TOWARDS A BENEFICIAL, SCIENTIFICALLY MEANINGFUL, AND APPLICABLE COMPATIBILITY-TESTING

Robert Zobel
Thomas Schwarz
Gareth Thomas
Volkswagen AG
Germany
Paper Number 05-0052

ABSTRACT

Compatibility is an issue that relates to the improvement of vehicle safety. After frontal and side impact self protection, partner protection, a key component of compatibility, represents the next step forward for passive safety improvement. Compatibility is complicated to achieve, because it requires world-wide industry to take steps in a similar direction. A harmonized approach is difficult to achieve because many differences in vehicle makes and models between the various fleets around the world exist. This leads to incompatibilities between vehicles in a global sense: Asian markets have a high market share of very small cars, the American market is characterized by a high proportion of LTVs and SUVs and the European market is somewhere between the American and the Asian markets.

It is obvious that a lot of requirements need to be fulfilled by a compatibility regulation which is; beneficial to the customer, which is scientifically meaningful, refers to front and side-impact and which is applicable for all markets and, last but not least, is considered to be fair by all manufacturers.

ACEA is not in the position to suggest a solution meeting all these requirements. However, some test results and observations which could contribute to a solution are presented in this paper.

The focus of most proposed compatibility procedures is to improve structural interaction in collisions involving passenger cars. A couple of conditions exist that influence the definition of a geometric zone for structural interaction. A zone for structural interaction has to ensure maximal interaction between passenger vehicles with other passengers vehicles, SUVs/LTV's and trucks (to be supported by under-run protection systems) can be achieved. This could represent a first step in increasing compatibility within vehicle fleets. Structural interaction is, in fact, the principle requirement for compatibility before the issue of stiffness can be solved. Keeping this in mind, ACEA drafted a road map chartering the path toward improved compatibility, which is presented in this paper.

INTRODUCTION

Accident Findings

There are two main areas of interest when discussing accidents: Single vehicle accidents and vehicle-to-vehicle accidents. In single vehicle accidents, the object is mainly rigid. All deformation energy has to be provided by the vehicle itself. In car-to-car accidents, both opponents provide deformation energy and the technical challenge is to enhance the interaction of both objects so that all available deformation energy is dissipated in a collision.

There is a clear finding in Europe and in the U.S regarding the distribution of single vehicle accidents and vehicle-to-vehicle accidents. Single vehicle accidents are of very high statistical significance and have to be taken into account when discussing partner protection and compatibility.

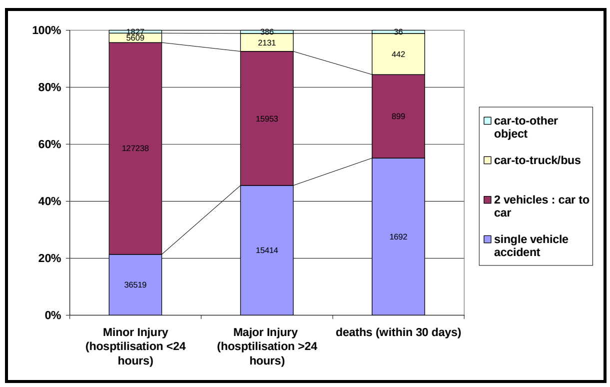


Figure 1: Distribution of car-to-car accidents, car-to-truck accidents and single vehicle accidents in Germany 2003 (StBA).

German data clearly indicates that single vehicle accidents are very relevant when considering fatalities Figure 1.

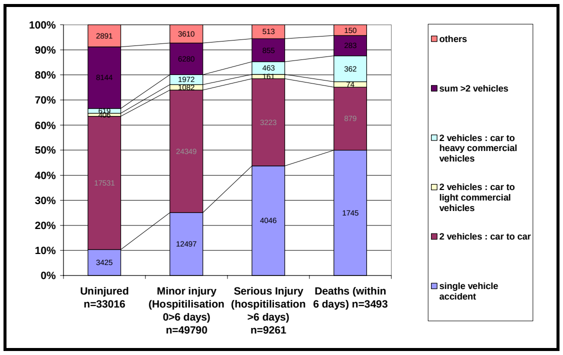


Figure 2: Distribution of car-to-car accidents, car to light and heavy truck accidents and single vehicle accidents in France 2003 (LAB).

The same observation holds true for France, approximately 50% of the fatalities occur in single vehicle accidents Figure 2.

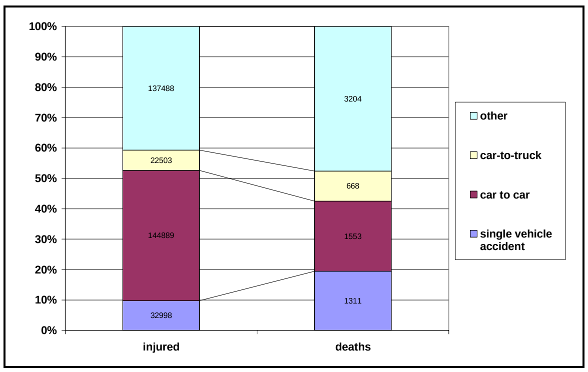


Figure 3: Distribution of car-to-car accidents and single vehicle accidents in Italy 2002 (ISTAT).

The situation is a little different in Italy regarding fatalities in single vehicle accidents and car-to-car accidents. Both accident configurations are of nearly equal relevance. In the Italian statistics, all participants in traffic accidents are included (pedestrians and motorcyclists are included). This explains the high percentage of "others" Figure 3.

When considering occupants that sustain injuries of lower severity only, the opposite observation is true. Car-to-car accidents are of higher statistical relevance than single vehicle accidents when considering less severe injuries. Severe injuries are somewhere in between, close to the distribution of the fatalities. The distribution of collision objects for occupants injured in accidents involving long term consequences can be estimated to more closely reflect the distribution for fatalities than for slight injuries. Unfortunately, the official statistics do not provide this information.

