Vehicle compatibility in car-to-car collisions

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Literature review in the framework of the European research project "Improvement of crash compatibility between cars", Workpackage 1

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Summary

Within the scope of the project “Improvement of crash compatibility between cars” scientific literature from the last 15 years was reviewed in order to obtain state-of-the-art knowledge about car compatibility in car-to-car accidents.

In the first place, current definitions of (in)compatibility were reviewed. There appears to be a variety of definitions and descriptions. They have in common that, while improving occupant safety is (traditionally) the main purpose of improving crash safety of cars, currently (improving) the safety of occupants of opponent cars is of equal concern. The first property (the level of occupant safety) is often called the level of self-protection, while the second item (the level of opponent safety) is often described as the level of aggressiveness of a particular car. Therefore, combining these different purposes in one, compatibility may be described as “the capability of cars to protect their occupants in crashes, while at the same time produce as less harm as possible to occupants of opponent cars”.

Literature was reviewed from three points of view: statistical, mechanical and geometrical. These views have been constructed to cover the whole range of scientific literature, including proceedings of international conferences as ESV, STAPP, IRCOBI and SAE and specialized journals.

From the *statistical point of view*, literature focussing on the influence of car mass (including mass ratio) and car size on the injury severity of occupants is reviewed; also several existing crash rating systems are discussed. In almost all reports, the main data sources used for analysis are statistical accident data.

The influence of car mass (and mass difference) appears to be very well documented, the main conclusion being that the fatality or injury risk is inversely proportional to car mass or mass difference: the heavier the car the lower the fatality or injury risk (and the higher the aggressiveness). Some authors also described the fact that in (frontal) collisions between cars of equal mass, the outcome was better for heavier than for lighter cars.

The question, whether the influences described above should be solely attributed tot car mass or to car size as well, was addressed by at least one author (Evans), who concludes that car mass appears the stronger influence of the two, though in most instances size and mass are interchangeable.

Crash rating systems are used to inform consumers about the level of occupant safety of different car models. Some of these systems are very sophisticated, using control variables such as exposure in traffic and driver sex and age. In general, ranking systems based on accident data reflect the important influence of car mass (and size) as described above.

From the *mechanical point of view*, literature is reviewed concerning the classical theory of mechanics of collisions (Newton mechanics). Conservation of momentum, deformation energy, and the difference

between plastic and elastic deformation are discussed and used by several authors to propose solutions to the problem of incompatibility. An example is the so-called bulkhead concept (Zobel): a maximum force level is defined for the front end of cars which should prevent intrusion into the compartments, after the crush zones of both or either car is fully deformed. Other solutions propose to change the force level of the crush zones of both cars to a more or less comparable (relatively low) level and also propose methods to measure these levels during crash testing (Steyer).

Crash testing is an important instrument for both measuring and improving the level of (occupant) safety of cars, as is currently successfully demonstrated all over the world by the existing (consumer) programmes. These programmes test (new) cars according to specific test protocols, in which the test speed is higher than the level of current legal requirements. They tend to work two ways: 1) the public awareness of the importance of crash safety of cars is increasing, and 2) improvement of the design by manufacturers is often reached without changing legal requirements.

In the chapter on the mechanical view, the importance of developing and applying mathematical modelling to analyse and ultimately improve crash safety of cars is reviewed, based on various types of models: lumped mass models en finite element method (FEM) models. These models are often used to develop new individual car design, but they will also be used to analyse the problem of incompatibility between different car types, or even models of the same car type in different collision modes. By using these instruments, numerous design and parameter changes may be analysed, without the need to build and crash test every separate item.

From the *geometrical point of view*, literature is reviewed in which authors describe influence on outcome of (differences of) car geometry and stiffness geometry that cause incompatibility. Regardless of mass and size differences, geometrical differences may also cause considerable harm. This is clearly demonstrated in case of collisions between “normal” cars and specific four-wheel-drive cars (and vans) having far stiffer longitudinals and greater structural height than these normal cars; but geometrical differences may also exist between cars of the same category, dependent on the type of collision.

Typical effects of geometrical differences are mismatches such as override/underride and fork effect of structures designed to (inter)act in case of collisions. Various solutions for this problem have been tested, such as increasing or lowering sill height (in side collisions) and increasing or lowering longitudinal height, bumper height and frontal stiffness. The difficulty of influencing (frontal or side) stiffness is that this property is almost never as homogeneous as it ideally should be, because of various local parts that contribute to the overall stiffness design. In view of the importance of geometrical properties, especially concerning stiffness, stiffness distribution, and alignment, the challenge for the coming years is to develop a test procedure of (new) car design that forces the future car fleet to converge to a far more compatible design than the current one.

In view of the various interacting influences as described in this literature review, it may be doubted that one single test may be developed for this

purpose, though some proposals already have been made, based on the current deformable barrier test, used for frontal impacts.
Probably, a second test is needed to better represent the opponent car when testing and monitoring the level of aggressiveness.

Contents

1. Introduction

This literature review on vehicle compatibility in car-to-car accidents has been performed in the framework of the project “Improvement of crash compatibility between cars” of the European Transport Research and Technological Development Programme. The SWOV Institute for Road Safety Research has been asked to carry out a review of literature published over the last 15 years.

1.1. Scope and setup of this report

This report covers deliverable D1 “Literature review” of the project “Improvement of crash compatibility between cars”. In this report a description is given of the state of the art on the subject of (in)compatibility between cars. The literature review consists of: a literature survey, a compilation of the results of relevant crash tests, and mathematical simulations performed in the past.

The objective of this literature review is to see how scientists define and tackle the problem of incompatibility between cars. It is based on scientific publications published over the last 15 years, including proceedings of international conferences as ESV, STAPP, IRCOBI and SAE and specialized journals.

The setup of this report is as follows. The compatibility problem is described from three points of view, the statistical, the mechanical and the geometrical. The statistical view (Chapter 2) has been divided into three sections. The first two sections cover two main explaining variables used to analyse car accident data, vehicle mass and vehicle size. The third sections deals with statistical methods applied to rank vehicle models on crash-worthiness and aggressiveness.

In the third chapter, the compatibility problem is described from a mechanical point of view. In the first section, literature is referenced in which the theory of classical crash mechanics is used to analyse compatibility. In the three remaining section of this chapter, the mechanical aspect of accident analysis, crash tests and computer simulation are dealt with. There is clearly a wide range of literature available in this area, but only those publications are referenced in which the authors themselves made a link to compatibility.

The last view concerns the geometrical aspect of compatibility. Subject of discussion here are differences in car geometry and stiffness geometry which lead to incompatibility in car crashes. Geometrical issues are also part of the views described in the previous two chapters. The reasons for devoting a separate chapter to the geometrical aspect is that differences in the geometry of cars, both car shape and stiffness distribution, play a key role in the solution of the incompatibility problem.

1.2. Definitions of (in)compatibility

The term “compatibility” is used in several publications. A strict definition of car compatibility has not been found, however. In this section a number of descriptions of the term are offered, all found in the publications studied in this literature review.

In the technical annex of the EU project “Improvement of Compatibility between Cars” the term “compatibility” has been defined as follows:

> “In protecting car occupants most activity has been associated with improving the occupant’s own car to aid his protection. In future, improvements should be possible from improving the front of the other car involved. The term ‘compatibility’ has been coined to describe this subject.”

F. Niederer et al. (1995) describe compatibility as follows:

> “In qualitative terms vehicles are denoted as collision compatible if their deformation characteristics are such that they do not impose excessive loads on the occupant compartment of the collision partner under a well-defined set of crash conditions. In particular, a collapse of the passenger compartment of the impacted car has to be avoided.”

Shearlaw and Thomas (1996) give a description of the opposite of compatibility:

> “Vehicles in collision can be said to be incompatible if the deformation and structural characteristics mean the occupant loads are unequally distributed between the vehicles.”

Schoeneburg et al. (1996) do not give a definition of the term compatibility but define the goal which should be achieved by research in this field:

> “The goal of compatibility is [...] to enhance partner protection without decreasing occupant protection or to optimize occupant protection in such a manner that the overall safety of the vehicle is maximized.”

They define partner protection as the ability of the other car which collides with the car under examination to avoid injuries in the car which it strikes.

Klanner, Felsch and Van West (1998) define compatibility as a vehicle feature which is related to both self-protection performance and aggressiveness towards the other vehicle. The level of aggressiveness depends on the amount of energy and the force transmitted to the other party but also on whether such transmission is distributed over a broad surface or has only a local quality.

2. The statistical view

Aspects of car compatibility have been studied with statistical methods in several countries. Various statistical methods have been used to find the relationships between vehicle properties and injury risk and injury severity.

A general problem of using statistical methods is the difficulty of isolating vehicle properties from other confounding variables, such as driver behaviour. The exposition used in the various calculated risk measures is also an issue of discussion.

Statistical analysis largely depends on the availability of bulk crash data. The structure of these data differs from country to country, or even within a country. Furthermore, there are of course large differences in the real situation between these countries, for instance in terms of vehicle population, layout of the infrastructure and demographic issues. These differences make it difficult to compare the findings of different statistical studies.

The various publications discussed in this chapter have been grouped into three subjects, vehicle mass, vehicle size and ranking. In the section on vehicle mass, the relationship between mass and injury risk/severity is discussed. In the section on vehicle size, the size of the vehicle is used as an explanatory variable for injury risk/severity. In the last section of this chapter the performance of individual cars concerning occupant safety in crashes is described, and on this basis different vehicle models are ranked.

