Rapid design of crash properties for safe automobiles : a conceptual approach

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Rapid Design of Crash Properties for Safe Automobiles: A Conceptual Approach

PROEFSCHRIFT

ter verkrijging van de graad van doctor aan de Technische Universiteit Eindhoven, op gezag van de Rector Magnificus, prof.dr. M. Rem, voor een commissie aangewezen door het College van Dekanen in het openbaar te verdedigen op

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door

Dirk Landheer

geboren te Nuenen

Dit proefschrift is goedgekeurd door de promotoren:

prof.dr.ir. R.F.C. Kriens

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Summary

The overall expanding mobility results specially on roads in a higher traffic intensity. Consequently, the risk to get involved in a crash increases. To cut back injuries ensuing from these crashes, the so-called "passive safety" of vehicles is improved. A minimum passive safety level is enforced governmentally via legislation. Passive safety in side impact crashes is now in the centre of attention due to the renewed Federal Motor Vehicle Safety Standard 214 for the United States and its for 1998 planned European counterpart.

Analysis techniques are applied to evaluate the crashworthiness of vehicle structures. In numerical crash simulations no physical structure (a proto-type) is required. The crash behaviour can be predicted via a numerical model of the structure. This enables the comparison of structural concepts at an early stage, so only the most promising are elaborated. This procedure is of special importance for side impact safety because as yet no general mature side impact safety concept exists. By application of the so-called "Quality Function Deployment" technique, criteria are assessed that have to be met for a sensible application of the procedure in the vehicle design process.

The generation of a numerical model of safety concepts, already before detailed geometric data is know, is discussed here. With such models, functions can be assessed that must be fulfilled by the elements of the vehicle structure. A proposal for a step-by-step procedure to design the vehicle structure is presented. After each step, a more detailed numerical model is used to verify if the formulated functions are fulfilled. The procedure is checked on meeting the criteria for a sensible application in the vehicle design process.

With common numerical tools the presented procedure is feasible. However, certain deformations require to much calculation time. As a consequence, the stated criteria are not always met. The in this project used FEM model of the Eurosid appears decisive for the time-step that is applied in the calculations. Thus, a change in the dummy model offers perspective to reduce the calculation time.

Samenvatting

De algemeen toenemende mobiliteit leidt in het bijzonder op de wegen tot een hogere verkeersintensiteit. Hierdoor neemt de kans op botsingen toe. Om letsels ten gevolge van deze botsingen te verminderen wordt de zogenaamde "passieve veiligheid" van de voertuigen verhoogd. De overheden hebben hierbij een sturende rol door middel van wetgeving. Passieve veiligheid bij zijdelingse botsingen is nu actueel door een vernieuwde Amerikaanse wet (Federal Motor Vehicle Safety Standard 214) en de voor 1998 geplande Europese wetgeving op dit punt.

Analysetechnieken worden ingezet om de botsveiligheid van voertuigconstructies te evalueren. Bij numerieke simulaties is hiervoor geen fysieke constructie (een proto-type) nodig. Via een numeriek model van de constructie is het botsgedrag te voorspellen. Dit schept de mogelijkheid concepten in een vroeg stadium te vergelijken en alleen de meest belovende uit te werken tot een constructie. Daar voor zijdelingse botsingen nog geen algemeen aanvaard veiligheidsconcept bestaat lijkt deze aanpak juist ten aanzien van dit type botsingen erg waardevol. Via een analyse volgens de "Quality Function Deployment" methode is vastgesteld aan welke criteria de gestelde aanpak moet voldoen voor een zinvolle toepassing in het voertuigontwerp.

Het genereren van een numeriek model van veiligheidsconcepten, reeds voordat de gedetailleerde geometrische informatie bekend is, wordt hier behandeld. Met een dergelijk model kunnen functies, te realiseren door bestanddelen van de voertuigconstructie, vastgesteld worden. Een voorstel om de constructie in stappen te ontwerpen wordt gepresenteerd. Na elke stap vindt met een geometrisch gedetailleerder numeriek model controle op vervulling van de functies plaats. Het voorstel is getoetst aan de eerder opgestelde criteria voor een zinvolle toepassing in het voertuigontwerp.

Met de huidige numerieke gereedschappen blijkt de voorgestelde aanpak mogelijk. Bepaalde vervormingssituaties vereisen echter nogal lange rekentijden. Hierdoor wordt nog niet in alle gevallen aan de gestelde criteria voldaan. Het in dit onderzoek gebruikte FEM-model van de Eurosid blijkt doorslaggevend voor de toegepaste tijdstap in de simulaties. Bijgevolg biedt wijziging van het dummy model perspectief om de rekentijden te reduceren.

Zusammenfassung

Die allgemein zunehmende Mobilität führt besonders auf den straßen zu einer höheren Verkehrsintensität. Folglich nimmt das Risiko in einem Verkehrsunfall verwickelt zu werden zu. Um Verletzungen infolge solcher Unfälle zu verringern wird die sogenannte passive Sicherheit von PKWs erhöht. Die passive Sicherheit wird mittels Gesetzgebung gesteuert. Heutzutage ist die passive Sicherheit, vor allem bei Seitenaufprallen, durch erneuerte Amerikanische und geplante Europäische Gesetzgebung, hoch aktuell.

Analysetechniken werden eingesetzt um die Aufprallsicherheit von Fahrzeugkonstruktionen zu beurteilen. Bei numerischen Simulationen wird anhand von numerische Modelle das Aufprallverhalten einer Konstruktion berechnet. Da hierfür keine materielle Konstruktion erforderlich ist, können verschiedene Konzepte in einer frühzeitigen Phase miteinander verglichen werden. Anschließend können nur die vielverspechendsten Konzepte zu einer Konstruktion ausgearbeitet werden. Da für Seitenaufpralle noch kein generelles Sicherheitskonzept auf dem Markt ist, sind diese Verfahren gerade für diesen Aufpralltyp besonders wertvoll. Durch eine Analyse, nach der QFD methode, sind die Kriterien denen die vorgestellte Vorgehensweise entsprechen muß, für eine sinnvolle Anwendung im Fahrzeugentwurf, festgestellt worden.

Die Entwicklung numerischer Modelle für Sicherheitskonzepte, in einer Phase inder detaillierte geometrische Daten noch nicht vorhanden sind, wird hier erörtert. Mit solchen Modellen können Funktionen, die von Teilen der Fahrzeugkonstruktion zu realisieren sind, festgestellt werden. Es wird eine Vorgehensweise vorgeschlagen inder die Konstruktion schrittweise entworfen wird. Nach jedem Schritt wird anhand detaillierter Modellen die Erfüllung der Funktionen kontrolliert. Diese Vorgehensweise wurde anhand der Kriterien für eine sinnvolle Anwendung im Fahrzeugentwurf geprüft.

Mit den heutzutage erhältlichen numerischen Werkzeuge ist die vorgeschlagene Vorgehensweise zu realisieren. Jedoch bestimmte Aufprallverformungen erfordern inakzeptabele Rechenzeiten. Folglich werden die gestellten Kriterien nicht in allen Fällen erfüllt. Das in dieser Studie benutzte FEM Modell des Eurosid Dummys erscheint entscheidend für den benötigten Zeitaufwand zu sein. Demzufolge bietet Änderung des Dummymodells Aussicht auf reduzierte Rechenzeiten.

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Chapter 1

General introduction

Since men's very existence, mobility has been an essential part of daily life. Collecting food and seeing fellow-men were the main motives for travelling. Originally, the mobility could only be achieved by going on foot. The most serious cause of getting injured as a result of walking itself was falling down.

Both velocity and mass, providing the total amount of kinetic energy of a travelling system, have increased enormously with time due to progress of technology. Deploying animals, such as the horse, for transport of men and luggage was the step that started this process. Nowadays, motor vehicles give the opportunity to travel over large distances with a speed of up to thirty times the speed of walking. Thus collisions in the horizontal direction are much more life threatening now than those resulting from falling down. The evolution of vehicles is largely coupled with a split of the infrastructure in sections with specific functions. For example, working, recreating, shopping and living areas can be distinguished. New travelling incentives are brought on this way. A massive use of motor vehicles, specially cars has been the result. This caused a shift in the relative importance

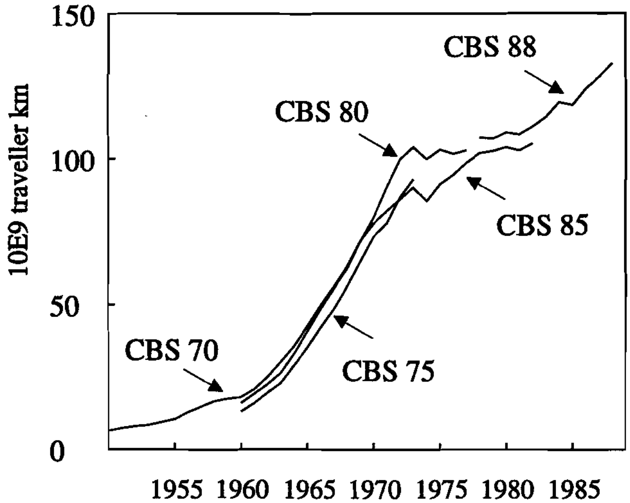


Figure 1.1 *Increased exposure to risks that go with travelling by car (based on CBS data concerning The Netherlands).*

of the traffic safety on injury risk (Figure 1.1). The relative influence of this shift on human health has grown considerably by the improvement in public health as an effect of better hygiene causing a reduction of infection risks.

From an economical point of view the social costs caused by traffic accidents are huge. For the United States of America these costs were estimated to be about $57,200,000,000.- in the year 1980 alone (Blincoe, 1983). Hence, even a small improvement in traffic safety saves the society a lot of money. To summarize, in the context of the common wish to enhance the quality of life, improving traffic safety is necessary.

1.1 How to reduce traffic injuries

Traffic safety can be influenced by four parameters, as shown in Figure 1.2. These parameters are: travelling needs, quality of vehicle and environment, trauma care, and human behaviour. The right hand side of the figure gives a survey of items you can control as individual. The items on the left hand side of the figure can only be influenced on a collective social basis. Changing the parameters has effect on one or more of the following four safety related fields:

Field 1 The *traffic intensity regulation field* embraces all activities that concern the amount of travellers on a certain location at a certain time. Main point here is keeping relative velocities within bounds. The distance between separate travellers, divided by their relative velocity, should provide a safe response time. Measures are: more travellers pro vehicle, more parallel travelling tracks, homogeneous distribution of travellers in time, or reduction of travelled kilometres.

Field 2 The *common safety field* contains active and passive safety aspects as described in the following two paragraphs. Within this field exploitation of safe travelling modes, in other words: organised utilisation of specific safe vehicles, is of concern. For instance, travelling per taxi, operated by a honest professional driver—familiar with the local infrastructure—instead of driving a small private car has a positive effect on active as well as passive safety.

Field 3 The so called *active safety field* is focused on crash avoidance. Active safety measures are always based on one or more of the following actions: human behaviour, vehicle quality or infrastructure improvement, as explained below. Considering human behaviour, better driving education can

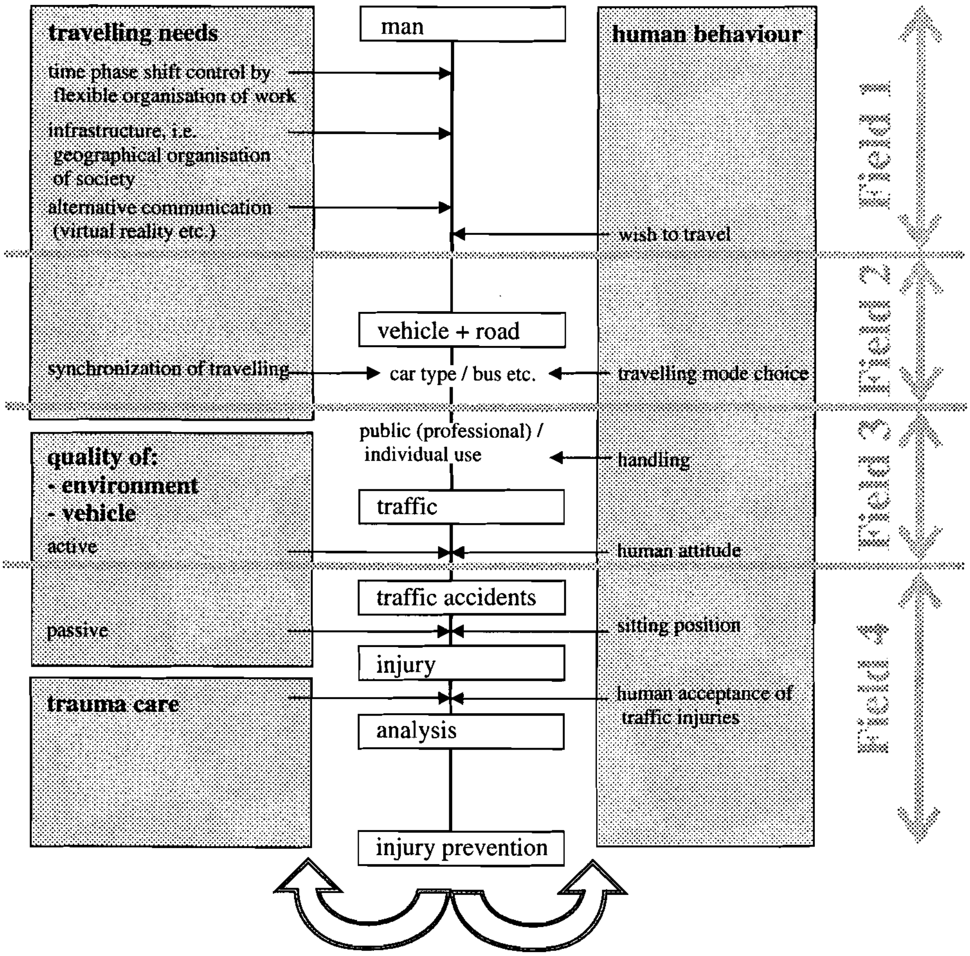


Figure 1.2 Traffic safety issue survey.

result in safer driving styles that are more predictable for other road users. Honest professional drivers should, nowadays, already have a safe driving style. Therefore, a shift from individual to professional traffic can reduce the occurrence of collisions. Improved vehicle dynamics and ergonomics ease the driving task. Clear view and visibility under all weather conditions is an important active safety quality for vehicle and environment. A predictable, conveniently arranged, infrastructure can help prevent the occurrence of dangerous situations (see also: Poppe and Prins, 1994). Easing the driving task, on the other hand, can have the negative side effect of an unjustified feeling of safety that results in reckless driving. A good example is the finally marginal effect of ABS (Anti-lock Brake Systems) on vehicle accident statistics, as displayed in an experiment carried out in Munich with taxis. Another example is the occurrence of mysterious accidents on monotonous roads. Traffic regulation in combination with governmental observing control, therefore, is an essential factor to achieve and maintain a certain active safety level.

Field 4 The so called *passive safety field* concerns the traveller protection capacity of full traffic systems in case of collisions. So, the deformation properties and shape of both colliding objects are variables. An alarming trend in this context is the private attachment of relatively rigid protuberances, such as bull bars on vehicles. Good passive safety requires compatibility of all travelling systems using the same track, including fixed obstacles near the track (Neilson, 1994). Characteristic example of a compatibility problem is the case of a car optimized for a frontal offset crash test, running into the relatively weak side structure of another car. Hence, only a well balanced legal requirement package has an over all positive effect on passive safety. Passive safety devices often are only effective if correctly adjusted. This imposes a huge responsibility on the traveller. A clear example of an often inadequate adjusted device is a head rest (van Kampen, 1993). The consequences of injuries can be influenced by medical treatment and rehabilitation services. Thus, the so-called "trauma care" is an important parameter in case injuries do occur in a crash.

The project described in this thesis is concentrated on the vehicle quality parameter in relation to the passive safety field.

1.2 Crashworthiness evaluation

Main criterium for the crashworthiness of a vehicle design is the protection capacity it offers to the occupants as well as to other road users. Priority should be given to the crashworthiness concerning frequently occurring crash situations because this has most effect on traffic safety. Two fundamentally different approaches for design evaluation exist. First, there is an approach which assesses the occupant protection capacity using accident statistics. A vehicle type related data base, linking injuries to crash specifications can indicate weak spots in a design. Since a representative amount of accident data is not available until several years after release of the vehicle on market, design improvements can only be implemented in later vehicle versions. To overcome this problem, a second approach for design evaluation is in use. This second—predictive—approach is based on accident simulations under conditioned circumstances.

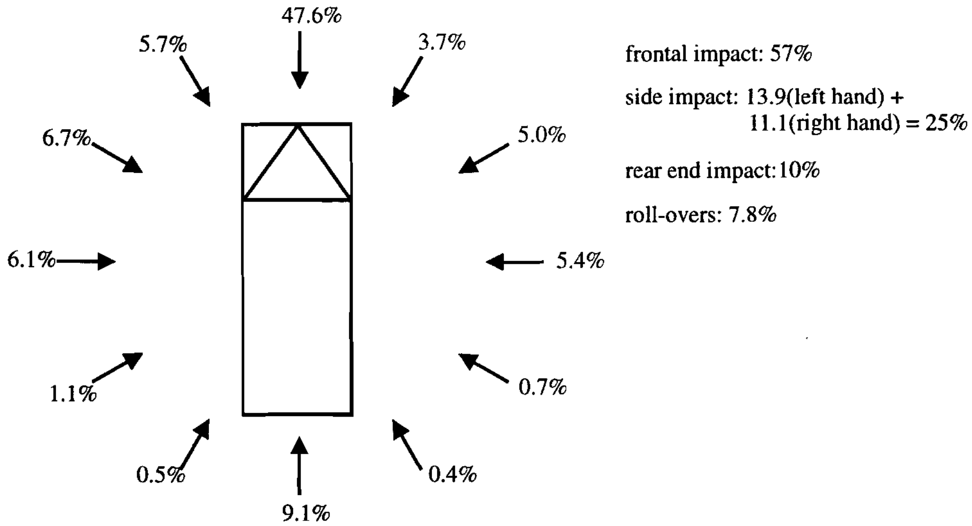


Figure 1.3 *Accident distribution by crash direction, 100% ≙ 8173 collisions (Schoon, 1988).*

collision type	collisions [%]	fatalities [%]
frontal impact	56.1	54.3
left side impact	13.8	19.2
right side impact	11.2	16.7
rear impact	9.1	-
multiple collisions	1.8	0.9
roll over	7.9	8.9
total	100 (~7669 cases)	100 (~114 casualties)

Table 1.1 *Driver data arranged according to impact type for the Netherlands (Schoon 1988).*

The collision simulation spectrum must be representative for the real situation on the road. Parameters are: the occupants, the impact location, speed and direction and the crash opponent. Figure 1.3 shows an accident distribution as a function of impact load direction and contact zone. It is important to realise that the accident distribution against impact direction does not coincide with the injury distribution. This is caused by the dependency of the above mentioned parameters combined with human biomechanics and the protection properties of existing cars. Comparing, for instance, the ratio frontal impact / left side impact occurrence with the fatalities ratio bearing on the driver, side impact collisions appear to be the most life threatening (Table 1.1).

When human load tolerance limits are exceeded, injury is caused. Accident simulation with a risk to exceed these limits can not be put through using human volunteers. For that reason human substitutes have to be used. Hence, knowledge of the mechanical behaviour of the human body under extreme load conditions, including the load tolerance limits, plays a key role in this validation approach of a design on crashworthiness. Colonel Stapp pioneered in research on the effects of mechanical force on living tissues (Stapp, 1967). As a volunteer in experiments, he sustained decelerations of 25 g average and 40 g peaks during a stop in 1.4 seconds from a velocity of 632 mph attained by a rocket sled in 1954. Nowadays, most legal requirements are based on experimental test configurations, using anthropomorphic test devices (dummies).

In the design concept stage, when no physical vehicle construction is available for experimental crash tests, numerical accident simulation techniques can inform the designer on the crash properties of a vehicle design. However, detailed numerical modelling is time consuming, requires much calculation effort and is only possible if all the geometrical data is available. Hence, a balanced compromise between numerical modelling effort and simulation accuracy, dependant on the design stage, is essential for achieving a high vehicle development efficiency.

A special hybrid technique is the Composite Test Procedure (CTP), combining numerical with experimental accident simulation as will be discussed in Chapter 2.

1.3 Design for crashworthiness

To avoid injuries in crashes, the vehicle design must prevent the surpassing of human load tolerance limits. Injuries occurring in traffic collisions can be classified by cause into three groups:

- penetration injuries (due to local high contact stress originated by sharp objects)
- non-penetration injuries: bone fracture or rupture of internal organs (due to contact forces resulting in body acceleration and/or compression)
- other injuries (due to chemical reactions or fire etc.)

Protection principles to avoid these injuries have been developed and are now commonly incorporated in vehicle concepts. All these protection principals are based on the following four methods for injury control (Wismans, 1994):

- Control of the *accident conditions* by changing the environment of the crashing vehicle.
- Improvement of the *crashworthiness* of the vehicle by means of the design of the vehicle construction.

- Control of the *occupant motion* during the crash by means of so called occupant restraint systems.
- Control of the *impact contact* between the body and the environment, i.e. distribution of the load over the contact area.

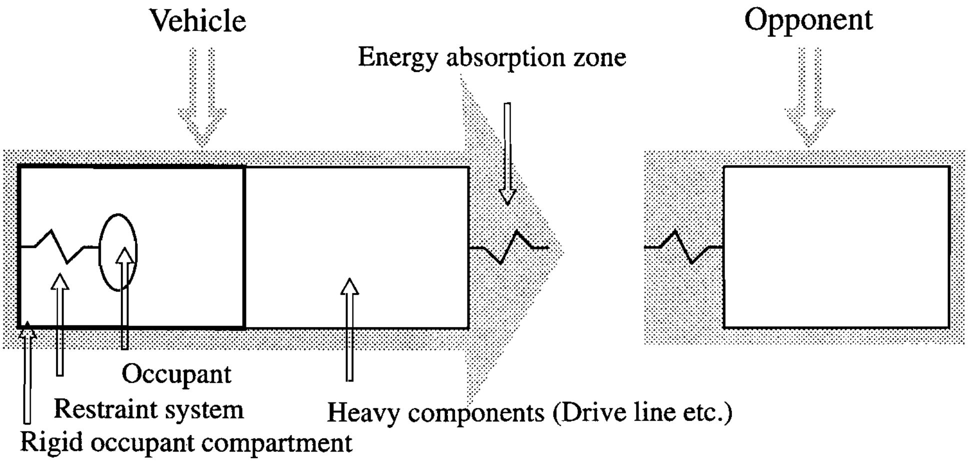


Figure 1.4 A common occupant protection principle.

Example of a protection principle developed for frontal and rear impact situations is illustrated in Figure 1.4. It consists of a rigid occupant compartment, the vehicle drive line, energy absorption zones and an occupant restraint system. When extending this principle to side impacts, the limited deformation length available in a vehicle side structure makes a shift of the energy absorption function to the crash opponent inevitable.

As a result of the successful implementation of the mentioned frontal impact protection principal, injuries caused in side impacts become statistically more relevant. Hence, new, more stringent, legal requirements for side impact safety have been formulated recently (Albers, 1994). A phased implementation of these regulations forces the car manufacturers to adapt their products quickly.

Several tools to enhance the side impact protection of existing vehicle construction concepts have been presented (Suthurst, 1985; Coo, 1991). These techniques are based on coarse numerical modelling of an existing vehicle structure. The vehicle models are fitted on experimental crash data. By tuning the properties of the construction components in the model, better performance of the concept is aimed. This approach fits well in the trend to minimize vehicle development time and costs, because the amount of necessary experimental test iterations is substantially reduced. However, when the concept properties are optimized and still no satisfactory safety level is reached, expensive options have to be implemented.

1.4 Context of this thesis

This thesis describes the development of a design-tool to support the generation and improvement of structural vehicle safety concepts and their efficient transformation into a vehicle body design. It shows how, with elementary numerical vehicle models, insight is gained into the effect of a safety concept on the kinematics of occupants in a crash and the accompanying loads. Appropriate load response characteristics that provide a safe vehicle deformation behaviour (controlled energy absorption) can be determined with these models.

The load response characteristics must arise from a structural vehicle layout to be designed. A step by step vehicle design procedure is proposed in which every step implies a more detailed description of the vehicle geometry. For each step an adequate expansion of the applied numerical modelling technique is specified to make it suitable for the validation of the laid down design decisions.

The contents of this thesis is centred on side impact crashworthiness. However, the design tool presented is also applicable to design vehicle structures that offer a planned protection in other crash configurations as well.

The contents of this thesis are structured as follows.

Chapter 2 gives a characterisation of side impact crashes and the related international legislation. Furthermore, an overview is given of the tools that are currently available to validate the side impact crashworthiness of vehicle designs and the goal of the research project, a design method, is formulated.

The process to achieve the stated goal is in fact a design problem in which the final product is a design procedure. First, this design problem is analyzed by applying the so-called Quality Function Deployment technique (Hauser, 1988). This process is described in Appendix A. Next, the design problem is functionally decomposed and a concept solution is put forward in Chapter 3.

The concept solution contains four design phases, distinguishable by the category of the design variables handled. Divided over the four design phases, the concept solution contains eleven modules with their specific input and output. In Chapter 4, each module of the concept solution is elaborated.

The validation of the practicability of the modules is discussed in Chapter 5. Furthermore, the performance of the solution is evaluated by means of the engineering requirements that resulted from the QFD analysis discussed in Chapter 3.

In Chapter 6, the significance of the evaluation of safety concepts in a very early stage is illustrated. Two side impact safety concepts are evaluated using elementary models of the type proposed in the developed design procedure.

In Chapter 7 the conclusions of this thesis are given. Suggestions for future research are also formulated.

Chapter 2

Tools to assess the side impact crashworthiness

2.1 Introduction

In the preceding chapter an occupant protection principle (see Figure 1.4) is discussed. First, the intended functioning of this principle during side impact load will be explained here. Next, the recently imposed legal side impact requirements are presented. Then, critical parts of current vehicles in relation to these requirements are reviewed. Current methods to investigate the deformation behaviour of these critical parts of vehicle structures, are then introduced. These methods come under the accident simulation category mentioned in Section 1.2. Finally, the limitations of these current methods in relation to the crashworthiness design process are stated and the aims of the research project dealt with in this thesis are formulated.

2.2 Characterisation of side impact crashes

Central point in a crash is to eliminate the velocity difference between occupant and crash opponent in such a way that human load tolerance limits are not exceeded. In the following discussion, the typical velocity of the major interacting parts during a side impact will be reviewed. Laboratory observation of side impacts has shown that the principle of this type of crash event can be described by the relative movement of the following vehicle elements and the occupant that are separated by deforming zones or free space (the numbers between brackets refer to Figure 2.1):

- vehicle bulk part that undergoes minor deformations (0)
- occupant (1)
- vehicle part contacting the occupant (2)
- side structure (3)
- vehicle part contacting the opponent (4)
- opponent contact face (5)
- opponent bulk part (6).

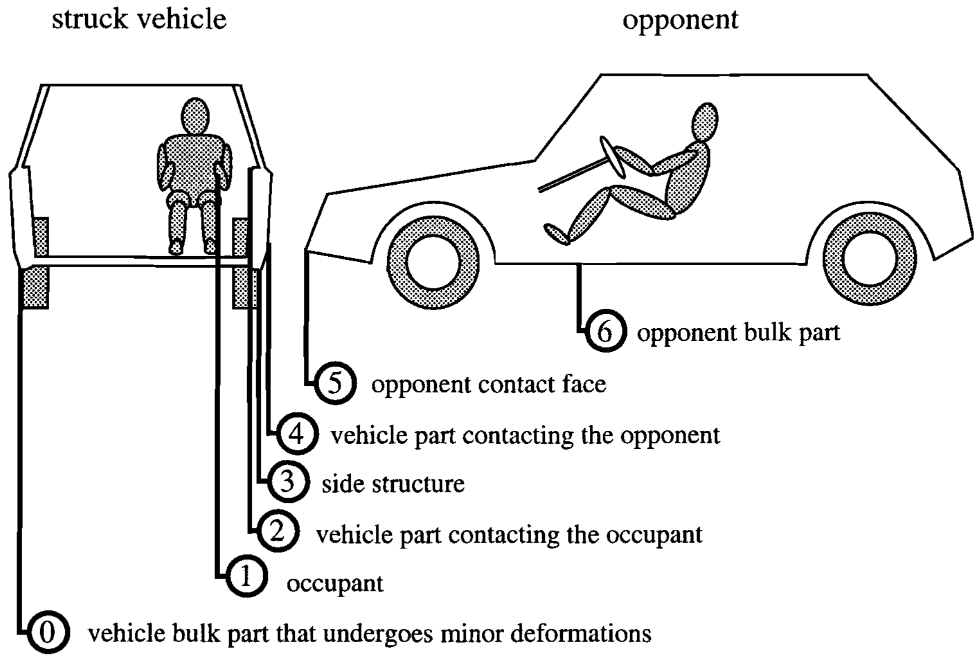


Figure 2.1 *The initial position of important vehicle elements and the occupant in a side impact.*

Figure 2.1 gives a survey of the initial position of these vehicle elements and the occupant. The opponent can be another vehicle as well as an immobile object such as a guard-rail. Considering the velocity development in time, the following crash stages can be distinguished:

Stage 1 After the first contact of the opponent with the vehicle, the contact area increases. This is the result of the deformation of the vehicle exterior surface as well as of the deformation of the opponent. At the same time the struck vehicle's side structure is accelerated. At this stage the occupant is not yet loaded. In case of a vehicle to vehicle crash with a crash velocity of around 14 meters per second, this first stage takes about 20 milliseconds.

Stage 2 At the moment that the initial gap between occupant and the vehicle side structure—in practice about 15 centimetres—is vanished, Stage 2 starts. The occupant is now being accelerated to the speed of the intruding side structure. First contact will occur either with the human pelvis or the thorax yielding a large difference in human body loads (Hobbs, 1989). The duration of this stage is about 40 milliseconds.

Stage 3 The occupant is now moving away from the contact zone, and may collide with the occupant seated at the non-struck side. Furthermore, the vehicle accelerates and the velocity of vehicle and opponent become identical. From that moment, no further vehicle intrusion takes place. In case of an impact with an immobile obstacle, this is the final stage of the crash.

Stage 4 If the vehicle is hit by a mobile object, such as another car, a fourth stage completes the side impact crash. During this stage, kinetic energy which could not be absorbed in the deformation of the structures is dissipated via the friction of the tires on the road surface.

Stage 2 of the impact process causes the most severe injuries. These injuries are the result of the contact of the occupant with vehicle interior at the struck side and, possibly, the direct contact with the striking object. In Stage 3, the survival space is further reduced. As a result of secondary contact with the interior, additional injuries can occur in the Stages 3 and 4. This secondary contact is involved with considerably lower velocity differences than in the primary impact.

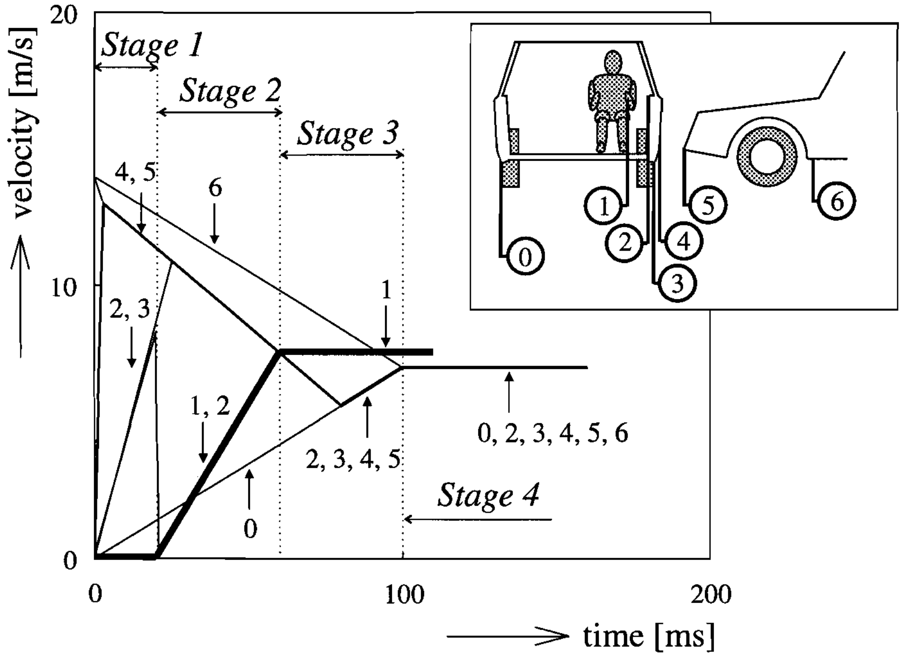


Figure 2.2 *Schematic velocity versus time diagram of a vehicle (a mobile object) impacting the side structure of another vehicle.*

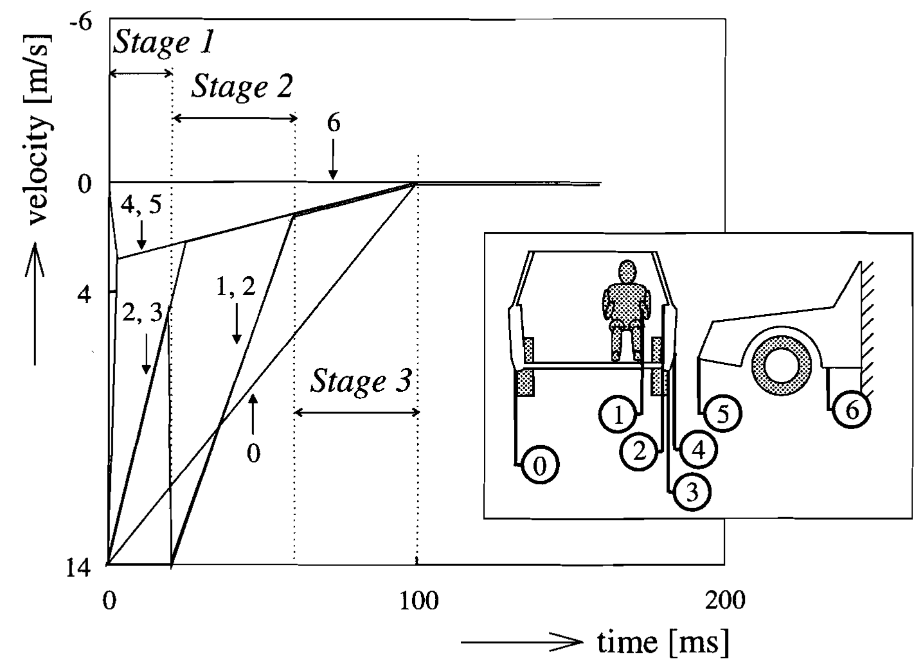


Figure 2.3 *A schematic velocity time diagram of an immobile object impacting a vehicle side structure.*

In Figure 2.2 and Figure 2.3 characteristic diagrams are given of velocities over time for a mobile and an immobile opponent, respectively. These diagrams are only meant as schematic illustrations of the side impact crash. Hence, characteristics such as vibrations and non-linear behaviour are not taken into account. Besides that, the common elastic rebound behaviour of the structure is not taken into account in the diagrams.

2.3 The side impact protection principle

The protection principle consists of the following elements (see Figure 1.4):

- an occupant compartment equipped with an occupant restraint system,
- an energy absorption zone.

Figure 2.1 illustrates that, in side impact crashes, the energy absorbtion zone to dissipate the crash energy is located between the "Vehicle bulk part that undergoes minor deformations (0)" and the "Opponent bulk part (6)". Evidently, the available deformation length of the side structure between the "Opponent contact face (5)" and the "Vehicle part contacting the occupant (2)" is rather small. This implies, that under the high force levels

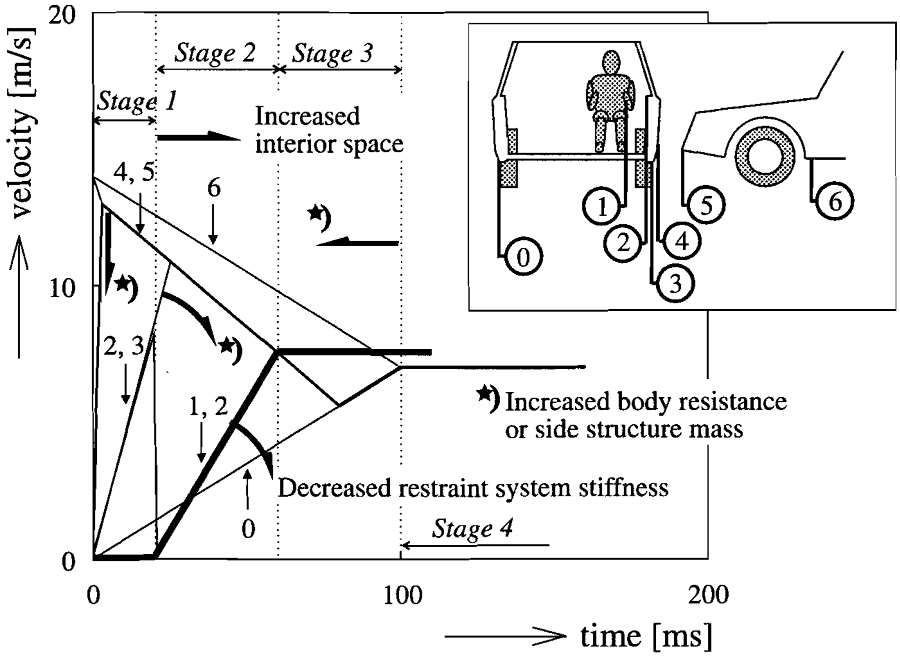


Figure 2.4 *Influence of the vehicle characteristics on the velocity development in a side impact.*

the occupant compartment does not behave as a fully rigid cage. The relative deformation resistance of the opponent and the struck vehicle body area near the contact zone induce an energy absorption ratio. Increasing the body deformation resistance has three important effects on the velocity diagram. First, the initial joint velocity of the side structure and the struck car contact face is decreased. Second, the slope of the side structure velocity curve is lowered. Third, the duration of Stage 3 is becomes shorter. These effects are illustrated in Figure 2.4. An other effect of increasing the body deformation resistance is a larger guaranteed survival space. The same effects are obtained if the share of the side structure in the total vehicle mass is increased.

The restraint system can for instance consist of crushable padding or an active system such as an inflatable airbag, placed between the occupant and the intruding side structure. Decreasing the restraint system stiffness decreases the slope of the occupant velocity curve. However, a minimum stiffness is required to prevent the restraint system from bottoming out leading to extreme acceleration levels for the occupant. However, increasing the padding thickness to prevent bottoming out comes at the expense of the gap between occupant and interior, resulting in an earlier initiation of Stage 2.

If the occupant could be seated at a safe distance from the intruding vehicle side, the restraint system could be fixed on the, relatively slowly accelerating, vehicle bulk part (0 in Figure 2.1). Such a construction is analogous to the widely used protection principle for frontal impacts.