Both sides of car safety (self- and partner-protection) should be taken into account when discussing safety enhancement. European car industry started its own compatibility research with the unanimous understanding that compatibility means an enhancement of overall safety of cars without compromising the existing safety level of cars provided to the cars' own occupants (self protection). The figures above, which reflect the accident environment in most developed countries, prove that a good balance between self and partner protection is a pre-requisite for an enhancement of the protection of passenger vehicle occupants.

Measuring Self Protection

Self protection is generally evaluated in crash tests and the dummy loads measured in the tests often form the basis of the safety evaluation. These parameters describe the risk faced by an occupant during a collision in the configuration tested. In fact, no vehicle occupant will ever be involved in an accident in a configuration identical to the crash-test. What is the real-world safety benefit e.g. of a rigid-barrier impact for an occupant involved in a collision with a tree? Is the amount of deformation energy available for this pole impact the same? Of course and unfortunately, the energy, dissipated in the front-end in a rigid barrier impact is an upper limit for the deformation energy available for an impact with a pole or tree. The tree may strike one longitudinal and miss the other, or the tree may strike the vehicle between the longitudinals. The deformation energy available within the longitudinals would not be available in this case as it is unlikely the cross beam could transmit the loading to both longitudinals.

When a rigid barrier is used, the amount of energy absorbed by the car is easily measured. It is almost equal the kinetic energy of the car before the crash (neglecting rebound). All energy has to be absorbed by the car, because the barrier does not absorb any energy. This was the reason that EES, the Energy Equivalent Speed, was formulated. The EES is the speed a car needs in an impact against a rigid barrier to absorb a certain amount of deformation energy.

D Deformation Energy

m Mass of Car

$$D = \frac{1}{2} * m * EES^2 \qquad (1)$$

$$\Rightarrow EES = \sqrt{\frac{2D}{m}}$$

The EES is a first approximation about the amount of self protection provided by a car. A couple of restrictions which apply to the statement mentioned above must be taken into account. However, it is a basis to ensure that a certain level of self protection is provided.

It was already mentioned that the EES can be easily calculated for a rigid barrier impact. However, barriers with deformable elements are being discussed for compatibility testing that absorb energy as well. The consequences this has for the EES have to be investigated.

Three types of barriers have to be distinguished:

- **Zero Deformation Energy Barrier ZDEB.**

This is, in-fact, a rigid barrier, as used in the U.S.

- **Limited Deformation Energy Barrier LDEB.**

This is a barrier that provides deformation energy, but the car will bottom out the barrier and the barrier behaves like a rigid-barrier at the end of the collision. (ECE R94 in Europe)

- **Unlimited Deformation Energy Barrier UDEB.**

This is a barrier that provides sufficient deformation energy, so that the car will never bottom out the barrier. This barrier never behaves like a rigid barrier.

Each of these barrier types are used or available as research tools. There are well known facts about these barrier types:

- Zero deformation energy barrier energy ZDEB:
  - Induces simultaneous/homogenous deformation-shear loads not activated
  - Cross-beams are not credited
  - Not representative of real-world car-to-car impact

- Limited deformation energy barrier LDEB:
  - Barrier provides shear loading only in the early stages of deformation, until bottoming-out occurs
  - To maximize energy dissipation within the barrier, wide load distribution in vertical direction is beneficial

- Unlimited deformation energy barrier UDEB:
  - Barrier always provides sheer load
  - Car will never bottom out barrier
  - Barrier never behaves like a rigid object
  - Is not representative of impacts with rigid objects

The purpose of this paper is not to discuss all these issue in depth. This list is also not complete. It only shows the main implications for compatibility. The question that has to be answered is; what is the influence of each barrier type on the EES?

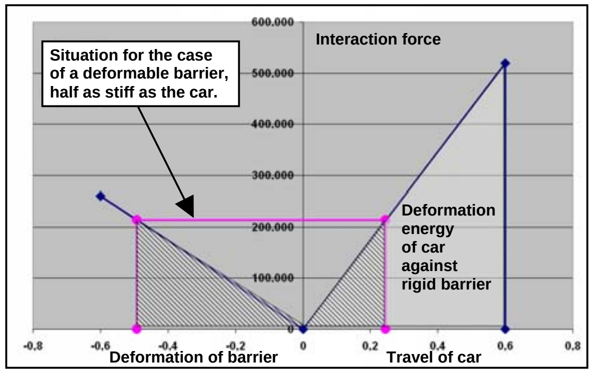


Figure 4: Energy distribution in the front-end of the car and in the barrier when the deformation characteristics of both objects can be described using triangular force-deflection-curves.

Figure 4 outlines the problem. A car, tested against a deformable barrier needs less deformation of its own structure to dissipate it own kinetic energy than a car tested against a rigid barrier. This means that the deformable barrier collision test self protection to a low degree than collision against the rigid barrier. The relationship between barrier stiffness and the self protection level of the car are easily computed using the following formulae.

c_{car} Stiffness of car

$c_{barrier}$ Stiffness of barrier

F Interaction force level

d Deformation travel

$$c_{car} * d_{car} = F = c_{barrier} * d_{barrier}$$

D Deformation energy in case of car-to-barrier impact

$$D = D_{car} + D_{barrier} = \frac{1}{2} * d_{car} * F + \frac{1}{2} * d_{barrier} * F$$

$$D = \frac{1}{2} * \frac{F}{c_{car}} * F + \frac{1}{2} * \frac{F}{c_{barrier}} * F = \frac{1}{2} * F^2 * \left(\frac{1}{c_{car}} + \frac{1}{c_{barrier}} \right)$$

Assuming there is force equilibrium at the interface between the car and barrier (action and reaction), the deformation travel of the car and the barrier is reciprocally proportional to the stiffness of the car and barrier, respectively. This allows the computation of the energy for a triangular force-deflection-curve. From this, the proportion of D_{car} compared to the total energy of the crash is easily computed.