2.1. Vehicle mass

Several authors have published papers on the relation between vehicle mass and injury severity. The research in these papers had its roots in the discussion in the mid 1980s about the safety implications of downsizing vehicles because of higher standards on fuel economy. Moreover, in the early 1990s the discussion on compatibility returned to the international agenda after this item had blossomed during the early 1970s but then disappeared at the end of the decade. Different statistical methods are used to quantify the risk of vehicle occupants being severely injured or killed in relation to the mass of the vehicle.

Evans & Frick (1993) used data of two-car crashes from the Fatal Accident Reporting System (FARS). From these they determined the relative risk, R, of a driver fatality in the lighter of two cars compared to the risk in the heavier car as a function of the ratio, μ, of the mass of the heavier to that of the lighter. The data was fitted to the functional relationship $R = A\ \mu^{u}$. An advantage of analysing the fatality rate as a function of mass ratio is that no exposure measure is required. The analysis was performed on several subsets of the available data, to determine the influence of safety belt use, model year, absolute mass of the involved cars, impact modus and driver factors.

Evans and Frick conclude that mass is the dominant factor on relative driver fatality risk when two vehicles of different mass crash into each other.

The influence of vehicle mass is expressed in two “laws” already published by Evans (1991):
- The lighter the vehicle, the less risk to other road users.
- The heavier the vehicle, the less risk to its occupants.

These two laws are also quantified. If somebody transfers to a car lighter by 1%, this person=s risk of death in a two-car crash compared to the risk of death to the other involved driver increases by between 2.7% and 4.3%. Correspondingly, if the driver of the other involved car transfers to a car heavier by 1%, this person=s risk of death compared to the other driver decreases by between 2.7% and 4.3%. When a car crashes into another twice its mass, driver fatality risk in the lighter car is between seven and 14 times that in the heavier.

In another paper, published in the *American Journal of Public Health*, Evans & Frick (1992) examined whether the relationship between fatality risk and car mass has changed over time. They used the same methods as discussed in the previous paper. They conclude that when a car of a given model year crashes into another car of the same model year but 50% heavier, the driver in the lighter car is more likely than the other driver to be killed by a factor of about 3.7 to 5.1 for 1966 through 1979 model year cars, 2.6 for 1984 model year cars, and 4.1 for 1990 model year cars. They explain the reduction of the relative risk in the mid 1980s models by the fact that lighter cars benefit earlier from vehicle redesign than heavier cars.

In the book *Traffic Safety and the Driver* (Evans, 1991) *Table 1* is discussed, published previously by Evans & Wasielewski (1987):

Mass (kg) car i		Mass category car j					
		m_1	m_2	m_3	m_4	m_5	m_6
500- 900	m_1	7,04	12,12	15,15	16,05	16,86	16,51
900-1100	m_2	5,06	9,78	11,88	13,38	14,58	14,68
1100-1300	m_3	3,50	5,33	7,79	9,48	9,30	9,36
1300-1500	m_4	2,14	2,67	4,83	6,06	6,94	7,12
1500-1800	m_5	0,98	2,04	2,57	3,56	4,34	5,01
1800-2400	m_6	1,00	1,48	1,86	2,61	3,01	3,46

Table 1. *Relative likelihood of driver fatality in a car of mass m_i involved in a crash with a car of mass m_j (from Evans & Wasielewski (1987).*

Table 1 shows the relative likelihood of driver fatality in a car of mass m_i involved in a crash with a car of mass m_j. The table contains six mass categories. The values are calculated on the basis of an exposure estimate based on the number of pedestrians killed in crashes involving cars in the six mass categories. All values are expressed relative to an arbitrary value of unity, namely a car in the heaviest mass category crashing into a car in the lightest category (m_6 x m_1). Evans (1991) concludes from the data that as a car’s mass increases, the fatality risk in that car decreases, but the risk in the other car involved increases. Another question Evans addresses is what happens to the net number of fatalities if a car driver transfers from a lighter car to a heavier car. Within the six mass categories there are 15 possibilities of a driver to move to a heavier car. An m_1 driver can move to an m_2-m6 car, an m_2 driver can move to an m_3-m_6 car etc. If, for example, an m_3 driver moves to an m_4 car, one can compare the combined risk of a crash against an m_1 car. The combined risk of an m_3 car crashing into an m_1 car is in this case $m_3 \times m_1 + m_1 \times m_3 = 3.50 + 15.15 = 18.65$. The combined risk of the m_4 car is in this case $m_4 \times m_1 + m_1 \times m_4 = 2.14 + 16.05 = 18.19$, which is slightly less. From the 90 comparisons, 15 possible moves x six cars, Evans finds 78 cases with a net fatality decrease.

Evans supports his conclusion that substituting a heavier car for a lighter car nearly always reduces the system-wide harm from two-car crashes with a relationship fitted by Joksch (1983) to state injury and fatality data. This relationship can be expressed as:

$$\mathbf{y = e^{-0.001102M_1 + 0.000441M_2}} \tag{1}$$

where y is the relative risk of an injury to an occupant in a car of mass m_1 when it crashes with a car of mass m_2 (masses in kg). If we add the risk to the driver in car m_1 to the risk to the driver in car m_2, we obtain an expression for the combined risk to both drivers. Doing so with the median values of the weight classes used by Evans (1991) in *Table 1*. One can do the same 90 comparisons he performed with his data. This leads to a net fatality decrease in 77 cases. However, the weight substitutions which do not lead to a net fatality decrease are in most cases different from the data from Evans (1991) than those obtained with the expression from Joksch (1983). According to Evans, these differences are spurious, especially as the regions where the inversions occur are different.

Evans (1991) finds another mass effect concerning cars of equal weight crashing into each other. From both the data in table 1 and the expression found by Joksch (1983), Evans (1991) finds that the heavier the car, the lower the risk of a fatality.

Substitution of m_2 by m_1 in *Equation 1* generates the following equation:

$$\mathbf{y = e^{-0.000661M_1}} \tag{2}$$

It follows from *Equation 2* that the relative fatality risk in a crash between two 900 kg cars is 1.8 times higher than in a crash between two 1,800 kg cars.

Thomas et al. (1990) analysed data taken from the computer file of the French national police. They sought to quantify the risks induced by varying levels of aggressiveness between cars involved in collisions with one another. The cars involved in head-on crashes were divided into three weight categories, < 800 kg, 801 to 1,000 kg and > 1,000 kg. For each weight category the internal and external safety was determined for two severity rates. The internal safety is associated with the occupants of the case vehicle, the external safety is associated with occupants of the opposing vehicle. Firstly, the internal and external safety were expressed in a mortality rate. Secondly, the internal and external safety were expressed in a severe injury rate. In this rate both fatalities and severe injuries were taken into account. The results are shown in Figure 1, taken from Thomas et al. (1990).

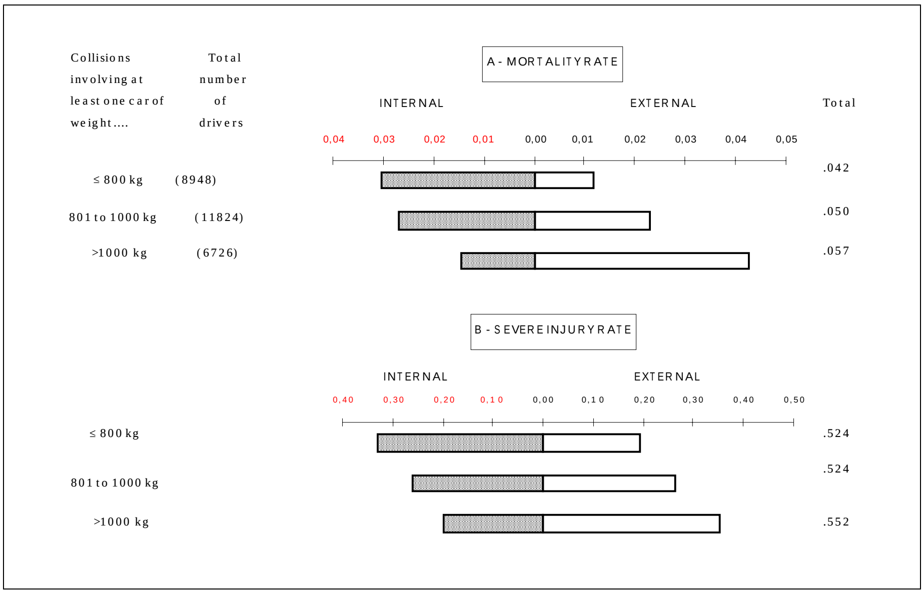

Figure 1. *Mortality and severe injury rate (internal and external) for 1000 belted drivers according to the weight of the car in head-on collisions (reproduced from Thomas et al., 1990).*

Figure 1 shows that the interior mortality rate decreases with the rise in mass of the cars in question (from 0.0305 to 0.0145). The corollary of this is the rise in the exterior mortality rate in the opposing vehicle (from 0.0119 to 0.0425).

For 1,000 collisions involving at least one light vehicle, the total number of fatalities in both vehicles is 42, as against 50 and 57 when the cars are in the average weight category or the heavy weight category respectively. When the severe injury rate is used, these figures vary only fractionally with the weight category.