2.4 Legal side impact requirements

In the United States, in 1966, the National Highway Safety Bureau was created by Act of Congress and it initiated a set of standards controlling the performance of vehicles in terms of their crashworthiness. Since then, various passive safety standards have come into effect all over the world. Unfortunately for the car manufacturers, these standards differ from country to country. This is especially the case for the current side impact safety regulations in preparation, that are discussed below.

United States of America
Already in 1973 the National Highway Traffic Safety Administration (NHTSA) introduced a side door strength requirement, formulated in the Federal Motor Vehicle Safety Standard FMVSS 214. Recently a full-scale test has been added to this standard. At the start of september 1996 all new vehicles in the United States must fulfil this extended safety standard.

The quasi static door strength test method
The FMVSS 214 prescribes minimum force levels for a quasi static perpendicular door intrusion test with a loading device consisting of a solid vertical circle cylinder, as shown in Figure 2.5. The diameter of this cylinder is 12 inches. The bottom of the cylinder must move in a horizontal plane 5 inches above the lowest point of the door. The upper part of the cylinder should reach at least half an inch above the bottom edge of the door window opening. Contacts with other structures above this bottom edge are not allowed. The door must be loaded at the centre, measured on a horizontal line 5 inches above the lowest point of the door. To prevent the test vehicle from moving, the sill of the side of the vehicle opposite to the side being tested must be placed against a rigid vertical surface. Besides, the vehicle must be fixed rigidly in position by means of tie-down attachments located at or forward of the front wheel centre line and at or rearward of the rear wheel centre line.

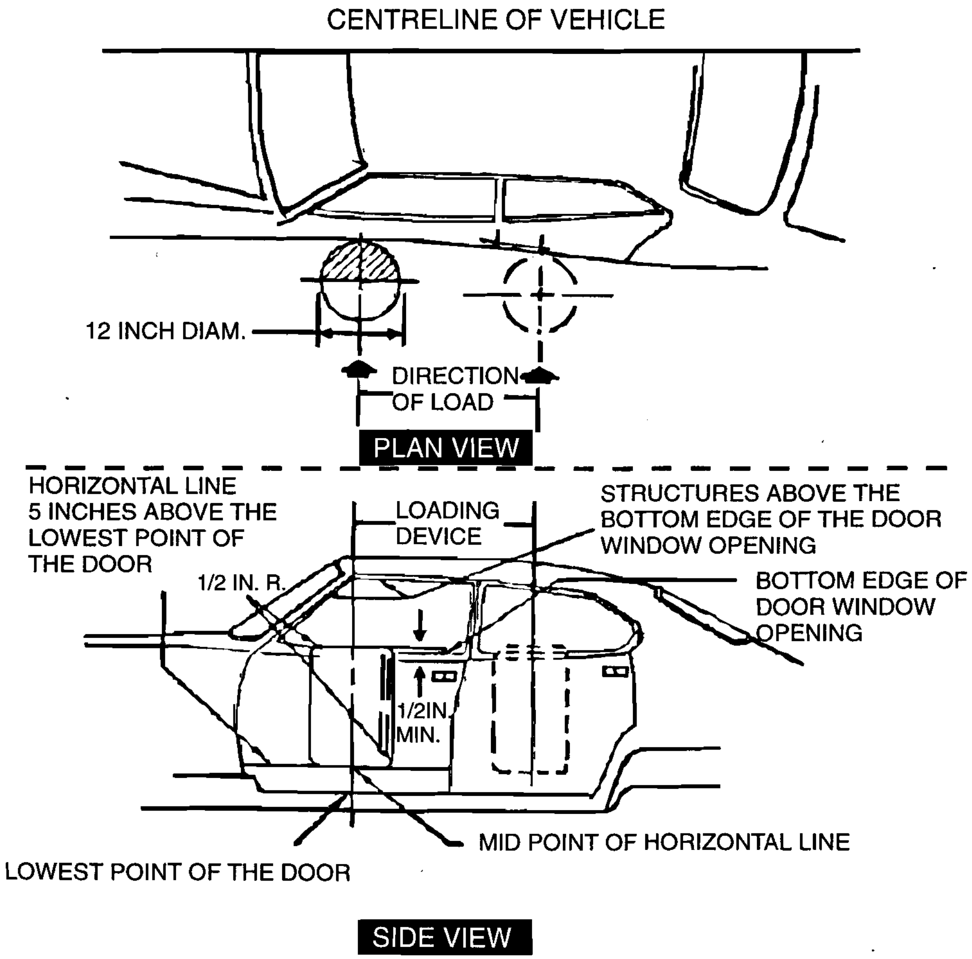


Figure 2.5 Outline of the door crush test (source INTEREUROPE).

initial crush distance [inch]	minimal force [pounds], seats not installed	minimal average force [pounds], seats installed
6 (152 mm)	2,250 average (10.0 kN)	2,250 average (10.0 kN)
12 (304 mm)	3,500 average (15.5 kN)	4,375 average (19.5 kN)
18 (456 mm)	7,000 peak (31.1 kN)	12,000 peak (53.4 kN)

Table 2.1 Resistance requirements for door intrusion test FMVSS 214.

Performance requirements for the quasi static test

Dependant on the seat presence, the average minimum crush resistance over the initial 6 and 12 inches of the intrusion is currently prescribed besides the minimum peak force level over the entire 18 inch of intrusion, as indicated in Table 2.1.

The full-scale test part of FMVSS 214

For the full scale test, the vehicle has to be equipped with two instrumented United States Side Impact Dummies (US-SID). One dummy must be seated at the front seat and one at the back seat, both at the struck side of the vehicle. The load is introduced by a barrier with a mass of 1368 kg. A deformable device is mounted on the front of this rigid barrier. This device is intended to represent the front-end structural stiffness and geometry of a typical American vehicle. The left edge of the barrier face at the front side of the vehicle must coincide with the so-called impact line at the moment of impact. This impact line is located 37 inches forward of the centre of the vehicle's wheel base. For vehicle's with a wheel base greater than 114 inches this line is located 20 inches rearward of the centre line of the vehicle's front axle.

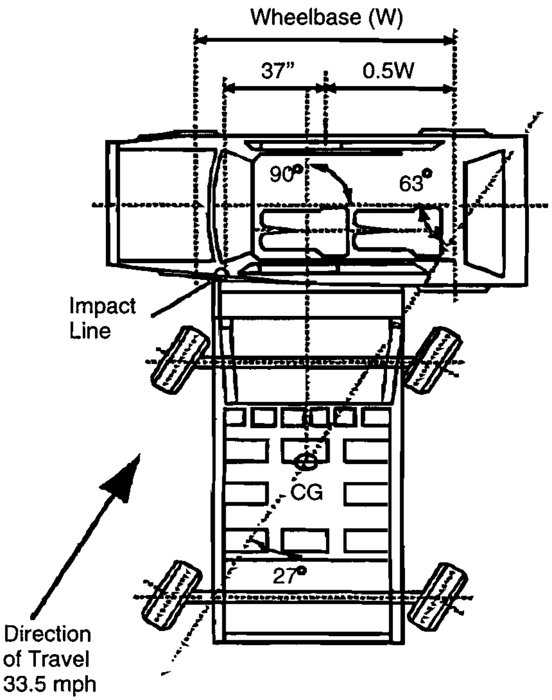


Figure 2.6 The full-scale test configuration (source INTEREUROPE).

It was decided to simulate a perpendicular 30 mph crash configuration in which the struck vehicle has a forward velocity of 15 mph. By applying a crabbed barrier movement, as shown in Figure 2.6., a synchronisation problem to achieve the correct impact location is avoided. This means that the line of forward motion of the moving deformable barrier forms an angle of 63 degrees with the longitudinal centre line of the stationary positioned test vehicle. The barrier front orientation is kept in parallel with the vehicle centre line by positioning all barrier wheels at an angle of 27 degrees relative to the centre line of the moving barrier. The barrier velocity, resulting from the following equation:

$$v_{barrier} = \sqrt{v^2_{test\ vehicle} + v^2_{bullit\ vehicle}} \tag{2.1}$$

is 33.5 mph (53.9 km/hour).

Performance requirements for the full scale test

Two load limits for the US-SID are stated. For the thorax, the so-called Thoracic Trauma Index (TTI(d)), an acceleration based empirical expression, shall not exceed 85 g for passenger cars with four side doors, and—as a compromise because of the difficult structural feasibility—shall not exceed 90 g for passenger cars with two side doors. The TTI(d) is given by the following equation:

$$TTI(d) = \frac{1}{2}(G_R + G_{LS}) \tag{2.2}$$

where "G_R" represents the highest value of either the upper or lower rib peak accelerations expressed in g's and "G_{LS}" represents the lower spine peak acceleration.

For the pelvis the peak lateral acceleration shall not exceed 130 g.

The FMVSS 214 contains additional requirements for all door latches and hinges concerning door opening during the dynamic test.

Europe

Concerning side impact safety, already in 1974 the European Experimental Vehicles Committee proposed a procedure to evaluate the protection capacity of passenger cars. However, it has taken more than two decades before in 1998 the first European side impact safety regulation will be enforced.

Full scale test method

The European regulation proposal consists of a dynamic full scale test. Most test criteria in this test are based on dummy readings measured with an EuroSID (European Side Impact Dummy) anthropomorphic test device seated on the driver's seat of the stationary vehicle to be tested. The driver's side of the vehicle is loaded, perpendicularly to its longitudinal centre line, with a 950 kg barrier. A deformable device is mounted on the front of the rigid barrier. This deformable device is intended to represent the front-end structural stiffness and geometry of a typical European vehicle. The barrier is propelled in a straight line, making the centre of the barrier move with a speed of 50 km/h in the direction of the seat reference point of the driver, as specified by the car-manufacturer.

Performance requirements

Five load limits for the EuroSID are stated. For the head, the so-called Head Performance Criterion (HPC) may not exceed the value of 1000 to pass the test. The HPC is given by the following empirical expression:

$$\mathbf{HPC} = \mathbf{max.}\left((t_2 - t_1) \cdot \left[\frac{1}{t_2 - t_1} \cdot \int_{t_1}^{t_2} \mathbf{a} \ dt \right]^{2.5} \right) \qquad (2.3)$$

where "a" is the resultant head acceleration expressed in g's,

t_1 and t_2 are points of time expressed in seconds to maximize the value of HPC, and $t_2 - t_1 \leq 36$ ms.

For the thorax, the maximum value of the deflection measured on each of the three dummy ribs may not exceed 42 mm. Additionally, the so-called Viscous Criterion (VC) defined as:

$$\mathbf{VC} = \left(\frac{\mathbf{D(t)}}{\mathbf{0.14}} \right) \cdot \left(\frac{\mathbf{dD(t)}}{\mathbf{dt}} \right) \qquad (2.4)$$

where "D" is the rib deformation, must be lower than 1 m/s. The factor 0.14 corresponds with the half width of the thorax in metres.

For the abdomen, a so-called Abdomen Performance Criterion (APC) is defined as:

$$\mathbf{APC} = \mathbf{max.}\left(|\mathbf{F_1(t)}| + |\mathbf{F_2(t)}| + |\mathbf{F_3(t)}| \right) \qquad (2.5)$$

with F_1, F_2, and F_3 being the three loads measured on the abdomen of the dummy. The APC is not allowed to exceed a peak force level of 2.5 kN.

For the pelvic region, a so-called Pubic Symphysis Peak Force (PSPF), measured on the pelvis of the dummy, may not exceed a level of 6 kN.

The European regulation contains additional requirements for the door-opening and rescue behaviour, for the general avoidance of injury potential in the interior especially to prevent penetration injuries and for the loss of fuel.

The performance requirements for the American and European full scale test are summarized in the following table:

	Europe	United States
Head	HPC < 1000	
Chest	VC < 1 m/s rib deflection < 42 mm	TTI < 85 g (four doors) TTI < 90 g (two doors)
Pelvic region	APC < 2.5 kN Pubic Symphysis Force < 6 kN	Pelvis lateral acceleration < 130 g

Table 2.2 Dummy load limits in the European and United States legal side impact requirements.

2.5 Critical parts of current vehicle designs

Dynamic (crash-)tests and quasi static (crush-)tests in accordance with the procedures as discussed in the preceding section are a final check if an integral vehicle design meets a prescribed safety level. Besides that, tests are in use to validate choices during the engineering design of the vehicle. For this purpose, quick validation techniques that indicate the functioning of the separate components that comprise the vehicle structure are required. The present section explains what the characteristic critical parts for side impact protection are. These critical parts require special attention in design validations.

The shape and speed of the intruding vehicle side structure in combination with the functioning of the restraint system are factors governing the outcome of a side impact.

The functioning of the restraint system depends upon the system itself and on the behaviour of the supporting structure. Especially, the dynamic behaviour of the door inner structure, supporting crushable interior padding, is of significance (Landheer, 1991; Wasko, 1991).

The shape of the intruding side structure contacting the occupant is of great influence on the injuring effect of a side impact (Hobbs, 1989). High load levels in the abdominal region for instance, can cause very life threatening internal injuries while comparable load levels in the pelvic region are much less alarming. This phenomenon is expressed in the requirements of the European side impact regulation. The interaction of the struck door with the door sill together with the deformation behaviour of the B-pillar greatly influence the intrusion shape in current vehicle designs.

The intrusion speed depends upon the construction supporting the part of the vehicle side structure loaded by the opponent. Especially, in a crash configuration with a stiff opponent as described by the FMVSS 214 dynamic test, relatively rigid parts of the vehicle body as the A-pillar have much influence on the intrusion speed. The relatively rigid bumper part of the deformable barrier face transfers the crash load to these stiff vehicle parts. The barrier front described in the European government bill is much weaker and therefore has a load spreading characteristic. The door sill plus its support in combination with the B-pillar are the other relevant elements in the velocity development of the side structure contacting the occupant. For compact class cars the rear wheel suspension is of influence as well because it forms a load transfer path to the opposite side of the vehicle.

2.6 Safety design validation

The validation procedures concerning passive safety—in use to support the vehicle design process—are intended to predict the behaviour of the vehicle or parts of it in the real life collision spectrum. Since the amount of tests that can be put through during the design of a vehicle is limited by cost and time, only a specific selection of test configurations is executed. Evidently, tests related to crash situations intended in legal requirements have priority above other tests. Measures to express the safety level of a certain construction that have to be established by the validation procedures are based on occupant loads or the crash behaviour of the construction itself.

2.6.1 Experimental safety design validation

Characteristic for experimental safety design validation is the obvious necessity for the availability of a real vehicle structure. Experimental tests in most cases are destructive. This implies that, for a given crash configuration, only one single test can be performed on each test-structure. Only in rare cases a deformed construction will be loaded a second time at another still undeformed zone.

If occupant load related measures must experimentally be established, a physical human substitute is applied. There are several anthropomorphic test devices available, each intended to be representative for a specific occupant group. The dummies are developed to predict human load for a specific crash configuration. The EuroSID and US-SID for instance, mentioned in Section 2.4, are meant to simulate the human body in side impact configurations only. In other load situations the dummy will not respond like a human body and no relevant load levels can be registered.

A main division can be made between full-scale, (sub)system and component testing. Characteristic for a full-scale test is that the system is dynamically loaded and force is built by the inertia effect of the loaded structure, not by compression of the structure. For that reason the full vehicle mass must be represented in the construction to achieve a realistic load situation. The load is introduced by a device representing the crash opponent. This device can be mobile (moving or stationary) or immobile, and rigid or deformable. The European side impact test proposal, described in Section 2.4, is a clear example of a full-scale test.

The other experimental validation techniques all have to do with a vehicle structure support that provides a reaction force during loading of the test object. This structure support can in some cases deform in a prescribed manner. Often only a part of the vehicle structure is tested. In this situation, dependant on the appearance of the test sample one speaks of a component or (sub)system test. As a consequence of the support the load can be applied dynamic (Crash) as well as quasi static (Crush). The door intrusion test as part of the United States side impact regulation, described in Section 2.4, is an example of a Crush test.

2.6.2 Numerical safety design validation

There has been an enormous expansion of computer capacity accompanied by continuing crash simulation software development for the last two decades. Furthermore, a lot of experience is gained on the numerical simulation of crashes. Accordingly, numerical crash simulation techniques nowadays form a substantial complement for experimental testing. A major advantage of numerical crash simulation technique compared to experimental crash simulation technique is that no real vehicle structure is necessary to perform a simulation. A division in two crash simulation categories can be made. One category is based on detailed geometry describing models, such as Finite Element Models (FEM). The other category is based on coarse geometry describing models as for instance multi body models. Only specific points of the structure can be recognized in this type of model. The essential difference lays is the input required in addition to the geometrical input, as will be explained below.

Detailed models
Using the finite element modelling technique, the mechanical behaviour of the standard shaped elements of which the structure model consists must be defined. For this purpose a

lot of constitutive materials models, i.e. functions that describe the relation between deformation and deformation resistance, are available in the current FEM software versions. For current vehicle bodies made of metal sheet, these material models have shown to be effective. The required material properties, expressed in parameters that can be substituted in the constitutive material models, are available in handbooks or can be established by simple material tests. In case of modelling composite material structures or for instance vehicle interior foam padding it requires more expertise to determine the material properties. Suppliers of these material types benefit by a situation that the mechanical behaviour of their products can be predicted as a function of the shape while less prototype components have to be generated. The car-manufacturer can accurately prescribe the desired material properties and an efficient co-operation tightening the relation with the supplier, can be achieved. Hence, it can be expected that quick progress will be made in the ability to model the mechanical behaviour of these materials.

Coarse models

Using the coarse modelling technique, the mechanical behaviour of the elements of which the model consists must be defined as well. Due to the non-standard shape of the elements no universal mechanical properties exist that are not geometry related. This means that the properties must be established for every new geometry. The advantage of the application of these coarse models compared to detailed models lays in the calculation time. If repeated calculations with the same model are put through to find model parameter settings that offer the desired crash behaviour, a short calculation time may save more time than the extra time that has to be invested first to establish the element properties of the coarse elements.

For prismatic shaped elements (columns and panels), e.g. by using so-called Super Folding elements, solutions are available to establish the mechanical properties for specific load cases (Wierzbicki, 1989).

For non-prismatic shaped elements or differing load cases, other, more time consuming methods are available. The component that the element of concern should represent in the numerical model can be loaded in an experimental test to assess its properties in specific load cases. For every load situation the accompanying properties must be assessed in a separate test. Analogously to this approach the coarse element could be modelled as a composition of finite elements and a numerical simulation of specific load cases could than be performed. This is sensible when repeated calculations are performed with the full vehicle model composed of the coarse elements. Valuable time can be saved in this way. However, when only one validation calculation must be made, it is more efficient to implement the detailed component model in the coarse full structural model. A library of specific component properties once established can be helpful for future numerical crash simulations.

The numerical occupant models form a special category. In fact it are all coarse numerical models of the dummies used in experimental testing instead of real human bodies. A complete detailed geometric FEM model of a human being is not yet fully developed since for the time being no direct applicable material models for the human tissues exist. However, should these models become available, a direct relation between internal occurring loads and mechanical damage will enable more accurate injury predictions.

2.7 The Computer-Controlled Composite Test Procedure

The Computer-Controlled Composite Test Procedure (CC-CTP) is a test method combining numerical and experimental validation tools in a process to predict the considered vehicle's side impact crashworthiness (Richter, 1991). This method was proposed by the Association des Constructeurs Europeans d'Automobile (ACEA), formerly the Committee of Common Market automobile Constructers (CCMC), as an alternative for the European—full scale test based—legal requirement (CCMC, 1989). The background for this proposal was the poor reproducibility of the full scale test measures. It appears that differences in dummy load measures of about a factor two are no exception in repeated full scale tests (Bourdillon, 1994).

The structure will be quasi statically loaded. Hence, the vehicle to be tested must be mounted on a support system. The roof rail and the door sill at the non struck side of the vehicle are supported in the lateral direction. The body shell fixation points for the front and rear wheel suspension systems are used to prevent the occurrence of vertical movement at the struck side of the vehicle and to restrict all the degrees of freedom at the non struck side.

Load is introduced by means of a linear guided lateral moving barrier called the Exterior Deformable Loading Device (EDLD). At the front of the barrier a deformable device, as used in the European full scale test, is mounted. The impact location is conform this full scale test as well. During a side impact, the interaction of the occupant with the structure takes place through contact zones located at the vehicle interior. In the CTP test rig these contact zones are loaded with so-called Interior Loading Devices (ILD's), one for the thoracic region and one for the pelvic region. The ILD's are rigid and have the shape of the concerned EuroSID sections for a specific load condition. A lateral guiding system prevents non-lateral movement of the ILD's (Figure 2.7).

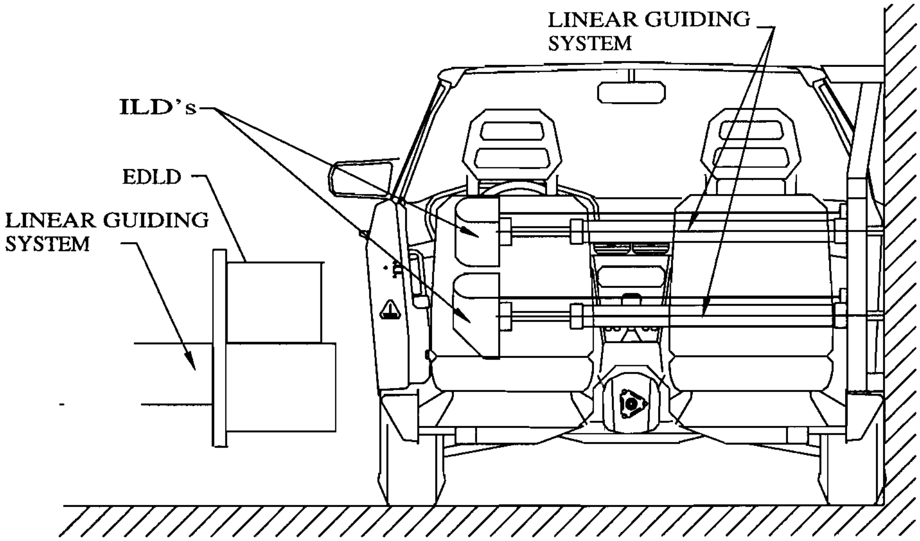
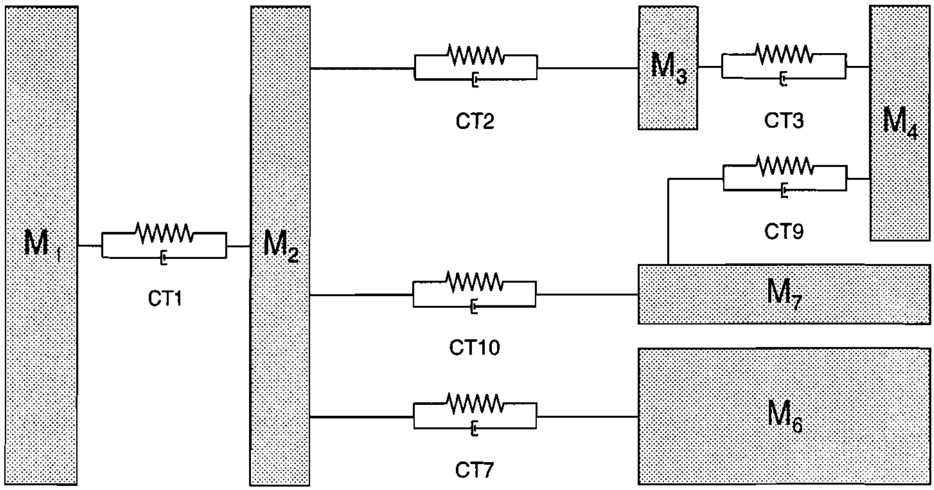


Figure 2.7 A vehicle on the CTP test rig.



M_1 =	mass of striking opponent	CT1 =	opponent contact
M_2 =	mass struck part of vehicle	CT2 =	thorax contact
M_3 =	rib mass	CT10 =	pelvis contact
M_4 =	spine mass	CT3 =	rib cage characteristic
M_6 =	mass of vehicle bulk part	CT7 =	vehicle body characteristic
M_7 =	pelvis mass	CT9 =	spine characteristic

Figure 2.8 The 1-dimensional model used in the CC-CTP.

The quasi static lateral displacement of the EDLD and both ILD's is on-line computer controlled. This on line control is based on a simultaneous numerical crash simulation. Preceding to the quasi static loading process, the masses of the elements in the applied numerical model are established. This simple one-dimensional model is illustrated in Figure 2.8. Validated properties of the dummy in the model exist for the simulation of a EuroSID (in drivers position as well as with the arm incorporated in the contact) and for the US-SID (Richter, 1991). The mass M_2 of the struck part of the vehicle is given by the following empirical relation:

$$M_2 = 0.033 \cdot M_{curb} + 7.3 \tag{2.6}$$

expressed in kg's, where M_{curb} is the mass of the complete vehicle. The mass M_6 of the vehicle bulk part is the difference between the mass of the complete vehicle and M_2 as expressed in the following equation:

$$M_6 = M_{curb} - M_2 \tag{2.7}$$

The required deformation characteristics for the numerical crash simulation are measured simultaneously on the test rig during the test. When the test is carried out all model parameters are known and dummy loads can be found. In the current CTP no head is modelled. This implies that no HPC value can be calculated. As a supplement to the CTP a separate head impact test on potential head contacting interior zones, for instance using the MIRA Free-Flight Headform Rig (Clemo, 1991), could be performed.

An important advantage of this hybrid approach is that no physical anthropomorphic test device is necessary. Only the occupant contact face shape has to be constructed physically.

2.8 An efficient procedure to design crashworthy vehicles

The tools discussed in the preceding section enable the comparison of structural concepts towards safety requirements via crash models in various stages of the vehicle design process. With an early comparison of concepts, it can be avoided that effort is put in a concept that—in a later stage—appears to require expensive measures to fulfil specific safety requirements or must even be rejected completely.

The research project dealt with in this thesis is focused on the efficient design of crashworthy vehicles. Therefore, the use of crash analysis models from first protection principal up to final geometric vehicle design via a safety concept stage is addressed. In the safety concept stage, functions that realise a protection principal are assigned to areas of the vehicle body with a step by step increasing detail-level. A convenient technique to come from a safety concept model for a protection principal to a more detailed modelling level, allowing an unambiguous vehicle geometry definition is presented. The final protection capacity of a vehicle design can then be achieved by tuning the structural properties within the detailed model.

Chapter 3

A method to design crashworthy vehicles

3.1 Introduction

Development of a crashworthiness integration method for the design of safe vehicles is in itself a design problem. Here a structured approach is chosen to analyze the design problem securing that it is well defined and understood. For this analysis the so-called Quality Function Deployment technique is applied. This technique, originated in 1972 at Mitsubishi's Kobe shipyard site and next developed at Toyota, is a basic design tool to accomplish design requirement specifications for the subsequent phases of product development (Hauser, 1988). The resulting engineering requirements, i.e. measurable design specifications to evaluate design propositions, are presented in this chapter. A description of the accomplishment of these requirements is given in Appendix A. After the design problem is functionally decomposed, a concept for a crashworthiness integration method suitable to fulfil the formulated engineering requirements is proposed. To tune vehicle design variables, the method contains four sequential design change levels differing in geometric detail. Finally, a function description for each design change level, will be presented. These function descriptions then form the basis for the accomplishment of the method discussed in Chapter 4.

3.2 The design specification

The analysis of the design problem with the QFD technique resulted in 32 engineering requirements. Five of these requirements are strongly related with "musts", i.e. user specified requirements that must be fulfilled to make a crashworthiness integration method applicable (see Table A.3). This implies that it is essential to meet the targets for these engineering requirements. An overview of the established engineering requirements labelled <e.r. number> with accompanying targets, is given in Table 3.1.

label	engineering requirement	unit	target
e.r.1	point modelling must be available	[-]	available
e.r.2	wire frame modelling must be available	[-]	available
e.r.3	surface modelling must be available	[-]	available
e.r.4	solid modelling must be available	[-]	available
e.r.5	geometry input must be graphically displayed	[-]	possible
e.r.6	geometry input must be graphically displayed on line	[-]	possible
e.r.7	time required for first geometric model generation	[hours]	< 2.5
e.r.8	time required to adapt a geometric model	[hours]	< 0.5
e.r.9	number of data conversions to create an analysis model	[-]	1
e.r.10	time required to generate an analysis model	[hours]	< 1
e.r.11	a concentrated load definition option must be available	[-]	available
e.r.12	a spread load definition option must be available	[-]	available
e.r.13	definition of loads as function of time must be possible	[-]	possible
e.r.14	an inertia load definition option must be available	[-]	available
e.r.15	number of detail levels	[-]	4
e.r.16	time required to adapt an analysis model	[hours]	< 0.5
e.r.17	number of data conversions to translate an analysis model into a geometric model	[-]	1
e.r.18	time to convert an analysis model into a geometric model	[hours]	1
e.r.19	deformation magnitude accuracy in the occupant contact area	[|deviation| in %]	< 20
e.r.20	time required for load card generation	[hours]	< 0.3
e.r.21	time required for load card adaption	[hours]	< 0.3
e.r.22	elapsed time for a crash calculation	[hours]	< 3
e.r.23	a deformation versus time output plot option must be available	[-]	available
e.r.24	an acceleration versus time output plot option must be available	[-]	available
e.r.25	a geometry state output plot option must be available	[-]	available
e.r.26	a deformation animation option must be available	[-]	available
e.r.27	number of structural points per occupied volume in the final output	[dm^{-3}]	> 2
e.r.28	number of colors in plots	[-]	> 2
e.r.29	hidden line removal in geometry plots must be available	[-]	available
e.r.30	a cross section state plot option must be available	[-]	available
e.r.31	a perspective geometry plot option must be available	[-]	available
e.r.32	time required to post process data	[hours]	1

Table 3.1 *The engineering requirements and the accompanying targets (grey cells contain targets for engineering requirements strongly related with "musts").*

3.3 The design problem functionally decomposed

Now that the design problem is formulated in a set of engineering requirements a decomposition of the design problem can be performed into specific functions that have to be fulfilled. In Appendix A Section A.2.2, the user requirements logically decomposed into the following five aspects of the design problem:

- **I** geometric vehicle model generation
- **II** analysis model generation
- **III** load definition
- **IV** (numerical) analysis
- **V** post processing

These aspects largely coincide with the functions to be performed. A graphical representation of the functional relations is presented in Figure 3.1. The words indicating a function are printed in bold letters and numbered from 1 to 11. The link of the functions with the summarised five aspects of the design problem is marked using the above specified Roman numbers. In this graphical representation the input to and output from each function are shown using arrows. The output format of one function must match the input format for the subsequent function.

It can be expected that the systematic application of the QFD technique to analyze a design problem results in a proper function description, not biased towards a specific predestined solution. The structured approach of our design problem indeed has resulted in a universal function description that for instance does not exclude experimental design validation techniques. Hence, this function representation is not specific for numerical solutions only. However, the engineering targets indicate an analysis model generation time of less than one hour what is quite short for the generation of a physical proto part.

It must be noticed that, during vehicle design, more detailed geometrical information comes available at every down stream phase of the process. Hence, starting with geometrical input of a coarse detail level, in loops geometrical data must be added to finally obtain a desired detail level necessary to fulfil engineering requirement e.r.19 and e.r.27. These loops must not be mixed up with the geometry correction loop indicated in the figure that is applied when a load response is rejected (Function 10). To keep the illustration clear, variation of load cases is not reflected. However, in a real vehicle design situation multiple loading requirements are stated. Hence, a vehicle model has to be accepted for all loading cases, i.e. all load cards with accompanying loading device models, before the geometric detail level is increased. This probably requires several geometry corrections within one detail level.

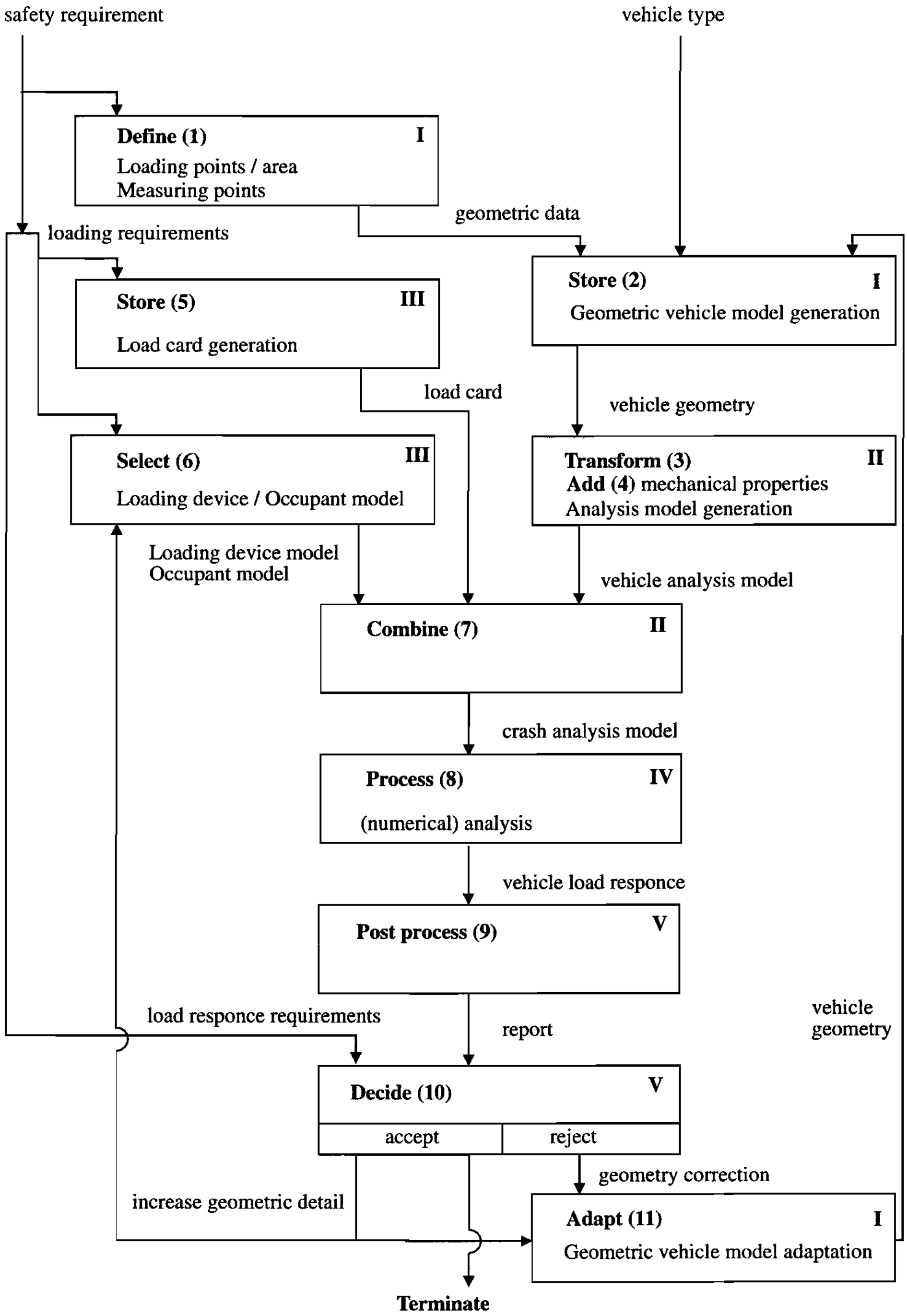


Figure 3.1 Functional decomposition of a crashworthiness integration method for the design of safe vehicles.

3.4 A concept for the crashworthiness integration method

The functionally decomposed representation of the design problem introduced in Figure 3.1 forms a framework for a solution. By specifying the form of the information flow that is represented by the arrows, a concept for the method originates from this frame. Central issue is to determine the contents of the geometric information flow. The geometric detail level of a vehicle concept is increased after the vehicle response to all considered load configurations is accepted, as can be seen in Figure 3.1 (Function 10). In the concept for the method now presented four separate geometric detail levels are proposed. If a next detail level is applied, in fact a new phase in the vehicle design process starts. In every subsequent phase, geometry parameters that describe the vehicle in more detail are frozen. Consequently, adaption of the design to influence the vehicles' crashworthiness is concentrated more on local structural properties after entering a new design phase.

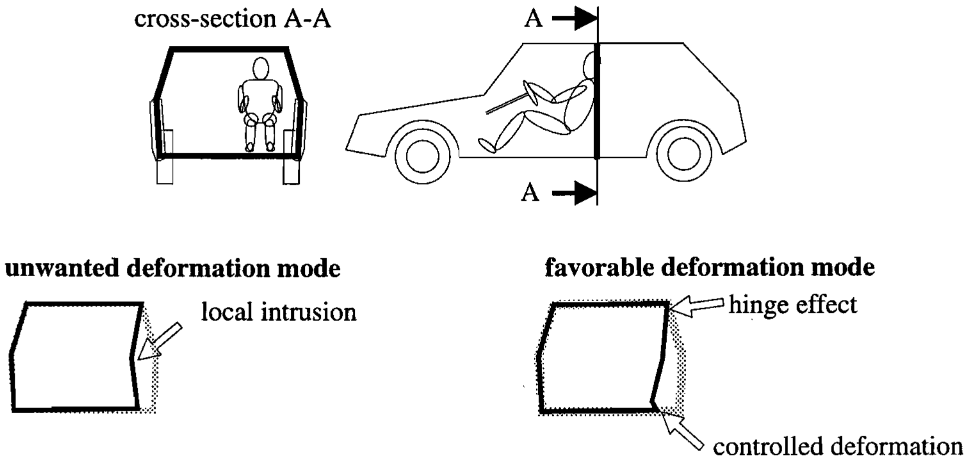


Figure 3.2 Example of deformation modes concerning side impact loads.

During a crash, a vehicle may deform in many different modes. Each of these depends on the layout of the specific structure. Not all deformation modes are equivalent. One is preferable to another. Figure 3.2 gives an example of two deformation modes for side impacts. In a sound safety concept the deformation mode offering the highest occupant protection is pre-programmed in the structure. Two steps are necessary to generate such a concept. First, in the initial design stage when many geometry parameters can still be varied, the preferred deformation mode has to be determined. Hence, the method starts with a geometric detail level that enables the description of structural deformation modes with a minimal amount of geometric data. The phase in which the vehicle geometry is described on this coarse level is indicated as Design Phase 1 of the method. Second, this deformation mode has to be programmed into the new vehicle structure by means of appropriate mechanical properties for the structural concept. The translation of the mechanical properties, appointed in the structural concept, into a vehicle design is planned in three succeeding phases. In these phases the structural concept is tuned by design adaptations starting with macroscopic structural characteristics. Input and output for the four design phases will now be discussed.