$$D_{car} = \frac{1}{2} * d_{car} * F = \frac{1}{2} * \frac{F}{c_{car}} * F = \frac{1}{2} * \frac{F^2}{c_{car}}$$

$$\frac{D_{car}}{D} = \frac{\frac{1}{2} * \frac{F^2}{c_{car}}}{\frac{1}{2} * F^2 * \left(\frac{1}{c_{car}} + \frac{1}{c_{barrier}} \right)} = \frac{\frac{1}{c_{car}}}{\frac{1}{c_{car}} + \frac{1}{c_{barrier}}} = \frac{1}{1 + \frac{c_{car}}{c_{barrier}}}$$

$$\frac{D_{car}}{D} = 50\% \quad \text{for} \quad c_{car} = c_{barrier}$$

$$\frac{D_{car}}{D} \approx 0\% \quad \text{for} \quad c_{car} \text{ large compared to } c_{barrier}$$

$$\frac{D_{car}}{D} \approx 100\% \quad \text{for} \quad c_{car} \text{ small compared to } c_{barrier}$$

For rigid barriers, c_{car} is negligible and the car has to absorb 100% of the deformation energy. If both car and barrier are similar, then only 50% of the kinetic energy has to be absorbed by the car. For a very deep barrier with unlimited available deformation energy, very stiff cars may deform the deformable element to a very large extent. In this case, little energy would be dissipated through deformation of the vehicle structure. High stiffness is not penalized by this barrier.

This can easily be transformed into the notion of EES. Considering the Barrier Impact Speed BIS, the following computation holds:

$$c_{car} = c_{barrier} \Rightarrow \frac{D_{car}}{D} = 50\% \Rightarrow \frac{EES}{BIS} = 71\%$$
$$\Rightarrow \quad EES = 39{,}6 km/h \quad \text{for} \quad BIS = 56 km/h$$
$$c_{car} \quad \text{large compared to} \quad c_{barrier} \Rightarrow \frac{D_{car}}{D} \approx 0\% \Rightarrow \frac{EES}{BIS} \approx 0\%$$
$$\Rightarrow \quad EES \approx 0 km/h \quad \text{for} \quad BIS = 56 km/h$$
$$c_{car} \quad \text{small compared to} \quad c_{barrier} \Rightarrow \frac{D_{car}}{D} \approx 100\% \Rightarrow \frac{EES}{BIS} \approx 100\%$$
$$\Rightarrow \quad EES \approx 56 km/h \quad \text{for} \quad BIS = 56 km/h$$

So, what remains in terms of self protection, when a barrier impact speed of 56km/h is used?: If both the car and the barrier have of the same stiffness, the EES (desribing the level of self protection) decreases to 40km/h.

Another question, often raised when discussing these different barrier types, is the question of mass influence. LDEBs, like the European R94 barrier, are often blamed for containing a mass dependency. The barrier provides a limited amount of deformation energy. A larger car, which has more kinetic energy at a given barrier impact speed BIS, receives a smaller percentage of its initial kinetic energy through deformation of the barrier than a small car. In absolute terms, both cars can absorb the same amount of energy but this amount represents a higher percentage of the initial kinetic energy of the small car. So, small cars are tested at a lower EES than large cars. **Figure 5** gives the relation.

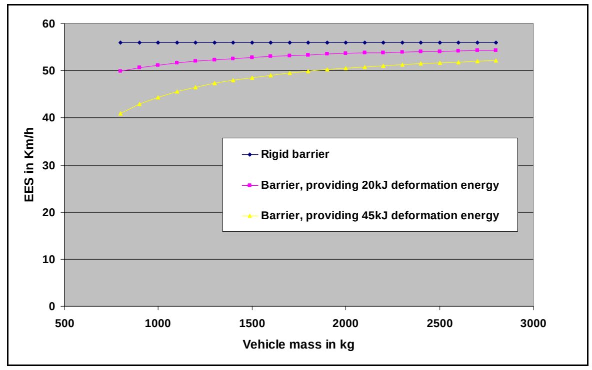


Figure 5: Mass influence of EES in LDEB barriers which provide a limited amount of deformation energy.

In the formula, previously presented, it was clear that the UDEBs (barriers providing an unlimited amount of deformation energy) provide deformation energy to the car depending on the stiffness of car and barrier:

$$\frac{D_{car}}{D} = \frac{1}{1 + \frac{c_{car}}{c_{barrier}}} \Rightarrow \frac{D_{barrier}}{D} = 1 - \frac{1}{1 + \frac{c_{car}}{c_{barrier}}} = \frac{1}{1 + \frac{c_{barrier}}{c_{car}}}$$

This formula clearly depends on stiffness. Unfortunately, there is a relationship between stiffness and mass, because car designers are not free to design cars with unlimited amounts of deformation travel. Therefore current cars have a similar degree of available deformation travel, which is nearly the same for all mass classes. This creates the mass influence for the UDEBs.

$$\frac{1}{2} * m_{car} * BIS^2 = D = \frac{1}{2} * F_{car}^{\max} * d_{car} \quad \text{for triangular force - deflection curves}$$

$$F_{car}^{\max} = \frac{\frac{1}{2} * m_{car} * BIS^2}{\frac{1}{2} * d_{car}} = \frac{m_{car} * BIS^2}{d_{car}} \quad \text{and} \quad c_{car} = \frac{F_{car}^{\max}}{d_{car}} = \frac{m_{car} * BIS^2}{d_{car}^2}$$

m_0 mass of average car simulated by barrier

d_0 deformation travel of average car simulated by barrier

Assumption : Not much difference between cars,

regarding deformation travel : $d_0 = d_{car}$

$$c_{barrier} = \frac{m_0 * BIS^2}{d_0^2} \quad \text{and} \quad \frac{c_{car}}{c_{barrier}} = \frac{m_{car}}{m_0}$$

$$\frac{D_{car}}{D} = \frac{1}{1 + \frac{c_{car}}{c_{barrier}}} = \frac{1}{1 + \frac{m_{car}}{m_0}} < 1 \quad \text{and} \quad \frac{EES}{BIS} = \sqrt{\frac{D_{car}}{D}} = \sqrt{\frac{1}{1 + \frac{m_{car}}{m_0}}}$$

Although there is no theoretical influence of mass, an influence is present driven by the fact that deformation travel of cars is limited. This influence is described in the following figure 6.

