THESIS FOR THE DEGREE OF DOCTOR OF PHILOSOPHY

in

Machine and Vehicle Systems

Structural Safety Design for Real-World Situations

Using Computer Aided Engineering for Robust Passenger Car Crashworthiness

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Department of Applied Mechanics
CHALMERS UNIVERSITY OF TECHNOLOGY
Gothenburg, Sweden, 2013

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ISBN 978-91-7385-904-2

Doktorsavhandlingar vid Chalmers tekniska högskola
Ny serie nr 3585
ISSN: 0346-718X

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Cover:
Basic model of how traffic environment affects the vehicle and human occupants in crashes (left), and simulation setup for oblique crashes described in Paper III (right).

Chalmers Reproservice
Gothenburg, Sweden 2013

You miss 100% of the shots you don't take.

- Wayne Gretzky

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Abstract

Road traffic continues to cause more than a million fatalities worldwide every year. Although many steps have been taken to improve occupant protection in car crashes, challenges still remain for car designers.

In the present study, real-world data derived from frontal crashes has been used as a base for identifying crash situations where occupants are severely or fatally injured in cars despite them having been awarded top-ratings in crashworthiness evaluation tests. One situation identified is small overlap crashes, where injuries are commonly related to intrusion. Another is large overlap situations, where injuries are not directly linked to intrusion but rather to vehicle deceleration and interaction with restraint systems.

The aim of the studies constituting this thesis was to develop design methods for robust crashworthiness of future passenger cars and propose solutions to mitigate injuries in large overlap situations. Research was performed using simulation models ranging from simple mass-spring elements to detailed Finite Element (FE) models of contemporary passenger cars.

A newly developed methodology has been proposed as a main contribution based on the research undertaken, in order to provide a comprehensive way of simulating and visualising structural robustness in car-to-car frontal crashes. The methodology was applied to identify worst-case scenarios both regarding intrusion (oblique small overlap scenarios) and deceleration (large, but not full, overlap scenarios). Further development of this methodology has been proposed in order to address issues of crash compatibility, as well as a tool for securing robustness in future mass reduction scenarios.

Another contribution is the proposal of an adaptive front structure to reduce passenger compartment deceleration levels by actively decoupling the front subframe on a contemporary passenger car in a range of frontal car-to-car crash scenarios. Results suggest a deceleration reduction potential equivalent to reducing the velocity change in a frontal crash by up to 44%.

The findings of the present study are compared to previous work and future applications are suggested.

Keywords: passive safety, crash simulation, structural robustness, frontal crashes, structural adaptivity, crash compatibility, small overlap crashes

List of appended papers

Paper I

Wågström, L., Thomson, R., Pipkorn, B. (2004) *“Structural adaptivity for acceleration level reduction in passenger car frontal collisions”* International Journal of Crashworthiness 9 (2), 121-127.

Paper II

Wågström, L., Thomson, R., Pipkorn, B. (2005) *“Structural adaptivity in frontal collisions: implications on crash pulse characteristics”* International Journal of Crashworthiness 10 (4), 371-378.

Paper III

Wågström, L., Kling, A., Norin, H., Fagerlind, H. (2013) *“A methodology for improving structural robustness in frontal car-to-car crash scenarios”* International Journal of Crashworthiness, International Journal of Crashworthiness 18 (4), 385-396.

Paper IV

Wågström, L., Kling, A., Norin, H., Fagerlind, H. (2012) *“A Correlation Study for Oblique Frontal Impacts with Focus on Small Overlap Situations”* In: Proceedings of the International Crashworthiness Conference (ICRASH), Milan, Italy.

Paper V

Wågström, L., Kling, A., Berge, S., Norin, H., Fagerlind, H. (2013) *“Adaptive structure concept for reduced crash pulse severity in frontal collisions”* International Journal of Crashworthiness (published online).

The appended papers were prepared in collaboration with the co-authors. The author of this thesis was responsible for planning the aim and scope of the papers, carrying out the numerical simulations, analysing the results, and stating the conclusions.