Design Phase 1

Geometric input for Design Phase 1 consists of spatial points characteristic for the vehicle to be developed and necessary to describe its macroscopic crash behaviour. Characteristic loading device and occupant simulation points can be enclosed in addition to these vehicle points. Within this phase, suitable deformation modes expressed as displacement programmes for the points are added to the geometrical data. Vehicle design variables that can be altered in Phase 1 are:

- the relative initial location of points representing vehicle parts or areas
- the planned relative displacement for spatial points during a crash
- occupant restraint system properties in the form of force-deflection curves
- the structural deformation resistance expressed as force-deformation functions for point pairs of the vehicle structure (notice that these load links that must make come true the planned deformation modes are merely an indirect vehicle geometry variable).

Design Phase 2

Within this phase the geometric crash load paths through the vehicle are set. Packaging data in the form of occupied volumes added to the output of Design Phase 1 will form the input of Design Phase 2. To fix the spatial paths additional points are defined. Besides that, the macroscopic mechanical properties of the loaded structural parts—as for instance the compression resistance of a cross-member in the floor-section—will be established. Next, a course cross section indication of the loaded parts is made. Vehicle design variables altered in Phase 2 are:

- the relative location of points representing vehicle parts or areas
- the shape of the load paths through the vehicle expressed as a composition of 1-dimensional sections
- the course shape of cross sections of prismatic load transferring structural components
- the occupant restraint system characteristics.

Design Phase 3

In this phase the geometric input consists of spatial points mutually coupled by mechanical properties combined with volumes available for defining load transferring vehicle parts, inside the vehicle contours. Within these volumes the shape of structural parts must now be established. To enable the evaluation of a designed shape of one part independent from the development of other parts, in this phase graduations in detail level over the vehicle will exist, i.e. certain parts are modelled in great detail while other parts maintain coarsely modelled. Vehicle design variables altered in Phase 3 are:

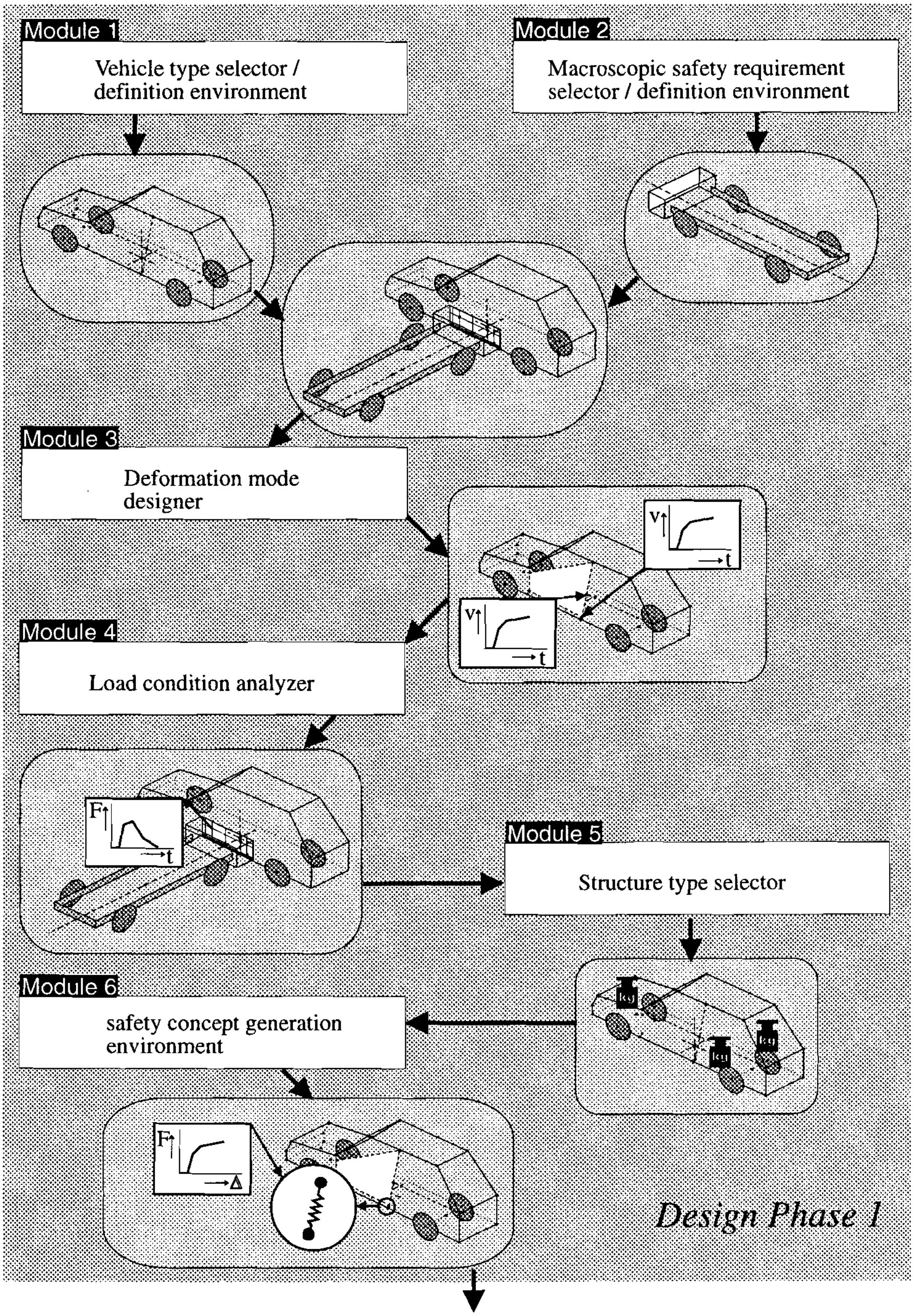
- the (curved) shape of the load paths through the vehicle
- the shape of cross sections of load transferring vehicle parts
- padding material properties and shape (airbags).

Design Phase 4

This design phase coincides with the final stage of the vehicle design. In this stage a detailed geometrical description of all vehicle sub-structures is available. Within Design Phase 4 fine adjustments can be made. In order to properly judge their impact on the crash behaviour a detailed geometric representation is necessary. In this phase detail level graduations will exist. Vehicle design variables altered in Phase 4 are:

- the local plate thickness
- the exact material properties
- the interior trim.

In Figure 3.3 and Figure 3.4 a schematic representation of the proposed concept for a crashworthiness integration method is given. The required input of the user and the output that should be generated by the system is presented below.



see next page

Figure 3.3 The concept for the crashworthiness integration method (Part 1).

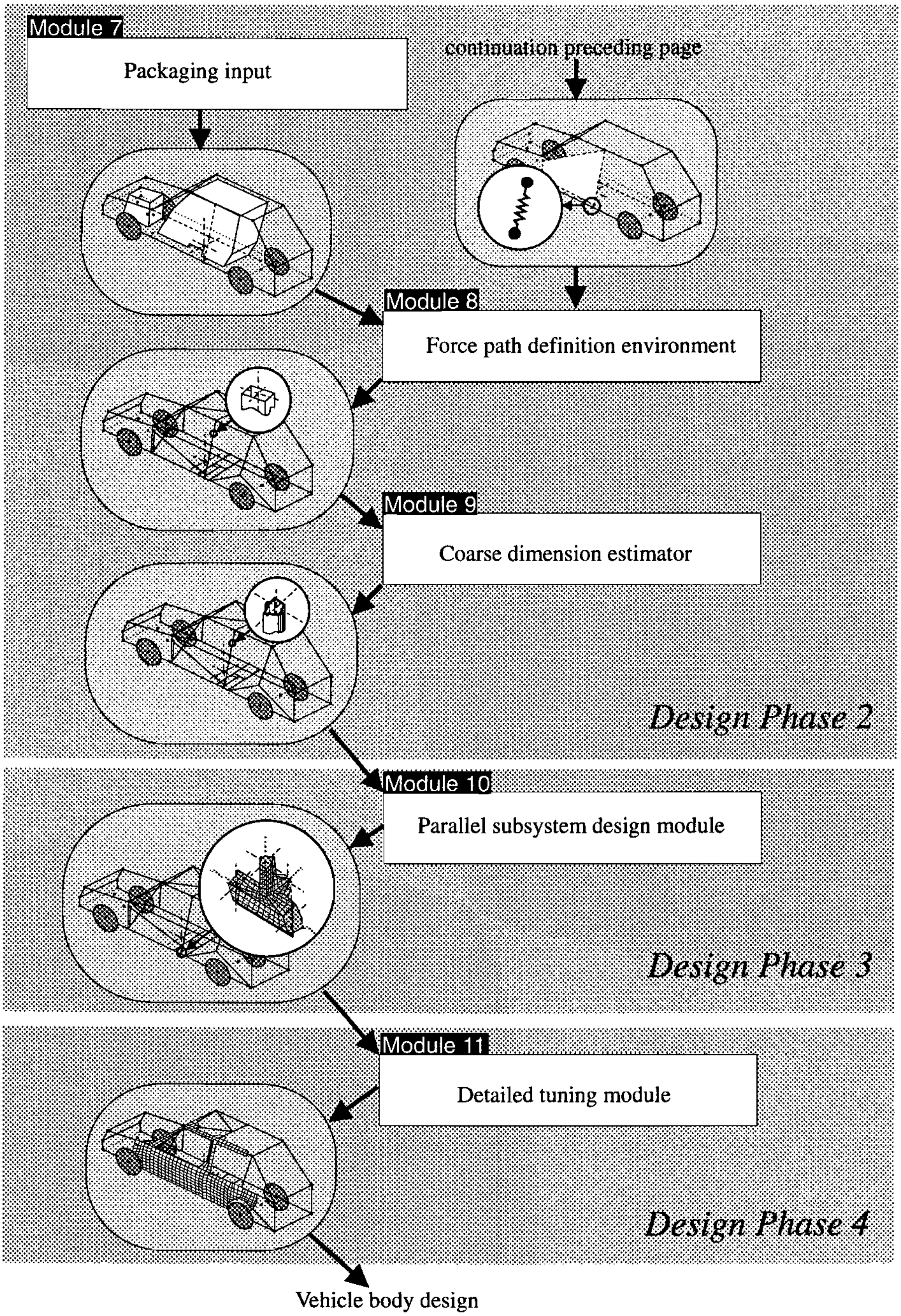


Figure 3.4 The concept for the crashworthiness integration method (Part 2).

Module 1

user input:

- specification of vehicle type and size

system output:

- set of characteristic spatial vehicle points (among which are restraint-system mounting points)
- estimate of the vehicle inertia properties expressed as discrete masses assigned to the points

Module 2

user input:

- selection of passive safety requirements that have to be fulfilled by the vehicle
- specification of load point positions to highlight the areas where load transfer from the environment into the vehicle structure is intended to take place

system output:

- a numerical model of the loading devices under consideration, correctly positioned to the characteristic set of spatial vehicle points
- measuring points for later checking if the safety requirements are met

Module 3

user input:

- deformation modes expressed as velocity time functions for groups of points that are intended to be part of rigid links (areas that do not deform) within the vehicle structure, in specific crashes (proposals must comply with law of conservation of momentum, this requires the mass specification generated by the system in Module 1)
- macroscopic occupant restraint-system deformation properties

system output:

- the system gives pass/fail on measuring point responses such as dummy readings, in numerical simulations
- a visual presentation of the imposed deformation process

Module 4

users input:

- a kinematic model with one internal degree of freedom for each crash type, composed of the rigid links represented as groups of points in Module 3 and constraints that represent connections between the rigid links
- velocity time functions for two of the rigid links so that a deformation of the kinematic model is specified
- macroscopic occupant restraint-system deformation properties
- kinetic conditions for the loading device according to the requirement (for instance: initial velocity for a barrier)

system output:

- the system gives pass/fail on measuring point responses such as dummy readings, in numerical simulations
- a visual presentation of the imposed deformation process
- the level of inertia forces that act on the rigid links in the defined constraints
- actual level of loads on the loading device contact face

Module 5

user input:

- more accurate specification of the mass of the rigid links based on the load levels calculated in Module 4 and derived from the choice of a structure type to be applied locally. The mass has to be distributed over the discrete points in the links because no structural elements have been defined yet.

system output:

- representation of the vehicle by the set of spatial points from Module 1, now with a more precise inertia property definition

Module 6

user input:

- specification of the location of areas where crash-energy must be dissipated in the vehicle
- force-deflection functions for these areas of crash-energy dissipation
- kinetic conditions for the loading device according to the requirement (for instance: initial velocity for a barrier)

system output:

- the system gives pass/fail on measuring point responses such as dummy readings, in numerical simulations
- a visual presentation of the developing deformation process of the defined mechanical model
- load levels in the connections between the rigid links represented as groups of points

Module 7

user input:

- specification of the contour of components including their orientation within the spatial point representation of the vehicle

system output:

- system marks the volume available for a structure to fulfil the functions laid down in the mechanical models developed for the different load cases

Module 8

user input:

- a first specification of the structural shape by definition of load paths consisting of prismatic rectangular beams that fit within the available volume, areas planned to undergo substantial deformations are still defined with springs.
- macroscopic occupant restraint-system deformation properties
- kinetic conditions for the loading device according to the requirement (for instance: initial velocity for a barrier)

system output:

- the system gives pass/fail on measuring point responses, in numerical simulations
- a visual presentation of the developing deformation process of the defined beam model
- load levels in the beam relative to their collapse loads

Module 9

users input:

- implementation of styling constraints, i.e. the shape of areas of the vehicle body that are directly in sight (not covered up), in the cross section shape of the beams resulting in adapted moments of bending inertia
- spring definitions for areas that are planned to undergo substantial deformations
- macroscopic occupant restraint-system deformation properties
- kinetic conditions for the loading device according to the requirement (for instance: initial velocity for a barrier)

system output:
- the system gives pass/fail on measuring point responses, in numerical simulations
- a visual presentation of the developing deformation process of the defined beam model
- load levels in the beam relative to their collapse loads

Module 10
user input:
- a 3-dimensional description of the structural elements that must offer the deformation properties, until now provided by the springs in the model
- a 3-dimensional description of the not prismatically shaped connections between the beams
- loads (force levels) that did act on areas showing minor deformations in the crash simulations performed in Module 9
- deformation levels of the springs in crash-simulations performed in Module 9

system output:
- the load response of these areas related to the presumed response as did occur in the crash-simulations of Module 9

Module 11
user input:
- a detailed geometry description of the loaded vehicle-side with special attention to the contact-surfaces that transfer the crash-loads. For the vehicle areas presumed to undergo only minor deformations the beam description can be maintained
- kinetic conditions for the loading device according to the requirement (for instance: initial velocity for a barrier)

system output:
- the system gives pass/fail on measuring point responses, in numerical simulations
- a visual presentation of the developing deformation process of the defined vehicle structure
- load levels in the beam relative to their collapse loads

Tools to realise the described information flow must be designed to transform this concept into a functional solution for our “design problem”. Accordingly, all items indicated in the boxes of Figure 3.3 and Figure 3.4, from now on marked as "modules" will be elaborated in the next chapter.

Chapter 4

The modules of the new design approach

4.1 Introduction

In the preceding chapter a new method was proposed, to generate safety concepts and implement them in the vehicle geometry from scratch. In this approach, suitable values for the geometric parameters to accomplish safety concepts in a vehicle structure are established via subsequent parameter alteration levels. Here, all eleven modules of the method, including the modules for preparation of the input for the first alteration level, are discussed, following the order of appearance in the method as presented in Figure 3.3 and Figure 3.4.

4.2 The vehicle type selector (Module 1)

A primary choice in a vehicle design process concerns the type of vehicle to be developed. This choice implies several constrains that have to be taken into account in generating the safety concept. Thus, the corresponding geometric information is required before suitable deformation modes for specific crash configurations can be established. In Module 1, i.e.: "The vehicle type selector / definition environment" the selected vehicle type is laid down as input for Module 3 of the crashworthiness integration method. For this purpose standard vehicle configurations can be defined by specification of the following characteristics:

1: the vehicle class (one of the following classes)[1]

city car
compact car / mid size car / limousine (dependant on wheel base)
station wagon
Multi Purpose Vehicle (MPV)
van
pick up vehicle

[1]Additional vehicle classes, based on the range here presented, can be defined by the user. An alternative wheel configuration for instance can require the definition of an extra vehicle class.

coupé
cross-country vehicle
cabriolet

2: the seats (a combination of the following seat variants)[2]
seat suitable for male/female (size 5, 50, 95%)[3] occupants
seat suitable for children
seat suitable for physically handicapped persons

3: the luggage capacity[4] (specified by the following three variables)
volume
box size[5]
mass

4: the doors (a combination of the following door variants)
occupant entrance
luggage entrance

5: the traction system (one or both of the following variants)
front wheel drive
rear wheel drive

6: engine location
front engine
mid engine
rear engine

7: gearbox location
front
mid
rear

The lists of specifications for each of the vehicle characteristics is based on common road vehicles. It is not intended to give a complete overview of all conceivable vehicle types.

[2]Special extra constraints as for instance space for hand luggage or a domestic animal inside the occupant compartment can be added by the user.

[3]The size of occupants is specified conform the method used to indicate the size of Hybrid crash dummies. (for example: a so called '5 percentile female occupant' is larger than 5% of the female population)

[4]Selecting the luggage capacity is necessary for instance to distinguish a luxury station wagon from a delivery station wagon. Two variable sets can be specified in case of a flexible luggage compartment, i.e. adjustable rear seats.

[5]The box size indicates the dimensions of a rigid element the vehicle can be loaded with as luggage. Volume, shape and entrance of the luggage compartment are constrained with this specification.

However, the methodology can always be extended with additional vehicle types. With the characteristics a first geometric specification of the vehicle is generated.

The representation technique must prevent the unintended introduction of extra geometric constrains due to a premature sketch of a structural shape. Such extra geometric constrains after all needlessly limit the solution space for the designer. In the primary development stage of a vehicle, were this specification is made, the precise shape and dimensions of structural elements are often not yet set. Hence, only the location of structural elements must be notated. Spatial points have no dimensions but only mark a location, so the geometry is represented by characteristic spatial vehicle points. An example of such a spatial point vehicle model for the vehicle characteristics specified in Table 4.1 is given in Figure 4.1. To keep the illustration clear, only occupant points corresponding with the driver—and not those corresponding with passengers—are reflected. Every point that is introduced adds information to the design.

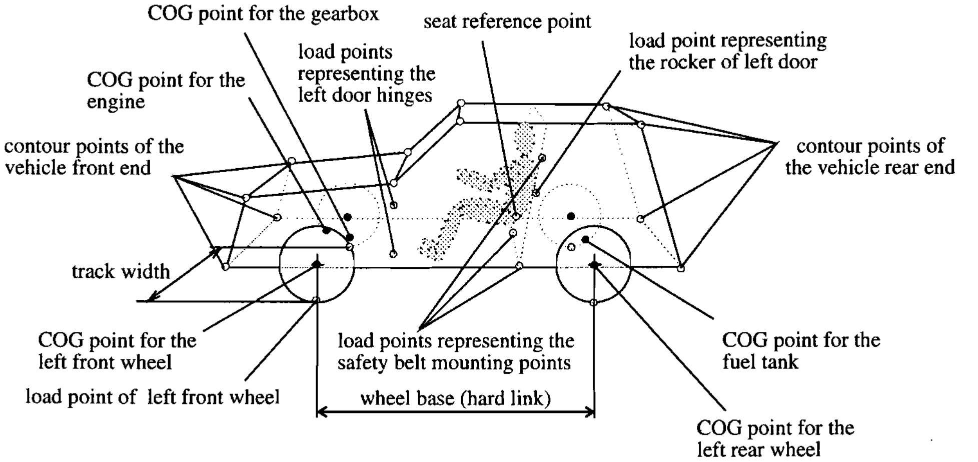


Figure 4.1 A first geometric specification of the vehicle geometry with spatial points.

The following three point classes are used:

1- "Centre of gravity(COG) points" representing the positions of the centres of gravity of vehicle components
2- "Contour points" showing the macroscopic shape of the vehicle structure, i.e. the location of the vehicle's structural elements
3- "Load points" representing the locations of areas transferring mechanical load.

characteristic	selection	value		
vehicle class	city car			
	compact, mid size car/limousine	wheel base (meters) 2.6		
	station wagon			
	multi purpose vehicle			
	van			
	pick up vehicle			
	coupé			
	cross-country vehicle			
	cabriolet			
seats	for female occupant	number: 2	size: 50 %	
	for male occupant	number: 2	size: 95 %	
	for child occupant			
	for physically handicapped persons			
luggage capacity	flexible	volume (liters)	box: h*b*l (meters)	mass (kg)
	with rear occupants	200	0.4*0.8*0.4	120
	without rear occupants	900	0.6*0.8*0.8	220
	fixed			
doors	occupant entrance	number: 2		
	luggage entrance	number: 1		
traction system	front wheel drive			
	rear wheel drive			
engine location	front engine			
	mid engine			
	rear engine			
gearbox location	front			
	mid			
	rear			

Table 4.1 Example of a vehicle type specification.

In Module 1, no structural connections are defined between the spatial vehicle points. The information on the vehicle shape is limited to the prescribed relative position of the points, the so-called 'geometric links'. If the undeformed relative position for a point pair is fixed, a so called “hard link” is applied. A design variable, indicated by a flexible relative position for a point pair, imposes a so called “soft link”. Besides with these geometric links, centre of gravity points are associated with inertia information, based on experience obtained in previous design projects.

Additional points to indicate non-standard components, important for the required crash behaviour, can be introduced in Module 1 as well. These vehicle components are identifiable by their relatively high weight, rigidity and/or large size. The battery pack of an electrically propelled vehicle, for instance, is such a special component. It largely contributes to the vehicle weight and besides, damaging of the battery pack in a crash is hazardous.

4.3 The macroscopic safety requirement selector (Module 2)

Besides the vehicle type specification, the passive safety requirements that must be fulfilled by the new vehicle have to be specified before suitable deformation modes can be set up. Requirements on a vehicle's passive safety consist of a prescribed loading procedure and a limit not to be exceeded by the response of the vehicle structure or occupant substitutes, the so-called 'load response restriction'. To enable the check of the response of a vehicle structure on prescribed load cases, spatial points must be added to the vehicle geometry specification. These additional vehicle points are specified within Module 2 called: "The macroscopic safety requirement selector / definition environment", as described in the present section.

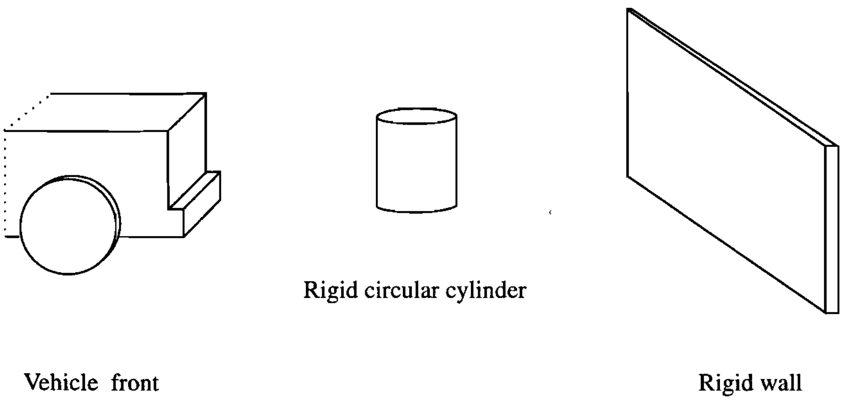


Figure 4.2 Examples of physical loading devices.

All current legal requirements on passive safety, are based upon the use of crash or crush tests. This implies that loads are always introduced via physical loading devices (see Figure 4.2). Hence, in the design methodology a numerical representation of these load introducing devices must be available. These numerical representations contain spatial points that must be positioned relative to spatial vehicle points conform the test regulations. Load points are added to the spatial vehicle model to transfer the imposed load into the vehicle structure (see Figure 4.3). Notice that a first crash related design decision is made by choosing the location of load transferring areas represented by the load points. The location of these points is not fixed relative to the loading device representing spatial points but can, as a design variable, be chosen within the limits of the contact face of the loading device. Load points for the transfer of frontal crash loads can for example be

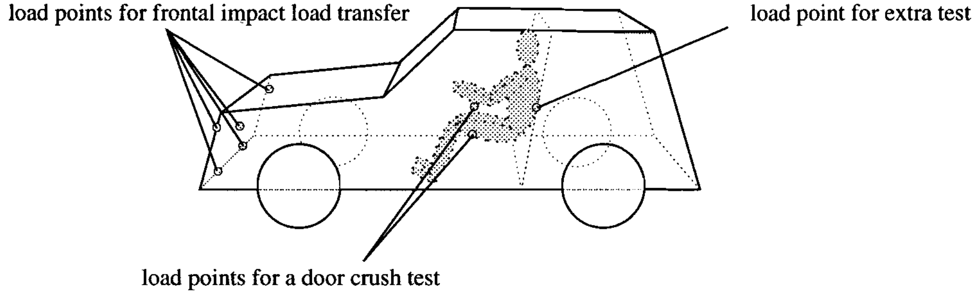


Figure 4.3 Examples of crash requirement specific spatial load points.

chosen to coincide with the end points of planned longitudinal members.

The use of numerical crash analysis enables the specification of additional safety requirements, based on theoretical load situations, without the necessity for the definition of a physical loading device. After all, a load can simply be imposed upon spatial points. In this situation only load points on the vehicle geometry have to be defined. These load points can then be actuated by imposing a nodal velocity or force in numerical crash simulations. This type of requirements is specially useful to compare vehicle concepts in an early stage because no data on contact surfaces is needed to transfer the impact loads.

The load condition depends not only on the load transfer area but on the load direction as well (see Figure 4.4). The load direction in a crash test depends upon the track for the loading device and its inertia properties. The properties of the track are represented in the model by boundary conditions for the spatial points of the loading device. The inertia properties are concentrated in the COG point of the loading device model.

The development of the vehicle load originates from a specified velocity time function or a prescribed initial velocity that provides a specific amount of kinetic energy for the loading device.

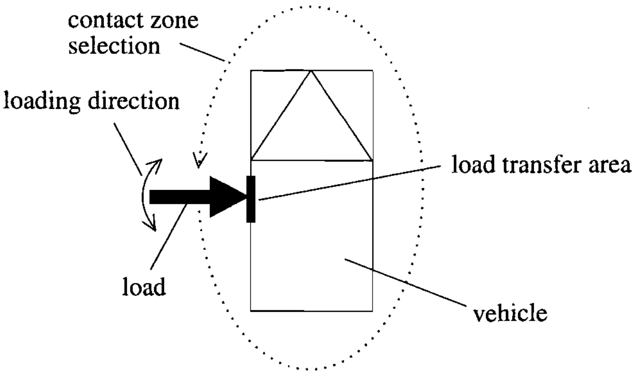


Figure 4.4 The crash load direction and the load transfer area.

Two forms of load response restrictions occur in passive safety requirements. One form is related to the loading device behaviour. For this loading device related response restriction form, the force development or energy transfer is checked on restrictions stated in the requirement. Hence, no extra spatial points in the geometric model are necessary for this response restriction form. The other form of a load response restriction that occurs is related to the vehicle structure behaviour at some distance from the direct contact area during the test. For the latter response restriction form the availability of additional vehicle contour points is required. Components intruding the vehicle interior are a threat to the occupants. Hence, intrusion levels are often restricted. An example is the limitation of the maximum steering wheel displacement in case of a frontal crash load according to the American standard FMVSS204 or its European counterpart, the EEC74/297. As a consequence, a point is required representing the steering wheel position in the spatial model to check the fulfilment of these standards (see Figure 4.5). Studying side impacts the intrusion profile of for instance the B-pillar near the occupant must be watched. Thus, contour points of the B-pillar are required in the model.

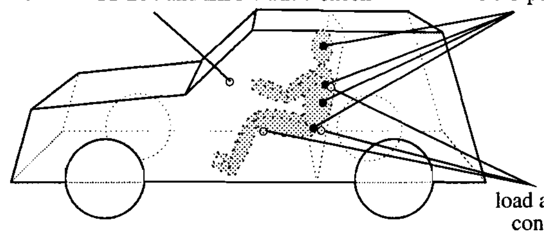


Figure 4.5 *Secondary load points and dummy related points.*

A dummy is used as a passive loading device to load the vehicle interior in several crash test configurations. The dummy is called a *passive* loading device while it is *actuated by* the vehicle structure, representing the secondary impact in a crash, and not the other way around. Spatial points to introduce the inertia properties and contours of dummies are added to the geometric model. The vehicle geometry describing spatial point model is expanded with secondary load points to support flat surfaces that represent dummy contact areas (see Figure 4.5). Dummy related response restrictions are in fact loading device related restrictions. Hence, no extra spatial vehicle points are necessary to measure the dummy response.

The requirements can now be specified as restrictions for the movement of and the force through the spatial points during specific load situations. Possible restrictions are maxima and minima for filtered or raw point movement related data such as a maximum HPC for the centre of gravity point of a dummy head (see Equation 2.3) and a minimum contact reaction force for a given deformation magnitude.

Two movement restriction types exist. One type limits absolute acceleration levels. To express the severity of occupant loads for instance, accelerations empirically interrelated with injuries levels are used. Thus, injury risks are estimated without actually determining the local physical loads. For the US-SID this is the only restriction type stated in the FMVSS 214 (see Table 2.2). This restriction type is applied to single points in the spatial point model.

The other movement restriction type limits relative displacements or velocities, and hence applies to point pairs in the spatial point model of a vehicle. The reduction of the size of the occupant compartment for instance, is the result of intruding components such as the B-pillar in a side impact. By monitoring the lateral movement of a point on the B-pillar compared to for instance the movement of a safety belt buckle mounting point, the magnitude of this intrusion can be compared with restrictions of this type. Another example concerns the door intrusion test according to FMVSS 214 as described in Chapter

2. In the numerical simulation of this test, the displacement of the COG-point of the loading device is measured relative to the spatial vehicle points that represent the points at which the vehicle body is tied on the test rig.

4.4 The deformation mode designer (Module 3)

Within the boundaries of the law of conservation of momentum a variety of deformation modes as response to specific load situations are conceivable for our spatial point vehicle model (Landheer, 1993). Equation 4.1 and 4.2 express the law of conservation of momentum for a spatial point model including the loading device with n points:

$$\text{lineair: } \sum_{i=1}^{n} m_i \cdot \vec{v}_i = \text{constant} \tag{4.1}$$

$$\text{angular: } \sum_{i=1}^{n} m_i \cdot \vec{r}_i \times \vec{v}_i = \text{constant} \tag{4.2}$$

where m_i and $\vec{v}_i$ are the mass and velocity of point i respectively, $\vec{r}_i$ is the position relative to a reference point.
In practice no considerable vehicle rotations result from the crash configurations discussed in Chapter 2 during Stage 1 and Stage 2 of the crash (see page 28). Transfer of momentum to the ground surface by means of friction is not taken into account in the equations. This change of momentum J (= impulse) can be expressed as:

$$J = M \cdot g \cdot \mu \cdot t \tag{4.3}$$

where M is the mass of the skidding vehicle
g is the acceleration of gravity
μ is the coefficient of friction to the ground
t is the period that the friction force acts on the vehicle.
It can be calculated that in the first 60 milliseconds of a crash, i.e. Stage 1 and Stage 2 of a side impact (see page 28), in which the injuries are originated a deviation of about four per cent in momentum occurs, which can be neglected.

In the current module, preferable deformation modes (i.e. crash load responses) for the load cases specified in Module 2, must be designed. For this purpose, the effect of the occurrence of specific deformation modes on the safety performance must be compared. A deformation mode of a vehicle can be characterised by the velocities of its spatial points during the crash (Landheer, 1996). Thus, a comparison is achieved by prescribing differing velocities for groups of points in numerical simulations to represent imposed rigid relative movements of parts of the vehicle. This technique of imposing a velocity function to investigate the accompanied dummy response is not new. In sled tests to validate seat belt systems for instance, a so-called crash pulse (deceleration-time function) is imposed on a rigid occupant compartment. However, only one rigid link (the sled) is imposed a movement. Besides that, in general the crash-pulse is a function established in experiments with an existing structural concept. Here, this concept has to be designed yet. Numerical simulations of this kind are performed as well. For instance in side impact airbag development, velocity-time functions are applied to reduce calculation time (Pilhall, 1994). Measuring points to check the fulfilment of response restrictions are available as spatial points in the model. To predict accompanying occupant loads, a contact that transfers forces from the spatial point groups to a numerical dummy is required. Because no detailed geometric information on the shape of contact surfaces is available yet, flat theoretical surfaces that contain a specified resistance against intrusion of dummy parts are attached to spatial vehicle points here.

If a specific deformation mode shows acceptable dummy loads and the response restrictions for the other measuring points are fulfilled, this mode is further investigated. A vehicle structure must be designed now that deforms conform the planned deformation mode as a result of the crash load. Mechanical properties of connections between the groups of points must be chosen qualitatively to realise the relative displacements of the vehicle points as a response to actuation of the load points, that form the interface between vehicle and loading device, by a prescribed velocity-time function. This qualitative characterisation of mechanical properties consists of the specification of degrees of freedom, i.e. allowed deformation in specific directions, and rigidity in other loading directions. A sound design should have only one degree of (deformation)freedom for a specific load situation, controlled by a total deformation resistance. This implies that the movement of all the point groups in the kinematic model is set, provided that the movement of two point groups is prescribed. Hence, to validate a kinematic model, velocities for two point groups must be prescribed. Selected deformation modes for specific load situations are programmed in the structure by application of this procedure.

The kinematic model obtained in this manner does not yet dissipate any kinetic energy. As a following step, in Module 6, a deformation resistance must be allocated to the deformable connections in the model.

4.5 The load condition analyzer (Module 4)

The deformation processes specified in Module 3 are accompanied by specific loads inside the structure. When these loads act upon the vehicle structure that has to be developed yet, it should not collapse. Deformations should be limited to the structural areas that are predestined to deform. The part of these loads that is originated by inertia effects is analyzed in Module 4 called: "The load condition analyzer". The information that comes available on the structural loads is used in Module 5 to establish an appropriate structure type for the vehicle.

In Module 4, the load points—via which the crash load is transferred into the elements of the kinematic vehicle model—are subjected to a time-dependant velocity in numerical simulations. The resulting loads in the deforming joints of the kinematic model act upon the structural parts of the vehicle that are considered not to deform. The deformation resistance of areas of the structure planned to deform is not yet set and hence, the contribution of their resistance to the structural load is not taken into account. As a consequence, lower load levels, only caused by inertia, are established than those that will finally act upon the structure. The deviation of the load levels in the contact-zone, at the load points were the time-dependant velocity is imposed, follows from the acceleration of the COG-point of the loading device, calculated in the numerical simulations. However, over which distance into the vehicle structure this extra load effects the load balance, depends on the deformation resistance distribution that has to be designed yet!

4.6 The structure type selector (Module 5)

Now that a spatial point model has been developed, deformation modes set up and structural load conditions are analyzed, the generation of safety concepts is started. As already emphasized in Section 4.2, inertia properties are based on information of previous vehicle designs and represented in the spatial model, concentrated in points. However, the inertia properties of vehicle structure parts depend on the structure types applied for the parts. Hence, specification of the planned structure types in the vehicle, according to the insights gained on load levels and available space so far, is required to update the mass distribution.

Common structures for components or the complete vehicle body are for instance:
- a box structure of plate material
- a space frame of thin-walled prismatic members that performs the load transfer function separate from a vehicles' contour styling
- a diecasted solid frame.

All structure types can be realized in several material types.

Besides crashworthiness, other vehicle characteristics must be considered as well to select the structure type because they are often decisive for this selection. If for instance a modular vehicle is developed, a central backbone on which several modules are mounted can be a design condition. Such a design condition implies a structure type as for instance a chassis with a specific mass distribution over the vehicle, that must be reflected by the inertia of the points in the spatial vehicle model. Besides, the relatively rigid chassis can be exploited as force transfer path through the vehicle in a safety concept.

To establish an estimate for the inertia properties of the spatial points in the model, data should be gathered on existing vehicle designs with similar vehicle structure parts, as specified in the structure type selector.

4.7 The safety concept generation environment (Module 6)

The kinematic model obtained in Module 4 is not yet able to dissipate any kinetic energy. In the next step a deformation resistance must be allocated to the deformable connections in the model to make them dissipate energy. It must be noted that a high deformation resistance does eventually lead to a heavy structure, especially for the structural parts that are predetermined to behave as rigid links in a safety concept. Consequently, the inertia properties of the spatial points that are applied as input for the numerical crash simulations have to be adjusted if an unconventional mass distribution is required.[6] A high vehicle mass has a negative effect on the vehicles' fuel consumption and its crash compatibility with other vehicles. Therefore, a minimum deformation resistance for the deformable connections must be established, to make the safety concept fulfil the requirements. To validate allocated deformation resistance settings, only an initial velocity or a displacement function for the centre of gravity of the loading device must be prescribed in the numerical crash simulations. No other velocity time functions may be imposed now.

In summary, after a suitable deformation mode has been chosen in Module 3, mechanical properties are defined to realise the prescribed velocity functions during a crash. In other words, the imposed velocity functions are replaced by energy dissipating connections between the rigid spatial point groups. Numerical simulations to study the effect of the defined connections can now be put through by only specifying a movement for the loading device. In this way requirements for structural properties that realise vehicle safety concepts are specified without taking into account the actual shapes of the load transferring structural elements.

[6]Moreover, in the long run heavier vehicles will lead to heavier loading devices, while the weight of loading devices in legal requirements is intended to be statistically related to the vehicles on the road.

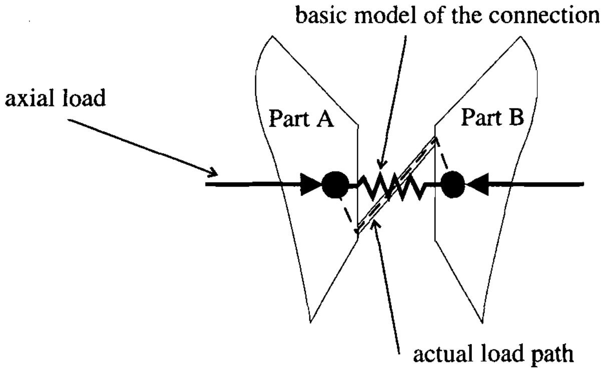


Figure 4.6 Load situation that causes a bending moment in the actual load path.

Since a single vehicle structure must fulfil the deformation programmes for all likely crash configurations, connections between points must act as rigid elements in one configuration whereas in another configuration, a certain deformation behaviour is required for that same connection. Tuning the mechanical properties of connections is necessary to achieve this behaviour. The detail level involved for the mechanical properties only makes sense if the geometric description takes into account realistic force paths that match a packaging plan. For instance, when an axial load is applied to a connection that represents a curved force path in reality, a bending moment must arise in the model (Figure 4.6). This is achieved in Module 8 by application of so-called beam elements that describe a structural shape in between the spatial points. However, first the spatial point model of the vehicle is expanded with packaging information in Module 7.

4.8 The packaging input module (Module 7)

A clear outline of the available space for the load transferring structural parts in the vehicle that embody the structural properties found in Module 6 must be established in Module 7, the so-called "The packaging input module". In the spatial point model, vehicle components are represented by points. Now, the main dimensions of these components such as the drive line and the engine have to be added to the model. Besides, space has to be reserved for occupants and luggage. The volume in between the vehicle exterior contour points (see Figure 4.1) that remains after subtraction of the occupant space, the luggage space and the volume of the components that form no load transferring part of the vehicle body, is available for load paths through the vehicle.