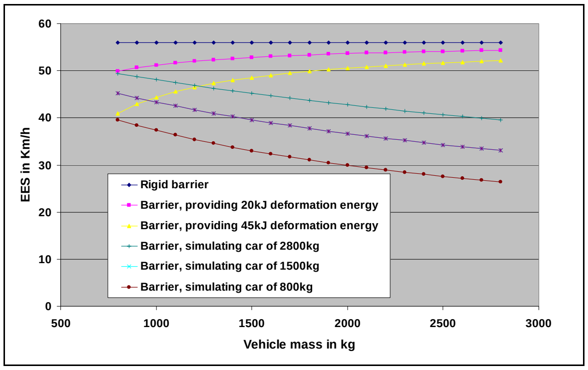


Figure 6: The inter-dependency between mass and EES for cars designed for collisions against LDEB barriers and ZDEB, respectively.

Concluding this section, the following can be stated:

A vehicle with lower available deformation travel has higher front-end forces and will be tested at a lower EES by an UDEB, a barrier providing an unlimited amount of deformation energy. As all of today's cars have nearly the same deformation travel (40cm...70cm), larger cars are stiffer than smaller cars and therefore will be tested at a lower EES level. This means that they will provide a lower self protection level.

Avoidance of bottoming out costs a high price leading to a reduction in self protection in single vehicle accidents and vehicle-to-vehicle collisions.

Referring to the bulkhead principle, compartment collapse can be avoided up to the sum of the EES of both vehicles in fixed barrier collisions. This is achievable because there is sufficient deformation energy available within the front-ends of both vehicles for this particular crash configuration.

If the EES of one of the vehicles is reduced, the maximum closing velocity, up to which achieving compatibility can be considered realistic, would be reduced as well [1].

Steps toward Partner Protection -
The Stiffness of the Crossbeam

ACEA conducted no own research on a special barrier, nor does ACEA wish to establish its own one. The main focus of ACEA is to discuss and evaluate the existing ideas of compatibility barriers. The member companies of ACEA do not have a common position on one barrier type or assessment procedure. The position of ACEA is that the current knowledge is not sufficient to make this decision. The calculation given in the previous paragraph is an example of such research. It is pure physics, so no decision was taken within ACEA about this issue. The common position of all partners is that self protection must not be compromised. The path to achieve this goal is still under discussion.

Besides the question, which barrier is the most appropriate one resulting in a maximum increase in safety, there is also the question of side effects that has to be studied carefully. So ACEA performed two test series.

A Rover 75, which was already tested in the previous EUCAR-project on compatibility, was tested by ACEA with three different crossbeams: A stiffened crossbeam, a serial crossbeam and a weakened crossbeam (Figure 7,Figure 10 and Figure 13). The idea was that homogeneity of front structures is beneficial. A crossbeam improves the distribution of forces exhibited by the front-end of a car and therefore the homogeneity, at least on the level of the longitudinals. This offered the opportunity to check how test procedures under consideration evaluated this change in front-end design.

Two barriers were used: The barrier designed by TRL with two deformable honeycomb layers of 150mm each and 125*125 mm² load cells, the FWB. The barrier, designed by French researchers, using a deformable layer with increasing stiffness, the PDB.

The results to these tests were presented to EEVC and IHRA to make them available to the scientific public. A brief overview of the results is given.

Figure 7: Rover 75 with a strengthened crossbeam.

After the crash with the full-width barrier, the crossbeam was deformed and creased in the middle Figure 8.

Figure 8: Rover 75 with a strengthened crossbeam after crash with FWB.

A surprising result was that the strengthened crossbeam was not stiff compared to the barrier. This indicates that our opinion about "stiff" crossbeams has to be revised with regard to load distribution.

Figure 9: Rover 75 with a strengthened crossbeam after crash with PDB.

The same figures are provided for the serial and weakened crossbeams:

Figure 10: Rover 75 with a serial crossbeam.

Figure 11: Rover 75 with a serial crossbeam after test with FWB.

Figure 12: Rover 75 with a serial crossbeam after test with PDB.

Figure 13: Rover 75 with a weakened crossbeam.

Figure 14: Rover 75 with a weakened crossbeam after crash with FWB.

It is not possible to provide the many different observations that could be derived from this test series. Only the main findings are reported:
Both barriers, when visually inspected, showed an imprint that reflected the different stiffness of the crossbeams.

When observing the deformation of the FWB, the visual inspection showed clearly the deeper imprint in the barrier by the stiffer crossbeam. After scanning the imprint, results showed that the strengthened crossbeam deformed 43.5% of the assessment area more than 150mm, the serial crossbeam only 28.2% and the weakened layer only 22.8%. (the assessment area was located between 330mm and 580mm of ground clearance).

Figure 15. Rover 75 with a weakened crossbeam after a test with the PDB.

The PDB distinguished the three crossbeams, when the deformation of the longitudinals was considered. The strengthened crossbeam induced a longitudinal deformation of 427mm, the serial crossbeam 354mm and the weakened 178mm. This was an evident result, because the stiffer the crossbeam, the more load the crossbeam can distribute to the longitudinals and the more the longitudinals will deform.

Although this indicated that the barriers behaved in an manner expected, all other assessment procedures under consideration (PDB assessment and TRL homogeneity assessment) failed [2,3].

This raised the question of the validity of force and/or deformation measurement. This was the reason, to conduct a second test series, discussing the reproducibility of the data. The question was whether the assessment procedures failed, because they were wrong or in a certain way misleading or because the data were too biased due to measurement problems, so that a test procedure is not able to derive a valid result.

Steps toward Partner Protection

The Reproducibility of Test Results in the FWB and PDB configuration

<u>Full Width Test FWB</u>

A full width test was already conducted at TRL in the United Kingdom. So another test was conducted at UTAC in France. The test was in fact a reproducibility test, examining the test procedure itself, the assessment procedure and the definition of the test procedure, whether another test institute is able to regain the results.

