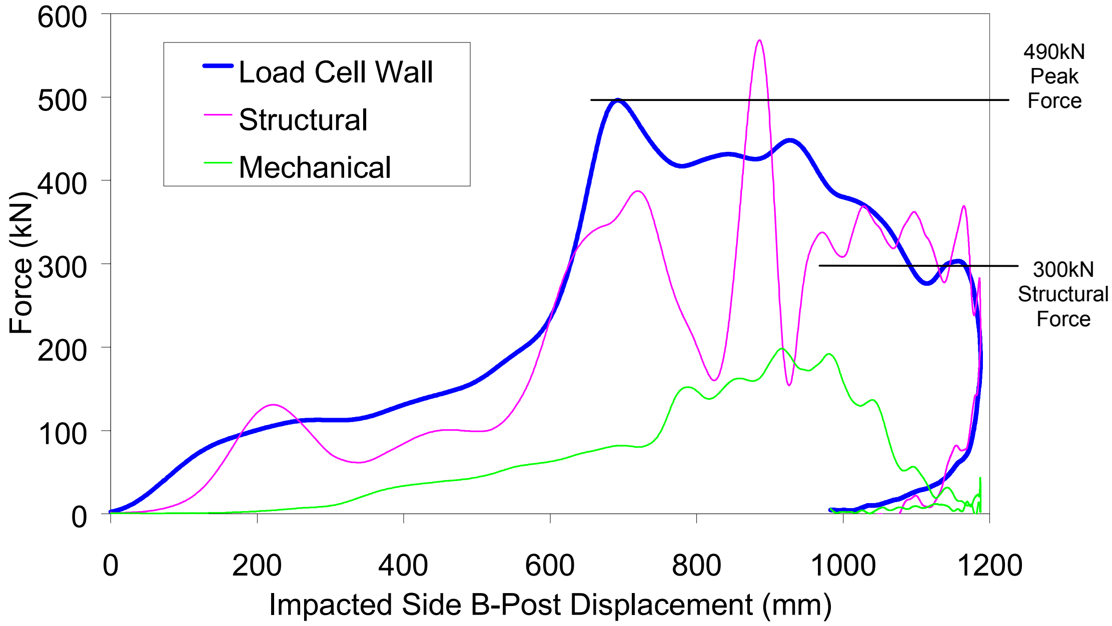


Figure 174: Total load cell wall force and its structural and mechanical components showing a peak force of 490kN at relatively low displacement of 680mm and an estimated end of crash structural force of 300kN.

The force displacement plot based on the peak cell forces is shown (Figure 175). Due to the body on frame design, the only significant force applied was directly in front of the LHS longitudinal rail of the frame, with a peak force of 199kN on an individual load cell. Away from the impact of the lower rail, the force distribution was typically 10kN per load cell in the area of the vehicle impact. In this test, the force on the outer row was lower than for the previous car tests, with a peak force of only 29kN.

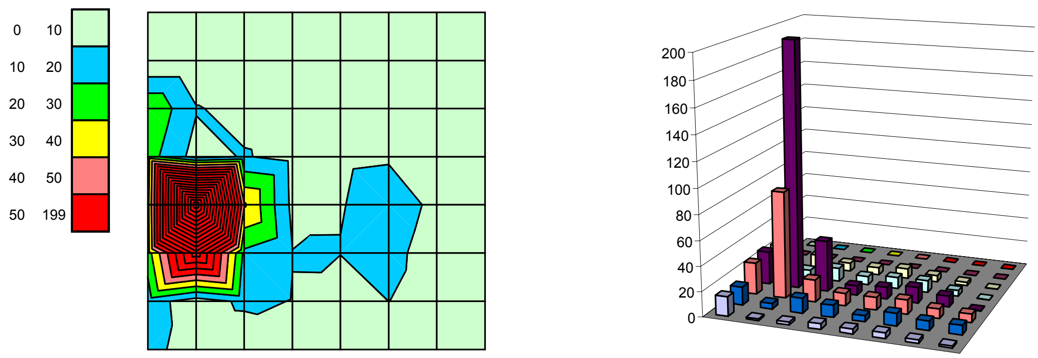


Figure 175: Force distribution based on peak cell forces showing the impact of the RHS lower rail.
Note: Due to the high peak forces, the scale has been altered in comparison to the previous tests.

6.5.11 EuroNCAP 64 km/h ODB IMPACT TESTS at BASt

In the Euro NCAP tests carried out by BASt, besides the measurements taken according to the Euro NCAP test protocol in force, further measurements are taken. These are the forces behind the deformable element and accelerations of structures of the car front, which can be considered as remaining undeformed during the impact test. The arrangement of force transducers behind the deformable element is shown (Figure 176). The measuring area of each force transducer is 125 x 125 mm.

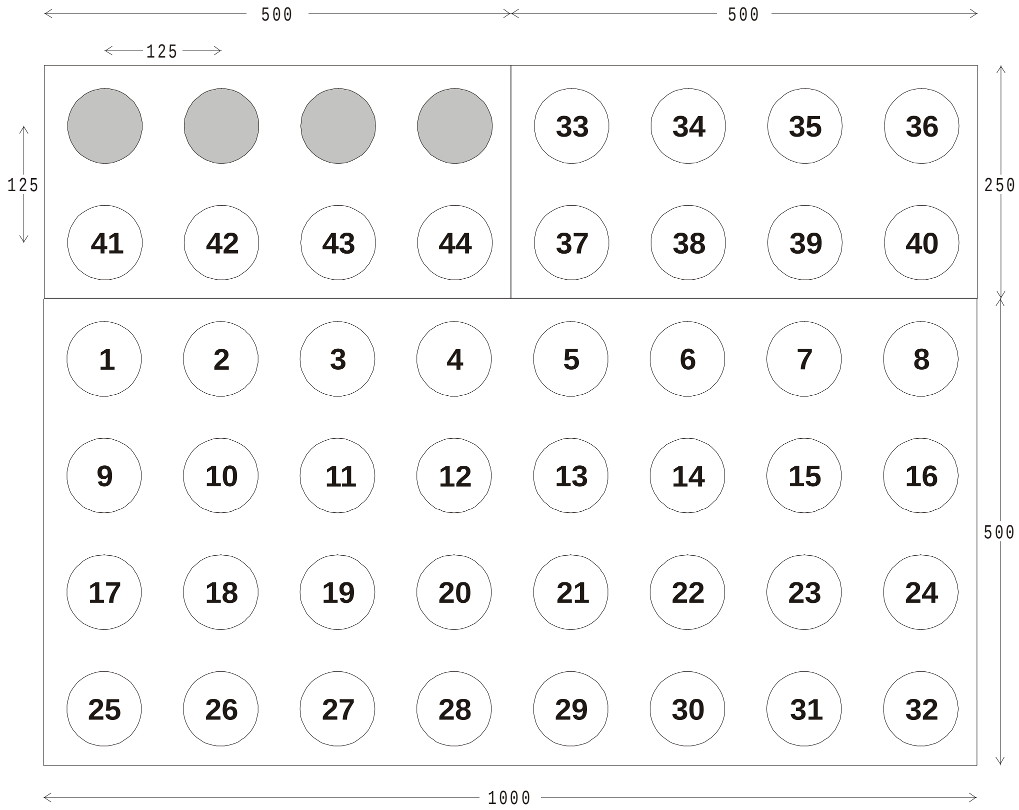


Figure 176: Arrangement of force transducers of the load cell wall

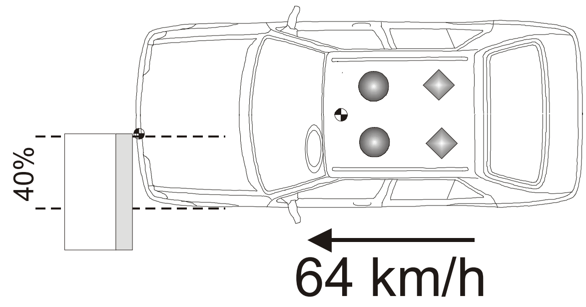
The deformable element mounted to force measuring wall is shown (Figure 177). The lowest row of the force transducers fits to the lower edge of the deformable element.