Figure 4.7 Example of a simple parametric solid model to specify the contour of a seat.

Although no detailed geometric description of the vehicle components is available in this stage of the vehicle design process, the main dimensions can already be extrapolated from the components applied in previous vehicle designs. To specify component contours, a simple parametric solid model representation (see Figure 4.7) is used to extend the spatial point model. If the new design approach is implemented, a data base with parametric models of standard components can be generated. By adaption of the parameters of these models according to the new vehicle requirements, the space occupied by the components is described quickly. A solid model representing the box size specified as part of the luggage capacity characterisation (see Section 4.2) has to be taken into account as well. After these solid models are positioned in the spatial point model of the vehicle, the remaining interspace available for the load transferring vehicle structure plus the occupants is established (Figure 4.8).

It is chosen not to reserve a fixed space for the occupants while their posture is flexible to some extent. Besides, the entrance to the vehicle interior must be considered as well. To take into account the volume claim of the occupants, dummies of the appropriate size[7] with an adjustable posture are applied (see Figure 4.9). Step-by-step simulating the entrance of the occupants by manipulation of the posture of the dummies will show possible space conflicts with the load transferring structure. These interference checks must take place in the next module called: "The force path definition environment". Besides that, a comfort aspect in the form of a certain amount of space around each of the occupants has to be accounted for.

[7]See the seats characterization in Section 4.2.

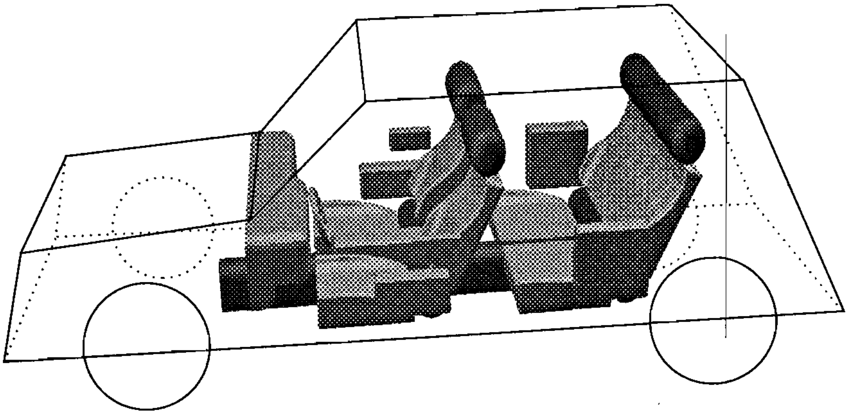


Figure 4.8 Example of a parametric packaging model for a vehicle interior.

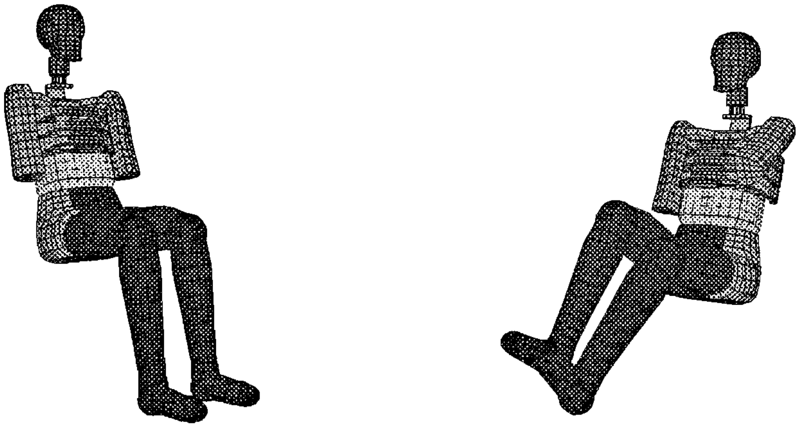


Figure 4.9 Dummy in two differing postures.

4.9 The force path definition environment (Module 8)

In this module the safety concept from Module 6 is transformed into a safety structure layout compatible with the packaging plan. Within the space available for load transferring structural parts as described in Module 7, now load paths are defined between the spatial vehicle points. Straight sections are applied as connective elements. This implies that extra points must be introduced for the representation of a curved connection (Figure 4.10).

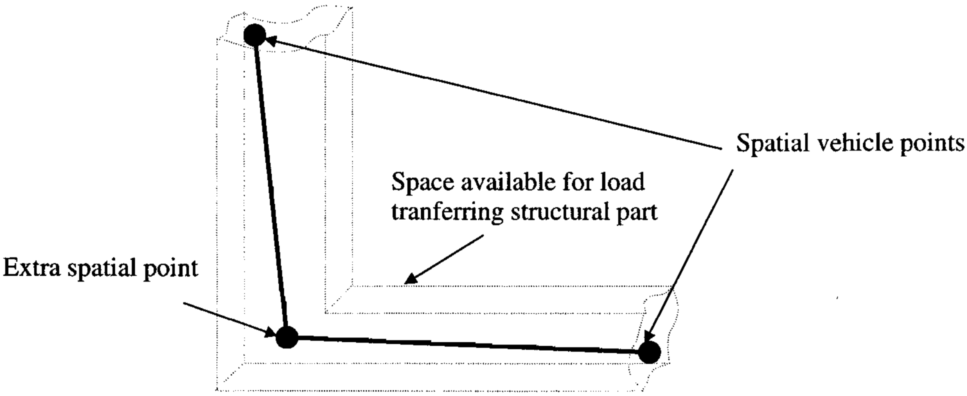


Figure 4.10 A curved load path between two spatial vehicle points.

As indicated in the safety concept, for specific parts of the structure only minor deformations are allowed under given crash conditions, i.e. they may not collapse as an effect of local buckling. These parts are represented as a composition of so-called beam elements in the numerical model. Beam elements represent simple prismatic shaped straight sections. Rectangular or circular cross-sections that fit in the space available for the load transferring structure, can now be specified for these beam elements.

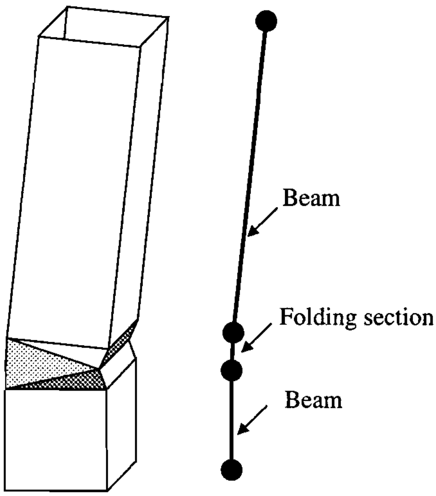


Figure 4.11 A folding section defined in between two beams to represent local buckling.

The relative movement of the aforementioned parts is constrained by intermediate connective elements. These connective elements are planned to undergo large deformations. Inhomogeneous deformations as a result of local buckling effects within these connections can be expected if sheet metal structural elements are applied. Hence, simple beam elements are not adequate to model these structural sections. By adding a so-called folding section to a composition of beam elements, a local buckling mode can be implemented (Figure 4.11). This method of modelling uneven deforming parts within a structure is already proposed for the development of longitudinal members by Sielaff (Sielaff, 1990). The position of a folding section predestines the location of a potential fold. As a consequence, it is vital for the reliability of the numerical crash simulations to ensure that this location corresponds with the natural buckling location. In the next module special attention is given to this aspect. The mechanical properties that have to be assigned to the folding sections must be considered as design variables. As upper limit for these design variables, the collapse properties of the adjoining prismatic sections have to be considered.

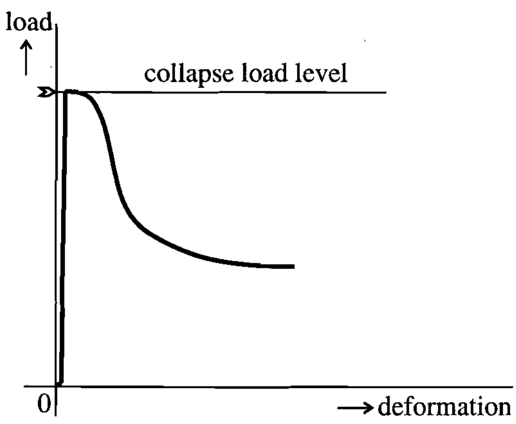


Figure 4.12 Characteristic typical for bending and compression of prismatic sections applied in vehicle structures.

The common deformation characteristic of thin walled prismatic sections always show a load climax that coincides with the initiation of a local folding process (Figure 4.12). This load climax can be established numerically for a given cross-section (Mahmood, 1986; Mahmood, 1988). If the deformation load levels of the folding sections are kept below the collapse load levels of the adjoining sections the resulting numerical model will show a realistic deformation behaviour.

In Figure 4.13 an example is given of an area were folding sections must be applied between two parts, in addition to the beams. The plastic hinge is represented via one folding section. The compressive member is composed of a series connection of beams and folding sections.

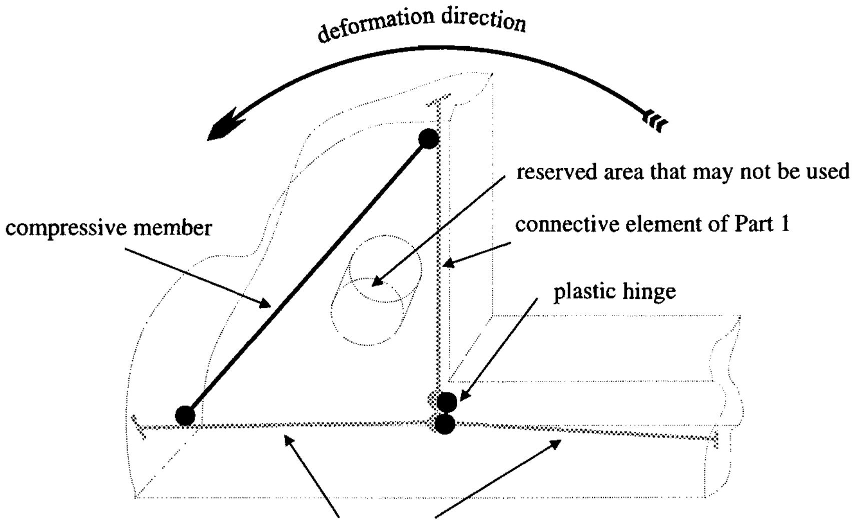


Figure 4.13 Example of a compressive member and a plastic hinge predestined to undergo large deformations.

The load path lay-out, set up according to the procedure discussed above, can now be validated against the desired behaviour as it was laid down in the kinematic model in Module 3. The maximum force and moment of bending levels for the individual beams that occur in the crash simulations have to be registered as input for Module 9.

4.10 The coarse dimension estimator (Module 9)

The lay-out of the crash load transferring vehicle structure is defined as a composition of spatially orientated beams. Each beam is assigned a rectangular cross-section that provides an indication of the collapse load levels, feasible within the constraints of the packaging plan. Based on these collapse load levels, the mechanical properties of the so-called folding sections are set. The actual maximum load levels within these beams, found in the crash calculations of Module 8, are now used to design a refined section shape, fitting in the styling plan. The sections must be designed to provide a collapse resistance that exceeds the calculated load levels.

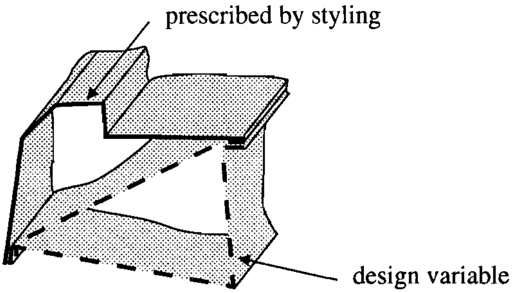


Figure 4.14 Cross-section shape with integrated styling aspects.

The styling data of the vehicle's exterior is available during this phase of the design process. The cross-section shape to be designed of beams, defined between spatial contour points that mark the outline of the vehicle, is partially restricted by this styling plan as can be seen in Figure 4.14. Suitable settings for the part of the cross-section shape that is not yet fixed have to produce an adequate collapse resistance of the beams. A computer program called Crash Cad (Impact Design Inc., 1995), can be applied to calculate the specific collapse loads that are associated with a defined cross-section shape.

Besides the strength requirements for crashworthiness, body stiffness characteristics are of concern as well, because they are of influence on the load levels occurring within the beams. The elastic deformations of structural sections are represented in the numerical model by bending of the beams. Hence, moments of bending inertia of these beams have to be updated to correspond with the refined section shape definition generated in the present module.

With the refined beam model crash calculations for the crash configuration range specified in Module 2, have to be put through. The post-processor used to visualise the deformations of the beam model should highlight eventual locations in which the collapse load is exceeded and hence, adaption of the beam cross-section design is necessary. In accordance with the approach of Mahmood (Mahmood, 1988), a proportional dependency of the crush initiation on the bending and compression load components is assumed. This implies that the total of the specific loads (bending moment around the principle axes M_s and M_t and the compression force F_r) weighted with the corresponding collapse load levels (M_{coll-s}, M_{coll-t} and F_{coll-r}) has to be calculated by this post-processor for every beam and checked on exceeding the value of 1, as expressed in the following equation:

$$\left| \frac{M_s}{M_{coll-s}} \right| + \left| \frac{M_t}{M_{coll-t}} \right| + \left| \frac{F_r}{F_{coll-r}} \right| < 1 \qquad (4.4)$$

If the condition expressed in Equation 4.4 is not fulfilled one or more of the collapse load levels must be increased by adjusting the cross-section of the beam in question.

4.11 The parallel subsystem design module (Module 10)

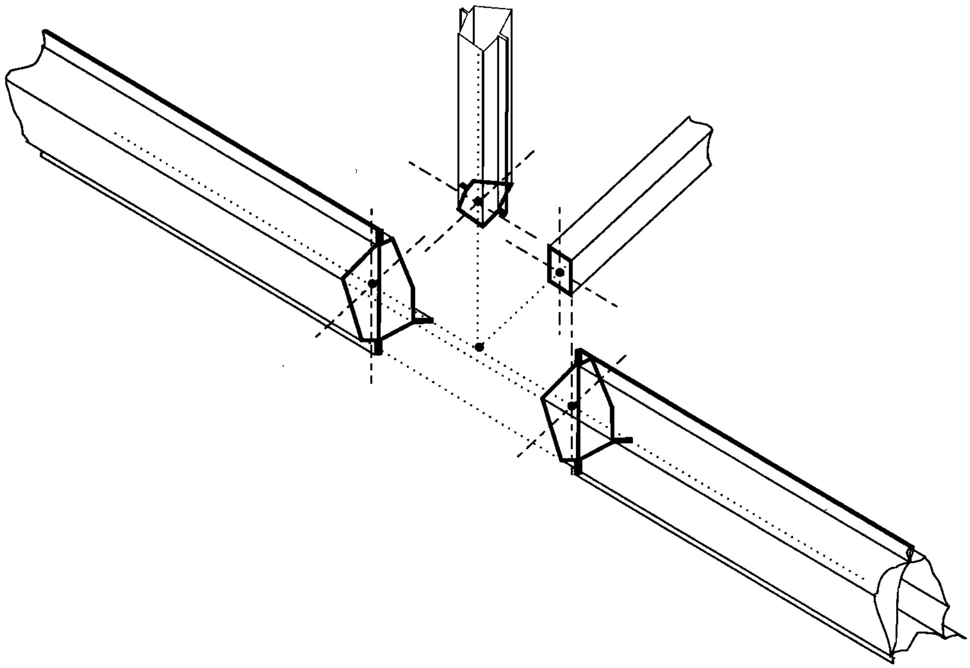


Figure 4.15 Geometric constraints imposed by the prismatic sections.

The coarse shape of the prismatic sections that are predestined to undergo only minor deformations is set in the preceding module. For the intermediate parts of the structure, deformation characteristics have been set. The cross-section shape and orientation of the adjoining prismatic sections act as geometric constraints for these parts as shown in Figure 4.15. Module 10 is meant to validate separate designs for the intermediate parts on their crash performance. For this purpose a so-called hybrid modelling technique is

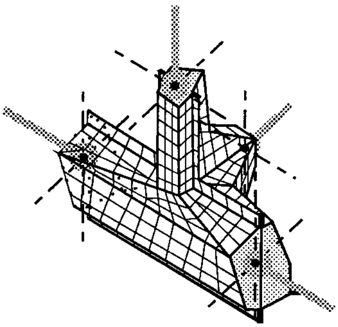


Figure 4.16 Example of a shell mesh representation of a part-design.

applied. By implementing a shell mesh representation as shown in Figure 4.16 of such a part-design in the beam model discussed in Module 9, the effect of the behaviour of the part on the crash performance of the car can be established in a direct way, i.e.: in contrast with experimental component tests the full vehicle model enables the validation of the design variant of the specific part in interaction with the rest of the structure. However, now that their shape is defined, experimental component tests could be performed to validate the numerical representation of these parts. In these experiments, the calculated deformations should be imposed on the part, to establish its actual deformation resistance.

As interface between the beam elements and the shell mesh, rigid bodies have to be defined as illustrated in Figure 4.17. Considering the accuracy of the crash simulation results, rigid bodies are allowed here because the sections represented by beams are designed to undergo only even deformations over their full length due to the crash loads, maintaining their specific cross-section shape.

In case a prismatic section is directly loaded, distortions in a cross-section, originated by this load can occur. Obviously, these distortions do not show in a beam model representation. Hence, these specific sections have to be modelled via a shell mesh as well. This prevents the overlook of these distortions that can result in unforeseen collapse of the prismatic section. Via a contact definition, detected intrusions of structural elements are followed by a reaction force resulting in the local deformation of the shell-model. Notice that shell-modelling of these sections is only necessary in crash simulations that represent the specified direct loading situation.

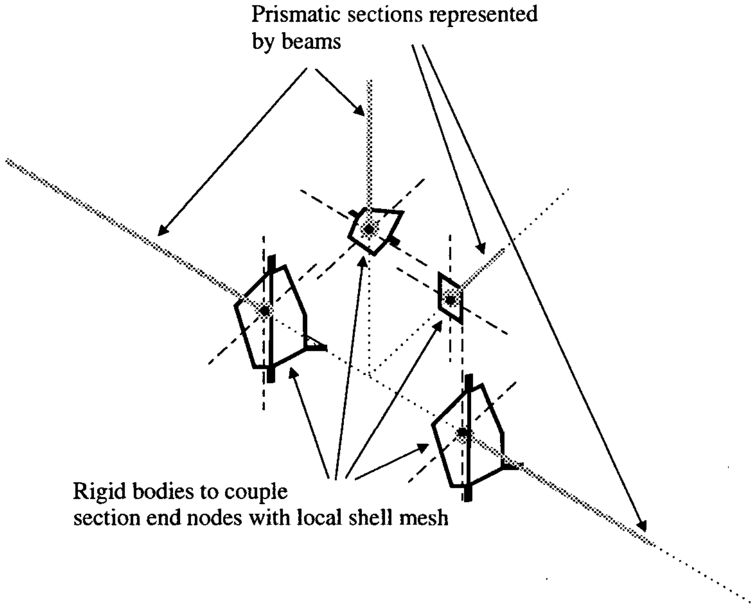


Figure 4.17 Rigid bodies to couple a local shell-mesh to the adjoining beams.

4.12 The Detailed tuning module (Module 11)

In this final module, detailed tuning of the structural elements has to take place. For this purpose all subsystems of the vehicle structure that do deform under the specific simulated crash configuration have to be represented as detailed shell mesh zones in the beam model. The meshes developed in the preceding Module 10 can be used here. For side impact crashworthiness simulations an example of such a hybrid model is given in Figure 4.18. In this model the exterior body panels are implemented as well to incorporate their positive effect on the deformation resistance. Besides that, the load transfer from loading device to the body structure takes place via these body panels and hence, the interaction of these panels with the structure can be validated. It implies that the contact-face of the loading device must be modelled with an accompanying detail level.

A major advantage of the application of the hybrid model is the reduced calculation time compared to simulations with a full shell model of the vehicle. A difference exists with the approach of substituting the vehicle structure opposite to the crash load area by a rigid body. That means, the effect of the introduction of an excessive deformation resistance in the deforming loaded part of the vehicle via a design adaption on the collapse margin of the rest of the vehicle structure is monitored easily as marked in Section 4.10.

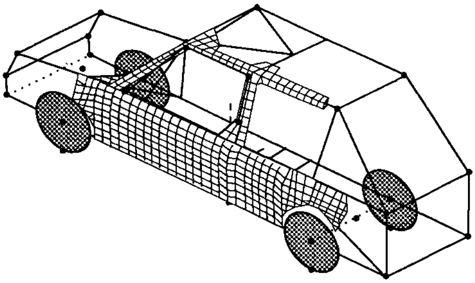


Figure 4.18 Example of a hybrid vehicle model with beam and shell elements.

After this final module has been put through, the same geometric detail level of the vehicle will be available as that required to generate a complete FEM model. However, the current vehicle design will incorporate a set of integrated safety concepts for specific crash configurations.

Chapter 5

Evaluation of the new design method

5.1 Introduction

The proposed modules presented in the previous chapter, are validated on their practicability in this chapter. The first step to a practical realization of the modules will be discussed here. For this first step, only commercially available software in common use within the automotive industry is applied. This enables a quick implementation of the method, partially or integrally, in existing vehicle design processes later on. The performance of the accomplished total system, composed of the modules, is evaluated by means of the engineering requirements as stated in Chapter 3.

5.2 The criteria for the evaluation

The engineering requirements stated in Chapter 3 have been drawn up without taking into consideration any modular structure in the design method. Because each engineering requirement refers only to a specific function in the design process, it is only of concern for a specific subset of the modules that are related to the function. Consequently, it is chosen to evaluate each module individually for those requirements that are related to its function. If for the fulfilment of a requirement more than one module within one design phase (Figure 3.3 and Figure 3.4) is of concern this is indicated. In Table 5.1, an overview is given of the modules that are related to each requirement.

label	engineering requirement	modules involved
e.r.1	point modelling must be available	1;6
e.r.2	wire frame modelling must be available	1;8
e.r.3	surface modelling must be available	1;7;8;10;11
e.r.4	solid modelling must be available	1;7
e.r.5	geometry input must be graphically displayed	1;8;9;10;11
e.r.6	geometry input must be graphically displayed on-line	1;8;9;10;11
e.r.7	time required for first geometric model generation	1
e.r.8	time required to adapt a geometric model	1;8
e.r.9	number of data conversions to create an analysis model	1
e.r.10	time required to generate an analysis model	1;2;3;8;10;11
e.r.11	a concentrated load definition option must be available	2
e.r.12	a spread load definition option must be available	3
e.r.13	definition of loads as function of time must be possible	2, 3
e.r.14	an inertia load definition option must be available	2
e.r.15	number of detail levels	3
e.r.16	time required to adapt an analysis model	3;8;10
e.r.17	number of data conversions to translate an analysis model into a geometric model	3
e.r.18	time to convert an analysis model into a geometric model	3
e.r.19	deformation magnitude accuracy in the occupant contact area	3;11
e.r.20	time required for load card generation	2
e.r.21	time required for load card adaption	6
e.r.22	elapsed time for a crash calculation	3;11
e.r.23	a deformation versus time output plot option must be available	3
e.r.24	an acceleration versus time output plot option must be available	3
e.r.25	a geometry state output plot option must be available	3
e.r.26	a deformation animation option must be available	3
e.r.27	number of structural points per occupied volume in the final output	8;11
e.r.28	number of colors in plots	3
e.r.29	hidden line removal in geometry plots must be available	3
e.r.30	a cross section state plot option must be available	3
e.r.31	a perspective geometry plot option must be available	3
e.r.32	time required to post process data	3

Table 5.1 Overview of the modules that are related to each engineering requirement.

5.3 Evaluation of Module 1: The vehicle type selector

The vehicle type selector module concept is based on a database containing spatial point models of reference vehicles for each vehicle type that can be selected in the design system. A parametric description of the relative point positions enables a quick scaling of the standard models towards the intended vehicle main dimensions by specifying parameters such as the wheel base. An analogy with the basic model generation method in

AURORA[8] (Hänschke, 1990; Heinke, 1990) exists although the accent in that method is placed more on structural appearance but less on the mechanical structure attachment locations in relation to the vehicle exterior contours. Besides that difference in accent, COG points definitions of vehicle components representing the concerning inertia force origination locations during a crash, as indicated in Figure 4.1, are missing in the AURORA basic model generation method whereas these points are vital to perform proper crash calculations. Nevertheless, users of comparable design systems could deploy an edited version covering the aforementioned issue to preserve the access to their existing source of basic vehicle dimensions.

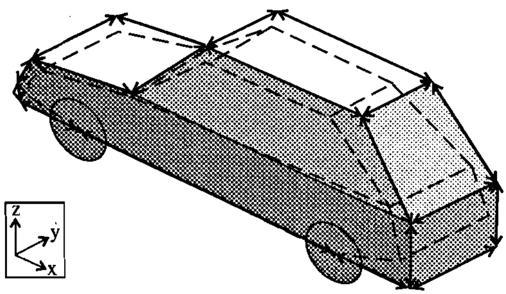


Figure 5.1 Example of the appearance of a basic parametric sight model.

To perform the validation of Module 1, as model of a reference vehicle, a CAD-representation defined with the CAD-software package Unigraphics is chosen. By applying parametric relations in this numerical sight model, dependencies in the vehicle main geometry dimensions can be expressed. Sizing up or down certain aspects of the vehicle geometry then requires only a mere change in the values of the parameters concerned (Figure 5.1).

The GFEM-unit in Unigraphics is applied to mark points in the model as spatial points for the analysis model. A serious disadvantage of Unigraphics version 10.4 is that the location of these points can not be linked with the parameter definitions in the CAD-module[9]. Consequently, the spatial points have to be re-positioned manually if parameter settings are altered. This is a time consuming operation. Via a Nastran-format output-file containing the coordinates of the spatial points, these base vehicle geometry dependant

[8]AURORA stands for: 'Automobiltechnisches, anwendungsorientiertes Entwurfssystem zur Optimierung der rechnergestuetzten Auslegung', a vehicle design system developed at the Berlin University of Technology

[9]In contrast with Unigraphics, other CAD-software packages such as I-DEAS from 'Structural Dynamics Research Corporation' enable the definition of parametric links for analysis models.

coordinates are entered into the numerical crash simulation environment of the Pam Systems International (PSI) crash analysis software package. Within Generis, the pre-processor of PSI, inertia properties are assigned to the spatial points. This completes the output for Module 1.

The choice to use Unigraphics as geometry representation tool implies that the following engineering requirements:

☑ e.r.1 point modelling must be available
☑ e.r.2 wire frame modelling must be available
☑ e.r.3 surface modelling must be available
☑ e.r.4 solid modelling must be available
☑ e.r.5 geometry input must be graphically displayed
☑ e.r.6 geometry input must be graphically displayed on-line

are automatically fulfilled for Module 1.

Concerning the interfacing between Unigraphics and PSI software, the following requirement:

☐ e.r.9 number of data conversions to create an analysis model [target < 2]

is not fulfilled because of the necessity of an intermediate Nastran data format in the conversion process. However, no additional input is required for the conversion steps and hence, this poses no serious problem.

If a primary spatial point model of a vehicle version that is available as parametric CAD-representation has to be generated, the correct settings for the parameters have to be defined. This turns out to be possible in less then two hours. However, the generation of a spatial point model of a complete new vehicle version requires more preparation-time. In particular establishing the inertia properties that have to be assigned to the spatial points in the analysis model appears to be time consuming. A suitable mass allocation can only be appointed to the discrete spatial points after basic assumptions are made on the structural composition of such a new vehicle. With the aid of data known from comparable structural constituents applied in other vehicles, such estimates can then be made. Consequently, engineering requirements:

⊠ e.r.7 time required for first geometric model generation [target < 2.5 hours]
⊠ e.r.10 time required to generate an analysis model[10] [target < 1 hour]

are fulfilled under the condition that the vehicle version to be developed is available as parametric CAD-representation already.

[10]Passing engineering requirement 10 for the here concerned detail-level depends not only on Module 1 but on Module 2 and Module 3 as well.

The adaptation of a primary spatial point model of a vehicle implies that either the locations of specific points have to be changed or extra points have to be added. This appears to be a matter of minutes in Unigraphics and hence, engineering requirement:

☑ e.r.8 time required to adapt a geometric model [target < 0.5 hour]

is fulfilled concerning Module 1.

5.4 Evaluation of Module 2: The macroscopic safety requirement selector

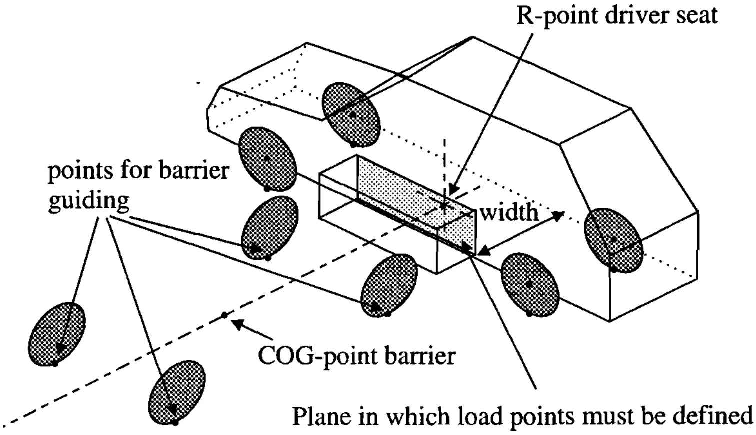


Figure 5.2 The European side impact loading device implemented in a layer of the Unigraphics parametric vehicle representation file.

The specification of a safety requirement as input for a vehicle design project calls for an unambiguous definition of the loading process and the accompanying restrictions to the vehicle response. These definitions obviously exist for legal requirements. An overview of legal requirements concerning side impact crashworthiness was already given in Section 2.4. Here, a representation of the loading device that can be combined with the spatial point model applied to represent the vehicle geometry in Module 1 is required. An easy way to ensure a proper orientation of the loading device model in relation to the vehicle model is to integrate the selectable loading devices in separate layers of the Unigraphics parametric model. Figure 5.2 gives an example for the European side impact loading device. The position of the barrier is coupled with the R-point location and the vehicle-body width.

For loading devices with a rigid contact face, a rigid body definition in the analysis model can be applied. However, if the loading-device-face is deformable, the relative intrusion of vehicle parts into the loading device implies a specific load distribution. Because points that have to be established yet as part of the design process are used to introduce crash loads into the vehicle model, the accompanying points on the loading-device-face with their specific deformation resistance must be defined in Module 3. Hence, for the latter loading devices, only the plane in which load-points must be defined can be appointed in the model. An example of such a plane is given in Figure 5.2.

Within the PSI numerical crash simulation environment, inertia loads as well as time dependant load functions—in the form of a prescribed force, moment or velocity—can be assigned to nodal points. Consequently, engineering requirements:

☑ e.r.11 a concentrated load definition option must be available
☑ e.r.13 definition of loads as a function of time must be possible
☑ e.r.14 an inertia load option must be available

are fulfilled. With the same function definition option, scalable force deformation relations can be defined. These scalable force deformation relations can be assigned to the load-point definitions in the loading-device-face later on. In most cases the force deformation relation is proportional to the surface size that is loaded. Only, if an atypical deformation mode for the loading-device is intended, extra attention must be given to its modelling. Hence, if the overall deformation resistance of the loading device part is available as a function, the analysis model of a deformable loading device can be completed quickly by selection of this function corrected with a scale-factor based on the surface size represented per load point.

Summarising, the action required to obtain a loading device definition in the analysis model consists of the following 5 steps:

1 - activating the specific layer in Unigraphics [< 1 minute],
2 - exporting the spatial points of the loading device along with the vehicle points as a Nastran-format output-file [requires no extra time],
3 - providing the loading-device points with the appropriate inertia properties and boundary conditions, selectable from a data-base [< 10 minutes],
4 - defining a rigid body of the points that represent the not deformable part of the loading-device [< 10 minutes],
5 - specifying the deformation resistance regarding the defined load points [< 30 minutes].

As a consequence, engineering requirement:

☑ e.r.10 time required to generate an analysis model [target < 1 hour]

is fulfilled.

To obtain an impact load, a relative movement of the loading-device model towards the vehicle model must be specified in a so-called 'Load Card', i.e. the addition of a line in the

PamCrash input-file. The numerical simulation of a crash load is obtained by the specification of an initial velocity for the COG-point of the loading device. The simulation of a crush load is obtained by the specification of a displacement-time or a load-time function for this COG-point. The preparation of either of these Load Cards takes in the order of a few minutes. Consequently, engineering requirement:

☑ e.r.20 time required for load card generation [target < 0.3 hour]

is fulfilled.

5.5 Evaluation of Module 3: The deformation mode designer

The design of a deformation mode implies that groups of the vehicle points defined in Module 1, that are planned to act as rigid links in this mode, must be specified. Within the PSI numerical crash simulation environment, these groups can be defined as rigid bodies. A velocity-time function can be assigned to points of the groups so that an unambiguous relative movement of the groups is prescribed, exactly in accordance with the intended mode. As a consequence, engineering requirement:

☑ e.r.13 definition of loads as function of time must be possible

is fulfilled.

In PamCrash a so-called 'Sliding Interface type 11' (body-to-plane contact model with user specified penetration response) can be applied to model the load transfer between moving point groups and an occupant model (ESI/PSI, 1994a). For this purpose, planes attached to point groups have to be defined in the PamCrash input-file. An interaction force as response to penetration of a plane by parts of the occupant model, as a function of penetration depth and volume, can then be calculated with interface type 11. This implies that engineering requirement:

☑ e.r.12 a spread load definition must be available

is fulfilled.

Beside that, in order to transfer loads to an occupant model, seat belts can be defined as beam-elements, attached to the safety belt mounting points shown in Figure 4.1. The shape and mechanical properties of occupant representing crash dummies in legal requirements is specified exactly in these requirements (see Section 2.4). This implies that a precise numerical model of these dummies can be defined. PSI supplies FEM-models of the EuroSID and the Hybrid3 dummy that are used here. Summarising, it can be concluded that all the following four detail levels:

- nodes (spatial vehicle points)
- beams (seat belt segments)
- planes (contact planes for Sliding Interface type 11)
- solids (occupant model)

can be applied in one analysis model. As a consequence, engineering requirement:

☑ e.r.15 number of detail levels [target 4]

is fulfilled. As an example, a primary numerical model to evaluate a specific side impact deformation mode is given in Figure 5.3 (Wijntuin, 1996).

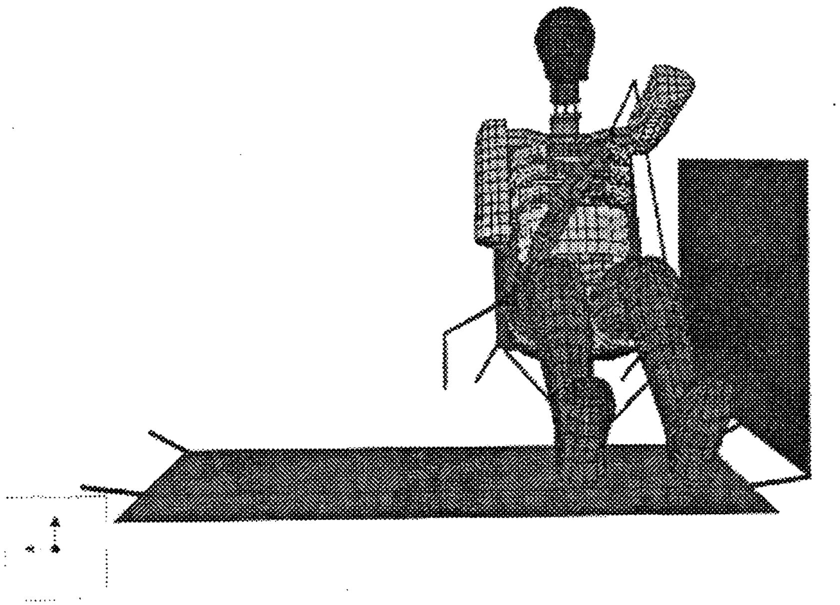


Figure 5.3 Example of a primary numerical model for the evaluation of a specific side impact deformation mode (Wijntuin, 1996).

Since the deformation behaviour is imposed on a theoretical vehicle model, the accuracy of the calculated crash process in the occupant contact area only depends on the accuracy of the applied dummy model. Hence, engineering requirement:

☑ e.r.19 deformation magnitude accuracy in the occupant contact area [target |deviation| < 20%]

is fulfilled. However, protection principles ensuing from biomechanical properties of occupants should form the basis for a sound vehicle design, not the behaviour of one particular dummy. Hence, it is of more importance that the load condition for different parts of the human body comes out of the simulations than that an accurate prediction of dummy readings is obtained. Until numerical dummies are available that enable the prediction of loads occurring within organs, the best we can achieve via numerical simulations is minimising the so-called 'Dummy Readings' that are empirically related to injuries. This implies that for an overall positive effect on safety a variety of dummies representing different occupants have to be studied.

The design of a deformation mode goes hand in hand with the generation of the accompanying analysis model because no structure has to be translated into numerical elements. However, this is only true on the condition that ideas on deformation modes are directly formulated in the preprocessor Generis of PamCrash. Otherwise, a transformation has to be performed that should be accounted for as analysis model generation time. This implies that engineering requirement:

☑ e.r.10 time required to generate an analysis model [target < 1 hour]

is fulfilled.

Concerning the calculation time, experiments with a flat intruding vehicle side as deformation mode for side impacts (Landheer, 1996) indicated a run time of about 3 hours on an HP715/50, not a 'high end' workstation. In Chapter 6, these experiments are discussed. It is likely that simulation of similar deformation modes requires a comparable run time. Hence, engineering requirement:

☑ e.r.22 elapsed time for a crash calculation [target < 3 hour]

is fulfilled for such modes.

The generation of a kinematic model after a proper deformation mode is established implies that constraints between separate spatial point groups must be set. This can for instance be spherical joints constraining three translational degrees of freedom, that will be represented by plastic hinges in the vehicle structure in Module 8. In most cases extra points must be added to the analysis model to mark the location of these joints. To put through these adoptions in the analysis model, nodal coordinates must be defined per added point and the points must be assign to a point group, what is a matter of minutes. Hence, engineering requirement:

☑ e.r.16 time required to adapt analysis model [target < 0.5 hour]

is fulfilled.

It is not possible to automatically generate an updated version of the geometric model in Unigraphics format via a transformation of the actual analysis model in PamCrash format with standard available numerical tools. This implies that engineering requirement:

☐ e.r.17 number of data conversions to translate an analysis model into a geometric model [target 1] is not fulfilled. To overcome this problem, the user should keep a journal of the changes made to the analysis model. With the aid of this journal, the adaptations to the geometric model for all deformation modes can be put through manually in one go as completion of Module 3. Of course, the time required for this manual adaptation of the geometric model depends on the complexity of the set up deformation modes. However, it can be expected that this will not take more then the 2 hours to generate a first geometric model in most cases. Consequently engineering requirement:

☐ e.r.18 time to convert an analysis model into a geometric model [target 1 hour]

is not fulfilled.