The test conditions were:

- Overlap 100%
- Speed 56km/h
- Load Cell Wall 16x8 Matrix @ 125mm2
- Deformable Face of Aluminum Honeycomb
- Barrier Faces
  150 mm @ 0.34 MPa
  150 mm @ 1.71 MPa
- Ground Clearance 50mm TRL and 80mm UTAC

The different ground clearance was consequence of the fact that the test procedure had changed during the performance of the two tests. Another difference between the two tests was the different ride height of the two cars. This resulted in a difference in impact point with respect to the grid of load cell attached to the rigid wall. The difference in impact point, measured with respect the lower edge of the load cell grid, was 46.5mm. The difference in impact points with respect to the ground was 16.5mm. In the TRL test, the car impacts the wall 46.5mm higher within the load cell grid than in the UTAC test. In other words, the assessment area of the TRL test was 46mm lower than in the UTAC test. The load cells were square with the dimension of each side equal to 125mm. 46.5mm reflects around a 30% overlap of the load cell in the vertical direction. This may have had implications for the force measurement. The implications for the deformation measurement can be considered to be negligible.

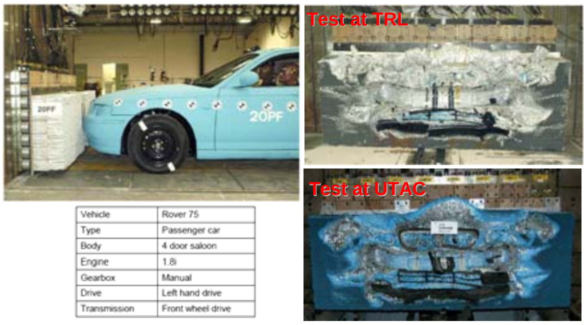


Figure 16: Reproducibility test - Rover 75 against a full width barrier.

Visual inspection of both barriers shows similar deformation behavior. Although there were differences in ride height, the imprints of the sub-frame, longitudinal and crossbeam were seen in both barriers. The deformation based results appear reliable, Figure 16. The force based results may be influenced by the difference in vehicle ride-height and barrier deformation.

The same holds true for the cars, Figure 17 and Figure 18. Visual inspection of both cars reveals a similar structural behavior. In both cars, similar welding spots of the Rover75 longitudinal failed.

Figure 17: Deformation of the Rover 75, tested in full width test at TRL.

Figure 18: Deformation of the Rover 75, tested in full width test at UTAC.

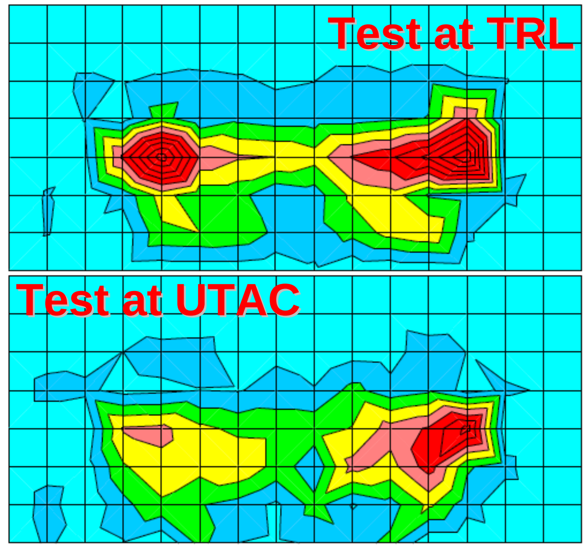


Figure 19: Force contour plots of the Rover 75, tested in full width test.

The imprint in the layer looks similar, but the force contour plots show differences. Figure 19 shows that longitudinal in the UTAC-test was deformed to a lesser degree than in the TRL-test, explaining the higher forces in TRL-test. But this is not reflected by compartment acceleration. The deceleration curves are similar, besides a difference in the peak acceleration over a time interval of 10 ms duration Figure 20.

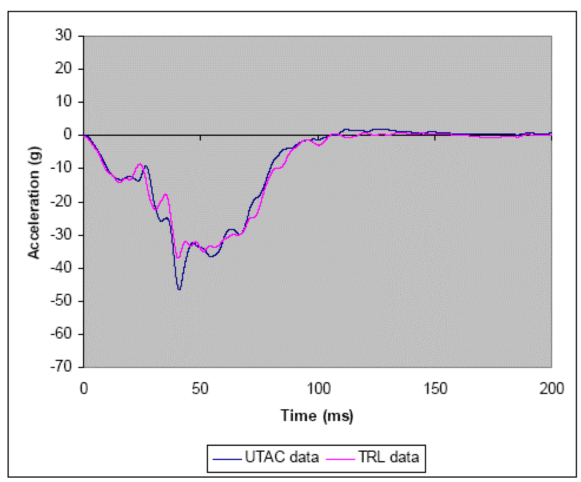


Figure 20: Acceleration plots of the Rover 75, tested in full width test.

Due to the differences seen in the force contour plot, a reaction of the homogeneity criterion would be expected. But the homogeneity criterion does not react significantly. The relative homogeneity is similar for both tests, although the TRL test shows a slightly better homogeneity Figure 21. The same holds true for the adjusted Average Height of Force AHOF. It was adjusted to the ride height differences mentioned above. The values for the AHOF were 411mm and 420mm for the test at UTAC and TRL, respectively.

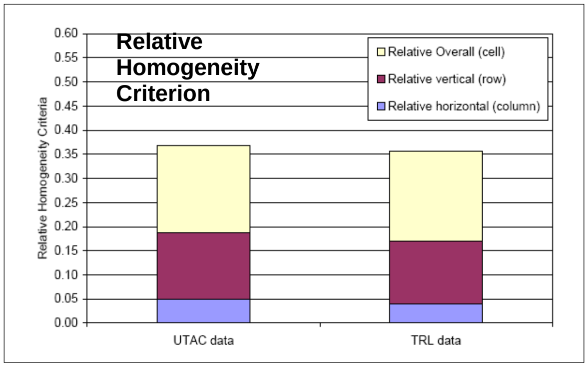


Figure 21: The relative homogeneity of the Rover 75, tested in full width test.[2]

There is an instability of the barrier deformation as well. The deformation of the barrier (vehicle imprint) is described by a cumulative curve in Figure 22. This describes the percentage of the assessment area which was deformed in each cumulative depth increment, from the wall to the front-face of the barrier (assessment area 1600mm wide with a lower limit at 330mm and an upper limit at 580mm). At 150mm (the interface plane between the two layers) this curve describes the percentage of the assessment area that has a completely deformed second layer. The assessment area corresponds to row 3 and 4 of the load cell wall (for a load cell wall ground clearance of 80mm) and the width covers all load cells hit by the longitudinals and in those in between. For force measurement, this adjustment was not possible.