Thomas et al. (1990) also present a graph in which the influence of the mass ratio is shown. Reproduced here as *Figure 2*, this shows that in a head-on crash between a car from < 800 kg category and a car from > 1,000 kg the interior safety of the < 800 kg car equals 0.421 and the interior safety of the > 1,000 kg car equals 0.147. In this case this can also be read as follows: the exterior safety of the < 800 kg car equals 0.147 and of the > 1000 kg car 0.421.

Both *Figures 1 and 2* show that the interior safety increases with higher weight and that the exterior safety decreases with higher weight of the involved car.

Figure 2 also shows, by the discontinuous line, that in crashes between cars from the same weight category the interior safety of the car decreases with an increase in the weight category. This effect, which is in disagreement with the findings of Evans (1991), according to Thomas et al. (1990) is due to higher closing speeds in the higher mass category and a bias caused by driver age.

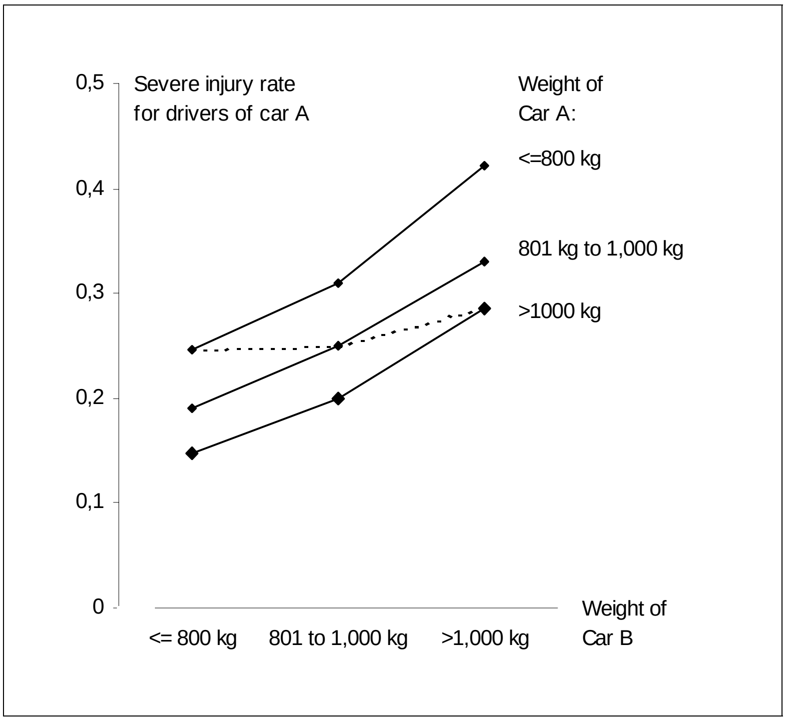

Figure 2. *Severe injury rate for belted drivers in car-to-car head-on collisions according to weight class (reproduced from Thomas et al., 1990).*

Like Thomas et al. (1990), Fontaine (1994) also used data from the French police to quantify the way in which explanatory variables affect the rate of severity. The rate of severity in this study is defined as the probability of a driver who is involved in a crash being killed. A logistic regression method is used, in the following form:

$$\mathrm{Log}\left(\frac{p}{1-p}\right) = \sum_{i=1}^{k} a_i v_i \qquad \textbf{(3)}$$

p = the probability of the driver being killed
C = constant
a_i = multiplying factor of variable i
v_i = the explanatory variable i.

The parameters of this model are estimated for front/front head-on collisions. The following explaining variables are taken into account:
- W_{int}, weight of the vehicle under consideration
- W_{ext}, weight of the other vehicle
- PW_{int}, power-to-weight ratio for the vehicle under consideration
- PW_{ext}, power-to-weight ratio for the other vehicle
- A_{int}, age of the vehicle under consideration
- A_{ext}, age of the other vehicle.

The factors linked to power-to-weight ratio and vehicle age are dropped by the model using the Somer's D indicator, resulting in the following relationship:

$$\mathbf{Log}\left(\frac{\mathbf{killed}}{\mathbf{not\ killed}}\right) = -\mathbf{1.3815} - \mathbf{0.00253} * \mathbf{W}_{\mathbf{int}} + \mathbf{0.000859} * \mathbf{W}_{\mathbf{ext}} \quad \textbf{(4)}$$

Equation 4 shows that the lighter the vehicle in question, the heavier the other vehicle, the greater the risk of being killed in the event of a head-on collision.

In the case of an impact between vehicles of the same weight W, *Equation 4* transforms into:

$$\mathrm{Log}\left(\frac{\mathrm{killed}}{\mathrm{not\ killed}}\right) = -1.3815 - 0.001671 * \mathrm{W} \quad \textbf{(5)}$$

The coefficient of variable W does not differ significantly from zero. Hence, no mass effect is found here in head-on crashes between cars of equal weight.

2.2. Vehicle size

In a paper entitled "Car size or car mass: Which has greater influence on fatality risk" Evans & Frick (1992) described whether car mass or car size is the causative factor for injury risk. A similar method to the one used in the previous section to quantify the relationship between relative injury risk R and size ratio v, defined as the ratio between the wheelbase of the two cars and v > 1. Data was extracted from the FARS. Evans and Frick come to the following finding:

$$\mathrm{R} = \nu^{\,6.76 \pm 0.24} \quad \textbf{(6)}$$

They also derive a relationship between car mass M and wheelbase W in the following form:

$$\mathbf{M} = \alpha\ \mathbf{W}^{\beta} \quad \textbf{(7)}$$

For cars of model year 1980 and later Evans & Frick (1993) find α = 109.1 ± 2.5 and β = 2.51 ± 0.024. The relationship between the relative injury risk R and the mass ratio μ for 1980 model cars or later reads:

$$\mathbf{R} = \mu^{u} = \mu^{2.75} \quad \textbf{(8)}$$

When *Equation 7* is substituted in *Equation 8* the following relationship is found between the relative injury risk and wheelbase ratio:

$$R = \left(\frac{M_2}{M_1} \right)^{2.75} = \left(\frac{\alpha \ W^{2.51}}{\alpha \ W^{2.51}} \right)^{2.75} = v^{6.9} \qquad \textbf{(9)}$$

The exponent of v in *Equation 9* is in close relationship with the exponent found directly from the data in *Equation 6* (6.8 ≈ 6.9). The much stronger relationship by wheelbase ratio than by mass ratio (Evans & Frick, 1992), as indicated by an exponent of 6.8 compared with 2.75, should not be interpreted to imply that wheelbase is a better explainer of fatality. Rather, the relationship between fatality ratio and wheelbase ratio follows directly from the relationship between mass and wheelbase. To support this view, Evans and Frick also analyse crashes between cars of similar wheelbase and different mass and crashes between cars of similar mass and different wheelbase.

In the first of these two analyses, 280 crashes between cars of model year 1980 and later were studied, whose wheelbases differed less than 1 cm. The value found for u, 3.76 ± 0.89, is according to Evans and Frick consistent with the value 2.75 ± 0.11 shown in *Equation 8*. The conclusion from this analysis is that when the wheelbase is held constant, mass influences driver fatality ratio just as strongly as when the wheelbase varies.

In the second analysis, where crashes between cars of similar mass but with a different wheelbase were studied, the wheelbase is not found to have any significant influence. The overall conclusion of Evans & Frick (1992) is that, in view of the large dependence of driver fatality risk on mass in two-car crashes, mass is the dominant causative factor, with size playing at most a secondary role.

2.3. **Ranking**

Krafft et al. (1991) used accident data from the Swedish National Bureau of Statistics (SCB) and injury severity data reported to Folksam Insurance Company to calculate a safety rating of car models. They used the accident data to calculate the relative injury risk R with the paired comparison method developed by Evans:

$$R = \frac{x_1 + x_2}{x_1 + x_3} \qquad \textbf{(10)}$$

where x_1 is the number of cases where both the driver in the specific car and the driver in the opposite car were injured, x_2 the number of cases where the driver in the specific car was injured but not the driver in the opposite car, and x_3 vice versa. The ratio R was calculated on the basis of all cars that were involved in a crash with the specific car, since the number of crashes between one car against one other car is too low. The used data contained 13,228 accidents over the years 1985-1989. The risk ratio was corrected for each 100 kg service weight by reducing or adding 0.05.

The injury severity is expressed by the Rating System for Serious Consequences (RSC):

$$\mathbf{RSC} = \mathbf{r_f} + (1 - \mathbf{r_f}) * (1 - \prod_{i=1}^{n} (1 - \mathbf{r_{id}})) \qquad \textbf{(11)}$$

where r_f is the risk of dying associated with an Injury Severity Scale (ISS) value, and r_{id} is the risk of being medically disabled as a result of a certain Abbreviated Injury Scale (AIS) level to body region i.

RSC is a scale from 0 to 1 which reflects the risk of either dying or sustaining permanent disability of at least 10% according to the procedure used by the Swedish insurance companies. The material used to calculate RSC was based on insurance claims reported to Folksam Insurance Company between 1976 and 1989. Only adult front seat occupants are included, and the total sample is 26,764.

To calculate the risk Z of receiving a disabling or fatal injury in accident, the injury risk R and the injury severity RSC were matched:

$$\mathbf{Z = R * mrsc} \qquad \textbf{(12)}$$

where R is the relative risk of being injured based on the paired comparison, and mrsc is the arithmetic mean of the risk of serious consequences in terms of death or disability. The value of mrsc was not correlated to service weight.