The information that is added to the original geometric model, is not necessarily an indication of the location of parts of the structure but can also mark imaginary folding lines, independent from the actual local presence of material. Two spherical joints that connect two point groups for instance, introduced to mark a folding line can be moved in the direction of a line through the joints without changing the implemented deformation mode. Hence, the spatial points that mark these joints are coupled via soft links.

The post-processor PamView is selected to process the analysis results of all numerical crash simulations in the here presented method. As a consequence, the following engineering requirements are fulfilled for every module but will only be mentioned here:

☑ e.r.23 a deformation versus time output plot option must be available
☑ e.r.24 an acceleration versus time output plot option must be available
☑ e.r.25 a geometry state output plot option must be available
☑ e.r.26 a deformation animation option must be available
☑ e.r.28 number of colours in plot [target > 2] (score > 10)
☑ e.r.29 hidden line removal in geometry plots must be available
☑ e.r.30 a cross section state plot option must be available
☑ e.r.31 a perspective geometry plots must be available

The required post-processing in the present module consists of:

- deformation versus time plots
- velocity versus time plots
- acceleration versus time plots
- momentum versus time plots
- force versus time plots
- deformation animations

The plots are used to check deformation modes on fulfilment of safety requirements and to compare the effect of different deformation modes. The momentum versus time plots are used to check conservation of momentum for the total crashing system. The deformation animations are intended to check if a deformation mode really coincides with how it was imagined by the designer.

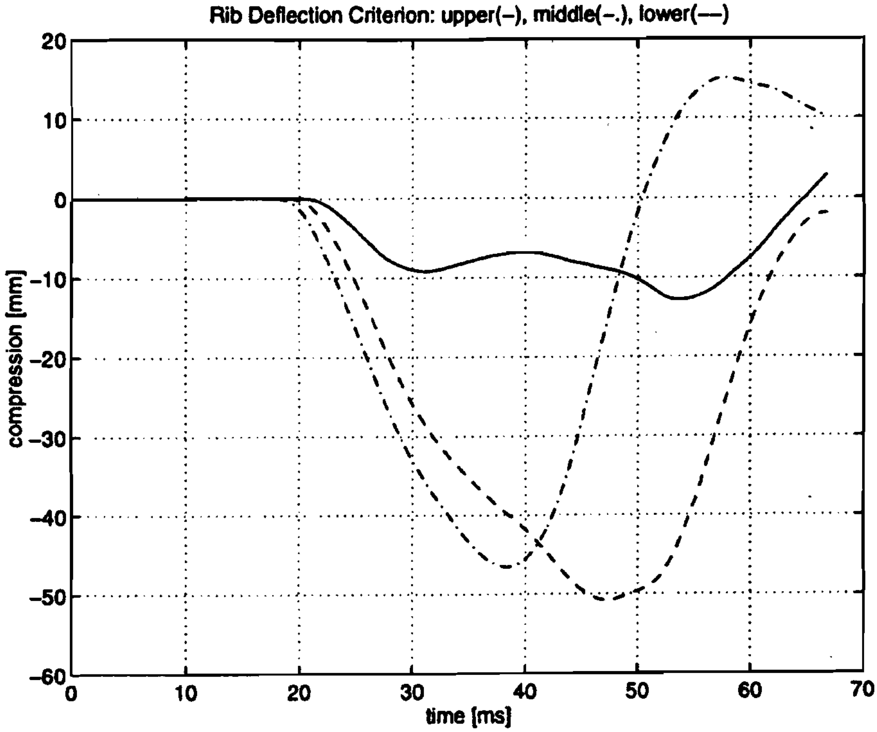


Figure 5.4 *Example of a time-history plot showing calculated rib compressions as a function of time (Wijntuin, 1996).*

As an example, a plot of the rib compression for the a specific side impact configuration, simulated with the model shown in Figure 5.3 is given in Figure 5.4. The generation of a time-history plot appears to take about 3 minutes. In many cases about 15 plots are sufficient to check the fulfilment of the safety requirements. Thus, the time required to post-process the simulation data of the present module can be generated within 1 hour. Unfortunately, with the current version of PamView, animations must be studied online and can not be stored to disk. Nevertheless, engineering requirement:

☑ e.r.32 time required to post process data [target < 1 hour]

is fulfilled.

5.6 Evaluation of Module 4: The load condition analyzer

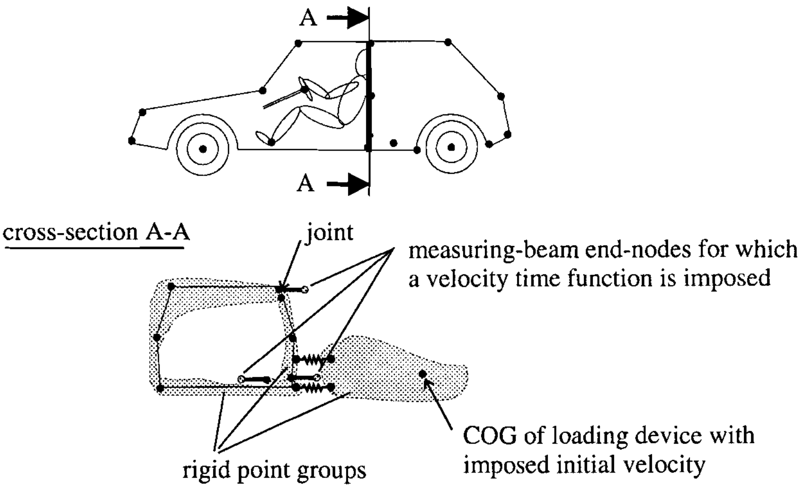


Figure 5.5 Example of the application of so-called 'measuring beams'.

The loads occurring in the joints between the rigid point groups can be written to hard-disk. The loads generated by the imposed velocity-time functions, introduced to compensate for the lacking deformation resistance of the deformation areas, can be established as well. This is achieved by imposing the velocity-time functions via a so-called 'measuring beam' that is mounted on a spatial point of the vehicle representation. The beam should be relatively stiff so that its deformation can be neglected. However, care must be taken not to choose a stiffness too high because this leads to a decreased time step in the numerical simulations that results in a longer crash calculation time (engineering requirement 22). The spatial point of a rigid point group on which a measuring beam is mounted, must be positioned near the expected location of a coupling with a neighbouring point group that must in the final structure provide the force. An example of the application of measuring beams is given in Figure 5.5 for a side impact deformation mode that was already presented in Section 3.4. The internal force of each measuring beam can be written to disk as well. With the post-processor PamView all the calculated force levels can now be plotted as a function of time.

5.7 Evaluation of Module 5: The structure type selector

The information necessary to select a proper structure type for the parts of the vehicle structure are:

- load levels that must occur during crashes in specific areas of the structure,
- data on structural alternatives for the parts such as costs, mass and required space.

The expected load levels have been calculated in Module 4. Data on what structural alternatives are available must largely come from engineering judgement of the design engineers. It is recommended that data on structural types that are applied is registered. This enables the designer in future to quickly get an overview of already applied structural alternatives that offer certain specified mechanical capacities.

After selection of a structure type for a part, the accompanying mass must be compared with the mass as it is represented in the spatial point model, generated in Module 1. Because that original spatial point model is based on a basic vehicle lay-out without special safety features, only deviations towards a higher mass can be expected. The masses assigned to the spatial points must be adapted to the insights gained in the current module.

A higher mass of a struck vehicle has a lowering effect on its velocity-change in a crash, as expressed in the law of conservation of momentum (see Equation 4.1). This implies that an underestimate of the effectiveness of the selected deformation modes is found in Module 3. Thus, the deformation modes that did fulfil the requirements with the original mass values will certainly fulfil the safety requirements with the new mass values for the spatial points.

5.8 Evaluation of Module 6: The safety concept generation environment

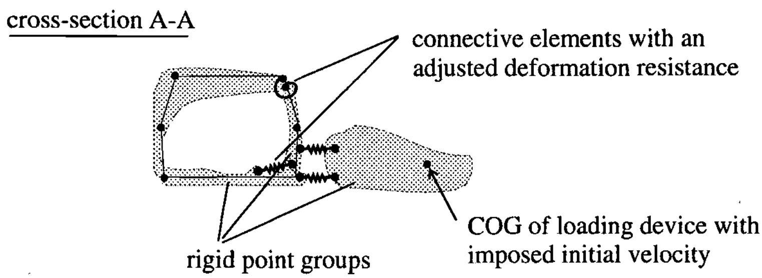


Figure 5.6 Example of the definition of energy dissipating connective elements between the spatial point groups presented in Figure 5.3.

The kinematic model designed in Module 3 consists of point groups that are constraint in their relative movement. In the current module, energy dissipating connections between these point groups must be defined to replace the so-called 'Measuring-Beams' with the originally imposed velocity-time functions (see Figure 5.5). Within PamCrash, several numerical elements are available that have an adjustable deformation resistance. Since the appearance of the deformable areas that are represented by these elements still have to be designed, only elements that do not require a shape description are appropriate. These elements connect a single point of one point group to that of another one by prescribing interacting forces and moments as a function of the relative position and movement of the two points. Depending on the intended properties, the following elements can be used:

- Spherical Joint or Flexion-Torsion Joint elements to model connections with one or more of three rotational degrees of freedom and all relative translations blocked
- Nonlinear Bar/Dashpot elements to model connections with one translational degree of freedom and all other relative displacements blocked
- Nonlinear 6DOF Spring/Dashpot elements to model connections with one to three translational and rotational degrees of freedom

The application of these element types is consistent with the geometric detail level of the spatial point model because only the two mounting points and up to three points to appoint the orientation of the elements must be specified. Consequently, engineering requirement:

☑ e.r.1 point modelling must be available

is fulfilled for the analysis model of the vehicle in Design Phase 1 (see Figure 3.3).

In this module, numerical simulations can be carried out to validate the sensitivity of a safety concept to variations in the severity of a crash load. For this purpose, the load cards—as they were defined in Module 2—must be modified. This is achieved by simply editing a scale factor for a load function or an initial velocity value in the PamCrash input file, a matter of a few minutes. Consequently, engineering requirement:

☑ e.r.21 time required for load card adaption [target < 0.3 hour]

is fulfilled.

5.9 Evaluation of Module 7: The packaging input module

The space available for the crash load transferring structural parts is characterised by its outlines. These outlines must be added to the spatial point model defined in Module 1. Therefore, the same CAD-software package Unigraphics as used to validate Module 1 is applied to validate the current module as well. The outlines are formed by the contours of components fitted to the vehicle and the exterior contours of the vehicle. The latter can be represented by planes fitted through the spatial contour points.

Since only the space occupied by components, and not their specific structure, is of concern for the packaging input module, a solid representation of these components is appropriate. Wijntuin (Wijntuin, 1995) demonstrates the representation of vehicle interior components as parametric solid models and the exterior as surfaces in Unigraphics. So, engineering requirements:

☑ e.r.3 surface modelling must be available

☑ e.r.4 solid modelling must be available

are fulfilled for the current module. The parametric definition of the interior components enabled a quick adaption of their shape to a specific vehicle layout, as it was already suggested in Section 4.8. Besides that, Unigraphics offers an interference option to check whether components intersect. This option automatically highlights areas of interfering components and in this way warns for design conflicts in an early stage. Figure 4.8 shows a vehicle interior thus obtained. Obviously, components of other vehicle compartments such as the engine compartment can be defined in the same way. If a library of parametric solid models of all standard vehicle components is available, this indeed enables a quick specification of a packaging plan. The compatibility of the packaging plan with the structure types for the vehicle parts as they are selected in Module 5 has to be taken into account during the generation of this packaging plan. Thus, the generation of the packaging plan starts at the same time as Module 5 starts.

An occupant model in Unigraphics is not available yet. Thus, a solid occupant model should be generated for various occupant sizes. The finite element occupant models available in the PamSafe software[11] can be used as a basis. Via the so-called 'Positioner' these models can be given a specific posture by specification of a rotational angle for the limbs around bio-mechanical joints (ESI/PSI, 1994b). If these joints are marked in a Unigraphics model the same posture adjusting operations can be put through simply by applying a rotation operation. Additional solids could be added to the occupant model that represent a standard free space for comfort of the occupant. Possibly, a cone shaped solid can be added to guard a minimum available sight for the driver.

5.10 Evaluation of Module 8: The force path definition environment

In this module, a load path layout must be generated that fulfils the planned mechanical properties. It is demonstrated by Du and Chon (Du, 1983) that with a beam-only model, the overall bending and torsional stiffness of a common vehicle body structure can be predicted with reasonable accuracy. Deviations between the calculated and the actual stiffness will always be negative because the stiffening effect of the sheet metal panels is

[11]PamSafe is a product from the PSI company

not considered in the numerical model. This implies that the actual loads within the individual beams in most cases are lower than the levels calculated for a given external load. Thus, the proposed representation of load paths as beams—coinciding with planned prismatic shaped members—in body parts that are considered not to deform is legitimate[12]. It implies that the actual collapse load of a body part is at least as high as the value with which is calculated in the current module. So, it is a safe estimate.

Based on the force paths layout formed by thin-walled beams of an existing vehicle design, a wire frame model is created in Unigraphics (Lambriks, 1994). The wires represent the neutral axes of the beams. This implies that a distance between the vehicle contour and the wires exists. A print out of this model, set up to carry out a side impact study, is given in Figure 5.7.

The mechanical properties of the beams are the result of rectangular cross-sections that must be set by the designer in the current module. The choice of the cross-section dimensions must be compatible with the packaging plan. By applying the so-called 'sweep along line' command in Unigraphics on the cross-sections for the accompanying wires, the contour(surface) of the space occupied by the beams in the structure is generated. An automatic interference check can then be performed. It is not necessary yet to define the shape of the connections between the beams. So, to save time it is skipped in this module. Consequently, the adaption of the geometric model only implies the definition of a new cross-section or the re-positioning of a wire what requires not more than a few minutes time. An example of the resulting representation of a connection between beams is given in Figure 5.8. So, it can be concluded that the following engineering requirements:

☑ e.r.2 wire frame modelling must be available
☑ e.r.3 surface modelling must be available
☑ e.r.5 geometry input must be graphically displayed
☑ e.r.6 geometry input must be graphically displayed on-line
☑ e.r.8 time required to adapt a geometric model [target < 0.5 hour]

are fulfilled for the current module.

For the transformation of this wire frame model into a PamCrash analysis model the GFEM-module in Unigraphics is applied. Each of the wires is modelled as a series of beam elements. Per straight wire section,

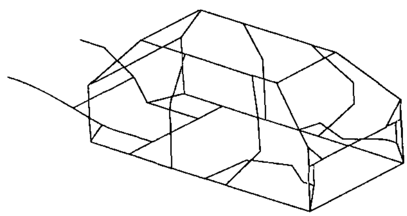


Figure 5.7 A wire frame model generated with Unigraphics.

[12]For this reason the discussion of the design process of the sheet metal panels is not given here. In relation to other vehicle requirements, such as acoustics, this process is for instance discussed by Herrmann and Helling (Herrmann, 1992)

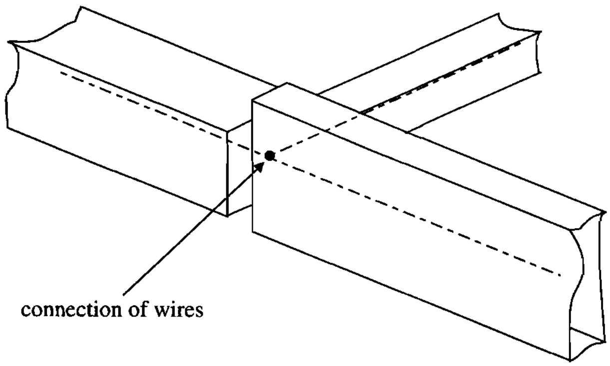


Figure 5.8 Example of the surface model of connected beams.

about ten beam elements must be defined. Since the length of such straight sections in common vehicle designs is about half a meter for these structural parts,

☑ e.r.27 number of structural points per occupied volume in the final output [target > 2]

is already fulfilled.

Within PamCrash, a specification must be assigned to the beams in so-called 'Material Cards'. This specification contains a cross-section description and a material property description. The mass of the vehicle body, in the preceding modules concentrated in the Contour Points, here has to be distributed over the beam end nodes. The proper weight of the beams themselves is already incorporated in the beam definitions. Therefore, it must be deducted on the mass of the Contour Points to be distributed. The transformation of the wire frame model is very time consuming because of the required mass allocation operation. It takes at least half a day to generate the first analysis beam model. However, this is inherent to the choice to generate a beam only model of the body. The following engineering requirements:

☐ e.r.10 time required to generate an analysis model [target < 1 hour]

can not be fulfilled. However, if a first analysis model is generated, adaption of cross-section definitions can be put through in a few minutes. So, engineering requirement:

☑ e.r.16 time required to adapt analysis model [target < 0.5 hour]

is fulfilled.

The response of the obtained beam model to an imposed side impact load is given in Figure 5.9. As could be expected, the response is quite unrealistic because the load levels in several beams exceed their natural collapse load. This is due to the fact that the model has not been equipped with so-called Folding Sections. The composition of Folding Sections to be implemented here should provide the energy dissipation pattern designed in

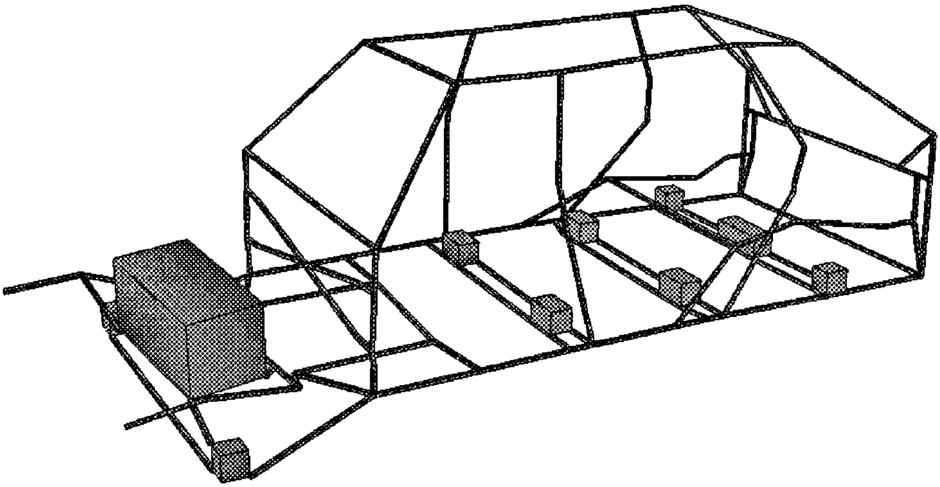


Figure 5.9 A beam only vehicle body model crashed with a NHTSA side impact barrier model (The boxes in the model represent components with a dense mass).

Module 6 and prevent overload of the adjoining beams. A more realistic response does indeed occur after folding sections in the form of 6DOF elements are added to the model as shown in Figure 5.10.

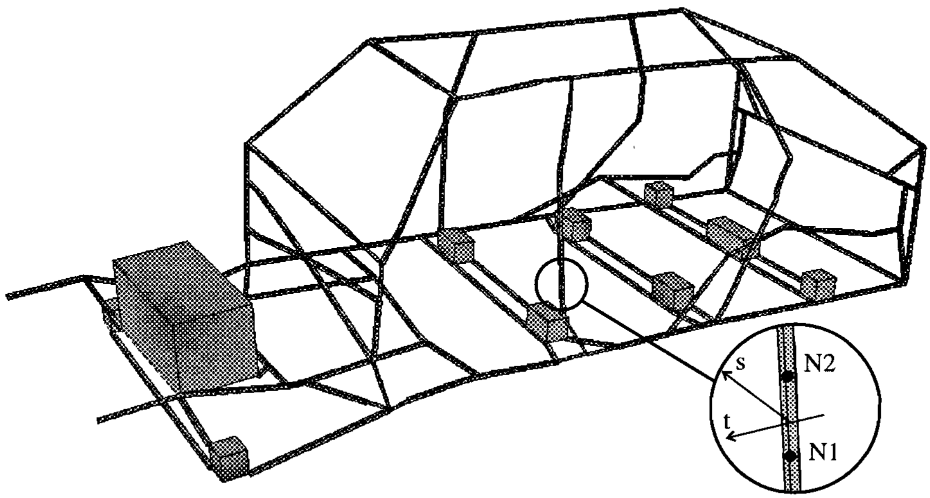


Figure 5.10 A vehicle body model with incorporated folding sections crashed with a NHTSA side impact barrier model.

The loads acting on the individual beams is stored during the crash simulations. This is required for the design of appropriate cross section dimensions for the beams, as described in the next paragraph. As an example, the loads acting on a beam element of the B-pillar located near the vulnerable abdominal region of the driver-body is given in Figure 5.11 and Figure 5.12. The principle axes are marked s-axis and t-axis in the plots where the t-axis approximately points towards the vehicle front-end. N1 and N2 represent the lower and upper beam-node respectively. A collapse of the B-pillar in this area would have a very unfavourable effect on the occupant loads. The plots demonstrate that the implemented folding sections cause a load reduction in the beam.

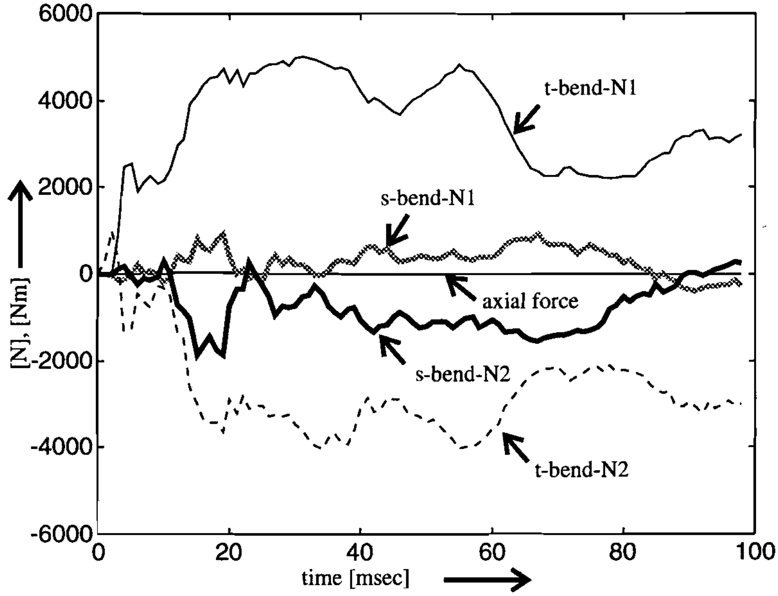


Figure 5.11 *Loads acting on the beam in the lower part of the B-pillar of the beam only model.*

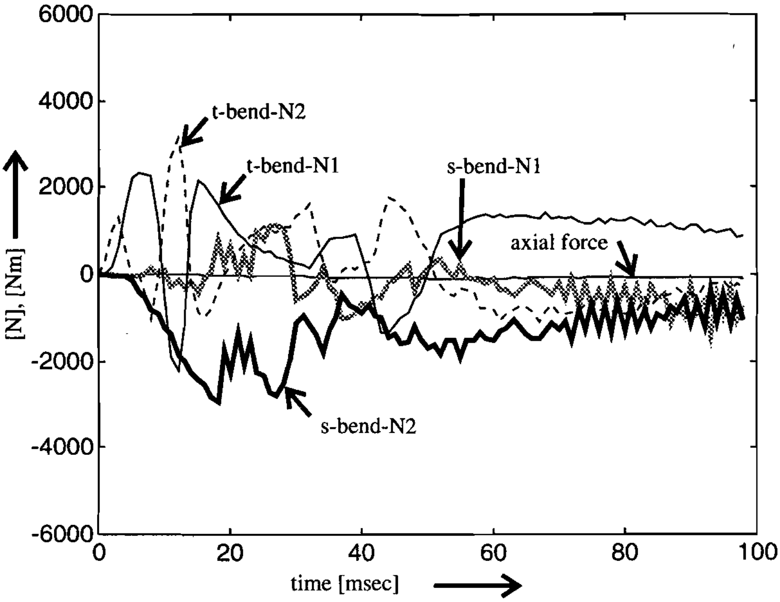


Figure 5.12 *Loads acting on the beam in the lower part of the B-pillar of the beam model with incorporated folding sections.*

5.11 Evaluation of Module 9: The coarse dimension estimator

In this module, appropriate cross-sections must be appointed to the prismatic sections of the vehicle body. The lay-out of these prismatic sections is set in the beam model, generated in Module 8. To obtain the same deformation mode that did occur in the numerical simulations with this model in a real crash, the characteristic collapse loads of the sections must exceed the calculated maximum load levels.

The collapse properties, inherent to a beams' cross-section, can be calculated with CrashCad, a software package that runs on a personal computer as well as on a Unix workstation. As illustration the calculation of the collapse properties for a simple rectangular cross-section, applied in the lower part of the B-pillar of the in the previous paragraph presented beam models, is discussed here.

First, the cross-section is drafted in the graphical interface that on-line displays the given geometry input. This implies that the engineering requirements:

☑ e.r.5 geometry input must be graphically displayed

☑ e.r.6 geometry input must be graphically displayed on-line

are fulfilled by CrashCad.

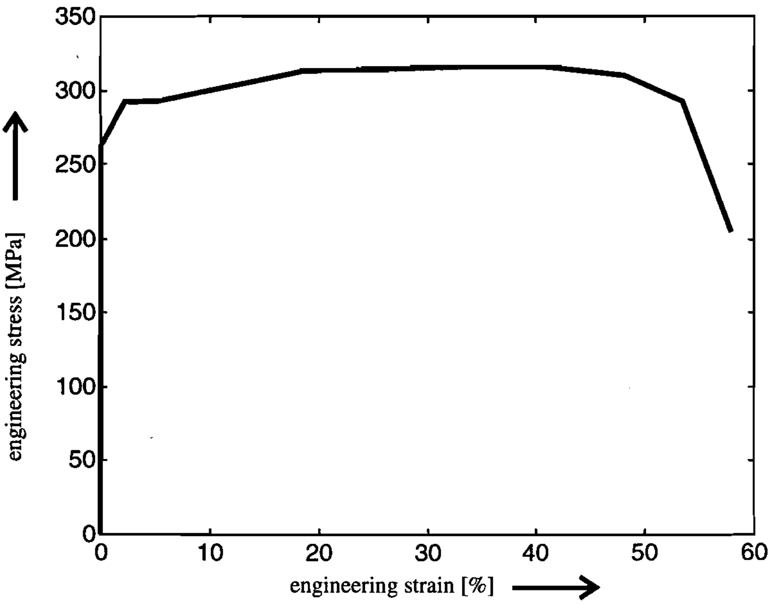


Figure 5.13 Tensile characteristic of the material for the B-pillar.

Second, the material properties must be defined. The tensile characteristic with Yield Stress, the Young Modulus and the Ultimate Tensile Strength used for this example, is given in Figure 5.13. A strain-rate dependency can be defined via the so-called "Cowper-Symonds law":

$$\frac{\sigma_{dyn}}{\sigma_{stat}} = 1 + \left(\frac{\dot{\varepsilon}}{D}\right)^{\frac{1}{p}} \qquad 5.1$$

where:
- σ_{stat} = quasi static stress
- σ_{dyn} = dynamic stress
- $\dot{\varepsilon}$ = strain rate
- D, p = constants that describe the strain-rate dependence.

Third, the crash resistance against axial compression and bending around the principal axes can be calculated. Per load case, the calculation takes about 0.2 seconds on a standard 486/50 personal computer. The results of the calculations are given in Figure 5.14 and Figure 5.15.

The prismatic sections defined as beams in the numerical model are intended not to collapse. Therefore, only the collapse load levels, the first peak in the curves, are required for the beam load analysis. The collapse loads are given in Table 5.2.

F_r (axial collapse force)	55 kN
M_s (bending collapse moment for s-axis)	6.1 kNm
M_t (bending collapse moment for t-axis)	4.7 kNm

Table 5.2 Collapse properties of the lower part of the B-pillar.

With these data, a so-called 'Weighted load' for the beam nodes can be calculated according to Equation 4.4 as a function of time. A calculation is performed on the beam load given in Figure 5.11(the beam only model) and Figure 5.12(the beam model with incorporated folding sections). The results are given in Figure 5.16 and Figure 5.17 respectively. As already suggested, the weighted load exceeds unit magnitude, i.e.: the collapse load level of the beam in the beam-only model is exceeded. Thus, the model without folding sections gave unrealistic results. The local collapse process of the B-pillar can not be represented by homogeneous deforming beams! However, the beam model with incorporated folding sections was permissible.

Other cross-section shapes can be analyzed in the same way. However, if another cross-section is designed to prevent collapse, it is important to perform extra calculations on the vehicle beam model with updated I-setting(moment of bending inertia setting) for the beams concerned, because other beam load levels may occur.

The weighted loads were presented here as curves. It would be very useful if the weighted load levels could be projected on the animation plots generated in the post-processor in the form of a colour code. The engineer would have a direct overview on the safety margin of his vehicle body design. After all, if the weighted load levels for the beams planned not to collapse appear low, an increased crash load will not directly destroy the implemented safety concept.

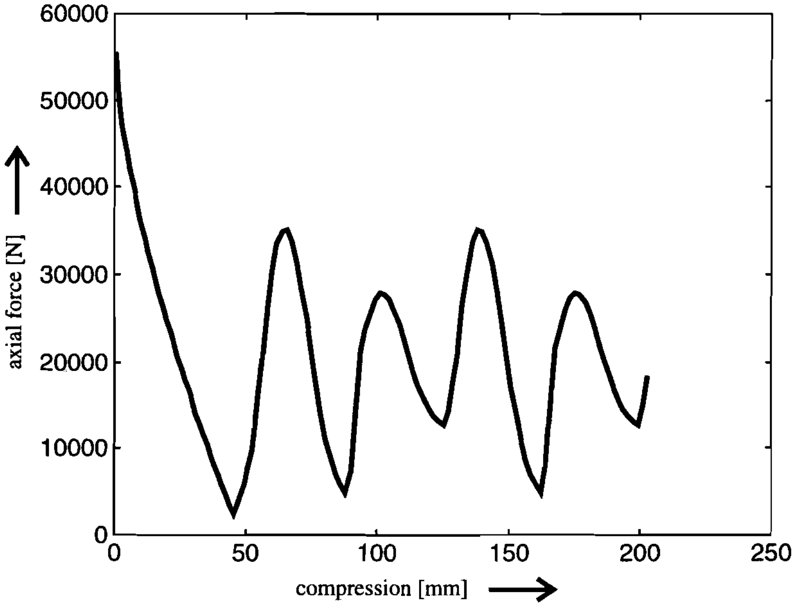


Figure 5.14 The resistance against axial compression.

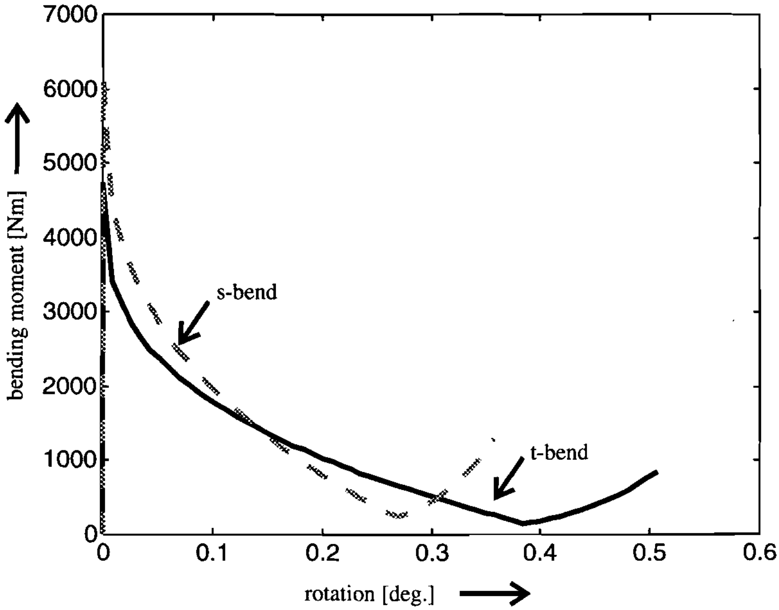


Figure 5.15 The resistance against bending.

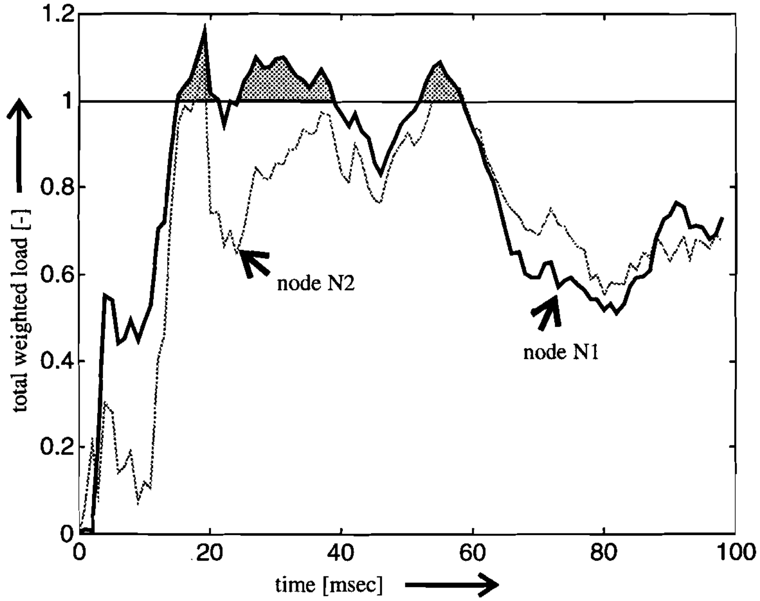


Figure 5.16 The weighted load for the beam in the B-pillar in the beam-only vehicle model.

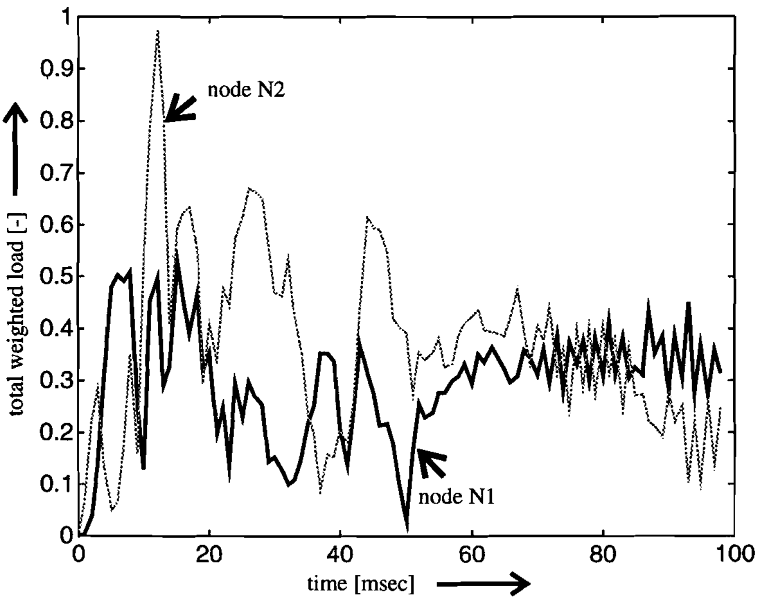


Figure 5.17 The weighted load for the beam in the B-pillar within the beam vehicle model with incorporated folding sections.

5.12 Evaluation of Module 10: The parallel subsystem design module

In Module 8, only parts with a prismatic shape that are intended not to collapse in the in Module 2 selected spectrum of crash configurations, are represented geometrically in the numerical model. The remaining parts are only represented by their intended mechanical behaviour. A matching geometry for these parts must be validated in the current module. With a shell mesh model, the geometry of these parts can be described. This is illustrated for a T-shaped connection of two prismatic sections in Figure 5.19. The procedure to generate such a mesh model contains the following steps:

- geometry definition with Unigraphics CAD software,
- mesh generation with the GFEM-module in Unigraphics,
- mesh data exchange to PamCrash, the crash analysis software, via a Nastran-format file.

The software choice implies that the following engineering requirements:

☑ e.r.3 surface modelling must be available
☑ e.r.5 geometry input must be graphically displayed
☑ e.r.6 geometry input must be graphically displayed on-line

are fulfilled.

Before load cases can be simulated with the shell mesh model, the sheets of which the part consists must be attached to each other and fixed to rigid bodies that represent the adjoining beam-ends. The mutual attachment of two sheets is often achieved by spot-welds. Such spot-welds can be modelled as rigid bodies (Oirschot, 1995). Unfortunately, the rigid body definitions can not be generated by the automatic mesh generator of the GFEM-module. This implies that every individual spot-weld must be defined manually in the preprocessor Generis of PamCrash. Consequently, the fulfilment of engineering requirement:

☐ e.r.10 time required to generate an analysis model [target $\leq$ 1 hour]

strongly depends on the complexity of the part.

The material properties to be assigned to the elements are the other design parameters besides the geometry that bring about the mechanical properties of the part. Definition or adaption of these material parameters is a matter of minutes. Therefore, concerning the material properties engineering requirement:

☑ e.r.16 time required to adapt an analysis model [target < 0.5 hour]

is fulfilled.

For the specific geometry, described by the shell mesh model, load-deformation curves can be established for the three mutually perpendicular axes that coincide with the local axes in the corresponding 6DOF element representation of the part in the vehicle beam-model, via numerical simulations.

As an example, one of the force-deflection curves for the shell mesh given in Figure 5.19 is calculated. For this purpose two rigid bodies are fixed by boundary conditions as illustrated in Figure 5.18. The third rigid body is loaded via a so-called measuring beam just as it was already applied in Module 4 (Hoofman, 1995). The calculated force-deflection curve for the T-direction is given in Figure 5.20.

To validate the shell-mesh model, experimental testing on a prototype part is possible in this stage because all necessary data is available now. For this purpose a prototype part of the T-shaped structure, displayed in Figure 5.19, has been constructed (see Figure 5.23). The rigid body definitions applied in the numerical simulations are simulated by a heavy 8 millimetre thick steel frame. A spherical joint is applied to enable the introduction of a load which only consists of a force in T-direction at a location coinciding with the measuring beam end node in the numerical model. For comparative reasons besides this structure with flat surfaces, an additional structure with curved surfaces is prepared for a test (see Figure 5.22). The later structure comes from a compact car where it connects the B-pillar with the door-sill. Both test pieces have been loaded on the test-rig showed in Figure 5.21. The measured force-deflection curves are given in Figure 5.26.