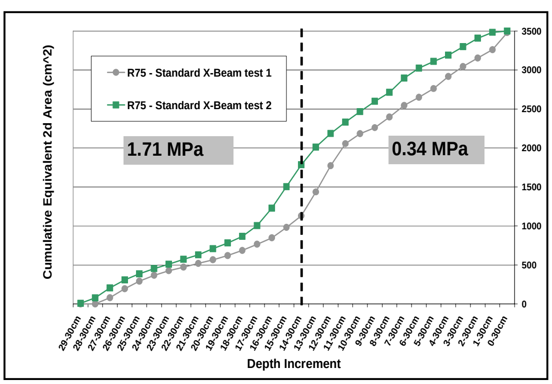


Figure 22: The deformation of the Rover 75, for tests carried out at UTAC and TRL, respectively.

Figure 22 shows a significant difference between the two cars, especially in the range of 20mm to 200mm. This has to be examined further on, because there is an expectation that deformation is a stable value. This test series indicates that this is not always the case.

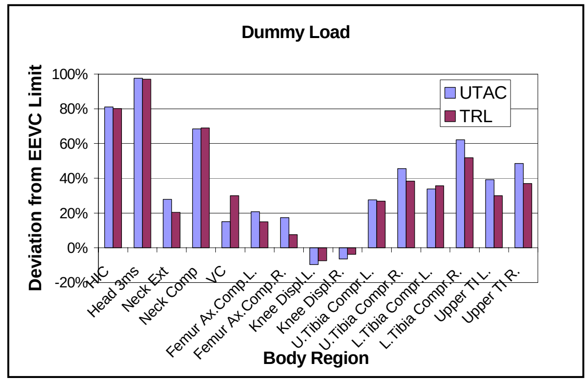


Figure 23: The dummy load of the driver of the Rover 75, tested in full width test.

Dummy loads were not the main focus of this test series but they were measured and documented in Figure 23and Figure 24. Roughly speaking, HIC and neck criteria are similar and the other body regions show differences, which are substantial in some cases.

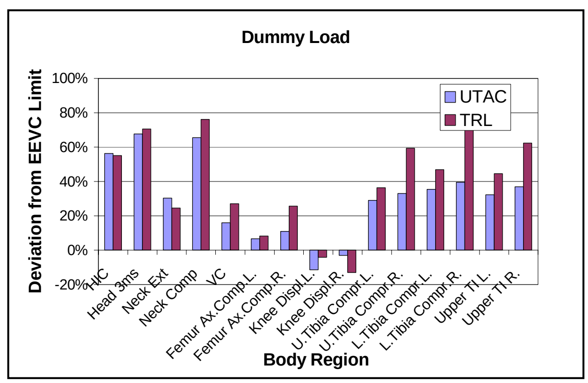


Figure 24: The dummy load of the passenger of the Rover 75, tested in full width test.

Progressive Deformable Barrier PDB

A Rover 75 test with PDB, conducted by ACEA at UTAC, already existed. So a second test was conducted at TRL in accordance to the PDB-test procedure.

UTAC test

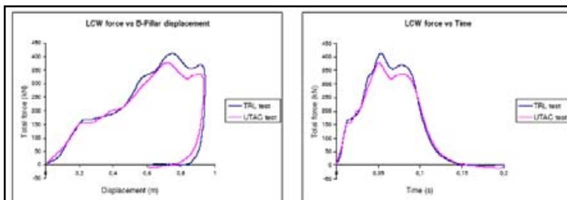
TRL test

Figure 25: The deformation of the Rover 75, tested in the PDB configuration.

There are clear differences in the deformation of the car. These are partially due to the car itself. The welding spots in the two cars, which were both manufactured in the same year, were different. So for all FWB-tested cars and the UTAC-PDB tested car, some of the welding spots failed.

Figure 26: The longitudinal of the Rover 75, tested with PDB.

The upper picture shows the behavior of the car tested at UTAC. This corresponds to the behavior of all longitudinals in FWB-testing. The lower picture shows the behavior of the car tested at TRL. The position of the welding spots was different for this car.



ROVER 75	Ec	Calculated energy	Maximum force
UTAC	220 kJ	211 kJ	378 kN
TRL	223 kJ	225 kJ	413 kN

Figure 27: The total force of the Rover 75, tested by PDB.

The total force plots, Figure 27, show slight differences which reflect the fact that the longitudinal behaved differently in both tests. The calculated energy was calculated based on the volume of the barrier deformation. These differences indicate that there are slight differences in deformation as well. Unfortunately, for the PDB, a curve comparable to Figure 22 is not available. However, there is a contour plot of the deformation available, Figure 28.

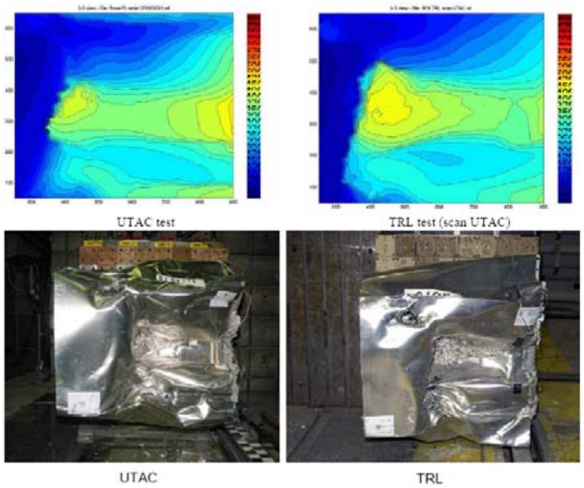


Figure 28. The deformation of the PDB against Rover 75.

The deformation plots reflect the different deformation modes of the longitudinals as shown in Figure 26. A higher degree of deformation of the longitudinal member means less penetration into the barrier which results in more being applied to the right edge of the PDB, which is loaded during the rotational phase of the car.