With the described method Krafft et al. (1991) calculated Z for 47 car models. They find a variation of more than five times between the best and the worst car model in the sample. They also find a clear correlation between vehicle weight and the safety level as measured with Z. This risk is not homogenous, however, in the sense that even among the small cars there are specific car models with a very low risk of death and disability, while among the larger cars there are examples of models with the safety level of small cars. This shows that weight of the vehicle is only one predictive factor among others.

Cameron et al. (1996) used a similar expression as Krafft et al. (1991) to calculate a measure to rate crashworthiness for different car models. The following expression was used by Cameron et al.:

$$\mathbf{C = R\ x\ \ S} \qquad \textbf{(13)}$$

where C (crashworthiness rating) is a measure of the risk of serious injury to a driver when the car is involved in a crash, R (injury risk) denotes the probability that a driver is injured during a crash, and S (injury severity) denotes the probability that an injured driver is killed or admitted to hospital.

Expression 13 is equivalent to *Expression 12* used by Krafft et al. (1991). However, Cameron et al. (1996) used different methods to calculate the injury risk R and the injury severity S. Both parameters were estimated by using a logistic model of probability of the form:

$$\log it(P) = \ln\left(\frac{P}{1-P}\right) = \beta_0 + \beta_1 X_1 + \ \ldots\ \beta_K X_K = F(X) \qquad \textbf{(14)}$$

The logistic regression model of the injury risk R was estimated from data on 288,612 drivers involved in tow-away crashes in New South Wales, Australia.
The conclusion is that driver sex and age, speed zone and number of vehicles, along with first-order interactions between speed zone and number of vehicles, sex and number of vehicles, age and sex, speed zone and age and speed zone and sex and second-order interactions between sex and speed zone and number of vehicles and sex and speed zone and age are all significantly associated with injury risk. These factors were included in the logistic model to calculate R.

The injury severity S (*Equation 13*) was based on 62,725 injured drivers in crashes in Victoria and New South Wales during 1987-94. In this analysis driver age and sex, speed zone and number of vehicles were also identified as affecting the result. First-order interactions between sex and age, speed zone and age, speed zone and number of vehicles and age and number of vehicles were found to have significant interactions with injury severity.

The crashworthiness rating for each car model was obtained for 109 car models by multiplying the individual injury risk and injury severity. The associated 95% confidence intervals were also calculated, to indicate that each crashworthiness rating is an estimate of the risk of a driver being killed or admitted to hospital in a tow-away crash.

A simpler approach to rank vehicles on occupant safety was applied by Tarrière et al. (1994). This paper examined 27 car models involved in head-on collisions. For each model an internal severe injury rate and an external injury rate was calculated. The internal severe injury rate of a particular vehicle was calculated as the number of drivers killed or severely injured seated in this car divided by the total number of involved drivers. The external injury safety rate is the number of fatalities and severely injured drivers seated in the opposite cars divided by the total number of involved drivers.

Van Kampen (1998) developed two metrics for expressing the passive safety of vehicle models. The metric for crashworthiness (EV) provides an indication of the occupants' safety for a certain vehicle model. This metric was calculated by dividing the number of severely injured and killed drivers of the subject vehicle model by the number of occurrences of the subject vehicle model in the used data file. The second metric, called the aggressiveness index (AV), expresses the degree of injury caused by a certain vehicle model to occupants of the opposite vehicle. This metric was calculated by dividing the total number of severely injured or killed drivers in the opposite vehicles by the total number of occurrences of the subject vehicle. Both metrics were calculated for those vehicle models occurring more than 100 times in a data file containing 5,680 accidents involving 11,356 vehicles with a vehicle mass less than 3,500 kg. The accidents were selected from the Dutch National Register on Road Traffic Accidents (VOR) over the year 1996. The vehicle data were linked to the accident data from the files of the National Vehicle Register (RDW). From the results a consistent relationship emerges: as vehicle mass increases, the crashworthiness index decreases and the aggressiveness index increases.

3. The mechanical view

In this chapter the mechanical aspect of compatibility is the subject of study. The mechanism of a two-car collision can be analysed using mechanical theory in terms of the interaction between two bodies each having their own mass distribution, stiffness properties and geometry.

3.1. Classical theory of impact

Newtonian mechanics is a relative simple theory used to describe and analyse the problem of incompatibility between cars. Before discussing a number of papers, the basic formulas of crash mechanics are given here (Goldsmith, 1960).

The collision of two non-rotating bodies at a point on the line connecting their centres of gravity is defined as central impact. The deformation history is envisaged as consisting of two subintervals, as shown in *Figures 3 and 4*: The approach period extends from the instant of contact to the point of maximum deformation, followed by a restitution period lasting to the instant of separation.

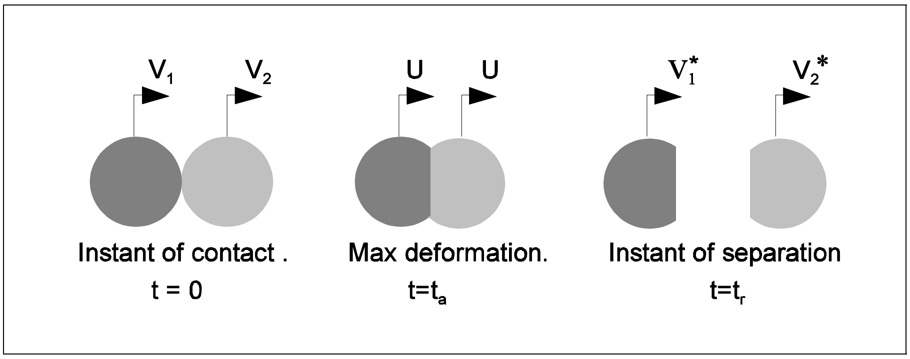

Figure 3. *Central impact of two translating spheres.*

During the period of approach ($0 \leq t < t_a$) both bodies deform. The deformation can be perfectly elastic, perfectly plastic or a combination of the two. At the end of the period of approach ($t= t_a$) both bodies travel with the same speed u. During the period of restitution ($t_a \leq t < t_r$) the elastic deformation recovers. The mathematical formulation of the classical theory of impact is based on the impulse momentum law and the law of conservation of mechanical energy. To describe loss of energy due to plastic deformation, the coefficient of restitution is defined as the ratio of final to initial relative velocity ($0 \leq k \leq 1$). The values of k=1 and k=0 denote the idealized concepts of perfectly elastic and plastic impact respectively.

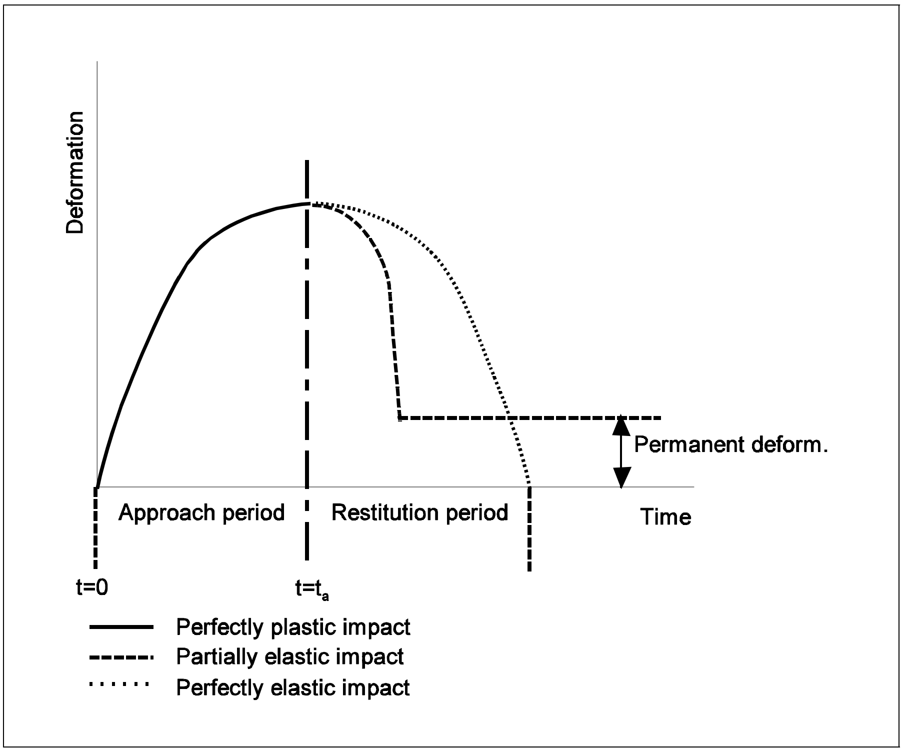

Figure 4. *Assumed deformation history in classical impact theory.*

The impulse momentum law is expressed in *Equation 15*:

$$\int_0^t \mathbf{F(t)dt} = \mathbf{m}_i(\mathbf{v}_i^* - \mathbf{v}_i) \qquad \textbf{(15)}$$

where m denotes the mass, v the initial velocity and v* the final velocity of a body, F the force and t the time. Application of *Equation 15* results in the following equations:

$$\int_0^{t_a} \mathbf{F_a(t)dt} = \mathbf{m}_1\mathbf{v}_1 - \mathbf{m}_1\mathbf{u} = \mathbf{m}_2\mathbf{u} - \mathbf{m}_2\mathbf{v}_2 \qquad \textbf{(16)}$$

$$\int_{t_a}^{t_r} \mathbf{F_r(t)dt} = \mathbf{m}_1\mathbf{u} - \mathbf{m}_1\mathbf{v}_1^* = \mathbf{m}_2\mathbf{v}_2^* - \mathbf{m}_2\mathbf{u} \qquad \textbf{(17)}$$

where F_a is the force during the approach period and F_r the force during the restitution period. From these *Equations 16 and 17* can be derived:

$$u = \frac{m_1 v_1 + m_2 v_2}{m_1 + m_2} = \frac{m_1 v_1^* + m_2 v_2^*}{m_1 + m_2} \quad (18)$$