Figure 177: Euro NCAP Frontal Impact Barrier mounted to the Load Cell Wall

The force measurements of three car models are presented: Mazda MX5, Mercedes SLK and Mercedes C-class. The Mazda MX5 and the Mercedes SLK are sports cars the Mercedes C-class is an upper middle class car type.

6.5.12 Mazda MX 5 against Offset Deformable Barrier (ODB)

Date	07.03.2002
Location	BASt
Topic Group	ODB Impacts
Mass Ratio	
Test Number	**EUMD05FO**



Vehicle 1:	Mazda MX5	Barrier:	EEVC Frontal
Impact Side:	Front		
Test Speed:	64km/h		
Overlap:	40%		
Test Mass:	1245 kg		
Dummies:	LHS – Hybrid III RHS – Hybrid III		

The test speed and overlap were within the specified tolerances. The vehicle acceleration at B-post on impact side against time is shown (Figure 178). The peak deceleration amounts to about 32 g, which is a normal high peak value.

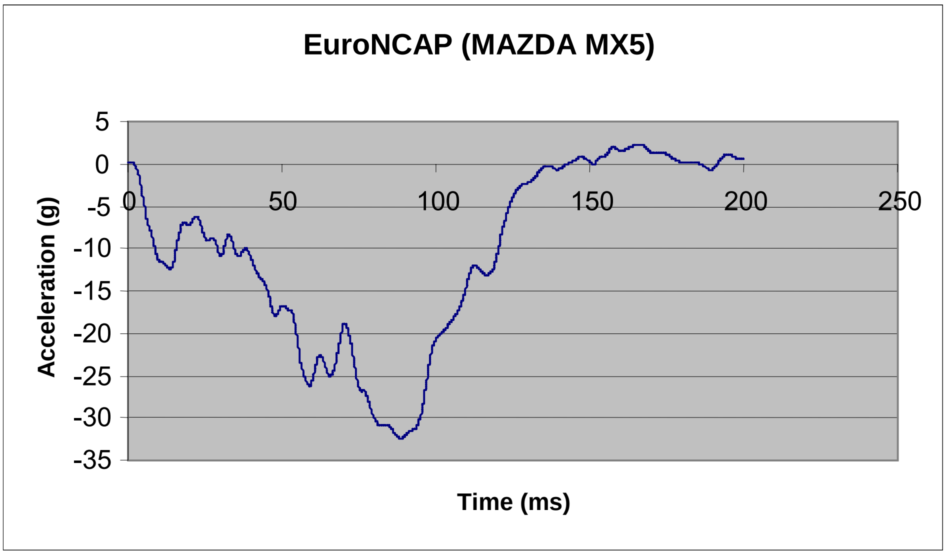


Figure 178: Vehicle Acceleration at B-Post on Impact Side Against Time

The next plot shows the comparison of the measured load cell force (LCW) behind the deformable element and the impact force of the Mazda calculated by multiplying the vehicle mass (including the engine) with the impact side B-post acceleration versus time (Figure 179). Both traces show similar characteristics, so together with the further measurements at different front components the measured signals seem to be suitable for future assessment approaches in EuroNCAP tests concerning vehicle compatibility

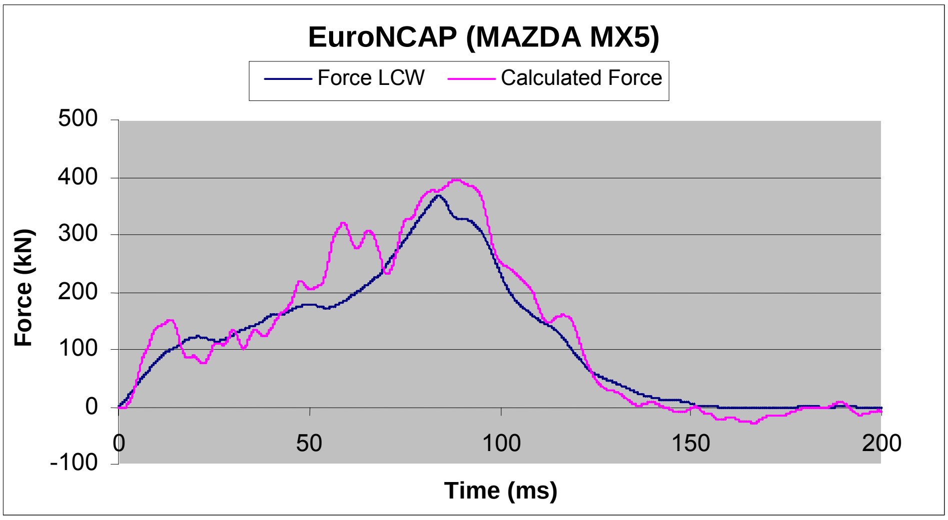


Figure 179: LCW Force and Calculated Force against Time

Towards the end of the impact the peak forces measured at the load cell wall increase substantially to about 370 kN (Figure 180). The relatively high final strength of the vehicle front is apparently the result of a front structure design to keep the compartment intrusions low as the good EuroNCAP (4 stars) assessment proves.

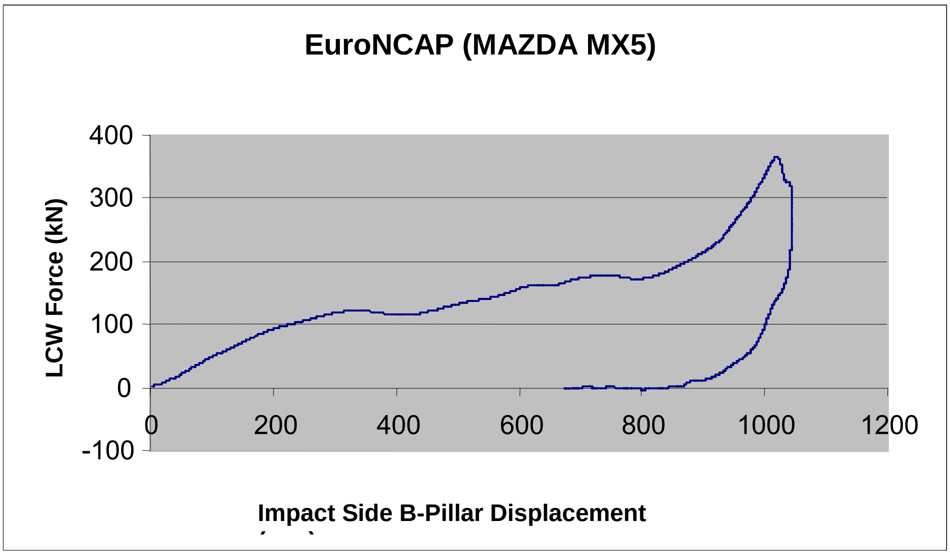


Figure 180: LCW Force against B-Pillar Displacement (mm), Test EUMD05FO

The peak load cell forces are mainly caused by the engine impact as the distribution of the peak forces illustrates (Figure 181). The impact forces caused by the engine are nearly ten times higher than the forces caused by the vehicle front structures. The relatively soft front structures on the one side and the small engine block show that this vehicle will apparently not show a compatible behaviour in car to car accidents.