ROVER 75	ADOD (X)	AHOD (Z)	Volume	Energy
UTAC test	238 mm	467 mm	146 dm^3	60,5 kJ
TRL test (scan UTAC)	236 mm	470 mm	138 dm^3	56,5 kJ
Variation	0,8 %	0,6 %	5,4 %	6,6 %

ROVER 75	PPAD
UTAC test	5,5
TRL test	5,7
Variation	3,5 %

Figure 29. Results to different assessment metrics, derived from the PDB-test against Rover 75.

Although the imprint looks different, the assessment does not react significantly.

The TRL test was scanned by TRL and by UTAC. The scans were similar, so that scanning can be understood as a stable measurement at a given barrier deformation.

When observing both vehicles (post-crash), the behavior of the longitudinals is clearly reflected by the deformation of the cars, Figure 30.

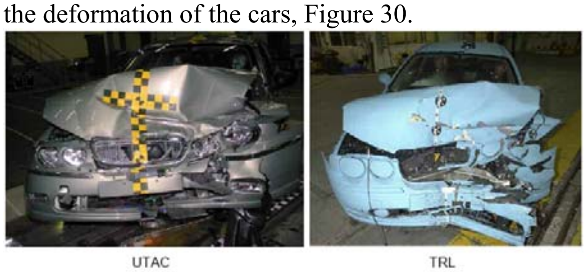


Figure 30. Post crash photographs of Rover 75's crashed at UTAC and TRL in the PDB configuration.

Performance parameter		Driver			Passenger			EEVC limits
		UTAC	TRL	Diff*(%)	UTAC	TRL	Diff*(%)	
Head	HIC	480	680	+20.0	364	469	+10.5	<1000
	3ms exceedence (g)	54.2	59.9	+7.1	43.7	56.3	+15.6	<80
Neck	Extension –Myoc. (Nm)	29.6	12.0	-30.9	34.4	11.7	-39.9	<57
Chest	Compression (mm)	28.1	38.6	+21.0	26.3	30.7	+8.8	<50
	Viscous criterion (m/s)	0.09	0.19	+10.0	0.11	0.15	+4.0	<1
Knee	Displacement left (mm)	-0.13	-0.63	+3.3	-0.36	-2.72	+15.8	<15
	Displacement right (mm)	-0.79	-0.71	-0.5	-0.63	-0.29	-2.3	<15
Upper tibia	Compression left (kN)	1.38	1.81	+5.4	1.61	1.63	+0.3	<8
	Compression right (kN)	1.32	1.61	+3.6	1.77	1.90	+1.6	<8
Lower tibia	Compression left (kN)	1.48	2.08	+7.5	2.04	2.18	+1.8	<8
	Compression right (kN)	1.61	2.18	+7.1	2.33	2.54	+2.6	<8
Tibia Index	Upper left	0.28	0.31	+2.3	0.31	0.30	-0.8	<1.3
	Upper right	0.29	0.37	+6.2	0.33	0.35	+1.5	<1.3
	Lower left	0.31	0.45	+10.8	0.15	0.19	+3.1	<1.3
	Lower right	0.31	0.24	-5.4	0.13	0.19	+4.6	<1.3

*Difference TRL compared to UTAC and expressed as a percentage of the EEVC limit

Figure 31. The dummy loads measured in the PDB-tests involving the Rover 75.

There are large differences noted for the head, neck and chest injury criteria for the two tests, with worst injury criteria differing by 30.9% and 39.9%, for the driver and passenger respectively. However, in all cases the test measurements did not exceed the EEVC limits.

These repeatability and reproducibility tests showed a couple of interesting results that were not obvious in the beginning. Together with the tests carried out with the Rover 75 with different crossbeams, they raised a lot of questions with regard to an assessment procedure to adequately predict the structural interaction potential of passenger cars.

The Roadmap

Together with these technical problems, there are a couple of problems that are related to the different traffic situation in the U.S., in Europe, in Asia and in the developing countries. The interests of car manufacturers diverge, depending on their model-mix. However, since more and more manufacturers tend to sell the most models in most world-markets, these differences diminish. Last but not least, there is also a concern about impacts with trucks. A car structure should be able to interact with a truck under-run protection system. These are the conflicts of goals that have to be solved by a compatibility test procedure.

Car mass and the type of car (e.g. passenger car, MPV, Mini-Bus, SUV etc) reflect customer demand. It is the unanimous position of automotive industry that compatibility requirements should be made in such a way that customer demand can be fulfilled in the future as well. A restriction of mass, for example, is unacceptable and makes no sense as long as trucks are still on the road. This statement is also true considering the structural design of a car. Requirements should address the vehicle performance and not restrict design possibilities.

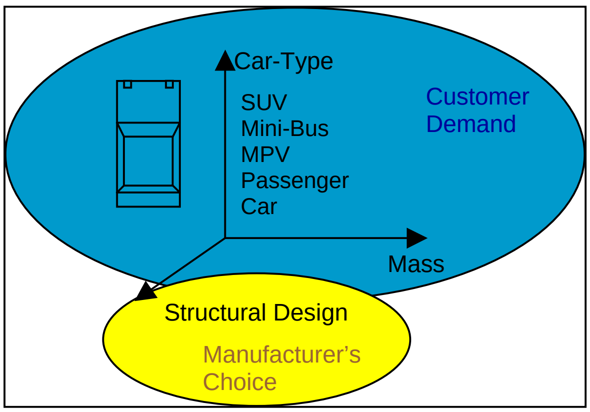


Figure 32: The dimensions of the challenge of compatibility.

Together with the dimensions of the car under consideration, the characteristics of all potential impact objects have to be taken into account as well. .