With the definition of the coefficient of restitution and *Equations 16-17* we find:

$$k = \frac{v_2^* - v_1^*}{v_1 - v_2} = \frac{\int F_r dt}{\int F_a dt} \quad (19)$$

From *Equations 18 and 19* the final velocities are computed in terms of the coefficient of restitution and the initial velocities:

$$v_1^* = v_1 - (1+k)\frac{m_2 v_1 - m_2 v_2}{m_1 + m_2} \quad (20)$$

$$v_2^* = v_2 + (1+k)\frac{m_1 v_1 - m_1 v_2}{m_1 + m_2} \quad (21)$$

The loss of kinetic energy (ΔT) is given by *Equation 22*:

$$\Delta T = T_i - T_f \quad (22)$$

where T_i and T_f are the amount of kinetic energy at respectively t=0 and $t=t_r$ (see also *Equations 23 and 24*).

$$T_i = \frac{1}{2} m_1 v_1^2 + \frac{1}{2} m_2 v_2^2 \quad (23)$$

$$T_f = \frac{1}{2} m_1 {v_1^*}^2 + \frac{1}{2} m_2 {v_2^*}^2 \quad (24)$$

From *Equations 20 to 24* follows:

$$\Delta T = T_i - T_f = (1 - k^2)\frac{m_1 m_2}{m_1 + m_2} * \frac{(v_1 - v_2)^2}{2} \quad (25)$$

As stated before, one can determine three situations, depending on the value of the coefficient of restitution:

- k=0, perfectly plastic impact; this case yields $v_1^* = v_2^*$, $\int F_r\, dt = 0$:

$$\Delta T = \frac{m_1 m_2}{m_1 + m_2} * \frac{(v_1 - v_2)^2}{2} \quad (26)$$

- $0 \leq k \leq 1$, partially elastic impact; this case yields $\int F_a\,dt > \int F_r\,dt$
- $k=1$, perfectly elastic impact; this case yields $\int F_a\,dt = \int F_r\,dt$ and $\Delta T = 0$.

Zobel (1998) used the classical theory of impact to show that when two vehicles of different mass collide at a closing speed v_c of twice their design speed v_B, then the available deformation energy is sufficient to sustain the collision without intrusion. v_B is the speed used in a successful crash test against a rigid barrier, hence the vehicle has sufficient deformation energy available to dissipate an amount of kinetic energy of 0.5 m v_B^2.

The amount of required deformation energy D is given by *Equation 26*, which is the amount of kinetic energy loss in case of a perfectly plastic impact. The available amount of deformation energy E is given by *Equation 23* by replacing the two velocities with v_B, which results in:

$$E = \frac{1}{2}(m_1 + m_2) v_B^2 \quad (27)$$

To prove that the available deformation energy E is greater than the amount of required deformation energy D, Zobel (1998) shows that D $\leq$ E as follows. A square is always positive:

$$(m_1 - m_2)^2 \geq 0 \quad (28)$$

By adding $4m_1m_2$ it can be seen that:

$$(m_1 + m_2)^2 \geq 4 m_1 m_2 \quad (29)$$

or dividing by (m_1 + m_2)

$$m_1 + m_2 \geq \frac{4 m_1 m_2}{m_1 + m_2} \quad (30)$$

Equations 26, 27 and 30 result in a basic finding on compatibility:

$$D = \frac{1}{2} \frac{m_1 m_2}{m_1 + m_2} v_c^2 \leq \frac{1}{2} \frac{m_1 m_2}{m_1 + m_2} (2 v_B)^2 \leq \frac{1}{2}(m_1 + m_2) v_B^2 = E \quad (31)$$

Equation 31 holds only when it is possible to compel both vehicles to deform. Zobel (1998) states that in real-world accidents this is not always the case. Sometimes both vehicles deform similarly, sometimes one of the vehicles is almost undeformed, while the other is completely destroyed. To force both vehicles to deform, the “bulkhead concept” is introduced. The bulkhead concept means that a maximum force level is defined for the front end of the car. A bulkhead has to be built which is able to sustain this maximum force. This bulkhead would avoid a compartment collapse, as long as one of the vehicles is still deforming.

The bulkhead principle has its limitations, however. Zobel (1998) shows that the test speed in rigid barrier tests for larger cars is limited to assure compatibility for cars having a mass ratio $\mu = m_l/m_s$ of 1.6. A mass ratio of 1.6 covers approximately 90% of the German real-world frontal collisions.

The available deformation energy D of a large car with mass m_l, designed for a barrier impact speed v_B can be computed in two ways:

$$\mathbf{D} = \frac{1}{2}\mathbf{m}_l \mathbf{v}_B^2 \qquad \textbf{(32)}$$

$$\mathbf{D} = \int \mathbf{F(s)ds} = \mathbf{F} * \mathbf{s}_l \qquad \textbf{(33)}$$

From *Equations 32 and 33* follows:

$$\mathbf{F} = \left(\frac{1}{2}\mathbf{m}_l \mathbf{v}_B^2\right) \Big/ \mathbf{s}_l \qquad \textbf{(34)}$$

When the large car collides with a small car and dynamic effects are neglected, then the principle action equals reaction can be used and the force acting on the large car equals the force acting on the small. The force acting on the small car can be computed as follows:

$$\mathbf{F} = \mathbf{m}_s * \mathbf{a}_s \qquad \textbf{(35)}$$

From *Equation 34 and 35*, *Equation 36* can be derived:

$$\mathbf{v}_B = \sqrt{\frac{\mathbf{2} * \mathbf{s}_l * \mathbf{a}_s}{\mu}} \qquad \textbf{(36)}$$

Zobel (1998) states that the deformation stroke of the large car, s_l, is limited to 0.7 m and that the maximum acceptable deceleration of the small car is 30*g*, in terms of acceptable dummy loads.
With $a_s = 30g$, $s_l = 0.7$ m and $\mu = 1.6$, it follows from *Equation 36* that the maximum test speed $v_B = 57.8$ km/h. From *Equation 36* it also follows that, given a certain value for as and s_l, an increase of the test speed v_B of the

larger car will result in a smaller value of the mass ratio μ, meaning that compatibility is only possible in a smaller vehicle range.

3.2. In-depth accident analysis

In the previous chapter, papers were discussed which deal with statistical analysis using more general information collected from real-world accidents. Another approach is to collect detailed information of real-world accidents to perform in-depth analysis.

Shearlaw & Thomas (1996) state that current crash testing does not consider vehicle safety in terms of the occupants of an opposing vehicle. Real-world accident data can be used to study effects of vehicle structure on the injury outcome in a number of impact types.

In this paper, presented at the 15th ESV conference in Melbourne, five cases of car-to-car crashes are used to demonstrate methods of studying compatibility in real-world accidents. The data of these accidents were collected within the UK Cooperative Crash Injury Study between 1992 and 1995. Basically four methods are described in the paper.

The first method Shearlaw & Thomas (1996) describe to study compatibility is to investigate accident types which lie on the extreme of the spectrum of incompatible crashes. An example of such a crash is a mid-sized saloon colliding with a van. From this type of accidents one can learn about incompatibility caused by difference in geometry, i.e. the mismatch of stiff structures in both cars.

The second method described is to study collisions between two apparently similar cars. The difference of deformation behaviour of both cars and the differences in the injury outcome can give insight into which design aspects cause incompatibility.

The third method consists of studying one particular model in one accident scenario. This can furnish information about the response to different striking vehicles

The last method described is to study crashes where both cars show good compatibility, despite mass differences. One can learn from these cases how compatible cars should be built.

3.3. Crash tests

Crash tests are used by several parties for a number of reasons. Car manufacturers perform crash tests to optimize designs and to prove that newly designed cars comply with governmental directives. Other institutes perform crash programmes to investigate the passive safety of new cars, and publish the results to inform the public. Examples of such crash programmes are:

- New Car Assessment Programme (NCAP).
  This programme is performed by the National Highway Traffic Safety Administration in the United States.
- Crashworthiness Evaluations.

This programme is performed by the Insurance Institute of Highway Safety (IIHS) in the United States.

- European New Car Assessment Programme (EuroNCAP).
  This programme is performed by a consortium of consumers associations, motoring organizations and governmental bodies in Europe.
- Australian New Car Assessment Programme (ANCAP).
  This programme is performed by the (NRMA) and supported by all Australian automobile clubs and the state government road and transport authorities in Queensland, New South Wales and South Australia.

There are differences in the test conditions used in these programmes. Also, there exist differences between the governmental directives in different countries. *Figures 5 and 6* show the test conditions applied in the EuroNCAP programme for cars with the steering wheel on the right side.