Both the calculated (Figure 5.20) and the measured curve (Figure 5.26) for the structure with flat surfaces show a peak in which a first fold is initiated, followed by a relatively constant deformation resistance. However, the experiment points out a higher average deformation resistance than the numerical simulation. A closer examination of the deformation process shows that an inverse fold is initiated in the experiment compared with the calculated deformation pattern (Figure 5.24 and Figure 5.25). This is probably due to imperfections in flatness of the surfaces of the proto-part caused during its construction. A structural shape that is sensitive to such small imperfections has an unpredictable response in a crash and therefore should be avoided in areas of a vehicle body intended to absorb crash energy.

By implementation of pre-formed folds or so-called fold initiators a high sensitivity of a structure to deviations in its shape can be avoided (v.d. Poll, 1996). In fact, a local deformation mode is pre-programmed by the curved surfaces in the second test piece. Consequently, the high peak for the initiation of a first fold is absent in the measured force-defection curve for this test piece (Figure 5.26). By application of such deformable areas instead of those requiring a peak load to initiate the deformation process in the vehicle body, adjoining areas intended not to deform are spared. This is illustrated for the same beam element in the vehicle model that is discussed in Paragraph 5.10. The weighted load given in Figure 5.17 can be reduced. This is achieved by replacing an entered force-deflection curve in the 6DOF element definition for the lower end of the B-pillar. For the T-direction of this element, the force-deflection curve originally containing a peak is replaced by a smooth force-deflection curve as shown in Figure 5.26. The resulting total weighted load is given in Figure 5.27. The reduction in load is evident. However, a disadvantage of such a smooth force-deflection curve is that in specific load cases where deformation of the 6DOF element is unwanted, the accompanying collapse load level that may not be exceeded is low.

If for all 6DOF elements in the vehicle model a geometric structure is found that meets the load-deflection properties defined in Module 8, a vehicle-body structure for which the numerical model is valid is established. Because the numerical model offered the desired crash protection capacity selected in Module 2, the established body structure design is effective. However, the deformation resistance of thin walled steel structures for one direction often depends on deformations in the other directions. This dependency can not be taken into account in the definition of a 6DOF element within PamCrash. In case such a combined deformation occurs a numerical model in which this deforming area is represented by a shell mesh can take into account the dependency. This type of numerical model is called a 'Hybrid Model' because it contains shell as well as beam elements. The beam model defined in Module 8 can be transformed in such a Hybrid Model by replacing the 6DOF elements that undergo a combined load by a shell mesh of the local structure. The shell mesh can be generated separately from the beam model as indicated for the mesh given in Figure 5.19. With the 'Positioner', a tool in PamCrash for the positioning of a dummy model in vehicle models, this shell mesh can be attached to the adjoining beam elements of the beam model via its rigid body definitions, after the concerned 6DOF element is removed. In Figure 5.28, a beam model with an implemented shell mesh representation of the B-pillar door-sill area is given as an example of a loaded hybrid model.

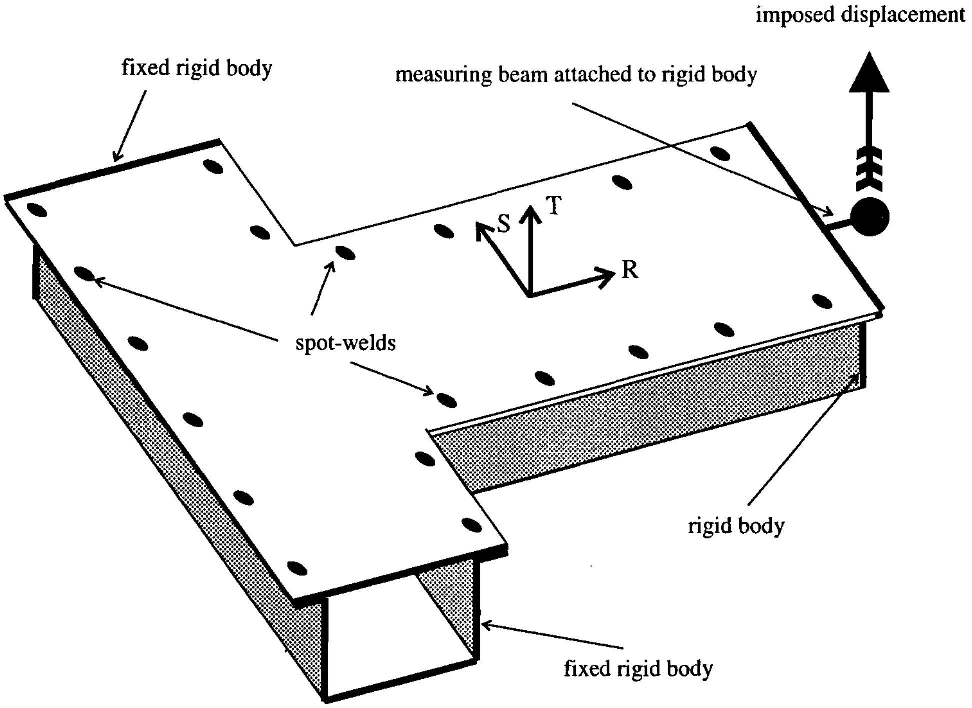


Figure 5.18 The rigid body definitions and the measuring beam in the analysis model of the T-shaped structure.

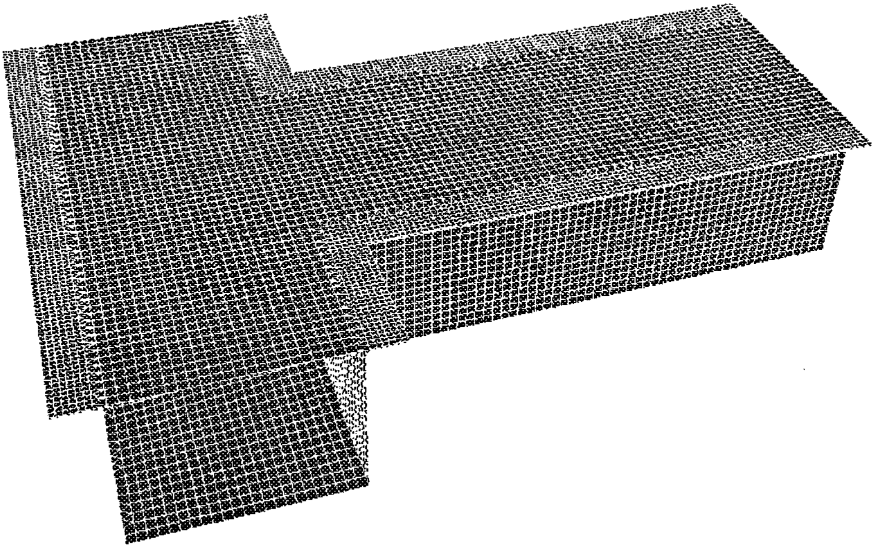


Figure 5.19 Shell mesh model of a T-shaped connection between two prismatic sections.

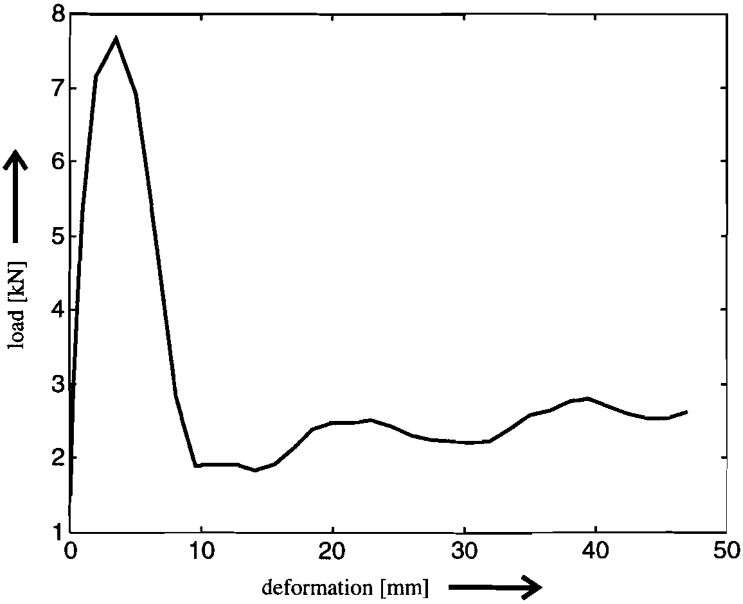


Figure 5.20 The calculated load response curve for the T-shaped structure.

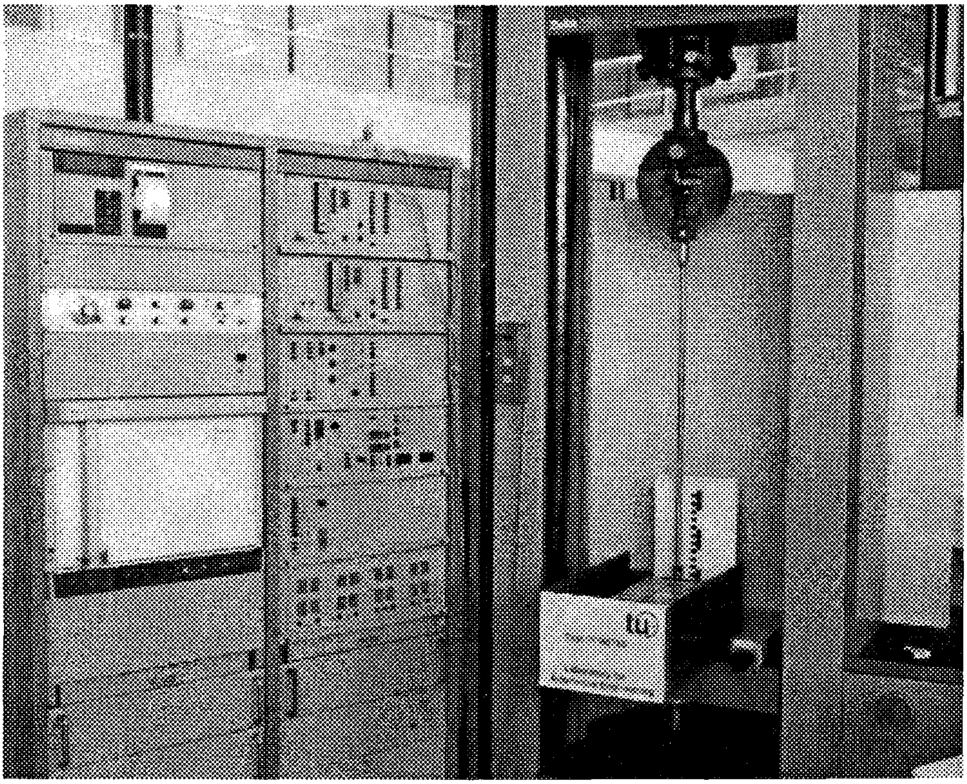


Figure 5.21 The test rig used to load the T-shaped structures.

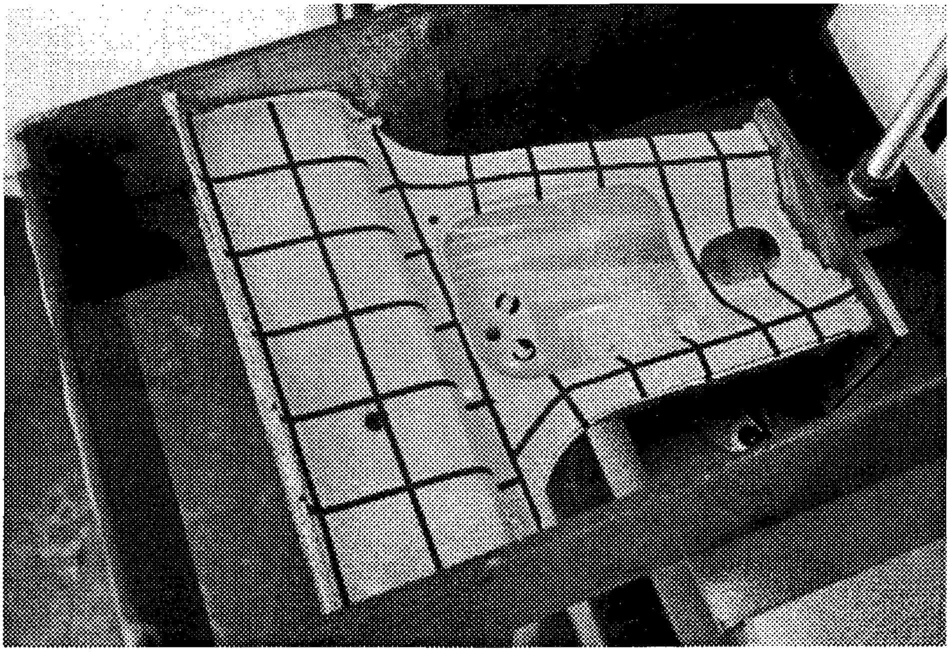


Figure 5.22 Test piece with curved surfaces.

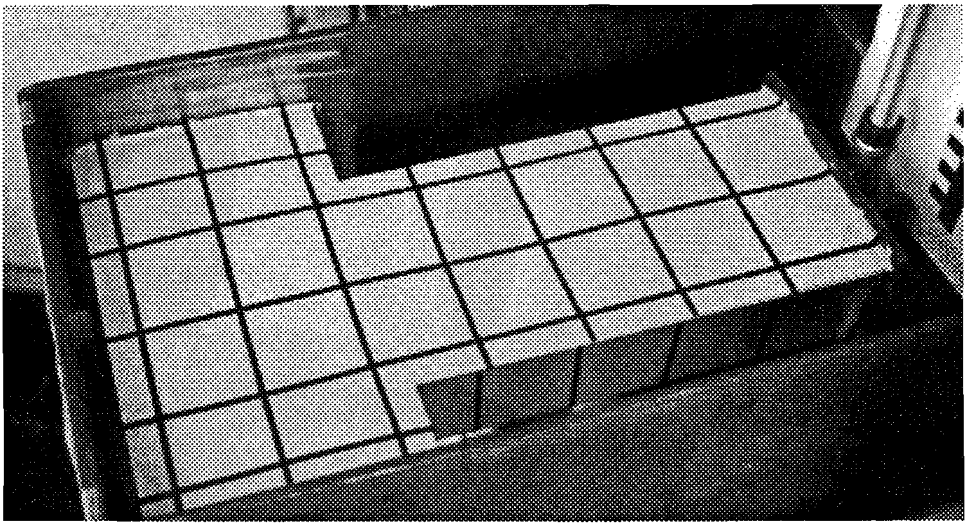


Figure 5.23 Test piece with flat surfaces corresponding with the shell mesh model.

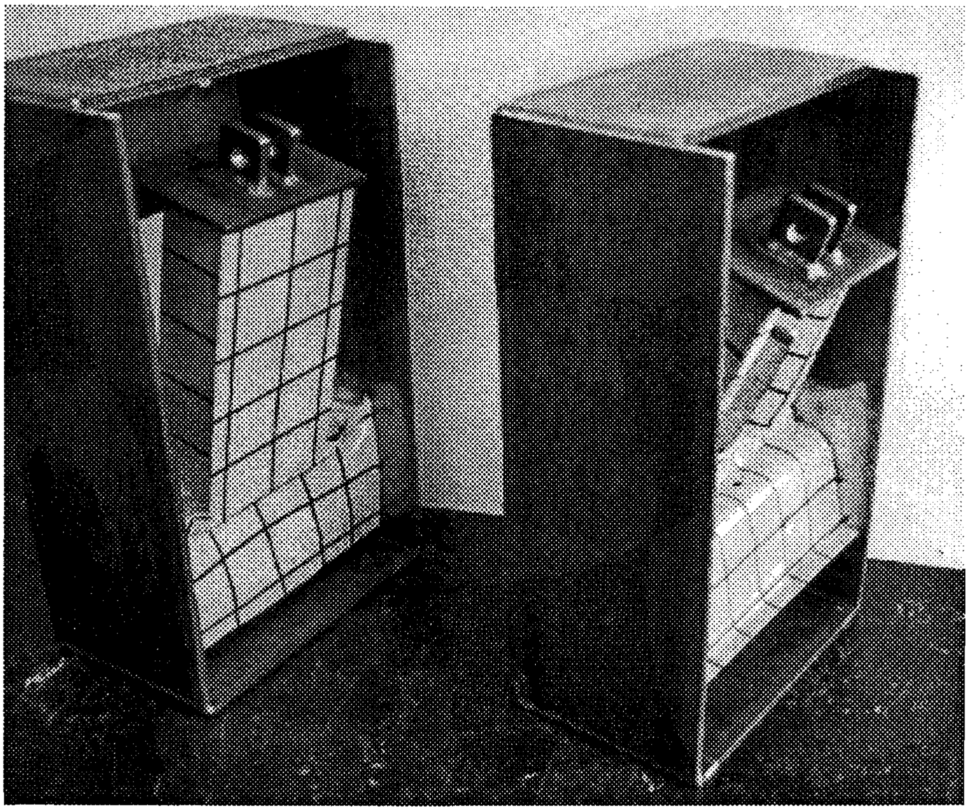


Figure 5.24 The test pieces after the test.

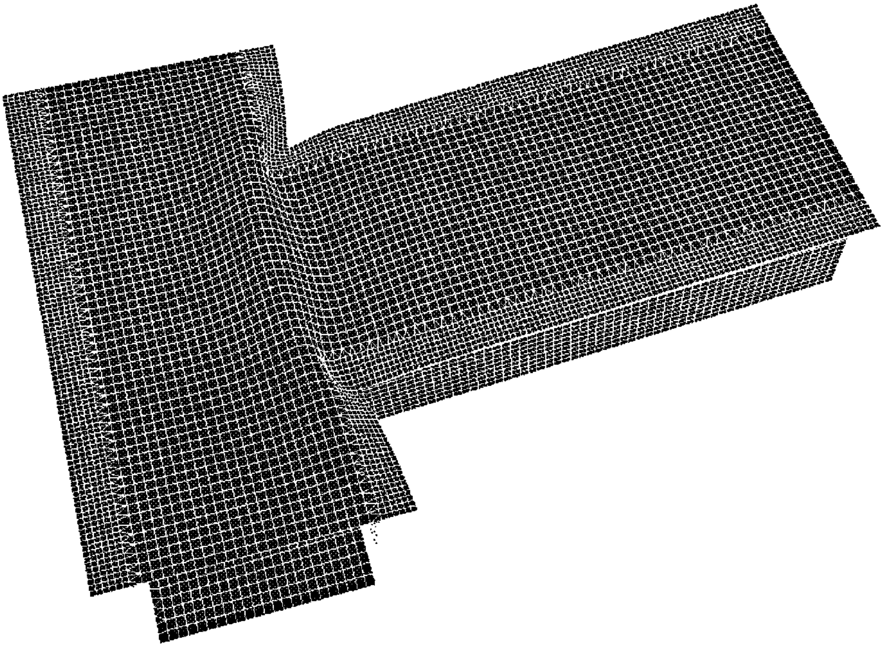


Figure 5.25 Deformed mesh of the T-shaped connection between two prismatic sections.

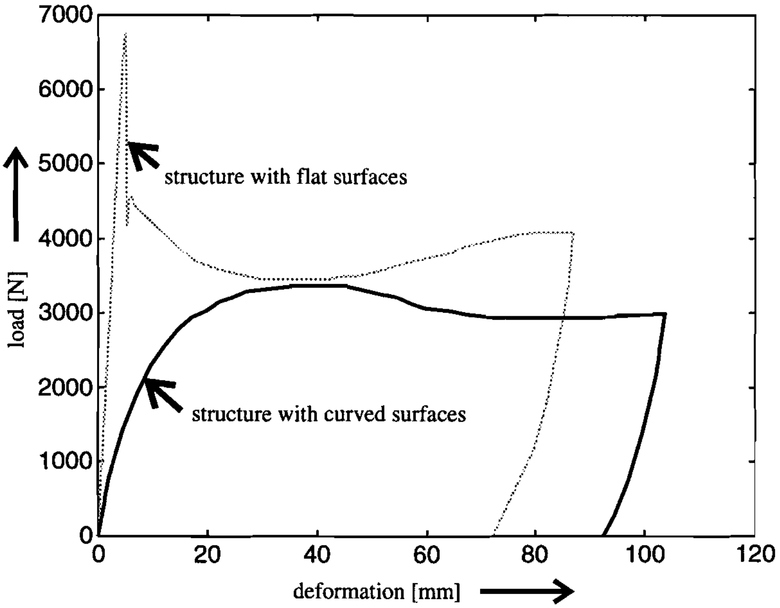


Figure 5.26 Measured force deflection curves for the two test pieces.

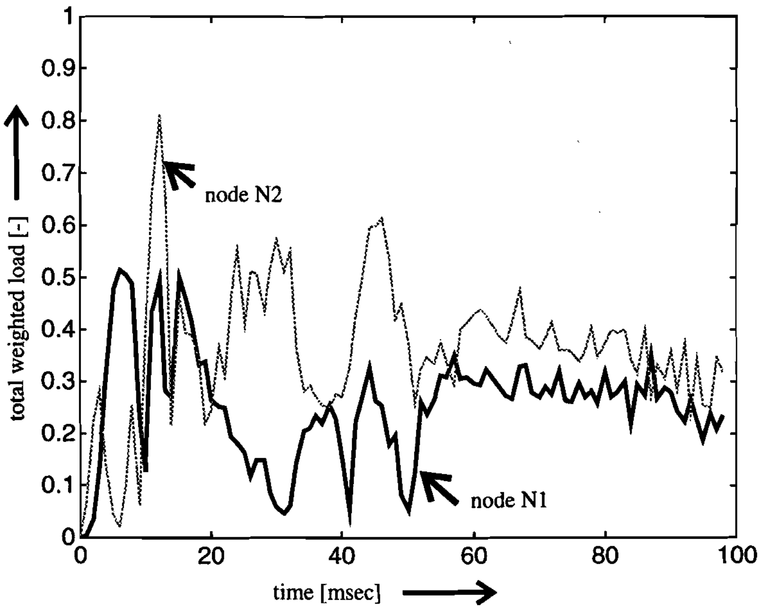


Figure 5.27 The weighted load for the beam in the B-pillar within the beam vehicle model with the smooth B-pillar door-sill deformation resistance curve.

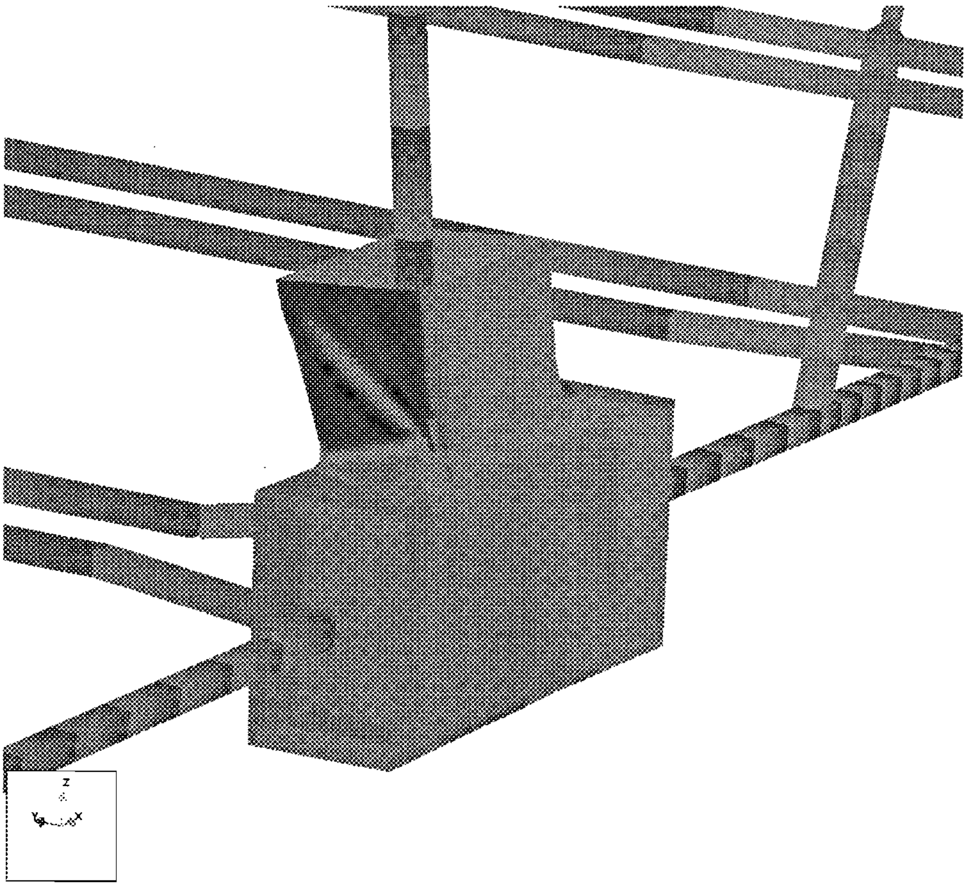


Figure 5.28 *Example of a hybrid model, the B-pillar door-sill area is modelled with shell elements within a beam model.*

5.13 Evaluation of Module 11: The Detailed tuning module

To generate the required hybrid numerical models in this module, the same technique can be applied that was used in Module 10. Moreover, the shell-mesh representations of areas that are generated in the preceding module can be utilized again here. This implies that engineering requirements:

☑ e.r.3 surface modelling must be available
☑ e.r.5 geometry input must be graphically displayed
☑ e.r.6 geometry input must be graphically displayed on-line

are fulfilled.

Besides these already meshed areas, now the deforming area of the vehicles' external surface, and all parts underneath involved in the deformation resistance via occurring contacts, must be represented by a shell-mesh. An automatic mesh-generator can be applied to create the mesh for these areas. However, so-called 'control node lists' must in this case be used at the edges to join the mesh with the already existing meshed areas. Where in the preceding modelling stages loads could only be introduced via discrete points within the vehicle representation, now a standard loading device model can be applied per crash configuration to load the vehicle model via its exterior surface, independent from the structural shape to be loaded. It is hard to give a general estimate for the time it takes to generate a complete hybrid model of the kind discussed here because this strongly depends on the complexity of the areas of the vehicle that have to be represented by a shell mesh. A smooth simple shaped area can be meshed with an automatic mesh generator in a few minutes after control node lists have been defined at the edges. A complex shape however requires a lot of additional manual input because automatic mesh generation can in this case only be applied after the definition of sub-areas, enclosed by control node lists, that can then be mesh subsequently. Moreover, the number of spot-welt definitions is of large influence, as already emphasised in the preceding paragraph. Consequently, engineering requirement:

☐ e.r.10 time required to generate an analysis model [target < 1 hour]

will be fulfilled occasionally.

Since the geometric data required to generate the described shell-mesh representation of the vehicle design is of the same detail level that is required to build a proto-type vehicle body, it is in this stage possible to validate the numerical simulation results against those of a full scale crash test. Within the framework of this research project no full-scale vehicle prototypes were built. Instead, a hybrid numerical model was generated of an already mature vehicle design. Of this design, the experimentally validated FEM shell-mesh model given in Figure 5.30 was already available for the LS-DYNA3D crash code. An example of a full-scale side impact validation test according to the NHTSA 214 test procedure is given in Figure 5.29. The validated FEM model is used as reference. The technique applied to generate a hybrid model afterwards from the detailed FEM model is described by Reilink (Reilink, 1995). Mainly, the transformation comes down to the identification of the structural areas showing insignificant deformations in crash-simulation runs and hence, can be represented by simple beam-elements and rigid bodies. A plot of the resulting reduced model for side impact simulations is given in Figure 5.31. Apart from the rigid body representation of the non-struck side of the vehicle, the model is comparable with a hybrid model created following the procedure described in this thesis. The definition of the non-struck side of the vehicle as one rigid body however, limits the value of the model. The user will not be warned when a collapse load level for a part in this rigid body is exceeded and the model starts giving incorrect simulation results. Apart from this difference, the run time and the accuracy of the hybrid model obtained by reduction of the FEM model or by expansion of the beam model will be similar.

The run time of the reduced model is cut back by a third compared to the reference FEM model. Reilink reports that a complete side impact simulation with the reduced vehicle model without occupants took 13.3 CPU hours on the Cray Y-MP stationed at Volkswagen AG in Wolfsburg. Consequently, engineering requirement:

☐ e.r.22 elapsed time for a crash calculation [target < 3 hours]

is with the current hard- and software still far from fulfilled for the detailed hybrid model.

The maximum size of the shell elements used in the hybrid model lays in the order of 5 centimetre. This implies that engineering requirement:

☑ e.r.27 number of structural points per occupied volume in the final output [target > 2 dm^{-3}]

is readily fulfilled. From this point of view a coarser mesh with less elements would be a possibility to further bring down the calculation time. However, the geometric shape of common vehicle designs does not allow a coarser mesh over a large area because this would significantly influence the accuracy of the simulation results.

The area of the vehicles' interior that contacts the driver in a side impact crash is situated around the B-pillar. Reilink found a maximum deviation of about 7 % for the calculated deformations in this area for the reduced model compared with the validated full-mesh model. Consequently, engineering requirement:

☑ e.r.19 deformation magnitude accuracy in the occupant contact area [target: |deviation| < 20%]

is fulfilled for the model without occupants.

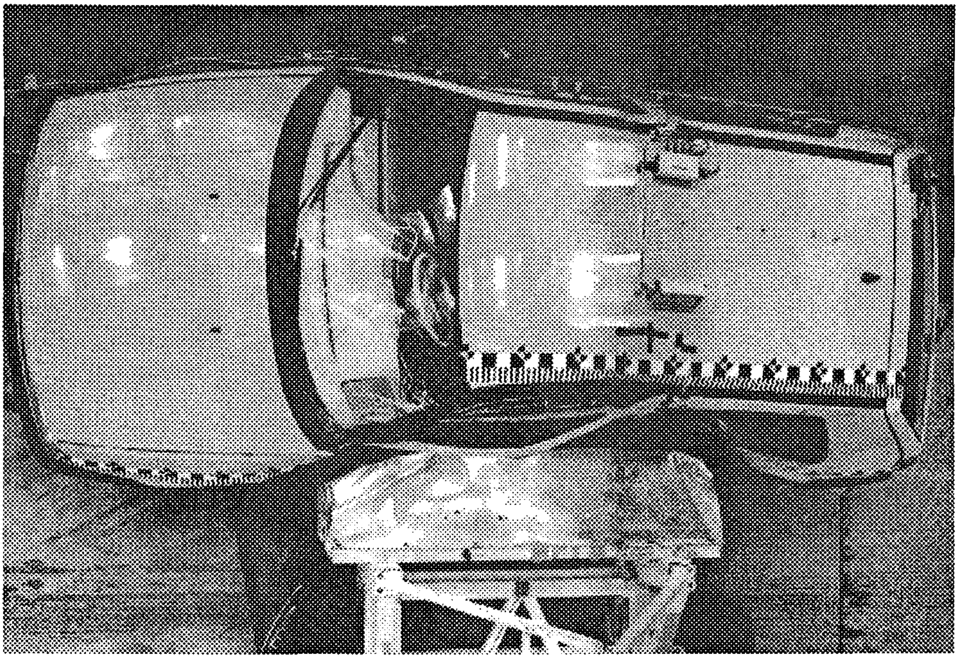


Figure 5.29 Experimental side impact test according to FMVSS 214 (Volkswagen AG, 1992).

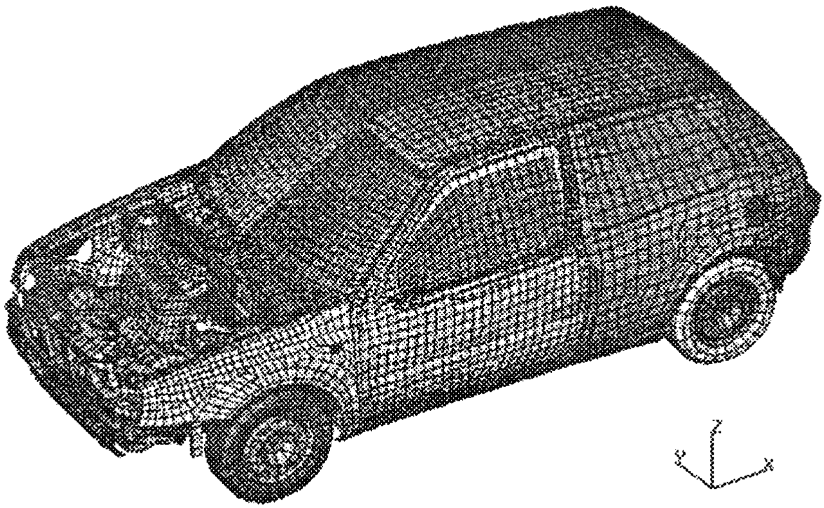


Figure 5.30 Full vehicle FEM mesh model (Reilink, 1995).

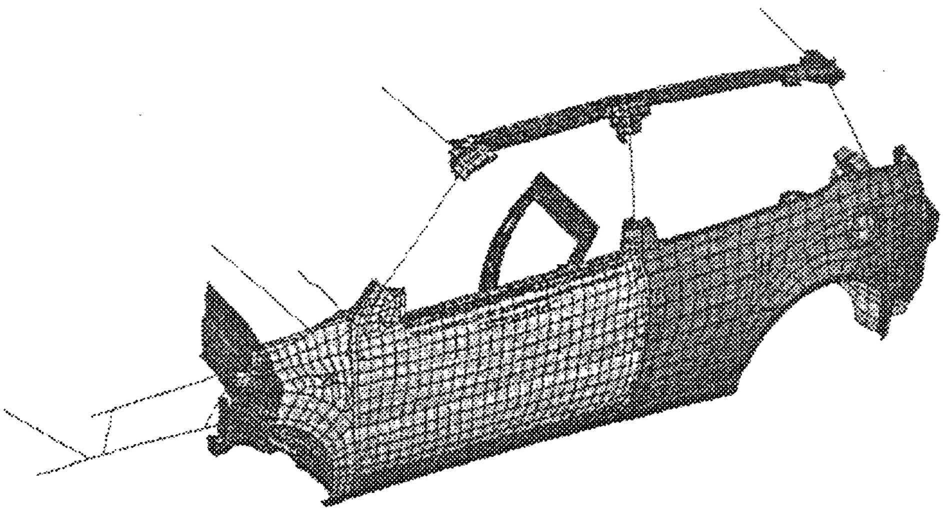


Figure 5.31 Reduced FEM mesh model (Reilink, 1995).

5.14 Discussion

It is illustrated that all modules of the new design method are practicable. The method complies with all engineering requirements that are strongly related with "musts" (see Table 3.1) and most other requirements. However, not all engineering requirements are yet met. Especially the time required for the generation of detailed analysis models (Modules 8, 10 and 11; e.r. 10) and the elapsed time for crash calculations with the hybrid models in Module 11 (e.r. 22) exceeds the targets.

The target for engineering requirement 10 in Table 3.1 is meant to appoint a period of time. Therefore, the performance of the method increases if the model generation is carried out by several engineers working simultaneously. When consistent control node lists are applied for the edges of model sections that must finally be joined together, this is indeed possible.

If the current rate of improvement of hardware and software continues to exist, it is expected that the calculation time will be reduced sufficiently in a few years time to meet the target values. In the meantime, the design method is nevertheless very useful because even the detailed calculations in Module 11 can be run over one night. This enables an evaluation of the effects tuning detailed geometric parameters in a day.

Chapter 6

Example: design and evaluation of two side impact deformation modes

6.1 Introduction

The present chapter illustrates the capabilities the design method offers to design and evaluate deformation modes. Two examples are given for a side impact crash configuration. Decisions on structural designs offering high side impact safety are rather difficult to take, because as yet no mature safety concept exists for this type of crash. Thus, these examples provide a real insight into the value of the method for product innovation and development.

6.2 Design objective

The vehicle type (Module 1)
A front wheel driven two-doors fast back vehicle with a wheel base of 2.6 meter is selected for this example (see Figure 4.1). It must offer space to 4 occupants. It is powered by a 1.6 litre engine in front, coupled with a manual gear-box. The total mass of this standard traction system is 241 kg. The total planned vehicle mass is 850kg.

The safety requirement (Module 2)
The European side impact safety regulation discussed in Section 2.4 is selected as safety requirement. For the calculation of dummy-loads a FEM-model of the EuroSID side impact dummy from the ESI company is used (see Figure 6.1). This numerical model enables the calculation of normal forces as well as tangent friction-forces in the contact area between dummy and vehicle

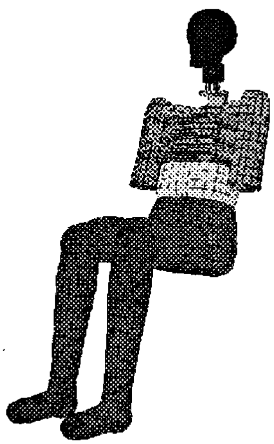


Figure 6.1 FEM model of the EuroSID.

structure that bring about the dummy response. No measuring points on the vehicle are required because the selected safety requirement is based on dummy readings only.

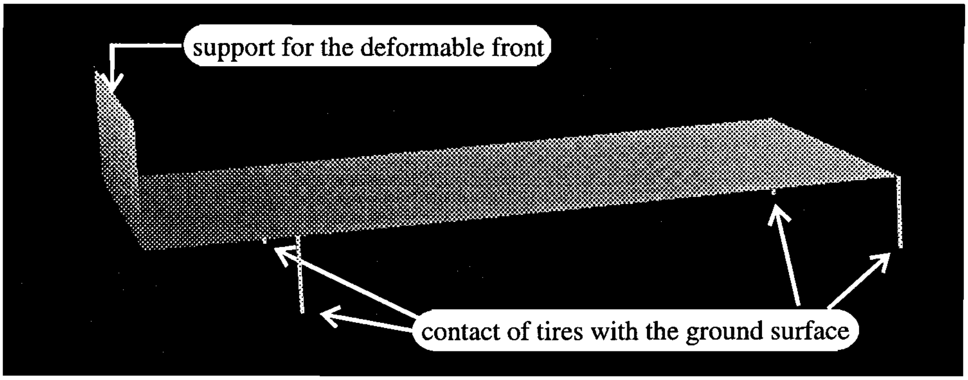


Figure 6.2 *Numerical model of the EEVC barrier.*

As loading device, a model of the EEVC barrier is applied. This model consists of a rigid body shown in Figure 6.2. This rigid body has the dimensions, and is assigned the inertia properties, that are prescribed in the side impact safety regulation. The support for the deformable front is modelled as a flat surface. The numerical elements that represent the deformable front still have to be defined because no choice on the load transfer to the vehicle has been made yet. However, the rigid part of the barrier model is positioned in such a way that the front-surface of the deformable front initially coincides with the planned vehicle exterior surface. This eases the interpretation of the graphical output of the numerical simulations. The contact of the tires with the ground surface is modelled in four points, one for each tire. In these contact points the tire friction as well as the normal force is modelled.

6.3 Protection principle

During a side impact, the occupants seated at the side of impact must be accelerated in a very short interval. Consequently, these occupants are loaded with high contact forces. To avoid injuries, these contact forces must be distributed over large contact areas thereby protecting the sensitive parts of the body. Besides that, the maximum amount of time from the available interval must be utilized to accelerate the occupant while this minimizes the required force levels. Two side impact deformation modes A and B, based on these principles, are discussed here. Deformation Mode B is an adapted version of Deformation Mode A. The design of these deformation modes comes under Module 3 of the method.