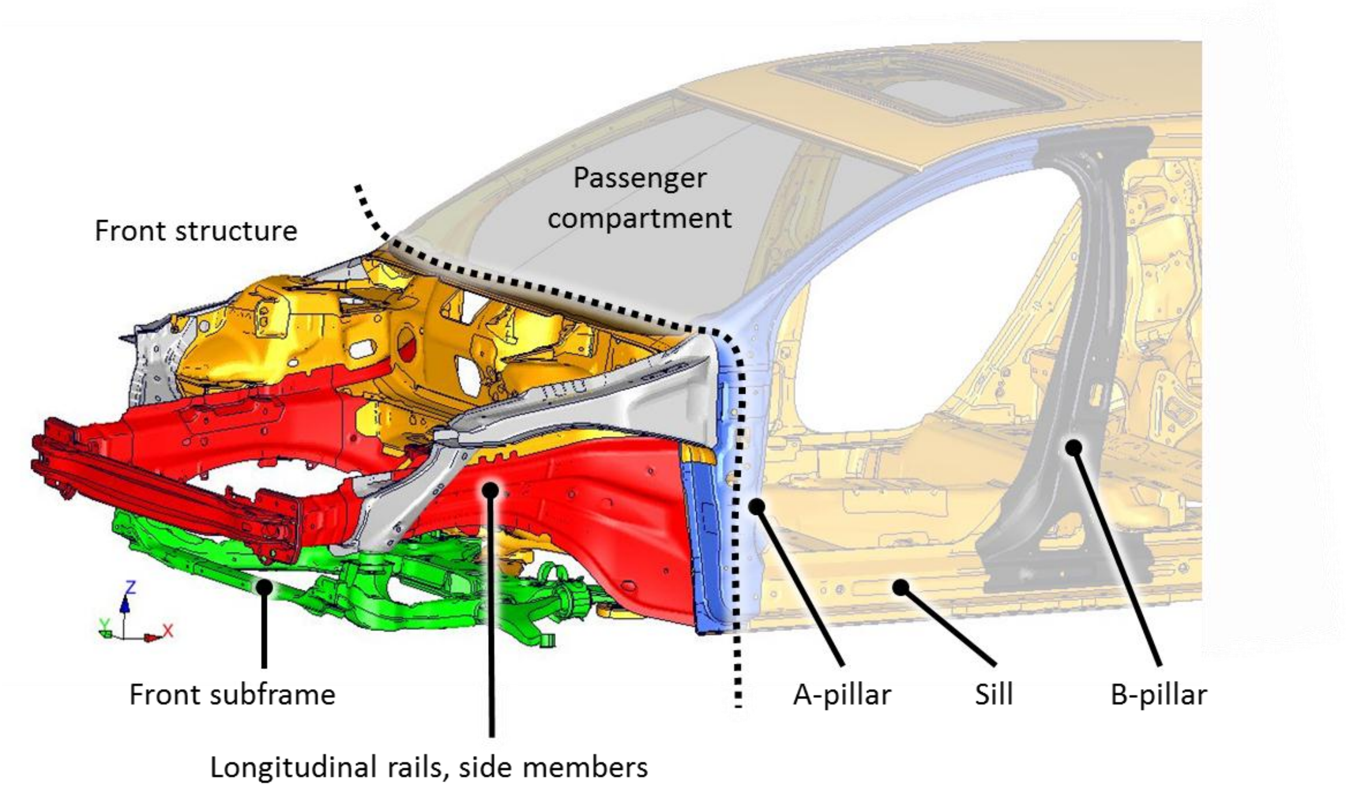
Table of contents

Definitions and abbreviations

Active safety	Measures taken to avoid or mitigate crashes
Aggressivity	Measured for a subject vehicle in terms of fatality or injury risk for occupants in opponent vehicles involved in the same crash
AHOF	Average Height of Force
AIS	Abbreviated Injury Scale
A-pillar	Vehicle structure connecting floor and roof, in front of front doors
B-pillar	Vehicle structure connecting floor and roof, rear of front doors
CAE	Computer Aided Engineering
CCIS	Cooperative Crash Injury Study
Crash	Dissipation of vehicle kinetic energy by structural deformation
Crash compatibility	Combination of self and partner protection in crashes
Crash pulse	Vehicle deceleration time history
ELVA	Advanced Electric Vehicle Architectures, research project acronym
Euro-NCAP	European New Car Assessment Programme
EVCR	Equivalent Velocity Change Reduction
EVERSAFE	Everyday Safety for Electric Vehicles, research project acronym
FE	Finite Element
FIMCAR	Frontal Impact and Compatibility Assessment Research, research project acronym
Frontal stiffness	Relationship between force and displacement during crush of frontal structures
FWDB	Full-width Deformable Barrier
FWRB	Full-width Rigid Barrier
GIDAS	German In-Depth Accident Study
HIII	Hybrid III crash test dummy
HOF	Height of Force
IIHS	Insurance Institute for Highway Safety
KW400	Work stiffness over the first 400 mm of vehicle front crush
LS-DYNA	Finite element solver used for crash simulations