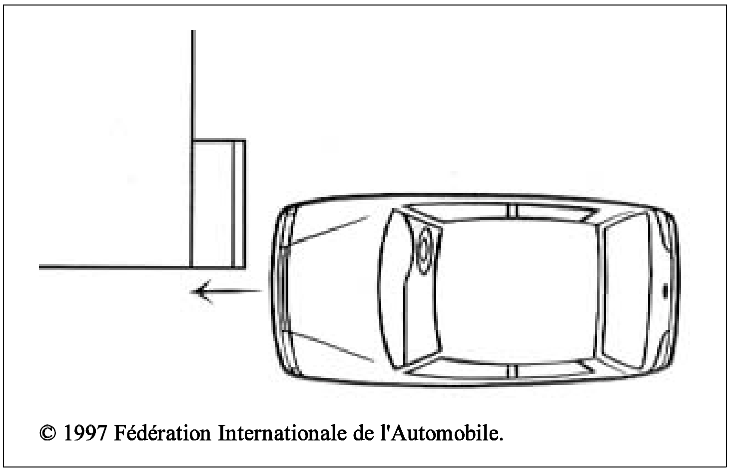

Figure 5. *The EuroNCAP offset deformable barrier test, vehicle speed 40 mph, overlap 40%.*

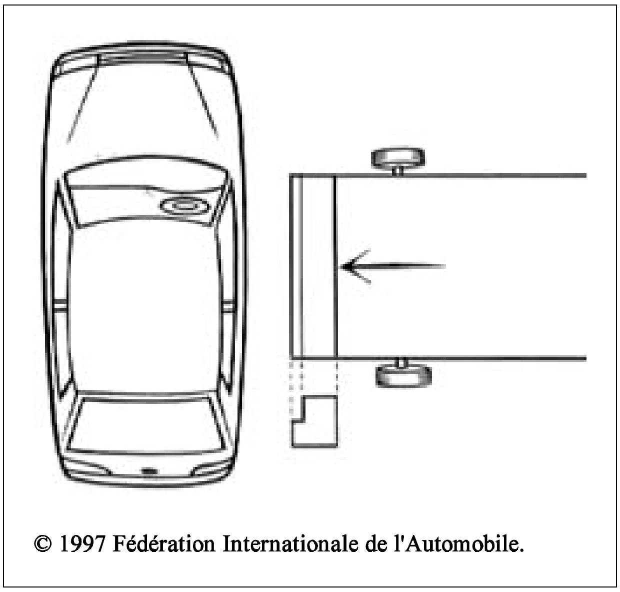

Figure 6. *EuroNCAP side impact crash test, barrier speed 30 mph.*

In the frontal impact of the EuroNCAP programme, the test car impacts against a deformable structure. The impact is across 40% of the test car's front and is intended to represent a crash with a car of equivalent size and weight. The offset test is always on the driver's side, where there is more risk of injury from the steering wheel and pedals.

For the EuroNCAP side impact test, a trolley with a deformable aluminium block is driven into the stationary test car driver's door at 50 km/h (30 mph). These test conditions are similar to test conditions required in European Directive 96/27/EC.

Additionally the EuroNCAP programme contains test procedures to investigate the injury risk of pedestrians hit by the tested vehicle and to evaluate child restraint systems. These subjects are beyond the scope of this report. Both the tests required by governments, such as 96/79/EC, 96/27/EC in Europe and FMVSS 214D and FMVSS 208 in the United States, and the tests performed in the New Car Assessment Programmes emphasize occupant protection and not aggressiveness or opponent safety. The test results are based on dummy readings and intrusion of the occupant compartment. A major concern during the development of these tests was that the test should be representative of real-world accident conditions.

Stcherbatcheff et al. (1989) analysed 39 tests with five different car models of the Renault range. Subject of this study was a test which could be used as reference frontal impact. They compared the results of six different frontal collisions: offset car-to-car impact with 50% overlap on the left side; 30^o angled barrier impact on the left side; 30^o angled barrier impact plus side wall; 30^o angled barrier impact with anti-slip system; offset half-barrier impact with 45% overlap on the left side; and impact against 0^o angled barriers. Furthermore, they compared these six impact configurations with accident data from the real world.

The collision between a car and a deformable moving barrier of given mass was intentionally not taken into account, for two reasons:
- This configuration would result in excessive scatter of the speed variation between vehicles tests, depending on their weight.
- Collisions between heavy vehicles and a fixed barrier at high velocity change would not be represented in this type of collision, due to excessively small velocity change in relation to a much lighter moving barrier.

Stcherbatcheff et al. (1989) find that the 30^o angled barrier impact seems most representative of real-world accidental conditions. As regards compatibility, they conclude that a single reference configuration in frontal impact cannot by itself solve the complex problems posed by inter-vehicle architecture and stiffness compatibility, especially in side impacts.

Klanner et al. (1996) state that the frontal impact test with a deformable barrier copies as closely as possible the real accident process. However, this test does not provide information on vehicle compatibility. In their contribution to the 15th ESV conference, Klanner et al. describe the use of the so-called “ADAC barrier”. This barrier is a three-stage barrier which has been based on the EEVC barrier. The ADAC barrier has been developed because the EEVC barrier needed three essential improvements:

- greater impact resistance and an increased barrier depth to prevent bottoming out
- greater energy absorption capacity to prevent blocking out
- a defined deformation in order to allow for a quantification of the absorbed energy and thus for the determination of the correct test speed as well as the aggressiveness and compatibility.

Klanner et al. (1996) used measurements of the horizontal deformation of the ADAC barrier to estimate the amount of energy absorbed by the barrier. The amount of energy taken into the barrier and the distribution of the deformation over the barrier face is a measure of the aggressiveness and compatibility of the tested vehicle. A uniform deformation pattern of the barrier indicates that the tested vehicle has a uniform stiffness distribution. When the barrier has high local deformations, the tested car has high stiffness variations, which is associated with an aggressive front end. Klanner et al. raise the question if one should define limits for barrier energy absorption as design targets: an upper limit to restrict aggressiveness and a lower limit to assure interior protection.

According to Zobel (1998), the ADAC approach leaves some open questions.

> “The force behind the barrier is a consequence of the interaction between the barrier and car. If the vehicle shows only little deformation, then the question remains as to whether this happened because the vehicle was only slightly stiffer than the barrier or whether it indicates an absolutely stiff vehicle. The deformable barrier will indicate a higher stiffness. But if this high stiffness occurs, it provides no information about the undeformed part of the vehicle. This information is needed when we think about the ability of vehicles to force potential opposing vehicles to deform and about the ability to be forced by opposing vehicles to deform.”

Kohlhoff & Bläser (1996) conclude that the offset deformable barrier (ODB) can offer advanced features for assessing the protective capabilities of passenger cars, including compatibility aspects. One advantage of the ODB is that it favours a frontal design with a good load spreading. The design of such a barrier is complex, particular if one wants to use the same barrier type to test vehicles from different weight classes. To achieve this, Kohlhoff and Bläser applied a different test speed and overlap for vehicles of different weight. In their attempt to determine the characteristics of the ODB, they used the results of computer simulation (see next section).

Steyer et al. (1998) describe two test methods to measure the crushing force in a car collision. The measurement of the crushing force is needed to verify that newly designed cars comply with their proposal of a crush force characteristic. The proposal consists of the regulation of the end-of-impact load, the so-called “compatibility load”, of 300 kN up to an EES of 55 km/h against a rigid wall or ODB. The first described method consists simply of measuring the force on a dynamometric barrier, which can be used only in a single vehicle crash. The second method is based on the principle of action and reaction. It is assumed that a number of discrete car components and areas of structures can be appointed which will act as a lumped mass during the crash. The deceleration force for each lumped mass can be calculated by multiplying the measured deceleration and its mass. The summation of these forces results in the overall interface force. The second method shows good agreement with measured interface forces with the dynamometric wall.

In their contribution to the 16th ESV conference, Wykes et al. (1998) also describe both methods mentioned by Steyer et al. (1998). They also find good agreement between load cell wall data and the interface force calculated from accelerometers from a frontal ODB test. Their motivation to investigate interface forces in car crashes is the observation that some authors have put forward proposals to control the global stiffness of the vehicle and hence to improve compatibility: Zobel (1998) the bulkhead principle, and Steyer (1998) the compatibility load. Wykes et al. raise two questions which need to be answered before either of these proposals can be pursued:
- What test could be performed to ensure that the occupant compartment could withstand the maximum crush force level?
- What should the crush force levels be?

Only the second question can be addressed with the described measurement methods, the first one remains open.

At the Transport Research Laboratory in the United Kingdom several full-scale crash tests were performed to study what aspects influences compatibility (Hobbs et al. 1996, Wykes et al. 1998) both in frontal and side impact. These tests mainly emphasize geometry aspects and structural interaction, and are therefore discussed in the next chapter.

3.4. Computer simulation

Computer simulation is taking an increasingly important role in vehicle design. Both lumped mass models and finite element method (FEM) models are used to investigate crash behaviour of new car design and to develop new test methods.

Schoeneburg et al. (1996) describe the development of the FEM model of the Audi A4 and the Seat Ibiza. The models were applied in the simulation of a side impact of the Audi A4 by the Seat Ibiza. The aim of the simulation was to analyse the compatibility of both cars. The simulation results were compared with the results of an experiment with precisely the same peripheral conditions as applied in the computer simulation. With regard to structural deformation, the simulation reveals the same tendencies as the experiment. There is also good agreement between the dummy loads in the experiment and the simulation. Only the head performance criterion (HPC) is 50% higher in the simulation than in the experiment. This was caused by the FE dummy’s head just touching the door frame, where it is just clear of in the experiment. Schoeneburg et al. conclude that simulation of car-to-car accidents as a means of investigation compatibility is a tool capable of analysing deformation behaviour and the resulting loads on dummies in the preliminary development phase for various vehicle structures.