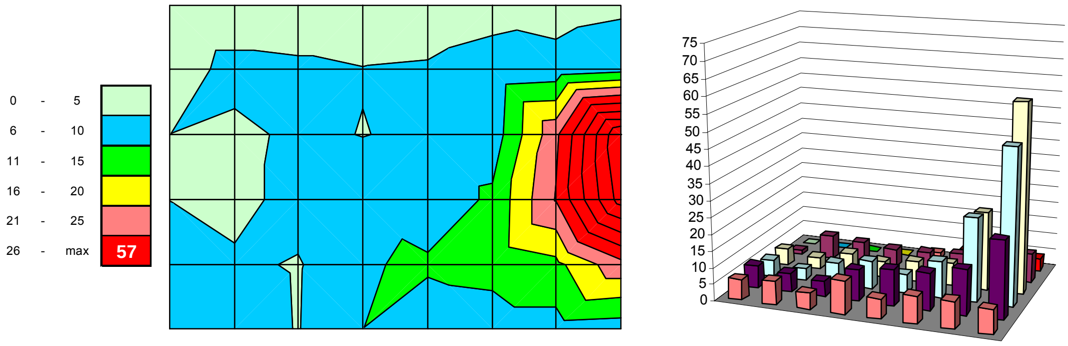
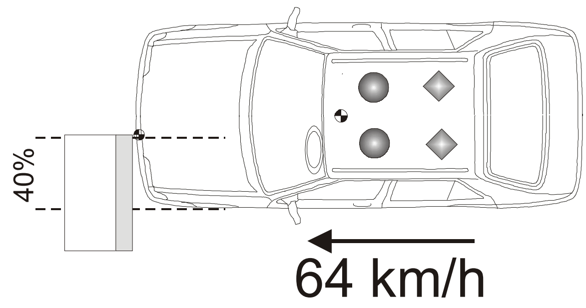


Figure 181: Force Distribution based on Peak Cell Forces, Test EUMD05FO

Mercedes SLK against Offset Deformable Barrier (ODB)

Date	15.03.2002
Location	BASt
Topic Group	ODB Impacts
Mass Ratio	
Test Number	**EUMC06FO**



Vehicle:	Mercedes SLK	Barrier:	EEVC Frontal
Impact Side:	Front		
Test Speed:	64km/h		
Overlap:	40%		
Test Mass:	1549		
Dummies:	LHS – Hybrid III RHS – Hybrid III		

The test speed and overlap were within the specified tolerances. The vehicle acceleration at B-post on impact side against time is shown (Figure 182). The peak deceleration amounts to about 54 g, which is an unusually high peak value. This high value is mainly caused by the intended direct force transmission of the left front wheel to the impact side door sill.

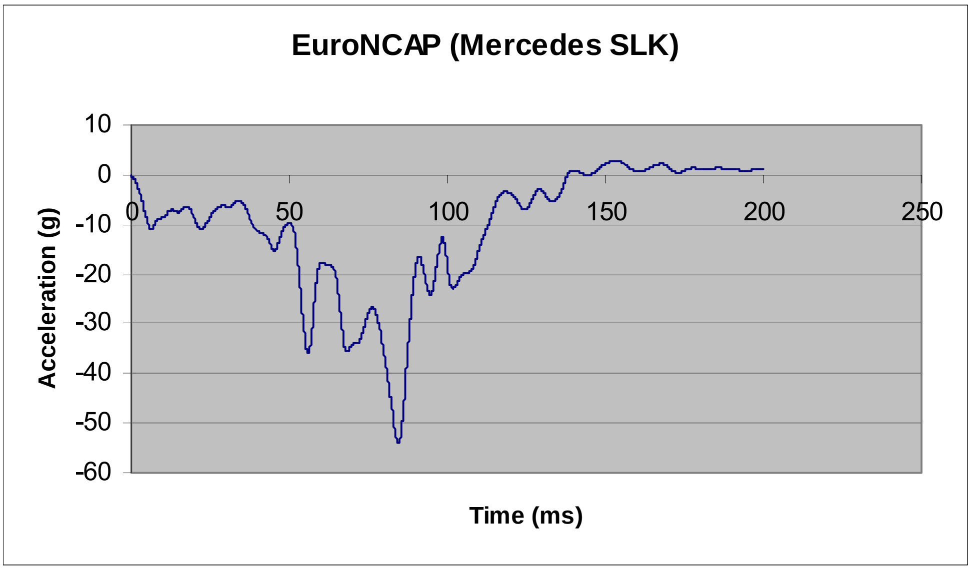


Figure 182: Vehicle Acceleration on Impact Side against Time

The next plot, shows the comparison of the measured load cell force (LCW) behind the deformable element and the impact force of the Mercedes SLK calculated by multiplying the vehicle mass (including the engine) with the impact side B-post acceleration versus time (Figure 183). Due to the above described load transmission through the door sill to the bottom of the B-post, where the impact side vehicle acceleration is normally measured, both traces can not be easily analysed together with the further measurements at different front components to assess vehicle compatibility

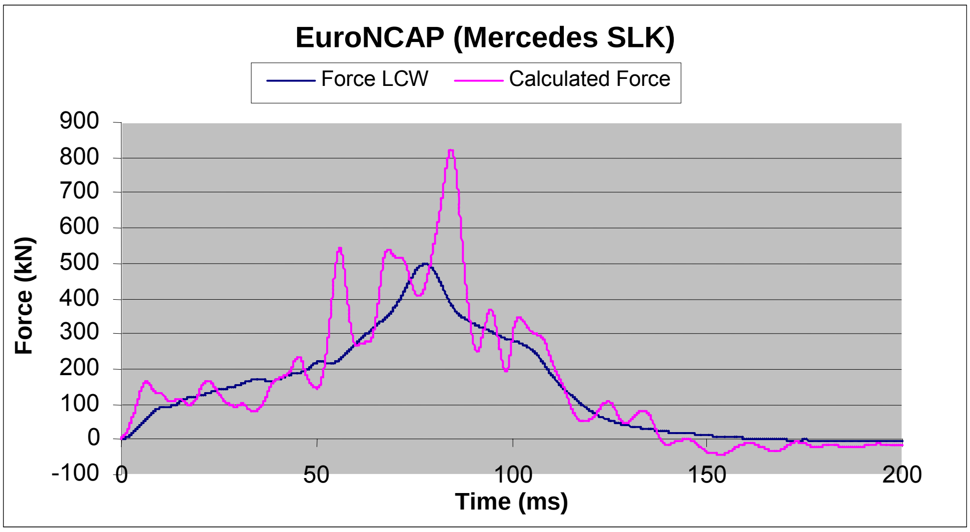


Figure 183: LCW Force and Calculated Force against Time, Test EUMC06FO

Towards the end of the impact the peak forces measured at the load cell wall increase substantially to the very high peak force of about 500 kN (Figure 182). This very high final strength of the vehicle front is apparently the result of a front structure design to keep the compartment extremely stable and intrusions low as the good EuroNCAP (4 stars) assessment, particularly concerning the compartment integrity, proves.