Deformation Mode A

The distribution of the occupant contact force over the body is the result of the shape and deformation resistance of the intruding vehicle side. In Deformation Mode A, a flat intrusion profile is examined. By pre-loading the relatively robust pelvic region of the occupant, his thorax is spared. In the deformation mode this pre-loading is realized by an inwards rotation of the vehicle side around an axis coinciding with the vehicle's roof rail. This behaviour of the vehicle side was recognized by Hobbs (Hobbs, 1989) to offer a positive effect on dummy loads in crash tests. To spread out the change of momentum of the occupant over time, the intruding contact zone is chosen to be deformable. For the part

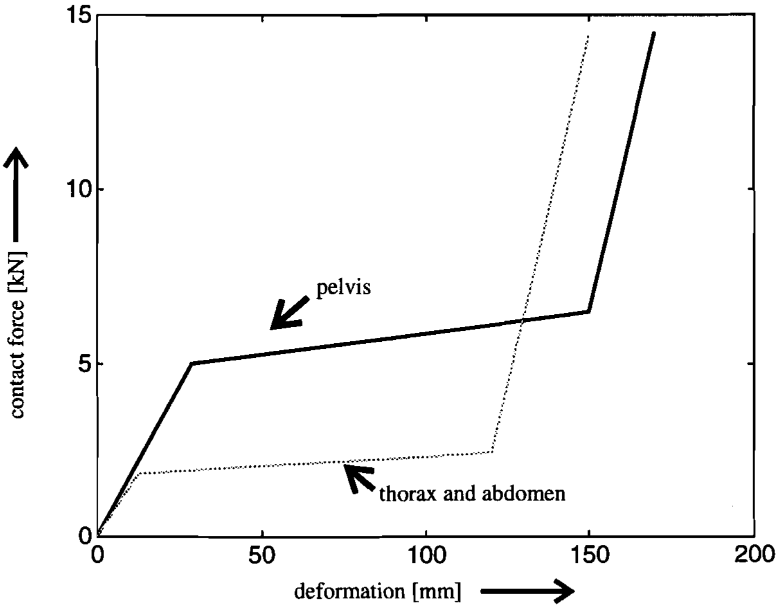


Figure 6.3 Interior deformation resistance for pelvic, abdominal and thoracic intrusions.

of the contact zone that contacts the thoracic region of the occupant, a lower deformation resistance is chosen than for the pelvic region. Crash and crush tests on experimental door assemblies (Landheer, 1991) indicated the characteristics shown in Figure 6.3 to be efficient. These characteristics will be applied here.

Deformation Mode B

A certain space between the driver and the side structure is required to enable comfortable driving. This gap implies a delay in the lateral acceleration of the occupants in a side impact. Consequently, for the first 20 milliseconds of the crash the pelvis is not loaded in Deformation Mode A. If these 20 milliseconds are added to the occupant acceleration interval, lower acceleration levels are sufficient to realize the required change of momentum for the occupant. In Deformation Mode B this option is exploited. Severy (Severy, 1976) suggested the positive effect of a moving seat in side impact crashes. In Deformation Mode B a lateral pre-acceleration of the occupant by its seat is examined. This pre-acceleration is achieved by a rotation of the seat (see Figure 6.5). This rotation causes an increased normal force between the backrest and the occupant. The increased normal force brings about a lateral acceleration of the occupant by means of a friction force. As an additional positive effect of the seat rotation during the initial phase of a crash, the seat backrest is positioned in between the intruding side structure and the occupant and hence, can act as a protective shield. To enlarge the transfer of momentum between seat and occupant, the backrest of the seat is equipped with padded lateral supports transforming it into a bucket seat.

6.4 The deformation modes numerically modelled

point groups

The deformation modes imply that the following parts of the vehicle must form rigid moving links in a crash:

- The vehicle side in the occupant contact area (mass M_2)
- The seat (in Mode A to avoid it penetrating the intruding vehicle side) (mass M_8)
- The roof, floor and non-struck side of the vehicle (mass M_6)

Consequently, the relative movement of units within these parts must be restrained in the numerical model. This is achieved by including the corresponding spatial points in rigid body definitions. In Figure 6.4, the planned relative movement of these rigid bodies is visualised for Deformation Mode A. In the figure, the points on which a velocity time function will be imposed to bring about the intended relative movement of the links that contact the occupant, are marked with a number. Figure 6.5 illustrates Deformation Mode B.

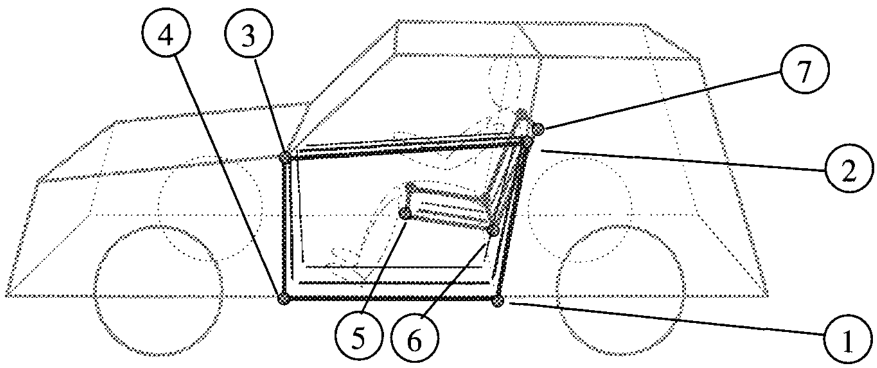


Figure 6.4 *Illustration of Deformation Mode A (the points on which a velocity time function is imposed are marked with a number).*

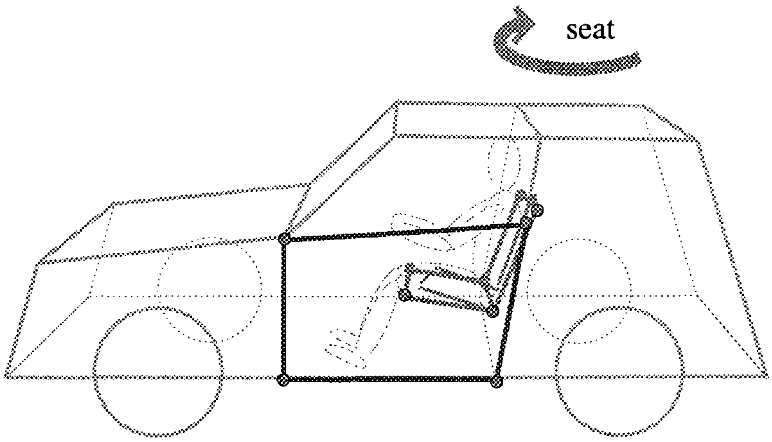


Figure 6.5 *Illustration of Deformation Mode B.*

To avoid unnecessary high load concentrations, the following heavy parts are defined as separate links (see Figure 6.6):

- The traction system (mass M_9, 241 kg)
- The left front-wheel suspension (mass M_{10}, 38 kg)
- The right front-wheel suspension (mass M_{11}, 38 kg)
- The rear-wheel suspension (mass M_{12}, 58 kg)

This allows a delay to be imposed upon the movement of these links, relative to the movement of the rest of the non-struck vehicle body.

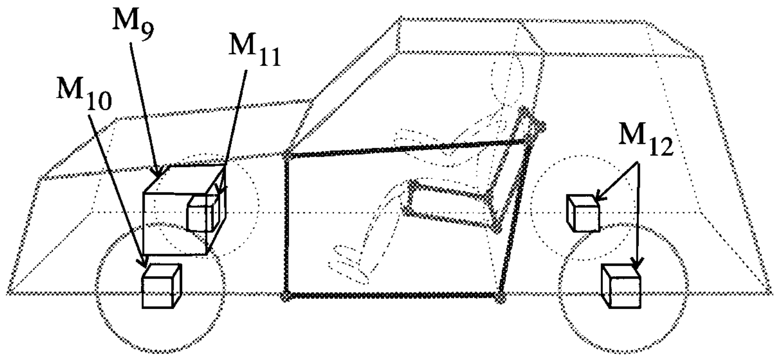


Figure 6.6 *Visualization of : - The traction system (M_9); - The left front-wheel suspension (M_{10}); - The right front-wheel suspension (M_{11}); - (M_{12}) The rear-wheel suspension.*

An estimate for the mass of the vehicle side hit by the EEVC barrier is found with Equation 2.6:

$$M_2 = 0.033 \cdot 850 + 7.3 = 35.35 \text{ kg} \tag{6.1}$$

The mass of the seat (M_8) in the model is chosen to be 10 kg (20 kg for the rotating seat). The mass of the remaining part of the vehicle can be calculated by deducting the known mass of the other links from the total vehicle mass as given in Equation 6.2:

$$M_6 = 850 - M_2 - M_8 - M_9 - M_{10} - M_{11} - M_{12} = 429 \text{ kg} \tag{6.2}$$

A simple numerical model of the driver seat and vehicle side structure, i.e. the parts that contact the occupant, is generated to determine the effectiveness of the proposed deformation mode. The seat model on which the EuroSID model is positioned consists of two rigid flat surfaces defined between six so-called "contour points". Between four other spatial points—the lower two at door sill height, the upper two at waist-line height—a rigid flat vehicle side that will contact the occupant model is defined as shown in Figure 6.4. The deformation characteristics as specified in Figure 6.3 have been taken into account via sliding interface definitions for the parts of the dummy-model with regard to the intruding flat vehicle side. For the seat, a penetration-response is defined in a similar way.

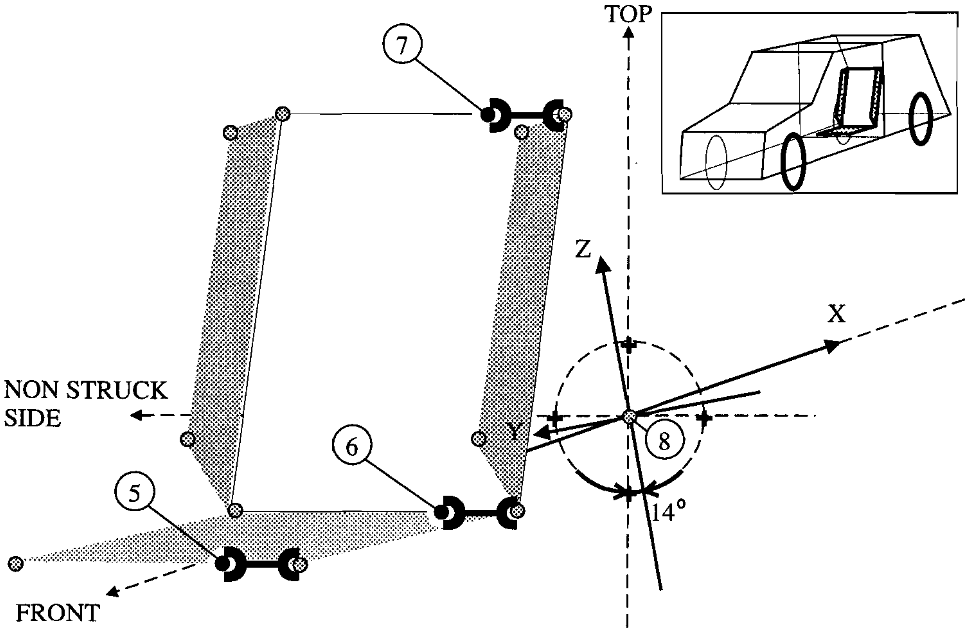


Figure 6.7 *Model of the moving seat in deformation mode B, on the numbered points a velocity time function is imposed.*

Lateral supports are added to the seat as illustrated in Figure 6.7 to simulate the bucket seat in Deformation Mode B. All grey points in the figure are part of the rigid body definition for the seat. On point 8, a boundary condition is applied that restrains all translations not in Y-direction (see Figure 6.7), and the rotation around the Y-axis, of the seat (M_8). Velocity time functions imposed on the points 5, 6 and 7 are transferred to the seat via beam elements. The accompanying loads in these beams, necessary to move the seat with the occupant, are stored during the calculation. With the boundary condition for point 8 and the velocity imposition for the points 5, 6 and 7 the movement of the seat is fully laid down.

velocity-time functions

For the imposition of the relative movements of the rigid links the following velocity-time functions are chosen:

- The vehicle's side structure (M_2) is driven via the four points 1 to 4 (see Figure 6.4). Each of these points is imposed a nodal velocity boundary condition in lateral direction, towards the non-struck side of the vehicle. The functions are chosen such that no twist of the vehicle side will occur since this would be in conflict with the rigid body definition for the side. A plot of the functions is given in Figure 6.8.

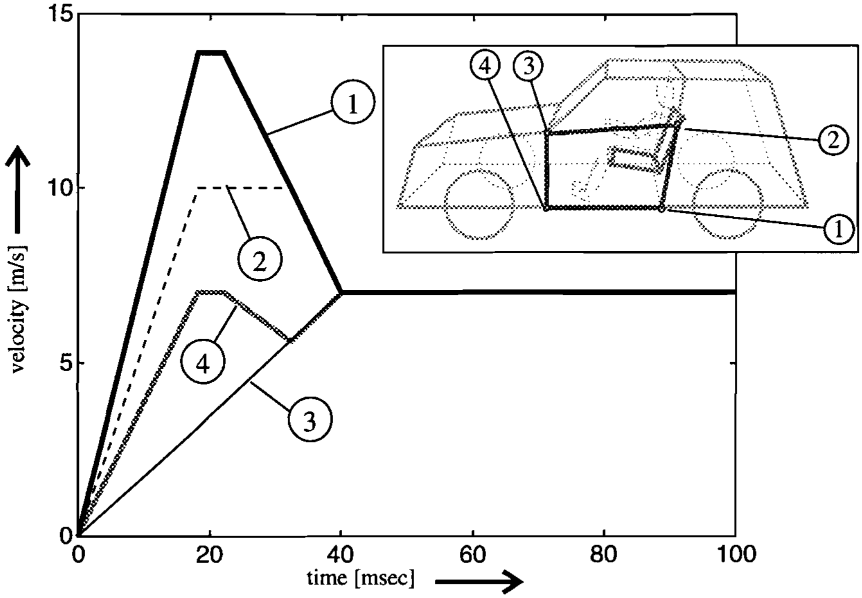


Figure 6.8 *The velocity-time functions for the vehicle side.*

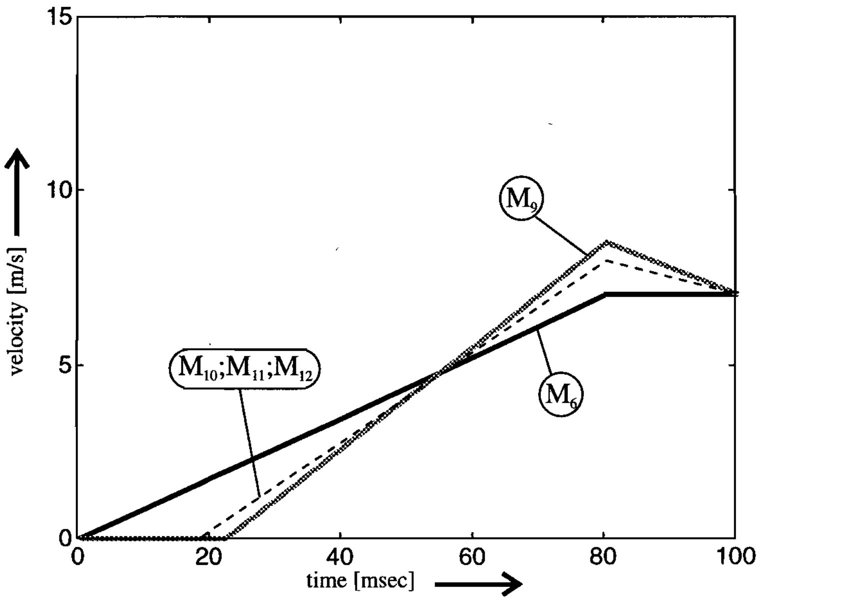


Figure 6.9 *Functions for: - the roof, the floor and the non-struck side of the vehicle (M_6);- the traction system (M_9);- the wheel suspension (M_{10},M_{11},M_{12}).*

- The lateral velocity of the roof, the floor and the non-struck side of the vehicle (M_6) must coincide with the lateral velocity of the struck side, at the location of the imaginary hinge at roof rail height. For the relatively heavy wheel-suspension and traction system a delay in the movement is imposed. A plot of the accompanying velocity time functions is given in Figure 6.9.

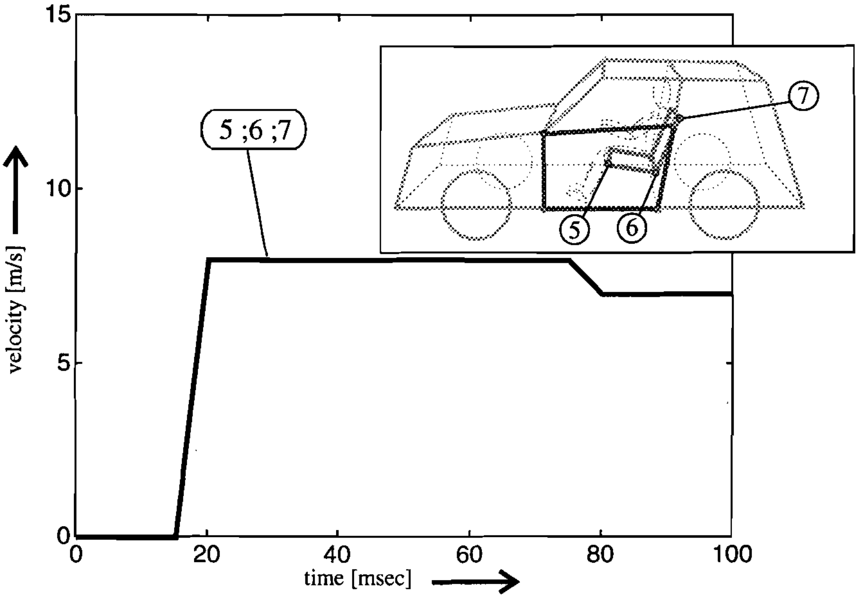


Figure 6.10 The velocity-time function for the seat in Deformation Mode A.

- For the seat, a distinction exists between the velocity-time functions in Deformation Mode A and the functions in Deformation Mode B. Figure 6.10 gives the velocity-time function for the pure lateral translation of the seat in Deformation Mode A. To achieve the seat rotation in Deformation Mode B different lateral velocities for each of the driving points 5, 6 and 7 (see Figure 6.7) are imposed. These velocity-time functions are given in Figure 6.11.

conservation of momentum

The deformation modes must be dynamically realistic. This implies that the modes must be made consistent with the law of conservation of momentum. This can be achieved by choosing the correct velocity-time function for the barrier. A large freedom exists for the choice of this function. However, a too slow ride down of the barrier will result in the penetration of the rigid barrier surface (that supports the deformable front-structure) through the vehicle side. This is physically impossible. By the incorporation of the barrier in the model on which the calculated velocity-time function is imposed, possible penetrations will be visualised in animations.

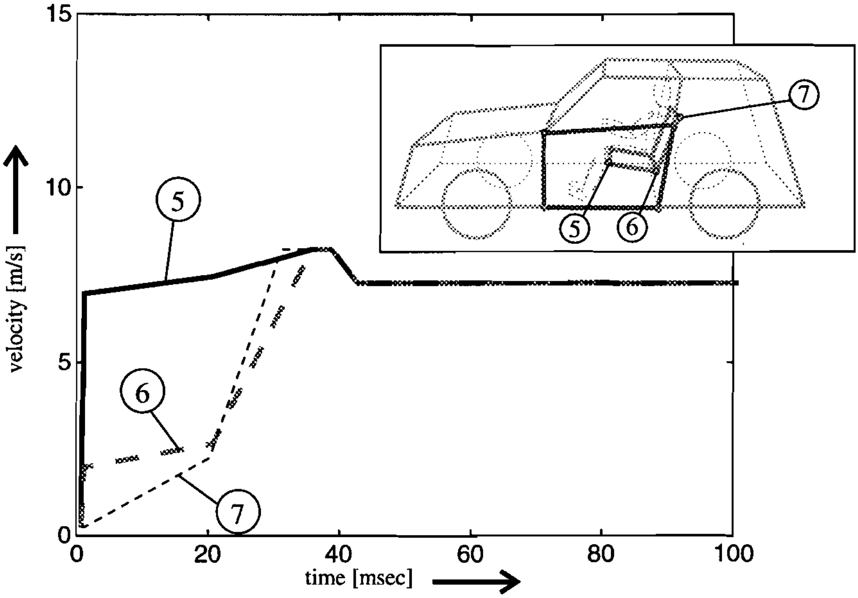


Figure 6.11 The velocity-time functions for the seat rotation in Deformation Mode B.

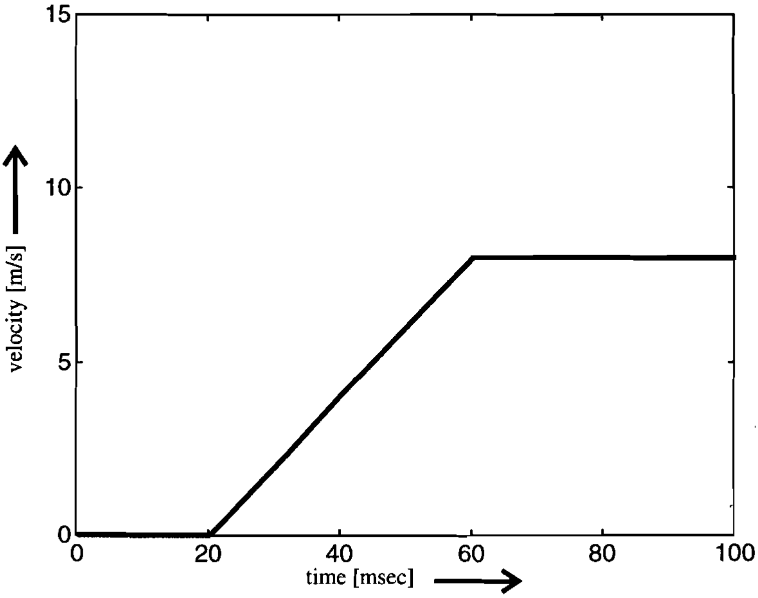


Figure 6.12 An estimate of the velocity of the loaded dummy.

In this example, only the conservation of momentum in lateral direction will be discussed. Before the time dependent distribution of momentum can be calculated, first an estimate of the dummy velocity is made (see Figure 6.12). Unlike the other links, the struck vehicle side (M_2) and the seat (M_8) move in a not pure lateral fashion. For the calculation of their share in the lateral impulse balance, the total mass of these links is distributed over the actuation-points of the concerned rigid bodies. The applied mass distribution is given in Table 6.1:

actuation point	1	2	3	4	5	6	7
mass [kg]	6	12	12	6	5	10	5

Table 6.1 Mass distribution of the vehicle side and the rotating seat for the calculation of their share in momentum.

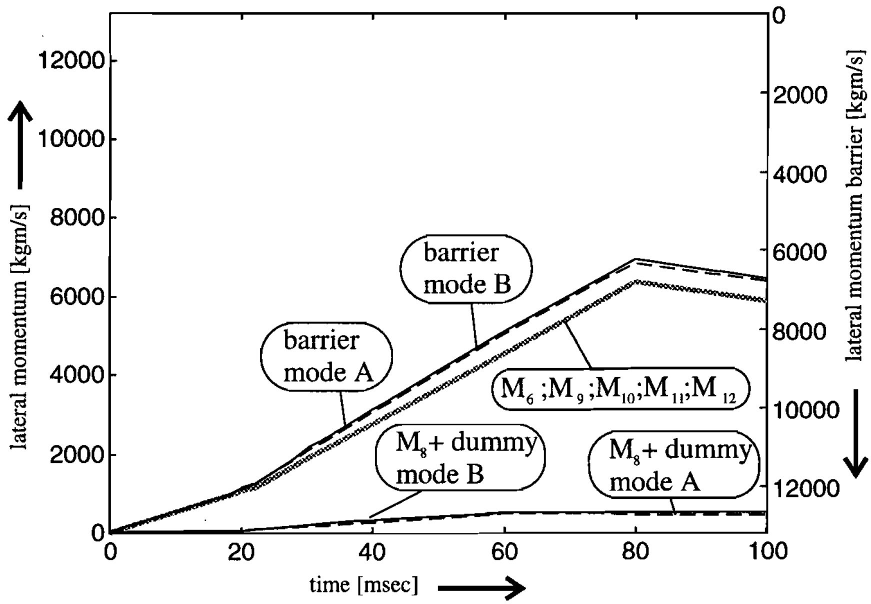


Figure 6.13 The distribution of momentum.

In Figure 6.13, the lateral momentum of:

- the dummy and the seat (M_8)
- the struck vehicle without seat (i.e.: M_6; M_9; M_{10}; M_{11}; M_{12})
- the total system without barrier
- the barrier

is given for both deformation modes A and B. It can be seen that the difference in distribution of lateral momentum between the two deformation modes is small. From the momentum curves for the barrier, the expected barrier velocity is calculated for the two deformation modes. The velocity-time functions, given in Figure 6.14, are used in the

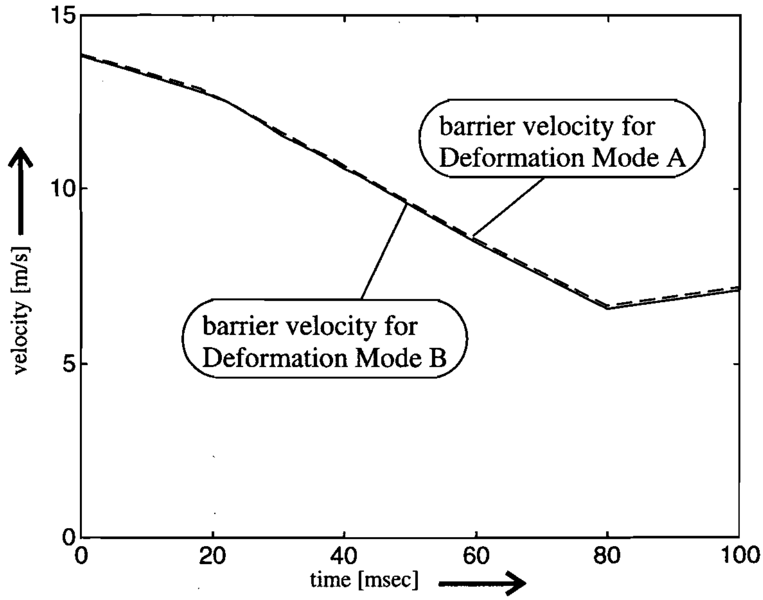


Figure 6.14 The velocity-time functions for the barrier.

crash-simulations.

6.5 Performance

The results of simulations with the discussed model settings are given in this section. The figures 6.15 to 6.18 show sequential states of the Deformation Mode A on the left and of Deformation Mode B on the right. The penetration of dummy parts into the flat side, as shown in the figures, represents the contact deformation.

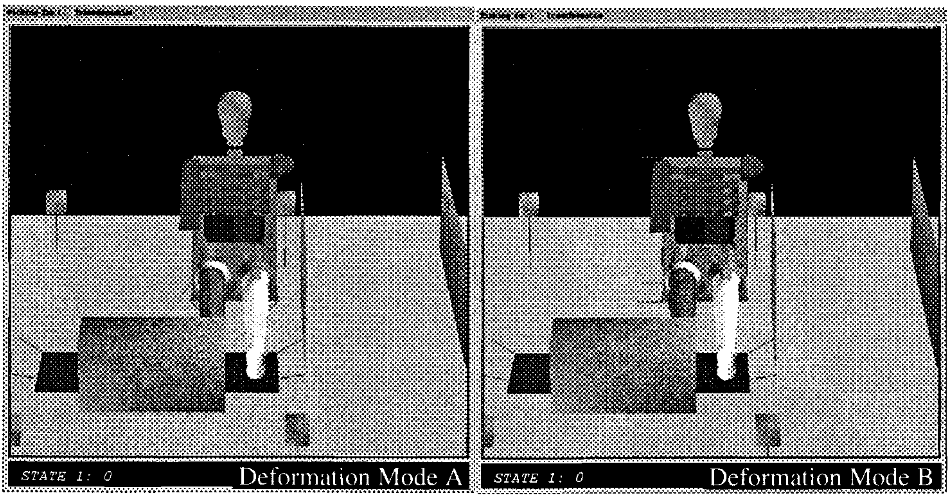


Figure 6.15 Initial state (0 msec).

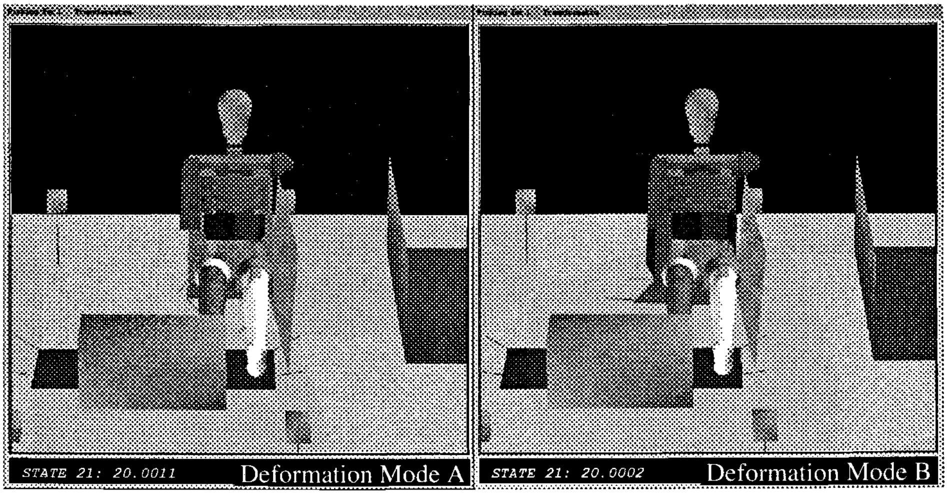


Figure 6.16 State after 20 msec.

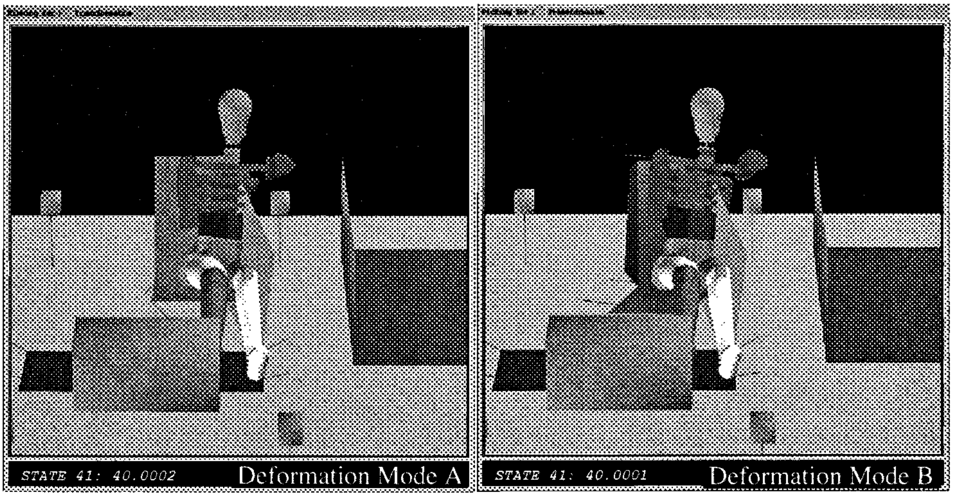


Figure 6.17 State after 40 msec.

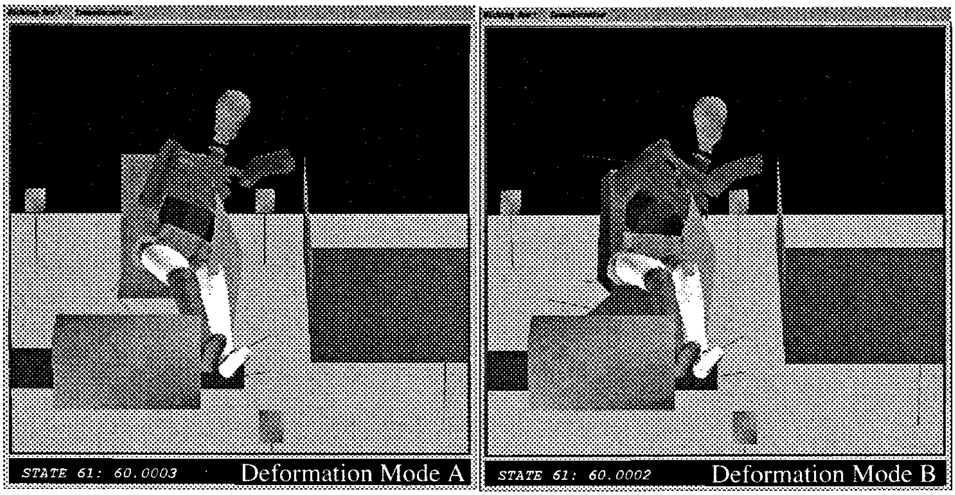


Figure 6.18 State after 60 msec.

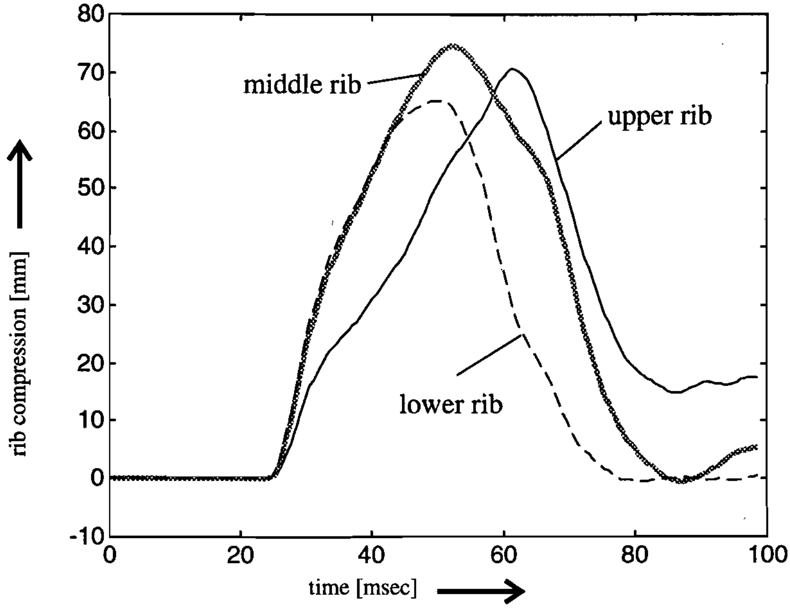


Figure 6.19 The rib-compression as a function of time for Mode A.

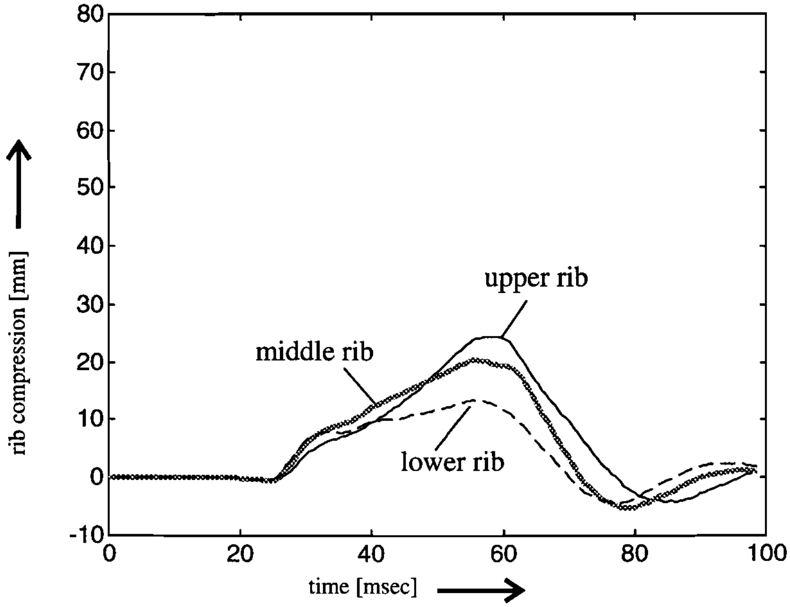


Figure 6.20 The rib-compression as a function of time for Mode B.

The compression in time of all three ribs of the EuroSID is given in Figure 6.19 and Figure 6.20. The maximum rib-deflection and the Viscous Criterion that are restricted in the european side impact safety requirement are established with this data. The performance of the modes on these criteria is given in Table 6.2:

mode	A	B	rib	limit
rib-deflection [mm]	75 (at 53 msec)	24 (at 58 msec)	middle upper	42
Viscous Criterion [m/s]	1.1 (at 58 msec)	0.16 (at 52 msec)	upper	1

Table 6.2 Rib loads for Mode A and Mode B.

With Deformation Mode B, the rib-compression related requirements are fulfilled while Deformation Mode A fails.

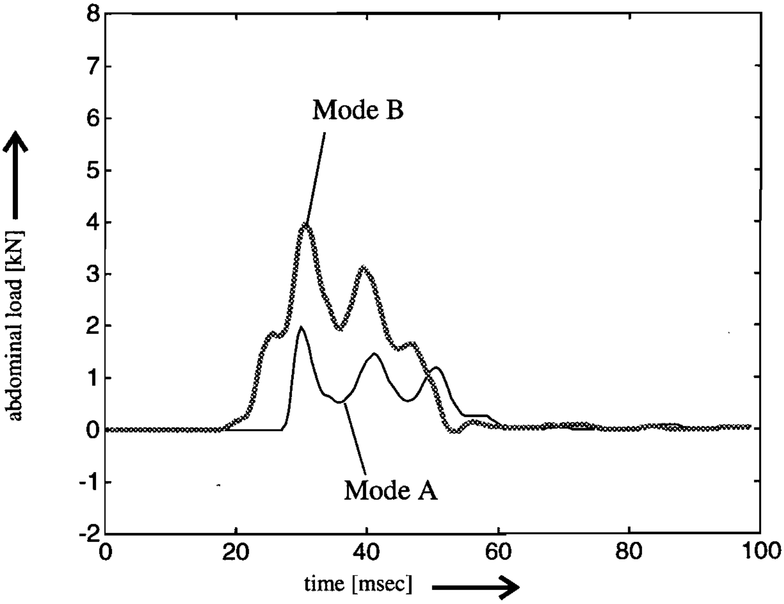


Figure 6.21 The abdominal load.

Figure 6.21 gives the abdominal force as a function of time. The peak value of this force, restricted in the Abdominal Performance Criterion, is given in Table 6.3:

mode	A	B	limit
APC [kN]	2.0 (at 30 msec)	4.0 (at 31 msec)	2.5

Table 6.3 The APC for Mode A and Mode B.