Lumped mass models are used for simulation of occupant behaviour. Mizuno & Kajzer (1998) used the simulation programme MADYMO to analyse the effect of stiffness variation of a mini car, mass 700 kg, on injury risks of the driver. They expressed injury risk in terms of HIC, chest acceleration, chest deflection, femur force, the maximum lower tibia axial force and the maximum lower tibia moment. The stiffness of the mini car was varied between 500 kN/m to 1000 kN/m. Two crash configurations were examined: a mini car crashing into a rigid wall, and an offset collision between a mini car and a large car. The initial speed of the mini car in the rigid wall crash was 50 km/h with 100% overlap. In the car-to-car crash, the crash speed of both cars was 50 km/h, the overlap of the mini car was 50% and the overlap of the larger car (mass 1,400 kg, stiffness 872 kN/m) was 40%.

From the rigid wall crashes Mizuno & Kajzer (1998) find that HIC and chest acceleration increase consistently with the stiffness of the mini car. The other four injury risk parameters do not change much with an increase of the stiffness of the mini car. From the calculated compartment intrusion the authors conclude that intrusion is a less important factor in determining the injury risk to the driver of a mini car, whereas the acceleration causes the majority of injuries.

From the simulations of the car-to-car crashes Mizuno & Kajzer (1998) find the following. As the stiffness of the mini car increases, the risk of injury to the driver in the large car tends to become greater. However, the injury risk of the large car driver is less than that of the mini car driver. The relevant injury criteria are lower than the tolerance level.

The car-to-car crash simulation was also used to investigate the effect of an additional crush space of the large car on injury risk. For this simulation the stiffness characteristic of the large car was changed without changing the front length of the large car. Into the force deformation curve (see *Figure 7*) a maximum force level of 200 kN over length *c* was introduced.

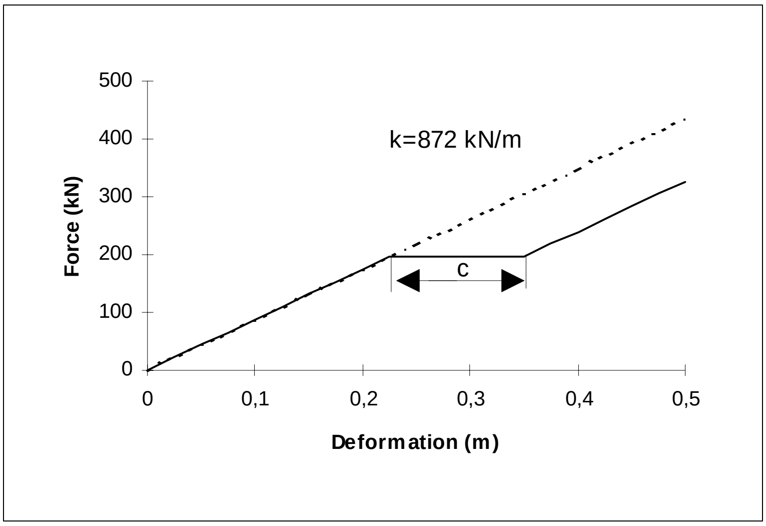

Figure 7. *Modified force deformation curve of the large vehicle (reproduced from Mizuno & Kajzer, 1998).*

In the performed simulations *c* was varied from 0 to 0.4 m and the stiffness of the mini car was varied between 500 kN/m and 1000 kN/m. These simulations show that the increase of *c* has an injury-reducing effect on the driver of the mini car by reducing the acceleration and the intrusion of the mini car. However, when the mini car is stiff, the risk to the driver in the large car becomes high, especially for injuries to lower extremities.

The study of Mizuno & Kajzer (1998) is a clear example of the strength of simulation for compatibility research. This method allows the investigator to study the effect of the variation of only one parameter. In an experimental setting the variation of one parameter is almost impossible.

Kohlhoff & Bläser (1996) used both FE methods and lumped mass methods to develop a deformable barrier test which account for compatibility. They describe the efforts of obtaining a single deformable barrier which gives similar deformation and deceleration profiles for vehicles of different mass, as in the case of a car-to-car collision (see also previous section).
Kohlhoff & Bläser (1996) developed a simple lumped mass model, consisting of one elastoplastic spring and a mass, of a large car (mass 1,652 kg) and a small car (mass 1,073 kg). The spring characteristics were based on the results of FEM simulations of a car-to-car crash of both car models and a car-rigid barrier crash of each car. The lumped mass models were used to determine a load deflection curve of the offset deformable barrier.

The National Highway Traffic Safety Administration (NHTSA) applied crash simulation, both FEM and lumped mass, within a large-scale systems model (Hollowell and Gabler, 1996). The model, referred to as Vehicle Research Optimization Model (VROOM) was based on the Safety Systems Optimization Model (SSOM), which was developed by Ford Motor Company in the 1970s and subsequently enhanced by the University of Virginia (Ford Motor Company, 1978; White et al., 1985; Gabler et al., 1994).
With VROOM, which still is under development, it will be possible to evaluate vehicle crashworthiness based on the safety performance of the vehicle when exposed to the entire traffic accident environment, i.e. across the full spectrum of expected collision partners, collision speed, occupant heights, occupant ages, and occupant injury tolerance levels. VROOM can be used to find the balance between two potentially conflicting objectives of optimal crash countermeasure designs: maximizing passenger protection in the vehicle under design, and optimizing compatibility with other vehicles in the fleet.

Results of one of VROOM's predecessors, the Model for Optimization of Safety Systems (MOSS), were published during the 14th ESV conference (Gabler et al., 1994). This paper discusses the outcome of the safety systems design optimization of a 3,000 lb production passenger car. The optimal structural design was characterized by a softer front frame and a stiffer rear frame. The softer front frame resulted in a subject vehicle which was less aggressive in collisions with other vehicles, especially when striking other vehicles in the side. This optimized design reduced the overall harm by 21% relative to the baseline design. These injury reductions were observed across all accident modes. The largest benefit was for the side impacted occupants of the other car for which injuries decreased by over 50%. Furthermore, the benefits were almost evenly distributed between the other car and the subject car. The distribution of the optimization benefits over the weight class of the other car shows that the vehicles of the 2,000 lb class take 26% share of the benefit, vehicles of the 4,000 lb class 17 % and vehicles of the 5,000 lb class 2%. The remaining 55% of the optimization benefits accrue to the subject vehicle of 3,000 lb, 46% when involved in a single-car accident.

Wykes et al. (1998) performed parametric studies, using full-car finite element models, to identify car structure characteristics which will improve compatibility in car-to-car side impact. They used a validated model of an European side impact test. A number of parameter sweeps were performed changing the following barrier characteristics:

- barrier centre impact point
- barrier mass
- barrier front face geometry
- barrier stiffness.

The results of this modelling study indicate that in order to improve compatibility for side impact, the bullet vehicle should be designed such that it engages the structure of the target vehicle more effectively, through improved geometrical interaction. This should be achieved without compromising the intrusion profile or causing excessive roll in the target car. Stiffening of the bullet car's upper load path without stiffening the lower path should be avoided.

Furthermore, Wykes et al. (1998) performed a simulation study with the MADYMO software package to investigate the influence of the deceleration pulse on restraint-system-related injuries from frontal impact. The reason for studying this effect is the assumption that a compatible vehicle will have minimized intrusion due to a rigid occupant cell, and therefore restraint-system-related injuries will dominate. The authors applied various simple-shaped, analytical and experimental deceleration pulses to an occupant compartment model consisting of a HYBRID III dummy held by a typical restraint system. Peak chest compression was used as the injury indicator.

The results of the Wykes et al. (1998) study indicate that in order to minimize chest injury the passenger compartment deceleration pulse should have a constant profile and ride-down distance should be maximized.

4. The geometrical view

The crash geometry is not discussed in this chapter, because it is not subject of this study. Clearly, vehicles should behave compatibly in all crash modes which occur in the real-world. In this project frontal and side collisions are considered. Subject of discussion here are differences in car geometry and stiffness geometry which lead to incompatibility in car crashes.

In the paper "Improved protection through greater compatibility between road vehicles", presented to the 14th ESV conference, Neilson (1994) addresses both the car geometry and the structural stiffness aspect of compatibility. Proposing to use the small car as the basic vehicle, he uses its structural layout to define an impact height band. Within this height band the initial contacts should occur on impact. The structures in this height band should be designed to take almost all of the impact interaction. In the paper a height band is suggested from 350 mm to 550 mm above ground for the fronts of all vehicles. The height might be raised up to 700 mm above ground when the crush depth is more then 300 mm, and should be maintained at 700 mm until a crush depth of 700 mm. The critical stiffness of the front of a small car should be the greatest that can be built into the lower sides of the passenger compartments of cars for the sake of car-front to car-side impacts. Furthermore, Neilson proposes that the fronts of all cars should be of similar size and crush stiffness.