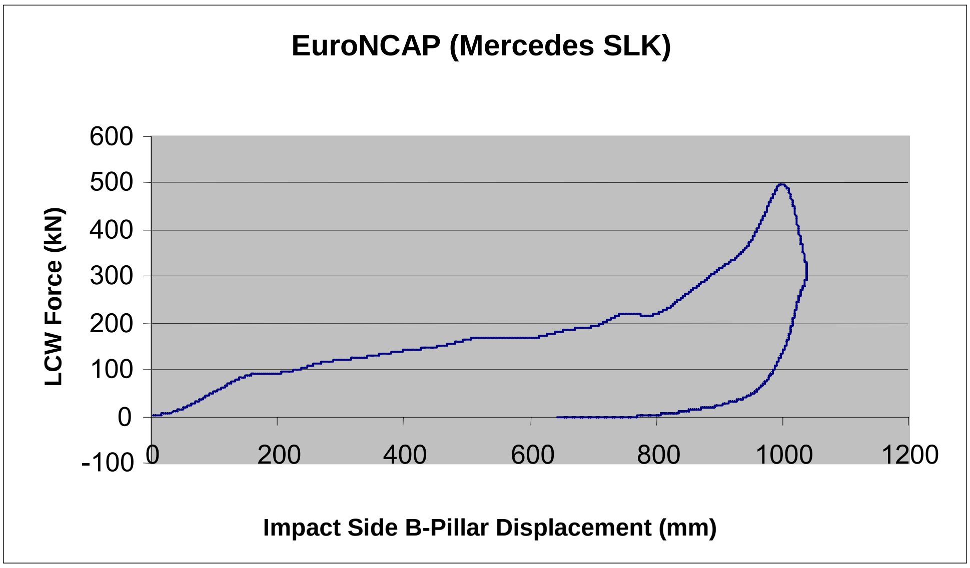


Figure 184: LCW Force against B-Pillar Displacement (mm)

Also for this sports car model the peak load cell forces are mainly caused by the engine impact as the distribution of the peak forces illustrates, see Figure 185. The impact forces caused by the engine are about 8 to ten times higher than the forces caused by the vehicle front structures and vary not as much as at the front of the MAZDA MX5. The softer front structures on the one side and the engine block show that this vehicle will apparently show only a limited compatibility behaviour in car to car accidents.

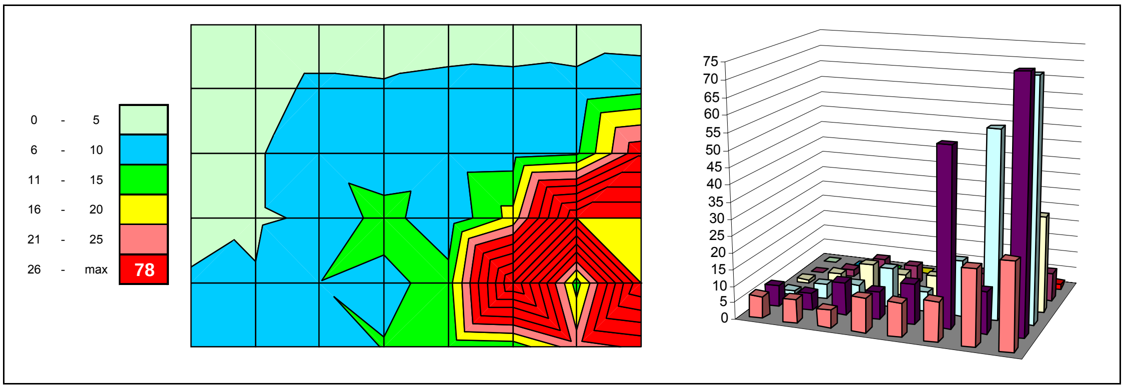
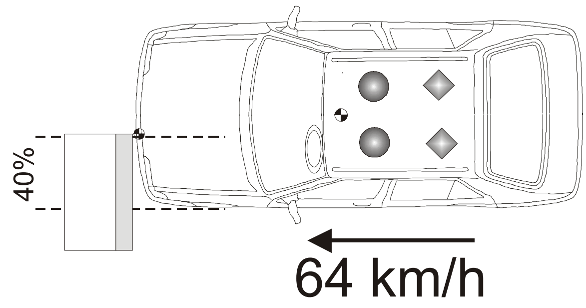


Figure 185: Force Distribution based on Peak Cell Forces, Test EUMC06FO

6.5.13 Mercedes C-Class against Offset Deformable Barrier (ODB)

The third car model tested is a Mercedes C-class, which represents a modern upper midsize vehicle.

Date	25.06.2002
Location	BASt
Topic Group	ODB Impacts
Mass Ratio	
Test Number	**EUMC07FO**



Vehicle:	Mercedes C-Class	Barrier:	EEVC Frontal
Impact Side:	Front		
Test Speed:	64km/h		
Overlap:	40%		
Test Mass:	1549		
Dummies:	LHS – Hybrid III RHS – Hybrid III Rear – P1,5 and P3		

The test speed and overlap were within the specified tolerances. The vehicle acceleration at B-post on impact side against time is shown (Figure 186). The peak deceleration amounts to about 33 g, which is a normal high peak value for this vehicle type. Also for this car the deceleration characteristics (superimposed by acceleration oscillations of higher resonance frequency) are caused by the intended direct force transmission of the left front wheel to the impact side door sill and the extremely stiff floor pan and the door sill.

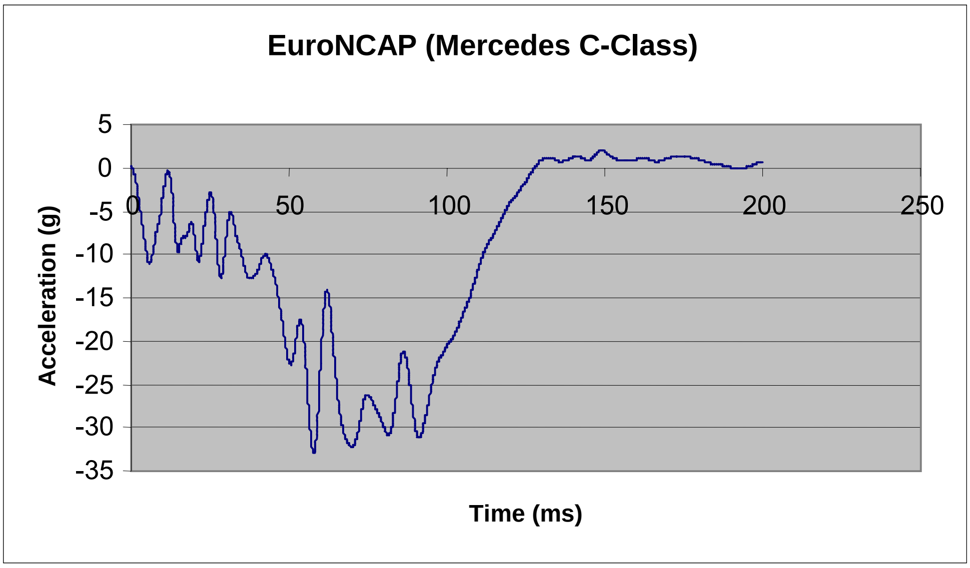


Figure 186: Vehicle Acceleration against Time, Test EUMC07FO

The next plot shows the comparison of the measured load cell force (LCW) behind the deformable element and the impact force of the Mercedes C-class calculated by multiplying the vehicle mass (including the engine) with the impact side B-post acceleration versus time (Figure 187). Like the Mercedes SLK both traces show significantly different characteristics. Again due to the above described load transmission through the door sill to the bottom of the B-post where the impact side vehicle acceleration is normally measured both traces can not be easily analysed together with the further measurements at different front components to assess vehicle compatibility.