MAIS	Maximum Abbreviated Injury Scale
MPDB	Moving Progressive Deformable Barrier
NHTSA	National Highway Traffic Safety Administration
ODB	Offset Deformable Barrier
OLC	Occupant Load Criterion
Partner protection	A vehicle's ability to indirectly protect occupants in opponent vehicles against injuries in car-to-car crashes (i.e., reduction in aggressivity)
Passive safety	Measures taken to reduce consequences of crashes
PDB	Progressive Deformable Barrier
Restraint system	Interior vehicle system designed to mitigate injuries in crashes, e.g., seatbelts, airbags and seats
Self-protection	A vehicle's ability to protect own occupants against injuries in crashes
SUV	Sport Utility Vehicle
Vehicle structure	Structural element of vehicle, e.g., body structure or front subframe
Vision Zero	Long-term aim for eliminating fatalities and serious injuries in road traffic
VPI	Volvo Pulse Index
WHO	World Health Organization

Visualisation of vehicle structures



Preface

The work presented in this thesis was carried out at Chalmers University of Technology in Gothenburg, Sweden and was divided into two periods.

The first work period was carried out at the Crash Safety Division of the Department of Machine and Vehicle Systems from 2002 to 2004 under the supervision of Dr. Robert Thomson and Dr. Bengt Pipkorn and was sponsored by Autoliv Research.

The second work period was carried out at the Division of Vehicle Safety of the Department of Applied Mechanics from 2011 to 2013 under the supervision of Dr. Hans Norin, Helen Fagerlind and Anders Kling. All studies during the second period were sponsored by Volvo Cars and conducted as a part of the Volvo Cars Industrial PhD Programme (VIPP) in association with SAFER – Vehicle and Traffic Safety Centre at Chalmers, Sweden.

Acknowledgements

The creation of this thesis would not have been possible without the support of numerous colleagues and family members. I would like to mention the people that have played the most important roles and at the same time apologize to anyone that I may have failed to mention:

My academic supervisor Dr. **Hans Norin**, for sharing your vast experience within automotive safety and posing important questions regarding the objective and contents of my work. I have fully enjoyed the creative atmosphere in our collaboration.

My assistant academic supervisor **Helen Fagerlind**, for providing a real-world perspective on my simulation studies as well as caring for the written quality of this thesis and included papers. You are always a source of optimism and you have fully understood the power of feedback and encouragement.

My industrial supervisor **Anders Kling**, and my companion since several years at Volvo Cars. You are one of the greatest safety enthusiasts around and I have had the privilege of learning from your knowledge that you share humbly and generously.

My examiner Professor **Mats Y Svensson**, who I have had the pleasure of interacting with during both phases of my doctoral studies. Your guidance towards completing this thesis has been valuable as well as inspiring to me.

My previous academic supervisor Dr. **Robert Thomson**, for enjoyable collaboration both for my studies up to the Licentiate degree and later in research projects FIMCAR and EVERSAFE.

My previous industrial supervisor Dr. **Bengt Pipkorn**, for guidance towards the Licentiate degree and sharing your experience within finite element simulations.

Adjunct Professor Dr. **Yngve Håland** for serving as an inspiration in terms of ingenuity and authority within the area of vehicle safety.

Professor **Per Lövsund**, my examiner for the Licentiate degree, for sharing your considerable experience in vehicle safety research and guidance towards research competence.

My group manager **Graeme McInally**, section manager **Anders Djärv**, director Dr. **Magnus Olsson** and former director **Hans Nyth** at Volvo Cars Safety Centre for constant support and for assisting me in keeping a balance between academic studies and development of even safer Volvo cars. I would especially like to thank my former group manager **Ragnar Crona** for looking beyond daily tasks and supporting me when the second half of my doctorate studies where set up.