Shearlaw & Thomas (1996) measured the height of the longitudinals and sills of 185 car models to study the geometrical mismatch of cars in side collisions. They find no clear relationship between vehicle mass and the two measured vehicle dimensions. The measurements of the sill top shows a considerable scatter in the data, with most points falling into a clearly defined broad band. Presumably the height of the sill top is designed for access considerations. The longitudinals measurements show more scatter than the sill measurements. Shearlaw and Thomas find that the sills and longitudinals of four-wheel-drive vehicles are located higher than the sills and longitudinals of normal cars. Four-wheel-drive vehicles are characterized by having considerably more ground clearance than conventional vehicles and frequently have a rigid chassis construction. According to Shearlaw and Thomas these four-wheel-drive vehicles present a considerable risk to the occupants of a conventional passenger car, not only because of the frequently greater mass they enjoy, but also by virtue of their potential to concentrate loading above the sills or longitudinals of the struck car. When the 4x4 vehicles were removed from the sample of 185 cars, the clearance of the bottom of the longitudinals was an average 36.3 centimetres, while the average ground clearance height of the top of the sill was 34.6 cm. In 64% of the possible car combinations in side collisions, the average longitudinal would not interact with the sill.

Kohlhoff & Bläser (1996) state that one aspect of compatibility is what one might call "stiffness distribution compatibility". A possible consequence of an unevenly distributed stiffness can be a front member penetrating the other car, resulting in a heavy loaded passenger compartment. This behaviour may also be the result of poorly connected load paths. An even distribution of stiffness will consequently result in a better energy absorption in case of a head-on crash. However, a stiffness distribution of vehicle frontal structure is never very homogeneous. Neither are frontal structures of different cars necessarily similar, in fact they usually do not match.

Hobbs et al. (1996) observe that in real-world accidents the impacting front structures do not interact properly. Due to lack of stiffness uniformity at the front of cars, it is usual for the stiff parts of one car to penetrate the weaker parts of the other. They also observe horizontal and vertical misalignment of the stiff structures in the opposing cars. A similar observation is made during an offset frontal car-to-car impact test between a small and medium-sized car. The test showed that problems with the interaction of the car's structures had a dominating effect on the outcome. This test was then repeated, replacing the medium-sized car with a car of the same mass which performed structurally well in the European offset deformable barrier test (Wykes et al., 1998). Since this car possessed a better frontal tie-up than the car it replaced, there was virtually no overriding. To study solely geometry, Wykes et al. (1998) performed a medium-size car-to-car offset test, using identical vehicles. The ride heights of both cars were modified to achieve a ride height difference of 100 mm. The deformation patterns of both cars were noticeably different, even though the vehicles mass, stiffness and occupant compartment strength were matched. Wykes et al. (1998) suggest that in order to achieve a compatible fleet it will be necessary to firstly establish good geometrical compatibility.

Steyer et al. (1998) give a simplified presentation of compatibility. They used a 50 % offset rigid barrier test as a reference, because this represents the ideal behaviour of the structure in terms of energy absorption. Compared to this ideal situation, they observe three major problems concerning the energy absorption in a real-world car-to-car crash:

- There is a lack of a plane interface between the two vehicles immediately after contact (fork effect).
- Overriding can occur.
- The two vehicles have different stiffnesses at the end of the impact.

Steyer et al. (1998) give a number of possible improvements. They propose to increase the number of load paths, to create a front face which spreads the load, to limit the load immediately after impact and to harmonize the end-of-impact load.

Crash statistics presented by Gabler & Hollowell (1998) demonstrate a clear incompatibility between cars and light trucks and vans (LTVs). Fatalities and injuries which arise from the incompatibility of LTVs and cars is a growing problem in the United States, because of the former's steadily increasing market share. Gabler & Hollowell (1998) do not attempt to assign what proportion of LTV aggressiveness is due to vehicle mass distribution, stiffness distribution or ride height geometry. But a comparison of these properties between cars and LTVs confirms that these two categories of vehicles are incompatible from a design point of view. More research is needed to determine the relationship between LTV design features and crash aggressiveness.

5. Discussion and conclusions

Although authors use different definitions or descriptions for the term “compatibility”, these definitions or descriptions have much in common. Car design should not only emphasize occupant protection but should also take into account safety of the occupants in the other car. That different definitions and descriptions are found in the literature illustrates the complexity of the problem of vehicle incompatibility in the current fleet.

One way to investigate compatibility is to analyse accident registration files with statistical methods. All discussed statistical studies show that there is a correlation between vehicle mass and the metrics calculated to rate the safety of vehicles. From this the following general relationship can be derived:

Heavier vehicles offer better protection to the occupants but are more aggressive toward occupants of the other vehicle. The reverse is true for the lighter vehicle, i.e. worse occupant protection, but less aggressive.

However, there are exceptions to this rule. Some vehicles perform better or worse than cars in the same weight category. Hence other variables, such as vehicle size, vehicle structure and vehicle geometry, also have an effect vehicle safety. From this arises the following recommendation:

As far as compatibility is concerned, the exceptions are of interest, since from these cars can be learned which parameters make the difference. Thus the exceptions should be found.

Cameron et al. (1996) states that the calculated crashworthiness ratings may differ due to other factors not collected (e.g. crash speed) in the data. This applies for the results of all statistical methods. Since the available data is restricted, it is very difficult to isolate vehicle properties. All other factors, such as driver behaviour, which can be highly correlated to vehicle type, are included in the results.

However, new opportunities for statistical analyses may emerge from using more information of the following three data types: vehicle property data, crash test data and data collected by means of in-car electronics.
Until now, researchers have used mainly vehicle mass and size as parameters for interpreting injury severity in car crashes. More insight in the compatibility problem can be gained by adding parameters such as body shape, engine orientation and the presence of an upper longitudinal into the analysis.
Linking crash test data, for example front-end stiffness determined from NCAP tests (Gabler & Hollowell, 1998) linked with accident data, can also provide more insight into the compatibility problem using statistical methods. Clearly there will be a several-year wait before a vehicle tested in any new car assessment programme will be found in sufficient numbers in an accident database to be used in a statistical analysis.
Theoretically, more detailed data can be collected by means of in-car electronic equipment. With this equipment data related to collisions, such

as crash speed and crash location, can be collected. Furthermore, one can gain insight in vehicle use characteristics such as mileage, speed behaviour and brake behaviour.
Practically, the use of this type of information faces legal and privacy problems. In summary, the following is recommended in relation to statistical studies and compatibility:

More detailed data on vehicle properties, driver behaviour and collision characteristics should be collected, for use in statistical analysis to study compatibility.

From the classical theory of impact the mass effect in a car-to car crash can be shown in a simple way. It follows from *Equations 20 and 21* that the lighter vehicle experiences a larger velocity change than the heavier car. *Equations 20 and 21* also show that the more kinetic energy is absorbed by both vehicles ($k \rightarrow 0$), the smaller the velocity change of both vehicles. From this observation the following can be concluded:

Given the mass effect, the technical limits of the mass ratio should be determined in such a way that the injury outcome of the occupants involved in a car crash is independent of the mass of their car.

The classical theory of impact neither describes the way the initial kinetic energy is transformed into deformation energy, absorbed by both car structures, nor the time history of the acceleration of both cars. Deformation of the car structure, resulting in intrusion, and the deceleration of the car are the main causes of injuries to car occupants. Deceleration and deformation are determined by the absolute value and the ratio of the stiffness of both involved vehicles. For this reason a number of authors suggest certain stiffness characteristics for the front end of cars (Zobel, 1998; Steyer et al, 1998; Kohlhoff & Bläser, 1996). The boundary conditions of stiffness characteristics of car front ends are given by the amount of energy which should be absorbed and the maximum deceleration level which an occupant of the lighter vehicle can survive.

More research is necessary to determine if a regulation of the front end stiffness will lead to a compatible car fleet, and if so, what shape the force displacement curve should have.

Deceleration and stiffness aspects of compatibility are studied by in-depth accident analysis, crash tests and computer simulation. The role of computer simulation in crash research is increasing. Not only the increase in computing power, but also the availability of simulation software and the growing knowledge of model development has led to many validated simulation models. Two model methods are used, the lumped mass method and finite element method. The strength of computer simulation is that one can investigate the effect of isolated parameters relatively easily, for example the stiffness of cars front end.

In-depth accident studies and crash tests emphasizing compatibility are still rare. The results of these studies show that both car geometry and stiffness distribution play an important role in the problem of car incompatibility. Misalignment of the load paths of cars involved in a car crash results in more intrusion of the occupant compartment than when the load path are aligned. This holds for both frontal and side impacts. The geometry and stiffness distribution aspect of compatibility is clearly demonstrated by the recently growing market share of light trucks and vans (LTVs) in the United States, as described by Gabler and Hollowell (1998). An important aspect in the design of a compatible vehicle is a homogenous stiffness distribution over the front end, to ensure that the load paths are aligned in any crash geometry (frontal, oblique, side and rear).

From the mechanical and the geometrical view the following can be concluded:

Stiffness, stiffness distribution and stiffness alignment are important keys to the solution of the incompatibility problem.

The challenge in the coming years is to develop a test procedure for new car designs which forces the future car fleet to converge towards compatibility.
The offset deformable barrier (ODB) test (European Directive 96/79/EC) for frontal impact testing should help to improve the interaction of frontal structures in car-to-car crashes. (Hobbs et al., 1996; Kohlhoff & Bläser, 1996; Klanner et al., 1996). Some authors (Wykes et al., 1998; Steyer et al., 1998) believe that two tests are necessary, one to test the crashworthiness and one to test the compatibility properties of the vehicle.

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