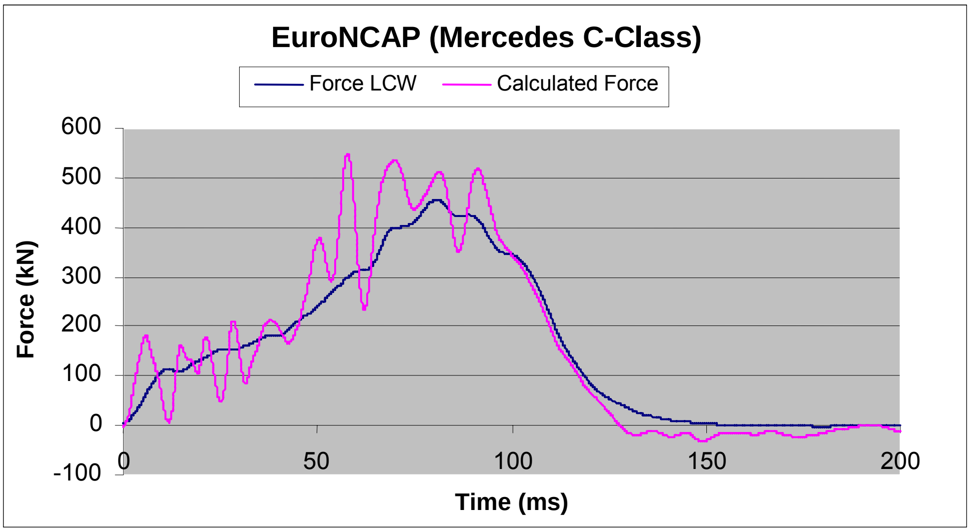


Figure 187: LCW Force and Calculated Force against Time, Test EUMC07FO

Towards the end of the impact the peak forces measured at the load cell wall increase steadily to the peak force of about 450 kN (Figure 188). This peak force value is in the lower range of the 1500 kg vehicle group. The final strength value of the vehicle front is apparently the result of a front structure design to maintain the compartment strength as the very good EuroNCAP assessment (5 stars), particularly concerning the compartment integrity and extremely low structural intrusion, proves.

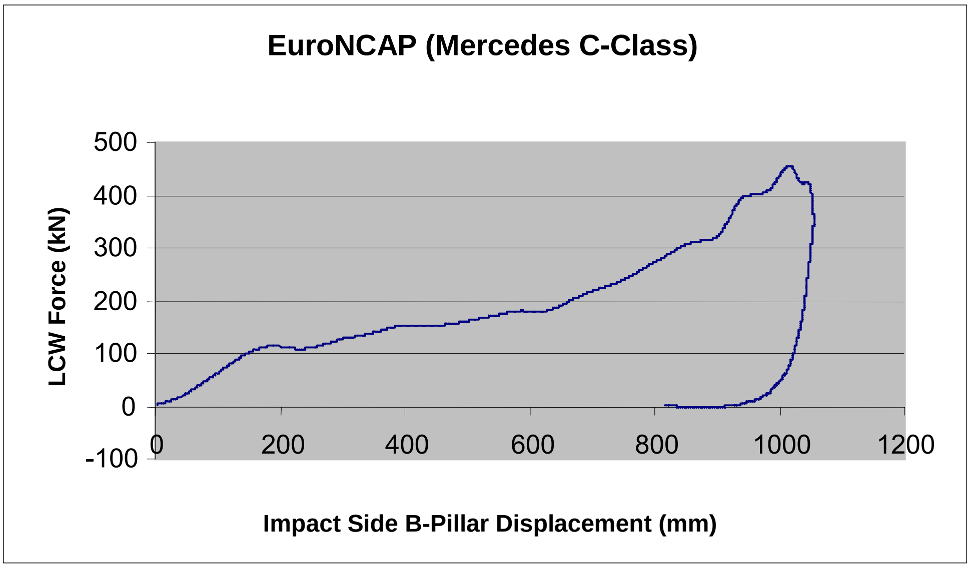


Figure 188: LCW Force against B-Pillar Displacement (mm), Test EUMC07FO

Also for this car model the peak load cell forces are mainly caused by the engine impact as the distribution of the peak forces illustrates (Figure 189). The impact forces caused by the engine are about 6 to 8 times higher than the forces caused by the vehicle front structures and vary not as much as at the front of the MAZDA MX5 and the Mercedes SLK. The softer front structures on the one side and the engine block show that this vehicle will apparently also show only a limited compatibility behaviour in car to car accidents.

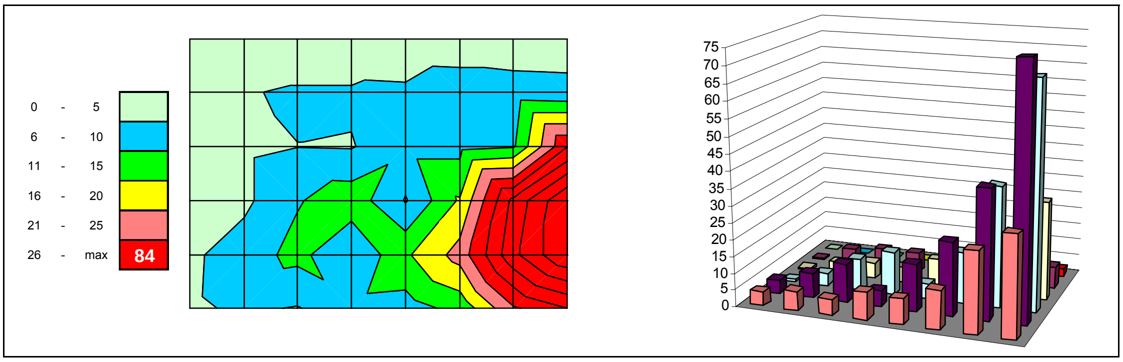


Figure 189: Force Distribution based on Peak Cell Forces, Test EUMC06FO

The force measurements with a load cell wall with an area of 125 x 125 mm behind the frontal offset deformable element, as used in EuroNCAP testing, show a reasonable resolution to detect stiffer vehicle front structures. The force distribution corresponds well to the deformation picture of the deformable element after the test. The bridging effect through honeycomb material pressed to the block could not be identified. It seems possible to define force characteristic and force distribution requirements for compatible vehicles.