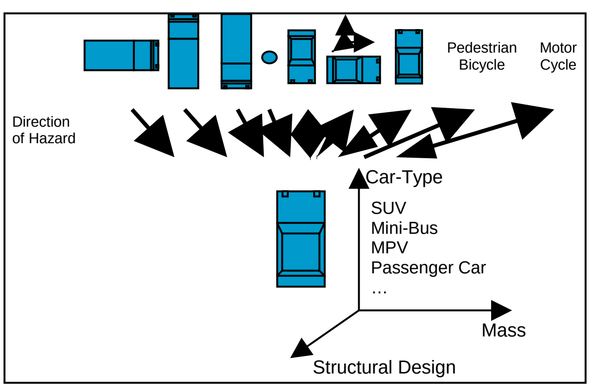


Figure 33: The opponents to be taken into account, when dealing with the challenge of compatibility.

Figure 33 shows the complexity of the challenge to improve compatibility. The idea is not to request a solution to all open questions in one big step. However, it is a reminder, not to worsen the situation in one of these configurations, when improving the situation in another configuration.

When looking at the players (or stakeholders), things become even more complicated, Figure 34. There is a lot of world-wide expectation with regard to compatibility with many parties contributing to compatibility research and decision making. This contribution has a multi-facetted political back-ground,

Figure **35**.

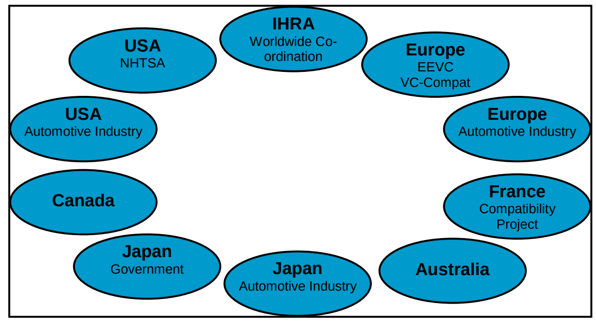


Figure 34: The players.

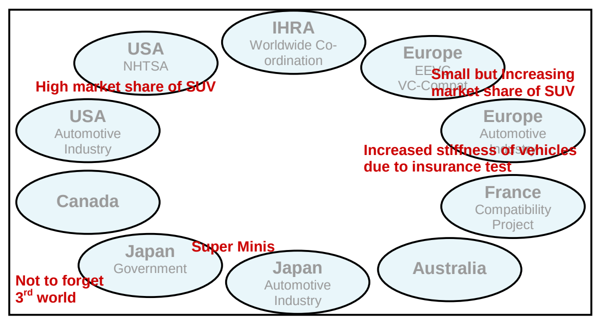


Figure 35: The political back-ground and the restrictions for the players.

In addition to the differences within the current fleets in different regions of the world, there is also different experience in crash testing and different research emphasis, Figure 36.

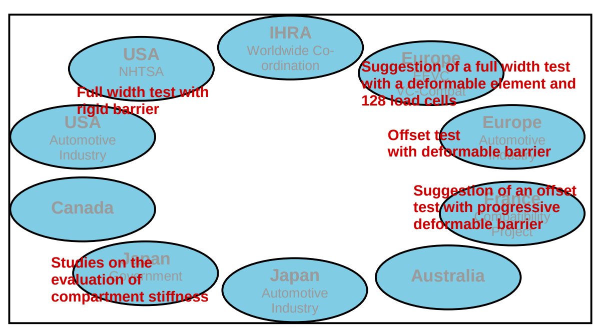


Figure 36: The current situation of convergence between the players.

Taking all of these different aspects into account, ACEA tried to find a step wise approach for compatibility. These are issues which are very difficult to achieve without compromising other goals, such as management of front end forces within the fleet. It is evident that this will never be solved completely, because force requirements between a car of 2000kg and above are definitely different to the force requirements of a car of 800kg. The details are discussed in former ESV papers by the authors. An agreement was made within the automotive industry at the very beginning, that improving structural interaction is the most appropriate first step to improve compatibility. It seems to be possible to achieve this goal without compromising other goals. The goal of structural interaction is in line with the ideas in the U.S.A.

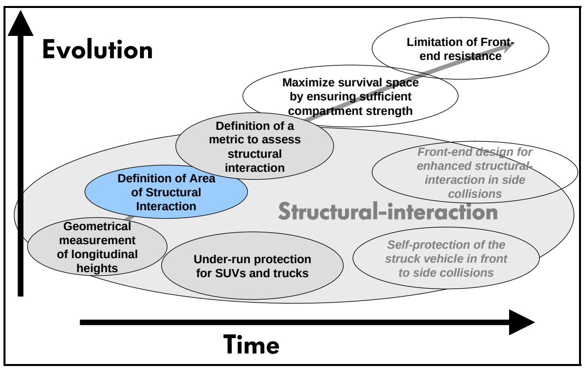


Figure 37: The Road Map.

CONCLUSIONS

- Single vehicle accidents remain a highly relevant collision mode which should not be neglected.
- Self protection should not be compromised by compatibility requirements.
- Tests with deformable barriers have to ensure that the self protection level of the vehicles tested is not reduced. For any mass class, for large and small cars.
- Mass dependent tests should be avoided.
- Tests should be able to detect stable crossbeams as a contribution to homogeneity.
- Current assessment procedures are not able to detect stable crossbeams.
- Assessment procedures have to be studied carefully, how they drive the development of the fleet.
- Reproducibility tests showed that there are still deficits as far as force measurement and deformation measurement is concerned. Further research is required in this area.
- The ACEA reproducibility test series is a worst case scenario. Repeatability tests with absolutely identical case vehicles at the same test institute should follow.
- Customer demand and, as far as possible, manufacturers choice regarding design should not be inhibited by compatibility requirements. Requirements should describe effects not prescribe design. Governmental requirements must be performance and not design-based. To encourage and not stifle innovation, government standards must regulate vehicle performance, but not vehicle design measures.
- Compatibility requirements should be introduced stepwise and in a world wide harmonized manner, because only a harmonized approach is able to result in compatibility fleet across the various world markets.

ACKNOWLEDGEMENTS

This paper is based on scientific research conducted within the ACEA, Sub-group Compatibility (SGC). The first co-author chairs the ACEA SGC and has attempted to summarize the results of numerous discussions that have occurred within the group. The authors wish to express their appreciation to the members for their personal contributions. Virtually all of the member companies believe that it is premature to formulate final conclusions on this complex subject.

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