On the APC, Deformation Mode B fails, while Deformation Mode A passes.

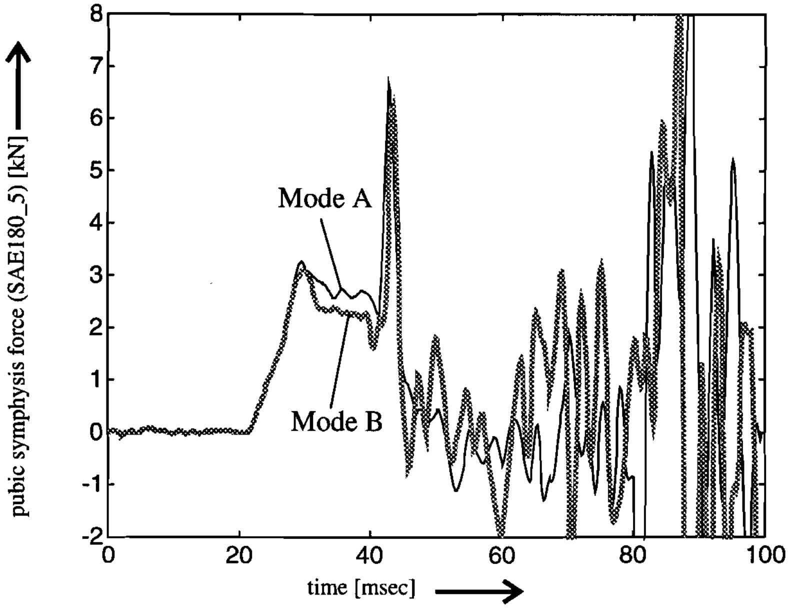


Figure 6.22 The pubic symphysis force.

A plot of the calculated pubic symphysis force as a function of time is given in Figure 6.22. A strong oscillation starts around 40 milliseconds after the start of the simulation. At this time, the pelvis reaches the same lateral velocity as the intruding vehicle side. From this perspective, a gradual decrease of the pubic symphysis force after 40 milliseconds could be expected. An error in the damping characteristics of the pelvic region of the dummy model however, could cause the oscillation-problem. Considering only the first 40 milliseconds of the simulation, the Pubic Symphysis Peak Force stays below the legal limit of 6 kN. Table 6.4 gives the values for the Pubic Symphysis Peak Force within this time interval:

mode	A	B	limit
PSPF [kN]	3.3 (at 29 msec)	3.1 (at 30 msec)	6

Table 6.4 The PSPF for Mode A and Mode B.

Besides the unexpected oscillations, another problem with the dummy model came out with the simulations. Finite elements within the dummy appear to be decisive for the time-step that is used in the calculations. Unfortunately, this time-step appears to be much smaller for Deformation Mode B than for Deformation Mode A (see Figure 6.23).

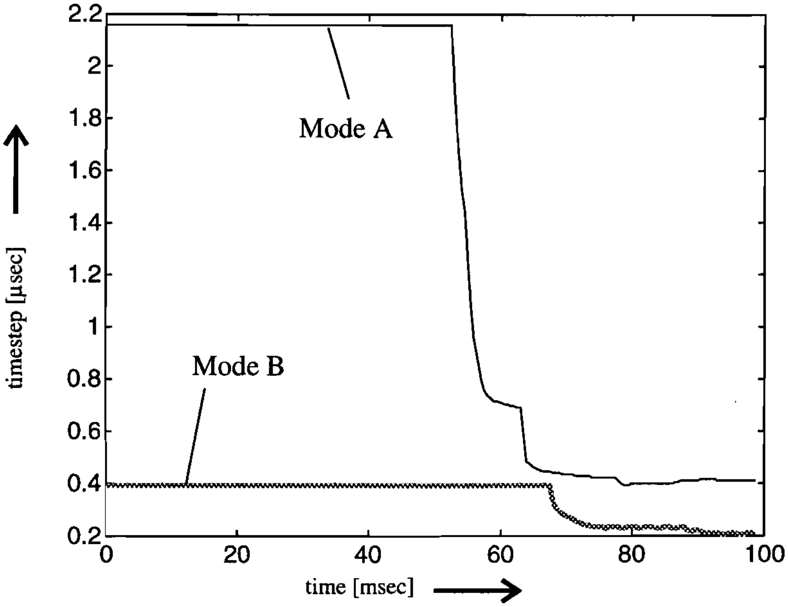


Figure 6.23 The time-step.

This implies that for this particular mode the time required to establish the maximum dummy-load levels is about a factor 6 more than the 3 hours stated in Engineering Requirement 22. Moreover, an instability occurs in the finite elements representing the rib-foam at about 60 milliseconds. This causes an unwanted reduction of the time-step. Consequently, it is recommended to investigate alternatives for the applied model of the EuroSID because it forms the bottle-neck for the calculation-time. Nevertheless, it is illustrated that the proposed design method enables the early comparison of safety concepts. The crash simulation models used for the comparison are based on main vehicle dimensions only, that are available already in the concept-stage of design. The geometric input used to build the models consists of the point groups combined with a few contact surfaces only. The estimate of the inertia properties is made with data available from existing vehicle designs. No further structural data was required to compare the performance of the deformation modes.

Chapter 7

Concluding remarks and recommendations

7.1 Conclusions

The protection a vehicle offers its occupants in a crash is a result of the vehicle mass, the deformation properties of the vehicle body structure and, the properties of the occupant restraint system. The deformation properties of the vehicle body structure arise from its geometric design, combined with the mechanical properties of the applied materials. The design of vehicle body structures such that crash-loads will result in a pre-programmed deformation, is addressed in this research project. A design approach making use of numerical modelling techniques is proposed. The level of geometrical detail of the applied model is increased step-by-step in this approach. From the research, several conclusions can be drawn:

- Simple kinematic models enable the validation and comparison of so-called "Deformation Modes" for vehicle structures in an early stage because only main vehicle dimensions are required as input.
- From the kinematic models, functions can be derived for the parts of a vehicle structure to cover the Deformation Modes. Parts of the structure that are intended to deform and parts intended to maintain their original shape in a crash are distinguished. The identification of these parts forms the basis for the generation of rapid analysis models.
- The functions can be quantified in deformation properties for structural elements that are intended to deform in a crash. The kinematic models can be expanded to mechanical models by representing the structural elements that are intended to deform with elements of so-called "1D material". Such mechanical models represent safety concepts. Numerical simulations can be carried out to validate the sensitivity of a safety concept to variations in the severity of a crash load and, to calculate the load levels that will occur within the structure.
- The functions, along with the calculated load levels, can direct the design of a suitable shape for the crash load transferring parts of the structure.
- With hybrid analysis-models based on the original mechanical model, these shapes can be evaluated before other parts of the vehicle are designed in detail.

- To quickly evaluate the effect of changes in already completed vehicle designs, the originally generated hybrid model can be used again. The model only has to be adjusted in conformance with the design-changes. This makes obsolete the usual reduction of a detailed FEM-model.

Thus, a sound safety concept can be identified and integrated into the vehicle design from scratch. This will increase the development efficiency because variations in a vehicle body concept can have much more impact on passive safety than a detailed optimisation of an inferior safety concept later in the development process. However, the FEM-model of the EuroSID that is applied in the project leads to an unwanted high calculation time for particular deformation modes.

Furthermore, it can be concluded from the comparison of two Side Impact Deformation Modes discussed as an example that:

- A pre-programmed rotation of the seat in the initial stage of a side impact can offer a substantial contribution to the occupant protection.

7.2 Recommendations

The loads acting on beam elements in a numerical model can be expressed in proportion to the characteristic load capacity for the given beam cross-section. These so-called "weighted loads" were presented in this project as curves. However, it would be very useful if the weighted load levels could be projected on the animation plots generated in the post-processor in the form of a colour code. The engineer would have a direct overview on the safety margin of his vehicle body design. Therefore, it is recommended to expand the post-processor software with this option.

Although with the applied FEM-model of the EuroSID all relevant dummy loads can be predicted in the models of increasing detail-level discussed in this report, it is recommended to investigate alternatives for this model because it forms the bottle-neck for the calculation-time in side impact simulations.

Appendix A

The QFD analysis

A.1 The Quality Function Deployment (QFD) technique

To efficiently generate a design, it is essential to have the design problem properly defined. Although, in the present case, a design method itself and not a physical product must be designed, a good understanding of the problem is the basis for the solution as well.

In Japan, the Quality Function Deployment method was developed by Toyota and its suppliers in the mid-1970s as an aid to analyze design problems. The method consists of the following six steps:
Step 1 Identifying the customer(s)
Step 2 Determining the customer requirements
Step 3 Determining the relative importance of the requirements
Step 4 Competition benchmarking
Step 5 Translating customer requirements into measurable engineering requirements
Step 6 Setting engineering targets for the design
More detailed descriptions of the method can be found in Hauser (1988) and Ullman (1992).

The QFD method as introduced by Hauser is primarily applied for design problems concerning product redesign. Hence, if a new product has to be designed from scratch not all steps are executable as described by Hauser. This is specially true with regard to competition benchmarking in Step 4, for a completely new product. In that situation benchmarking can only be performed on specific functions of the product. Nevertheless, the main lines of the method form a suitable basis for the development of new products as well. Although no physical product but a method must be designed, the QFD method is applied now step-by-step to analyze the stated problem. To avoid mixing up the customer with the buyer of a car, the term *user* of the method is applied here instead of *customer.*

A.2 The QFD technique applied to the indicated design problem

A.2.1 The identity of the user

All engineers involved in the development of a new car will benefit from a good understanding of the crash protection related functions of its separate components. This will facilitate the communication with crashworthiness specialists within the engineering and testing departments and, therefore, increase the vehicle quality. However, those involved with the design of the vehicle body must be regarded as the users or customers of the method to be developed. It is common practice that representatives of the analysis department, familiar with numerical simulation techniques, join a vehicle development project team. The analysts, with their skills to handle numerical analysis, can then support the designers. Hence, the analysts can be seen as users as well.

A.2.2 The user requirements

To establish the user requirements, the following procedure has been followed. First, a literature study has been carried out to access the state of the art in crashworthiness design and passive safety validation. Second, several representative users from NEDCAR and VW have been consulted (Landheer, 1992a; Landheer, 1993). Third, because a lot can be learned from imperfect designs, engineers involved with crash testing have been questioned as well (Landheer, 1992b). Fourth, attention is given to the biomechanic aspects of the crashworthiness design problem by consulting specialists in this particular field. This effort has resulted in the following list of user requirements:

- The method must be applicable already in the primary vehicle design phases in which vehicle body concepts are generated
- The method must provide the possibility to compare the protection capacity of safety concepts through all stages of vehicle design
- The method must quickly demonstrate the effect of load condition alterations
- The method must quickly demonstrate the effect of design alterations
- A clear relation between models employed and the construction geometry must exist to ease the interpretation of crash simulations
- The model accuracy must have such a level that with the aid of the simulation results it can be guaranteed that a specific safety potential (as for instance specified in the regulations discussed in Section 2.4) is available
- The method must be compatible with the existing engineering design process

To get a better overview, the list of requirements will now be organised according to the product aspects they affect. The requirements are reformulated making them more specifically oriented towards the product aspects. Most requirements do have impact on more than a single product aspect. The user requirement labels <u.r. number> in the following list correspond with the labels in Table A.1.

product aspect: <u>Geometric vehicle model generation</u>

- u.r.1 Course geometry definitions must be feasible[13]
- u.r.2 Different detail levels within one geometric vehicle description must be feasible
- u.r.3 Adaptation of the geometry must go quickly
- u.r.4 A geometry and materials definition must be possible in such detail that a certain passive safety potential can be guaranteed
- u.r.5 The geometric model generation must be compatible with all phases within engineering design processes

product aspect: <u>Analysis model generation</u>

- u.r.6 A course analysis model definition must be feasible[14]
- u.r.7 It must be possible to introduce external loads[15]
- u.r.8 Analysis model generation must be quick
- u.r.9 It must be possible to take into account inertia effects of not yet modelled construction parts[16]
- u.r.10 It must be feasible to include different levels of detail within one model[17]
- u.r.11 Adaption of a model must be quick
- u.r.12 It must be possible to translate an (adjusted) analysis model into a geometric vehicle model
- u.r.13 A geometry and materials description must be possible in such detail that a certain passive safety potential can be guaranteed

product aspect: <u>Load definition</u>

- u.r.14 Crash condition related load must be generated with common tools

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[13]Detail level comparable with that of a draft.

[14]The analysis modelling technique must be compatible with the course geometric detail level as mentioned in u.r.1.

[15]With external loads, loads introduced by the contact face of a loading device are meant.

[16]In the primary vehicle design stages many parts of the vehicle are not yet geometrically defined. However, from experience with previous designs a good estimate of their mass can be made. This data must be implementable in the analysis model.

[17]In a vehicle development process the vehicle parts are not designed completely parallel in time. Hence, it must be possible to describe vehicle parts in a detail level that depends on the stage of development they are in, within one vehicle analysis model.

u.r.15 Alteration of load definitions must go quickly

product aspect: (numerical) Analysis

u.r.16 The analysis software must be capable of analysing models containing coarse as well as detailed geometric data

u.r.17 A crash calculation must be quick

u.r.18 The accuracy of the calculations must be such that the safety potential can be guaranteed with certain confidence

product aspect: Post processing

u.r.19 The deformational behaviour must be visualised in a way that vehicle parts in study can be recognized

u.r.20 It must be possible to compare the protection level of a construction graphically with that of another design variant

u.r.21 The performance on safety criteria must be determined quickly[18]

u.r.22 It must be possible to display the shape of a deformed construction

A.2.3 Arrangement of the requirements

Not all the requirements summarised in the preceding section are of the same importance. Some requirements must be fulfilled to make a crashworthiness integration method applicable. These requirements are indicated as so-called **"musts"**. Five of the user requirements are musts as will be explained now.

User Requirement 1 (u.r.1): Course geometry definitions must be feasible

The method is intended to be applied already in the primary stages of vehicle design in which body structure concepts are generated. In these primary design stages, only macroscopic dimensions as for instance wheel base, seat reference point location, waist line height and total length of the vehicle are set. The shape of the body's local cross-sections depends on design variables not yet fixed, such as the location of door hinges, the dimensions of the interior padding and trim and, the dimensions of the exhaust-pipe et cetera. Hence, detailed geometry description forced by the method in the primary design stages would make the user spend time which may later appear to be superfluous. For that reason the method must allow a course geometry description.

User Requirement 6 (u.r.6): A course analysis model definition must be feasible

To investigate the influence of variations in geometric design variables on the crash behaviour of a particular design, repeated generation of analysis models is required. Hence, the primary analysis models must be compatible with the detail level indicated in

[18] Safety criteria are for instance a HPC or a maximum intrusion.

User Requirement 1. Otherwise, the user is forced to put in so much effort to translate the quickly generated coarse geometric models into analysis models that, due to time pressure, iterations to tune the design variables are omitted. However, a not optimal value chosen for a design variable in this stage of design can hardly be corrected later on. For that reason it is essential for the method that User Requirement 6 is met and correspondingly this requirement is marked as must.

Users Requirement 7 (u.r.7) It must be possible to introduce external loads
Crash loads are always introduced via a load transferring contact area. Hence, it must be possible to analyze this contact development in the method.

Users Requirement 10 (u.r.10) It must be feasible to include different levels of detail within one model
In a vehicle development process often vehicle parts are designed not all at the same time. Hence, in a complete vehicle analysis model it must be possible to describe the separate vehicle parts in a detail level that depends on the stage of development they are in. Otherwise, a more detailed part can only be analyzed separately from the less detailed part of the vehicle. This would imply the definition of complex boundary conditions and the interpretation of the results projected on the full vehicle.

Users Requirement 22 (u.r.22) It must be possible to display the shape of a deformed construction
Geometric information is the central input variable that must be tuned following the method. Hence, the deformed vehicle shape is the primary geometry related output as fits best in the imagination of the designer. Besides, this geometric information contains all data necessary to generate interesting deformation related details. Consequently User Requirement 22 is marked as must.

The remaining requirements are so-called **"wants"**. The better these wants are fulfilled, the more convenient the developed method will be. However, sometimes "want" combinations claim contradictorily design measures. In other words, improving the performance of the method on one requirement implies a decrease in performance on another requirement. In such a situation, the priority must be given to the most important wants. Hence, in this step of the QFD method weight factors are established, in order to express the relative importance of the wants. A pairwise comparison technique will be used here. By comparing the requirements two at a time, giving the more important a score 1 and the other a 0, a rating can be realized. Formula (A.1) gives the amount of combinations when N is the number of "wants".

$$\textbf{Amount of combinations} = \frac{N \cdot (N - 1)}{2} \tag{A.1}$$

For the current problem, with N = 17, a total of 136 combinations can be made. In Table A.1 the result of application of the comparison technique is presented. As can be

	u.r.2	u.r.3	u.r.4	u.r.5	u.r.8	u.r.9	u.r.11	u.r.12	u.r.13	u.r.14	u.r.15	u.r.16	u.r.17	u.r.18	u.r.19	u.r.20	u.r.21	%
u.r.2	≠	1	0	1	1	1	0	1	0	0	1	1	0	0	0	0	0	5
u.r.3	0	≠	1	1	0	1	1	0	0	0	0	1	1	0	0	0	1	5
u.r.4	1	0	≠	1	1	0	1	1	1	1	1	1	1	0	0	0	1	8
u.r.5	0	0	0	≠	1	0	0	1	0	0	1	1	0	0	0	0	0	3
u.r.8	0	1	0	0	≠	0	1	1	0	1	1	1	1	0	0	0	0	5
u.r.9	0	0	1	1	1	≠	1	1	1	1	1	1	1	0	0	0	1	8
u.r.11	1	0	0	1	0	0	≠	0	0	1	1	0	1	0	0	0	0	4
u.r.12	0	1	0	0	0	0	1	≠	0	1	1	0	1	0	0	0	0	4
u.r.13	1	1	0	1	1	0	1	1	≠	1	1	1	1	1	1	0	1	10
u.r.14	1	1	0	1	0	0	0	0	0	≠	1	1	1	0	0	0	1	5
u.r.15	0	1	1	0	0	0	0	0	0	0	≠	1	1	0	0	0	0	3
u.r.16	0	0	0	0	0	0	1	1	0	0	0	≠	0	0	0	0	1	2
u.r.17	1	0	0	1	0	0	0	0	0	0	0	1	≠	0	0	0	0	2
u.r.18	1	1	1	1	1	1	1	1	0	1	1	1	1	≠	0	0	1	10
u.r.19	1	1	1	1	1	1	1	1	0	1	1	1	1	1	≠	0	1	10
u.r.20	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	≠	1	12
u.r.21	1	0	0	1	1	0	1	1	0	0	1	0	1	0	0	0	≠	5

Table A.1 *Pairwise comparison of the wants (in the grey cells user requirement numbers are given, "musts" have been left out).*

seen, only the user requirements that are wants are included in the table. The total scores are expressed as percentages of 136.

The pairwise comparison technique indicates User Requirement 20 (≠It must be possible to compare the protection level of a construction graphically with that of another design variant) as the most important "want".

A.2.4 Competition benchmarking

Benchmarking can now be performed to check an existing solution on its capabilities to fulfil the requirements. For each customer requirement the solution will be rated on a scale from 1 to 5, where

1 = the solution does not meet the requirement at all.
2 = the solution meets the requirement slightly.
3 = the solution meets the requirement to some extent.
4 = the solution meets the requirement to a large extent.
5 = the solution fulfils the requirement completely.

It is common practise that with a detailed FEM model containing in the order of 100.000 elements it is possible to realistically simulate the deformation process of a crashing vehicle. However, the required calculation time of more than a day on super computers for such 100,000 element models has resulted in the general application of model simplifications based on foreknowledge of the vehicle deformation process. As an example the results of a benchmark carried out with a 20,000 element FEM model for a side impact simulation will be shown here. In Table A.2 the score of the solution is given per requirement. By multiplying the scores with the weight factors established in the preceding section (see the column with the % header in Table A.1) and summing up the results for all "wants" a total score for the solution is generated. Concerning the "musts" the following notation is used in the weighted score column: - fail; + pass; ± potential to pass after improvements have been made.

u.r. No	user requirement	weight factor	FEM score	FEM score weigh-ted
1	Course geometry definitions must be feasible	*	1	-
2	Different detail levels within one geometric vehicle description must be feasible	5	2	10
3	Adaptation of the geometry must go quickly	5	1	5
4	A geometry and materials definition must be possible in such detail that a certain passive safety potential can be guaranteed	8	5	40
5	The geometric model generation must be compatible with all phases within engineering design processes	3	2	6
6	A course analysis model definition must be feasible	*	1	-
7	It must be possible to introduce external loads	*	5	+
8	Analysis model generation must be quick	5	3	15
9	It must be possible to take into account inertia effects of not yet modelled construction parts	8	5	40
10	It must be feasible to include different levels of detail within one model	*	3	±
11	Adaption of a model must be quick	4	2	8
12	It must be possible to translate an (adjusted) analysis model into a geometric vehicle model	4	5	20
13	A geometry and materials description must be possible in such detail that a certain passive safety potential can be guaranteed	10	5	50
14	Crash condition related load must be generated with common tools	5	4	20
15	Alteration of load definitions must go quickly	3	2	6
16	The analysis software must be capable of analyzing models containing coarse as well as detailed geometric data	2	1	2
17	A crash calculation must be quick	2	1	2
18	The accuracy of the calculations must be such that the safety potential can be guaranteed with certain confidence	10	5	50
19	The deformational behavior must be visualized in a way that vehicle parts in study can be recognized	10	5	50
20	It must be possible to compare the protection level of a construction graphically with that of another design variant	12	5	60
21	The performance on safety criteria must be determined quickly	5	3	15
22	It must be possible to display the shape of a deformed construction	*	5	+
	Total	100%		399

Table A.2 Benchmark results for fine mesh FEM modelling (musts are marked by an asterix (*) in the weight factor column).

The benchmarking results show that a good performance is achieved in particular on the post processing aspect of the problem. It is important to realise that the total weighted score with a maximum of 5· 100 = 500 is not more than an indication of the fulfilment of "wants". As can be seen the "FEM" solution fails on Must 1 and Must 6 as expected. Besides, the foreknowledge of the vehicle deformation process used to restrict the amount of elements does not exist for new vehicle concepts. Consequently, fine mesh FEM modelling is no solution to our design problem.

A.2.5 Measurable engineering requirements

In Step 5 of the QFD technique engineering requirements have to be formulated. In contrast to most of the customer requirements these requirements must be quantitatively measurable. They can be used evaluating various design concepts. A list of 32 engineering requirements labelled <e.r. number> is given below. The units for the engineering requirements are included as well. Targets for the engineering requirements will be formulated in Step 6 of the QFD technique as discussed in Section A.2.6. However, “available” respectively “possible” are the inherent targets for the direct “available/not available” and ”possible/impossible” discriminating requirements. Hence, the targets for these requirement types are already known in Step 5 of the QFD technique.

e.r.1	point modelling must be available	(available/not available) [-]
e.r.2	wire frame modelling must be available	(available/not available) [-]
e.r.3	surface modelling must be available	(available/not available) [-]
e.r.4	solid modelling must be available	(available/not available) [-]
e.r.5	geometry input must be graphically displayed	(possible/impossible) [-]
e.r.6	geometry input must be graphically displayed on line	(possible/impossible) [-]
e.r.7	time required for first geometric model generation	[hours]
e.r.8	time required to adapt a geometric model	[hours]
e.r.9	number of data conversions to create an analysis model[19]	[-]
e.r.10	time required to generate an analysis model	[hours]
e.r.11	a concentrated load definition option must be available	(available/not available) [-]
e.r.12	a spread load definition option must be available	(available/not available) [-]
e.r.13	definition of loads as function of time must be possible	(possible/impossible) [-]
e.r.14	an inertia load definition option must be available	(available/not available) [-]
e.r.15	number of detail levels	[-]
e.r.16	time required to adapt an analysis model	[hours]
e.r.17	number of data conversions to translate an analysis model into a geometric model	[-]

[19] Most modelling techniques use their own input and output data format for geometric and material information. As a consequence, generation of an analysis model out of a geometric model often implies data conversions.

e.r.18	time to convert an analysis model into a geometric model	[hours]
e.r.19	deformation magnitude accuracy in the occupant contact area	[%]
e.r.20	time required for load card generation	[hours]
e.r.21	time required for load card adaption	[hours]
e.r.22	elapsed time for a crash calculation	[hours]
e.r.23	a deformation versus time output plot option must be available	(available/not available) [-]
e.r.24	an acceleration versus time output plot option must be available	(available/not available) [-]
e.r.25	a geometry state output plot option must be available	(available/not available) [-]
e.r.26	a deformation animation option must be available	(available/not available) [-]
e.r.27	number of structural points per occupied volume in the final output	[dm^{-3}]
e.r.28	number of colours in plots	[-]
e.r.29	hidden line removal in geometry plots must be available	(available/not available) [-]
e.r.30	a cross section state plot option must be available	(available/not available) [-]
e.r.31	a perspective geometry plot option must be available	(available/not available) [-]
e.r.32	time required to post process data	[hours]

Analogue with the user requirements, not all the engineering requirements summarized are of the same importance. To express the relative importance of the engineering requirements weight factors are established for each engineering requirement. These factors are based on the customer requirement weight factors. The strength of the relation between customer requirements and engineering requirements for each of the 704 requirement pairs is expressed in a level value. A subdivision in the following four relation levels is used:

9 = strong relation
3 = medium relation
1 = weak relation
0 = no relation at all.

By multiplying the relation strength expressing value for each requirement pair with the concerning customer requirement weight factor and summarising the outcome per engineering requirement, the weight factors are found. The engineering requirement will be indicated as "must" marked by an asterix(*) if the engineering requirement is strongly related with a "must" customer requirement. In all other situations a chosen weight factor of 15 will be applied for a "must" customer requirement. The weight factors for the engineering requirements including the applied relation strength level values are presented in Table A.3.

	u.r.1	u.r.2	u.r.3	u.r.4	u.r.5	u.r.6	u.r.7	u.r.8	u.r.9	u.r.10	u.r.11	u.r.12	u.r.13	u.r.14	u.r.15	u.r.16	u.r.17	u.r.18	u.r.19	u.r.20	u.r.21	u.r.22	
weight u.r.	*	5	5	8	3	*	*	5	8	*	4	4	10	5	3	2	2	10	10	12	5	*	
e.r.1	9	3	0	1	1	9	0	0	1	3	0	0	3	0	0	1	0	0	0	0	0	0	*
e.r.2	9	3	0	1	3	9	0	0	0	3	0	0	3	0	0	1	0	0	0	0	0	3	*
e.r.3	1	3	0	9	3	1	0	0	0	3	0	0	3	0	0	0	0	0	0	0	0	3	246
e.r.4	1	3	0	1	1	1	0	0	0	3	0	0	3	0	0	0	0	0	0	0	0	3	176
e.r.5	3	0	3	0	9	3	0	1	0	0	0	0	0	0	0	0	0	0	0	0	0	0	137
e.r.6	3	0	3	0	9	3	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	132
e.r.7	3	0	3	0	3	3	0	0	0	0	0	0	0	0	0	0	0	0	0	3	0	0	150
e.r.8	1	3	9	0	3	3	0	0	0	0	3	0	0	0	0	0	0	0	0	3	0	0	177
e.r.9	0	0	0	0	3	1	0	9	0	0	3	3	0	0	0	0	0	0	0	0	0	0	93
e.r.10	0	0	0	0	3	3	3	9	0	0	3	0	0	0	0	3	0	0	0	0	0	0	162
e.r.11	1	0	0	3	0	9	9	1	0	3	0	0	3	9	3	1	0	0	0	0	0	0	*
e.r.12	0	0	0	3	0	3	3	1	0	0	0	0	3	9	3	1	0	0	0	0	0	0	205
e.r.13	0	0	0	0	0	0	0	1	0	3	0	0	0	9	3	0	0	0	0	0	0	0	104
e.r.14	0	0	0	0	0	3	0	1	9	0	0	0	3	9	3	3	0	0	0	0	0	0	212
e.r.15	1	9	0	0	3	1	0	1	3	9	0	0	0	0	0	1	0	0	0	0	0	0	*
e.r.16	0	0	0	0	3	3	0	1	0	0	3	0	0	0	0	0	0	0	0	0	0	0	71
e.r.17	1	0	0	3	3	1	0	0	0	0	9	9	0	0	0	0	0	0	0	0	0	3	180
e.r.18	0	0	0	1	3	0	0	1	0	0	1	9	0	0	0	1	0	0	0	0	0	3	109
e.r.19	0	1	0	9	1	3	0	0	0	0	1	0	9	0	0	3	0	9	0	0	0	3	360
e.r.20	0	0	0	0	3	1	0	0	0	0	0	0	0	9	0	1	0	0	0	0	0	0	71
e.r.21	0	0	0	0	3	1	0	0	0	0	0	0	0	0	0	1	0	0	0	0	0	0	26
e.r.22	0	0	0	0	9	3	0	0	0	0	0	0	0	0	0	0	9	0	0	0	0	0	90
e.r.23	0	0	0	9	3	0	0	0	0	0	0	0	0	0	0	1	0	0	0	9	0	1	206
e.r.24	0	0	0	9	3	0	0	0	0	0	0	0	0	0	0	1	0	0	0	9	0	1	206
e.r.25	0	0	0	9	3	3	0	0	0	0	0	0	0	0	0	1	0	0	9	9	0	0	326
e.r.26	0	0	0	1	3	3	0	0	0	0	0	0	0	0	0	1	0	0	1	1	0	3	131
e.r.27	1	0	0	9	3	9	0	0	0	0	0	0	0	0	0	9	0	0	3	1	0	9	*
e.r.28	0	0	0	0	1	0	0	0	0	0	0	0	0	0	0	0	0	0	1	0	0	1	28
e.r.29	0	0	0	0	1	0	0	0	0	0	0	0	0	0	0	0	0	0	3	0	0	3	78
e.r.30	0	0	0	3	3	3	0	0	0	0	0	0	0	0	0	0	0	0	9	9	0	3	321
e.r.31	0	0	0	1	3	1	0	0	0	0	0	0	0	0	0	0	0	0	3	0	0	3	107
e.r.32	0	0	0	0	3	3	0	0	0	0	0	0	0	0	0	0	0	0	0	0	9	3	144

Table A.3 *The relation strength between the engineering requirements (first column) and the customer requirements (first row). The weight of the engineering requirements is given in the last column.*

A.2.6 Engineering targets for the design

Due to the non standard type of design problem handled here (a completely new product), as already indicated in Section A.1, absolute targets for a number of requirements expressing a limit in the form of a minimum or maximum value are difficult to formulate. On the other hand, a target in the form of a value to be maximized or minimized, can often be stated. Nevertheless, it is chosen to generate absolute targets as directives for the crashworthiness integration method. These targets have to be reconsidered if later on in the design process it appears that they are unfeasible. Besides, absolute targets expressed in the form "possible" or "available" occur, as already mentioned in Section A.2.5.

The targets are based on a full vehicle body model study situation. As a reference model adaption case, adding a side intrusion limiting door beam is chosen. The result after application of Step 6 of the QFD technique, i.e. after setting the targets, is presented in Table A.4. The grey cells contain targets for engineering requirements strongly related with “musts”, i.e. a score 9 in Table A.3.

In Section A.2.4, which described Step 4 of the QFD technique, a detailed FEM option was checked on fulfilment of the customer requirements. In Step 6 a FEM option is checked on fulfilment of the engineering requirement targets as explained below. The results of this check are presented in Table A.4 in the right most column.

The performance check of the FEM option on the geometry generation aspects (e.r.1 through e.r.8) is based on the utilisation of the CAD software package Unigraphics[20].

The performance on analysis model generation aspects (e.r.9 through e.r.21 excluding e.r.15 and e.r.19) is based on application of the GFEM module of Unigraphics, i.e. a mesh generation tool, and Generis, a preprocessor included with Pam-Crash[21] crash analysis software.

The indication of the elapsed time for a crash calculation (e.r.22), the number of detail levels (e.r.15) and the accuracy (e.r.19 and e.r.27) are based on an example of a FEM study where a 17,000 elements model calculation of a side impact is performed on a Cray-YMP (Steyer, 1991).

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[20]Unigraphics is a registered trademark of Electronic Data Systems Corporation.

[21]Pam-Crash is a registered trademark of Pam System International.

The 17,000 element model described by Steyer is based on a model tuned on experimental crash results. This tuning is not intended to take place in the early vehicle design stages in the method to be developed. Besides, our approach is to finally tune the geometry in a way that the (numerically) predicted crash behaviour is actualised and not the other way around. Hence, data on the time required to obtain this complete vehicle body model is not suitable for this benchmark. However, our experience with FEM modelling of vehicle components justifies the following estimation. Starting with a complete CAD file available of for instance an A-pillar door sill area, a FEM model can be generated in approximately three hours. Consequently, modelling of a complete vehicle body including its complex local contact properties takes at least in the order of several days. The target for Engineering Requirement 10 is far from reached. Besides, the generation of a CAD file containing the geometric definition of a complete vehicle body with a detail level suitable to base a primary FEM model on, takes at least a week. This means that the target for Engineering Requirement 7 is neither reached.

The performance check on post processing aspects (e.r.23 through e.r.32) is based on the application of the post processor Pamview (version 1.2) included with Pam-Crash.

label	engineering requirement	unit	target	score FEM
e.r.1	point modelling must be available	[-]	available	available
e.r.2	wire frame modelling must be available	[-]	available	available
e.r.3	surface modelling must be available	[-]	available	available
e.r.4	solid modelling must be available	[-]	available	available
e.r.5	geometry input must be graphically displayed	[-]	possible	possible
e.r.6	geometry input must be graphically displayed on line	[-]	possible	possible
e.r.7	time required for first geometric model generation	[hours]	< 2.5	> 40
e.r.8	time required to adapt a geometric model	[hours]	< 0.5	0.5
e.r.9	number of data conversions to create an analysis model	[-]	1	3
e.r.10	time required to generate an analysis model	[hours]	< 1	> 30
e.r.11	a concentrated load definition option must be available	[-]	available	available
e.r.12	a spread load definition option must be available	[-]	available	available
e.r.13	definition of loads as function of time must be possible	[-]	possible	possible
e.r.14	an inertia load definition option must be available	[-]	available	available
e.r.15	number of detail levels	[-]	4	2
e.r.16	time required to adapt an analysis model	[hours]	< 0.5	1
e.r.17	number of data conversions to translate an analysis model into a geometric model	[-]	1	3
e.r.18	time to convert an analysis model into a geometric model	[hours]	1	1
e.r.19	deformation magnitude accuracy in the occupant contact area	[deviation in %]	< 20	17
e.r.20	time required for load card generation	[hours]	< 0.3	0.5
e.r.21	time required for load card adaption	[hours]	< 0.3	0.5
e.r.22	elapsed time for a crash calculation	[hours]	< 3	16
e.r.23	a deformation versus time output plot option must be available	[-]	available	available
e.r.24	an acceleration versus time output plot option must be available	[-]	available	available
e.r.25	a geometry state output plot option must be available	[-]	available	available
e.r.26	a deformation animation option must be available	[-]	available	available
e.r.27	number of structural points per occupied volume in the final output	[dm^{-3}]	> 2	25
e.r.28	number of colors in plots	[-]	> 2	> 2
e.r.29	hidden line removal in geometry plots must be available	[-]	available	available
e.r.30	a cross section state plot option must be available	[-]	available	available
e.r.31	a perspective geometry plot option must be available	[-]	available	available
e.r.32	time required to post process data	[hours]	1	0.2

Table A.4 The engineering targets (grey cells contain targets for engineering requirements strongly related with "musts").

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Abbreviations

ACEA	Association des Constructeurs Europeans d'Automobile
APC	Abdomen Performance Criterion
CC-CTP	Computer-Controlled Composite Test Procedure
CCMC	Committee of Common Market automobile Constructers
CTP	Composite Test Procedure
EEVC	European Experimental Vehicles Committee
EDLD	Exterior Deformable Loading Device
EuroSID	European Side Impact Dummy
FEM	Finite Element Method
FMVSS	Federal Motor Vehicle Safety Standard
HPC	Head Performance Criterion
ILD	Interior Loading Device
NHTSA	National Highway Traffic Safety Administration
PSPF	Pubic Symphysis Peak Force
QFD	Quality Function Deployment
TTI	Thoracic Trauma Index
US-SID	United States Side Impact Dummy
VC	Viscous Criterion

Curriculum vitae

Dirk Landheer was born 1967, in the Netherlands. He studied Mechanical Engineering at Eindhoven University of Technology. In 1991, he received his M.Sc. degree in Mechanical Engineering after a final project on side impact protection of passenger cars at Volvo Car bv. This final project established his interest in vehicle crashworthiness. Therefore, he chose to participate in a project on crashworthiness design of the Laboratory for Automotive Engineering at the Eindhoven University of Technology. The results of the latter project on crashworthiness design are reported in this thesis.

Stellingen

bij het proefschrift:
"Rapid Design of Crash Properties for Safe Automobiles:
A Conceptual Approach"
van Dirk Landheer

1 Het is uit energetisch oogpunt verstandiger in auto's het klimaatbeheersingssysteem te optimaliseren dan de aandrijflijn.

2 Het huidige marketingbeleid (pull) is een bedreiging voor het milieu; een behoefte aan nog te ontwikkelen produkten wordt immers geforceerd terwijl de klant de praktische beperkingen ervan nog niet overziet.

3 Ten onrechte wordt door velen in Nederland de montage van een 'bull bar' geassocieerd met verhoging van de botsveiligheid (dit proefschrift)

4 De compatibiliteit van voertuigtypen ten aanzien van passieve veiligheid is alleen relevant wanneer er een gerede kans bestaat dat de verschillende voertuigtypen onderling botsen. Deze kans is te sturen door verkeersstroomregulering (dit proefschrift).

5 De consument moet bij de keuze van een voertuig de vrijheid hebben veiligheid of een andere kwaliteit te laten prevaleren; de agressiviteit ten aanzien van andere verkeersdeelnemers dient hierbij wel aan restricties gebonden te zijn.

6 De verzekeringspremie zou gekoppeld moeten zijn aan het veiligheidsniveau van de verzekerde auto.

7 Het uitblijven van accijns op kerosine (vliegtuigbrandstof) leidt tot onnodige milieuverontreiniging en stimuleert het ontstaan van een hierop afgestemde infrastructuur.

8 Aan in het nederlands toegepaste engelse termen wordt vaak een meerwaarde toegekend die door engelstaligen niet wordt begrepen.

9 Een heffing voor de verwerking van afval dient niet bij afval-inzamelpunten plaats te vinden daar dit het ontstaan van zogenaamd 'zwerfafval' in de hand werkt.

10 Door de moderne elektronische communicatiemiddelen wordt versturen van goederen bemoeilijkt; het verminderde aanbod van poststukken leidt tot minder brievenbussen die bovendien minder frequent gelicht worden.