6.5.14 Force Limits

The results of the analysed tests have been combined with results available to TRL from other offset deformable barrier impact tests. As the main aim of this proposed test procedure is to set limits on the global stiffness of vehicles the following chart details the global stiffnesses measured when assessed against the vehicle test mass. Examination of the data shows that the peak forces lie in the range from 200kN to 500kN (Figure 190). From this information, a first estimate for a maximum force limit could be 400kN and a minimum of 300kN. To determine whether these suggested values are appropriate and practicable much further work is necessary to address issues such as the passenger compartment strength, the deceleration pulse and the need to increase the crush depth in heavier cars.

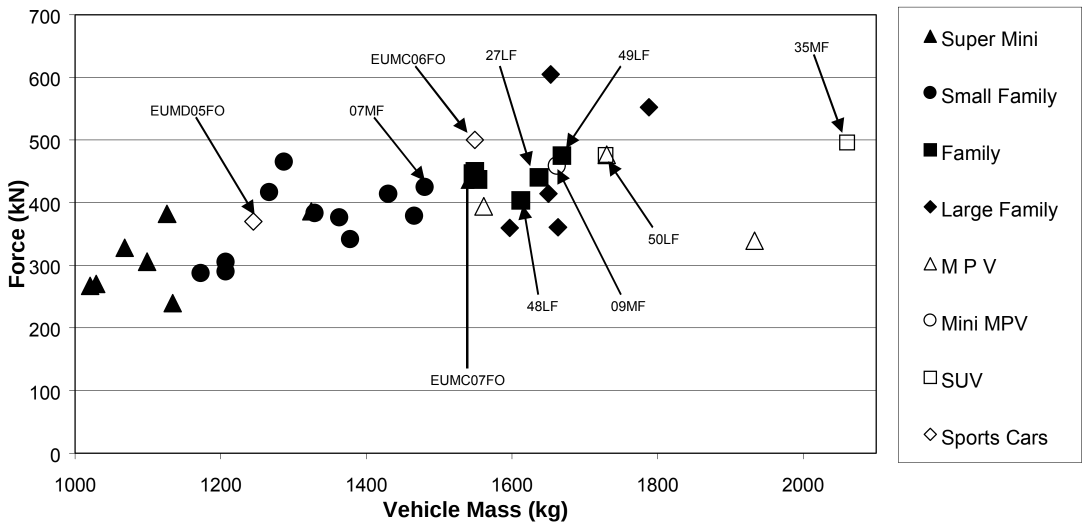


Figure 190: Peak force against vehicle mass from offset deformable barrier impact showing the range of forces for vehicle type.

6.5.15 Conclusions

- LCW measurements were taken for 10 vehicles varying in mass from 1245 to 2060 kg. The peak load cell wall forces measured for the vehicles tested were between 400kN and 500kN, which was within the range measured for previous tests for vehicles of similar mass.

- By using data from accelerometers mounted on the vehicle the contributions of the LCW force from the deceleration of the transmission package (mechanical forces) and occupant compartment (structural forces) were calculated. The force from the deceleration of the occupant compartment was typical between 60 to 70 percent of the global peak force recorded by the load cell wall.

- The vertical distribution of the peak cell forces varied from test to test. Examination of the vehicles post test indicated that this distribution was in some cases influenced by the interaction of the engine and crossbeam with the load cell wall edge. This observation should be taken into account if it is proposed that the vertical force distribution measured in this test should be used as a criterion to control compatibility, as it may invalidate such a criterion.

- Although the peak force applied by the body on frame SUV was less than some large family cars, the peak load cell force distribution measured was extremely inhomogeneous as the majority of the load was applied to a single load cell by the vehicle's lower rail.

7 SUMMARY OF CONCLUSIONS

The conclusions for each of the work packages, namely, accident analysis, benefit analysis and crash testing are summarised below.

7.1 ACCIDENT ANALYSIS

For GB and Germany, it was confirmed that the compatibility problems for car to car frontal impacts are structural interaction, stiffness matching and compartment strength.

Poor structural interaction was seen to occur in a number of different ways, namely the fork effect caused by lateral misalignment and under/override caused by vertical misalignment. Two types of the vertical misalignment problem have been identified, static and dynamic. Static misalignment is caused by an initial geometric mismatch of the vehicle's structures. Dynamic misalignment occurs for structures, initially approximately aligned, deforming to become misaligned during the impact.

For GB, poor structural interaction was found to be a major problem. Of the 162 cases examined only 2 had structural interaction that could be described as reasonable. However, some of the cases had poor structural interaction caused by low overlap, which improved compatibility is not expected to address. 100 (62%) cases had structural interaction problems that improved compatibility is likely to address. For Germany, it was found that structural interaction problems could probably only be quantified using detailed case studies. Unfortunately, unlike the UJK, detailed case studies were not performed for all the selected cases. However, it is intended that this should be done in the VC-COMPAT project.

For GB and Germany stiffness mismatch / compartment strength was found to be a large problem. For GB, the problem magnitude was quantified by identifying the cases where there was a significant intrusion difference between the colliding vehicles. In the data sample there were 78 cases where at least one of the vehicles had intruded and therefore it was possible to identify an intrusion difference. A significant intrusion difference was identified in 68 percent of these cases indicating that stiffness mismatch / compartment strength is a large problem. For Germany, the problem magnitude was quantified by identifying the cases where one vehicle had injury causing intrusion and the other no intrusion. In the data sample there were 76 cases where at least one of the vehicles had intruded. From these 76 cases, 33 (43%) had no intrusion in one vehicle and injury causing indicating that stiffness mismatch / compartment strength is a large problem. It should be noted that the extent to which poor structural interaction contributed to this problem is unknown.

For Germany, from the 135 accident cases examined it was found that the 14 MAIS 3+ injuries correlated well with compartment intrusion, i.e. no MAIS 3+ injuries occurred unless the compartment had intruded. This confirms the results of previous studies that show that intrusion is the major cause of deaths and serious injuries (Wykes 1998). However, there was no correlation of MAIS 3+ injuries with the Vehicle Deformation Index (VDI).

For GB, structural interaction problems were also identified in some single vehicle accidents indicating that a benefit from improved compatibility could also be expected in this type of impact.

7.2 BENEFIT ANALYSIS

For GB the potential benefit of improved frontal impact compatibility for car occupant casualties involved in frontal impact collisions was estimated to be:

- *some* reduction in injury risk for between 415 (45%) and 567 (61%) fatalities per year (currently out of 931 frontal impact car occupant fatalities per year on average[11]).

- *some* reduction in injury risk for between 8216 (66%) and 10470 (85%) seriously injured casualties per year (currently out of 12385 frontal impact seriously injured car occupant casualties per year on average).

For GB the benefit has been estimated for one particular type of accident only, namely a car frontal impact with one other car. For this accident type there were on average 254 fatalities and 5557 serious injuries annually in recent years in GB. From the analysis performed, using the assumptions that optimistically 'compatible' cars should prevent contact related injuries and pessimistically 'compatible' cars should prevent injuries caused by intrusion up to a given impact severity, the following predictions were made:

- If it is assumed that improved compatibility offers increased protection for all impact severities, it is predicted that between 102 (40%) and 152 (60%) fatalities and between 587 (11%) and 1605 (29%) serious casualties would be prevented.

- If it is assumed that improved compatibility offers increased protection up to an impact severity of 56 km/h ETS, it is predicted that between 25 (10%) and 46 (18%) fatalities and between 389 (7%) and 1167 (21%) serious casualties would be prevented. It should be noted that compatibility is expected to offer some benefit above an impact severity of 56 km/h ETS, so these predictions are most likely low.

It should be recognised that much further benefit can be expected for other accident types, especially car to vehicle frontal impacts, most likely car frontal collisions with roadside obstacles and possibly for side impacts as well. The seriously injured casualty category defined to the Police's injury severity rating covers a wide range of injury severities. It should be noted that the benefit from, for example, reducing a MAIS 4 serious injury to a MAIS 2 serious injury is not accounted for in the analysis performed.

For Germany, the potential benefit of improved compatibility for car occupant casualties involved in frontal impact collisions based on accident data for the year 2000 has been estimated to be:

- some reduction in injury risk for between 9,317 (33%) and 18.736 (67%) seriously injured car occupants per year, (there were 27,967 frontal impact car occupant seriously injured casualties in the year 2000).

An estimate was also made for fatalities. However, it is possible that this result was not statistically significant as the GIDAS database, on which the analysis was based, contained only 33 fatalities for this impact configuration. Noting this caveat, the estimate was:

[11] This figure is adjusted to remove the effect of cars greater than 7 years old.

- some reduction in injury risk or for between 287 (14%) and 572 (28%) fatalities per year, (there were 2,066 frontal impact car occupant fatalities in the year 2000).

7.3 CRASH TESTING

The conclusions are listed below for each of the different types of tests performed.

Full width deformable barrier test to assess structural interaction

- Two tests using a Mondeo car were performed to help in the redesign of the barrier face in order to overcome the problem of small stiff protruding structures forming preferential load paths. The second test demonstrated that the redesigned face overcame this problem, whilst still achieving the aims of the initial barrier face which were:
    - To prevent unrealistic decelerations at the front of the car.
    - To attenuate the engine inertial loading
    - To have a similar compartment deceleration to an equivalent rigid wall test.

- The multiple loads of the Opel Astra and modified Astra could be identified from the homogeneity of the load cell wall (LCW) force distribution recorded in the full width tests. A difference was distinguished between the Astra and modified Astra, the modified Astra showing better homogeneity for the LCW force distribution, which is consistent with the better structural interaction seen in the modified car to car crash test. However, the engine subframe to lower rail shear connection was not loaded as much in either of these tests compared to the car to car tests. This indicates that the full width test may not generate as much shear force across this type of connection as in a car to car impact.

- The LCW results from the Renault Laguna II test showed that the Laguna II did not exhibit good stiffness homogeneity. This was due to the lower rails bottoming out the barrier and applying large loads directly on the load cell wall and the low loading applied by the centre of the bumper and subframe crossbeams due to their failure. The bottoming out of the lower rails formed preferential load paths, which most likely reduced the load applied by other structures, such as the subframe. The stability of the lower rails was most likely helped by the good vertical connections. The formation of a preferential load path was also seen in the Rover 75 test, in which one lower rail bottomed out the barrier.

- Ideally, a test method to evaluate compatibility needs to be able to deform a car as much as it is deformed in accidents so that all the possible load paths and the shear connections between these load paths are exercised. The tests performed in this project have shown that the frontal unit deformation achieved may not be sufficient to adequately check all these load paths and the shear connections, especially if they are positioned some distance behind other paths, for example, a subframe positioned more than about 150 mm behind the front of the lower rails.

PDB test to assess structural interaction and frontal unit energy absorption

It should be noted that the purpose of the PDB test is to assess structural interaction and the frontal unit energy absorption up to an Equivalent Energy Speed (EES) of 50 km/h. An impact speed of 60 km/h was calculated to give a vehicle EES of 50 km/h, which takes into account the energy absorption of the barrier and the vehicle stiffness. A fixed overlap width of 750 mm is used to ensure that the barrier generates the same load for cars of different widths.

- The use of the load distribution on the LCW behind the PDB does not appear to give an accurate enough measure of a car's stiffness homogeneity and hence is not worth pursuing further as an assessment method. This is because problems similar to those encountered with the full width tests, such load cell bridging caused by the shear strength of the honeycomb, occur to some degree with this test. This conclusion is supported by a separate French study, which found an uneven load distribution was recorded on the load cell wall for an impact against the PDB using a trolley with a flat rigid face.

- In the Mondeo test a part of the barrier remained attached to the car after the test. This would cause severe difficulties in measuring the barrier final deformation profile objectively, which the PDB approach is completely reliant upon. For this test the version 6 of PDB was used. Version 7 of the barrier has a thicker front sheet, which may reduce or solve this problem. The lack of penetration of the barrier front sheet in the test with the Volvo S80 indicates the improved performance of version 7 of the barrier in this respect.

- The PDB barrier was defined to represent an average car and its stiffness is such that bottoming out is unlikely, even for large cars with a homogeneous front end. However, on the Range Rover test this barrier bottomed out. The implication of this should be considered in relation to future regulations and consumer testing.

- The Smart is a very light car. It was judged to be a non aggressive car based on the shape of the barrier deformation after the impact, even though its stiffness is very high.

- For the Volvo S80 the deformation shape of the PDB was relatively homogenous, so based on a subjective assessment of the barrier deformation this car was judged to be non aggressive.

Car to car test

- The Toyota Yaris has a design consisting of one main load path, the lower rails. The Renault Clio has a multi-level load path design. Examination of the cars prior to the test showed that there was good structural alignment between them, which indicated that good structural interaction between the lower rails might be expected. However, poor structural interaction was seen in the test caused by dynamic effects. When the Toyota Yaris lower rail impacted against the Renault Clio one, the Yaris lower rail bent upwards resulting in poor interaction with the Clio structure. As a result of this the Yaris occupant compartment intruded significantly and became unstable. The Clio compartment performed well without significant intrusion.

- A comparison of the relative performance of the Yaris and Clio in the car to car test and the EuroNCAP tests based on the intrusion measurements showed that the Clio performance was slightly worse in the car to car test compared to the EuroNCAP test. In contrast, the Yaris performance was significantly worse in the car to car test. It is believed that the main reason for this difference was the change in the structural performance of the Yaris, namely the lower rail, caused by poor structural interaction, which in turn was a result of the Yaris having a design based on a single main load path.

EuroNCAP test LCW measurements

- LCW measurements were taken for 10 vehicles varying in mass from 1245 to 2060 kg. The peak load cell wall forces measured for the vehicles tested were between 400kN and 500kN, which was within the range measured for previous tests for vehicles of similar mass.

- By using data from accelerometers mounted on the vehicle the contributions of the LCW force from the deceleration of the transmission package (mechanical forces) and occupant compartment (structural forces) were calculated. The force from the deceleration of the occupant compartment was typical between 60 to 70 percent of the global peak force recorded by the load cell wall.

- The vertical distribution of the peak cell forces varied from test to test. Examination of the vehicles post test indicated that this distribution was in some cases influenced by the interaction of the engine and crossbeam with the load cell wall edge. This observation should be taken into account if it is proposed that the vertical force distribution measured in this test should be used as a criterion to control compatibility, as it may invalidate such a criterion.

- Although the peak force applied by the body on frame SUV was less than some large family cars, the peak load cell force distribution measured was extremely inhomogeneous as the majority of the load was applied to a single load cell by the vehicle's lower rail.

8 RECOMMENDATIONS

The recommendations resulting from each of the work packages, namely, accident analysis, benefit analysis and crash testing are listed below.

Accident analysis

For Germany, it is recommended that further analysis should be performed to quantify the magnitude of the structural interaction problem. For both Germany and the UK, it is recommended that further accident analysis should be performed in the future to check that the conclusions of this work are still valid, as the vehicle fleet is constantly changing. Additional accident variables such as improved deformation measurements and harmonised impact severity measures would help future analyses.

Benefit analysis

In order to obtain a more complete benefit estimate for GB, it is recommended that a similar benefit analysis to that performed for the car frontal impact with one other car or van type of accident should be conducted for other car frontal impact accident types. For Germany, it is recommended that an analysis to estimate the benefit of improved compatibility, in terms of the number of lives saved as opposed to the reduction in injury risk, should be performed.

The benefits predicted are largely dependent on the assumptions made for how ‘compatible’ cars will perform. Hence, it is recommended that once more about a ‘compatible’ car’s performance is known, the assumptions made should be refined and the analysis repeated.

Crash testing

It is recommended that principles on which the full width and PDB tests are based should be validated, i.e. for the full width test is the homogeneity measured on the LCW representative of a car’s structural interaction potential and similarly for the barrier deformation measured in the PDB test.

For the full width test objective assessment criteria require development. At present differences in the performance of the vehicles are based on subjective analysis of the load cell wall force distribution. Criteria should be developed to evaluate and quantify the changes observed between different vehicles. This will require additional crash test data to be generated from a larger range of vehicle designs to validate the procedure and set definitive limit values.

For the full width Laguna and Rover 75 tests preferential load paths were formed because the lower rails bottomed out the barrier. This most likely reduced the load carried by other structures set further back in the car, such as the subframe, resulting in the reduction of the homogeneity of the load recorded on the wall. A study should be performed to address the following questions:

- Does the current barrier design give a representative homogeneity measure for cars with high local stiffnesses?

- Approximately, how far back can a secondary load path be positioned from the front of the main load path and still be able to contribute significantly to improving a car's compatibility?

For the PDB test the current assessment criteria require further development. At present these criteria are based on the shape of the barrier face final deformation. A formula has been developed to assess the barrier deformation in terms of its height and depth, but limit values for these parameters still need to be defined. The test data collected in this project completes a crash test matrix, which will form a useful data set for future work to continue the development of the current assessment criteria. However, further data will be required to validate the procedure and set definitive limit values.

The current PDB barrier face is based on the stiffness of a small to medium European car. The suitability of the current barrier face design should be considered in terms of its likely effect on the future vehicle fleet design and the vehicle classes which are likely to be included in future regulatory and consumer testing. An example of a parameter that should be investigated is the stiffness distribution between its upper and lower sections.

The Renault Clio to Toyota Yaris crash test demonstrated the poor structural interaction performance and possibly low compartment strength of the Toyota Yaris. It is recommended that this car could be used as a possible benchmark to help verify the full width and PDB tests and set limit values for structural interaction performance for the proposed assessment criteria following verification of the Yaris compartment strength.

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10 APPENDIX 1 TABLES FOR BENEFIT ANALYSIS FOR GB

Table 51: Upper and lower limits for accident parameters used to identify accident cases in which the casualties are expected to experience a potential benefit, for car frontal impacts with a Public Service Vehicle (PSV) or Heavy Goods Vehicle (HGV).

Accident parameter	Upper (somewhat optimistic) limit	Lower (somewhat pessimistic) limit	Basic sample for all frontal impacts with a PSV / HGV
Principle direction of force (pdf) (o'clock)	10,11,12,1,2	11,12,1	10,11,12,1,2
Overlap	Include ≥ 20%	Include ≥ 30%	All
Multiple impacts	Exclude cases in which a significant[12] side impact occurred.	Exclude cases in which a significant side impact occurred and cases in which the other impact was judged to be more injurious than the frontal impact.[13]	All multiple impacts with frontal as the initial point of contact.
Rollover	Include all, with or without rollover	Include only those where rollover was less injurious than the frontal impact.	All, with or without rollover
Impact severity	Include all with ETS up to 56km/h and 50% of those more severe.	Include all with ETS up to 48km/h and 50% of those more severe.	All with known ETS.
Underrun	Include 20% of underrun cases, plus all without underrun	Include all with no underrun present	All
Change in Velocity (Δv)	All	Known to be ≤56km/h	All

[12] "Significant side impact" is defined as having a CDC extent code of at least 2.

[13] "Less injurious" assessment is based on the vehicle examiners' judgement of the relative likelihood of a particular part of the accident causing the serious injuries.

Table 52: Upper and lower limits for accident parameters used to identify accident cases in which the casualties are expected to experience a potential benefit, for car frontal impacts with a wide object.

Accident parameter	Upper (somewhat optimistic) limit	Lower (somewhat pessimistic) limit	Basic sample for all frontal impacts with a wide object
Principle direction of force (pdf) (o'clock)	10,11,12,1,2	11,12,1	10,11,12,1,2
Overlap	Include ≥ 20%	Include ≥ 30%	All
Multiple impacts	Exclude cases in which a significant[14] side impact occurred.	Exclude cases in which a significant side impact occurred and cases in which the other impact was judged to be more injurious than the frontal impact.[15]	All multiple impacts with frontal as the initial point of contact.
Rollover	Include all, with or without rollover	Include only those where rollover was less injurious than the frontal impact.	All, with or without rollover
Impact severity	Include all with ETS up to 56km/h and 50% of those more severe.	Include all with ETS up to 48km/h and 50% of those more severe.	All with known ETS.

[14] "Significant side impact" is defined as having a CDC extent code of at least 2.

[15] "Less injurious" assessment is based on the vehicle examiners' judgement of the relative likelihood of a particular part of the accident causing the serious injuries.

Table 53: Upper and lower limits for accident parameters used to identify accident cases in which the casualties are expected to experience a potential benefit, for car frontal impacts with a narrow object.

Accident parameter	Upper (somewhat optimistic) limit	Lower (somewhat pessimistic) limit	Basic sample for all frontal impacts with a narrow object
Principle direction of force (pdf) (o'clock)	10,11,12,1,2	11,12,1	10,11,12,1,2
Damage pattern	Damage width ≥750mm **OR** ≤750mm with a mid-point offset ≤700mm	Damage width ≥750mm	All
Multiple impacts	Exclude cases in which a significant[16] side impact occurred.	Exclude cases in which a significant side impact occurred and cases in which the other impact was judged to be more injurious than the frontal impact.[17]	All multiple impacts with frontal as the initial point of contact.
Rollover	Include all, with or without rollover	Include only those where rollover was less injurious than the frontal impact.	All, with or without rollover
Impact severity	Include all with ETS up to 56km/h and 50% of those more severe.	Include all with ETS up to 48km/h and 50% of those more severe.	All with known ETS.

[16] "Significant side impact" is defined as having a CDC extent code of at least 2.

[17] "Less injurious" assessment is based on the vehicle examiners' judgement of the relative likelihood of a particular part of the accident causing the serious injuries.

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The conclusions and recommendations of this report have been reviewed by EEVC WG15 members.