My skilled and hard-working colleagues at the frontal impact analysis group at Volvo Cars Safety Centre over the years for providing a pleasant and challenging work environment, especially the people that have been my closest colleagues working with crash simulation: **Peter Appelgren**, **Anders Carlsson**, **Petter Hörling**, **Per-Henrik Myrefelt**, **Fredrik Tidborg**, **Peter Sandqvist**, **Anders Sandahl**, **Weijia Wu** and **Babak Nejabat**.

My other colleagues at Volvo Cars Safety Centre, especially section manager **Jan Ivarsson**, senior technical leader and adjunct Professor Dr. **Lotta Jakobsson**, senior technical advisor **Thomas Broberg**, technical leader **Stefan Berge** and technical experts Dr. **Johan Jergeus** and Dr. **Mikael Ljung-Aust**. You all represent extraordinary competence in vehicle safety engineering that has been a highly valuable asset to me during my studies.

Senior technical leader Dr. **Mikael Fermér,** serving as a role model in combining academic merits with industrial excellence. You played an important part in encouraging me to continue towards this thesis.

All my colleagues at Chalmers University of Technology, especially associate Professor Dr. **Karin Brolin** for assistance with Paper III that needed two major revisions before finally being accepted for publication, Professor **Jac Wismans** for collaboration in ELVA, and assistant Professor Dr. **Johan Davidsson** for assistance regarding teaching.

My former and current doctorate student colleagues, especially **Fredrik Jenefeldt**, Dr. **Fredrik Törnvall** and Dr. **Weijia Wu** with whom I had the pleasure of sharing office space with during my first period 2002-2004. The same goes for my colleagues at Injury Prevention 2011-2013: Dr. **Katarina Bohman**, **Stina Carlsson**, Dr. **Anna Carlsson**, **Kristian Holmqvist**, **Aleksandra Krusper**, **Manuel Mendoza-Vazquez**, **Krystoffer Mroz**, **Jóna Marín Ólafsdóttir**, **Ruth Paas**, **Isabelle Stockman** and **Jonas Östh**.

The SAFER organisation directed by **Anna Nilsson-Ehle,** for making my time here pleasant and inspiring.

My master thesis students **Yannick Schroeder**, **Oscar Centeno**, **Nilesh Dharwadkar** and **Krishna Prashant Adivi** that I had the privilege to supervise on their way to becoming excellent engineers.

Finally, the **Wågström** clan: my parents **Ingrid** and **Sören**; my sister **Linda** and brother **Pär-Johan** with families. Thanks for everything since day one.

My dear family **Arezo**, **Jonathan**, **Elias** and **Alicia**. This is for you.

Göteborg, September 30, 2013

1 Introduction

The World Health Organization (WHO) estimates annual road fatalities to more than 1.2 million globally (World Health Organization 2013), making traffic injuries the leading cause of death among young people aged between 15 and 29. The WHO reported data from 2006 amounting to more than 100,000 fatalities in India, close to 90,000 in China and more than 40,000 in the United States. Furthermore, the WHO stated in the same report that “while road traffic death rates in many high-income countries have stabilised or declined in recent decades, data suggest that in most regions of the world the global epidemic of traffic injuries is still increasing”.

In the European Union, road transportation claims more than 30,000 lives annually as reported by the European Commission (2013). Approximately half of the European road fatalities represent occupants in passenger cars (DaCoTA 2011). The design of passenger cars must therefore constantly be reviewed to investigate whether there are further steps that can be taken in order to reduce the number of road fatalities.

A number of countries and organisations have adopted visions for eliminating fatalities and serious injuries in road traffic, often referred to as Vision Zero (Tingvall and Haworth 1999, Johansson 2009). It has been identified that countermeasures may be feasible both by protecting occupants in the event of a crash (passive safety) and by avoiding or mitigating a crash (active safety) as described by Eugensson et al. (2011).

1.1 Basic model for occupant protection in passenger car crashes

This thesis is focused on passive safety of passenger cars and uses a basic model of how car occupants interact with the traffic environment via the vehicle in crashes as illustrated in Figure 1. In the first layer of this model, the outcome for the occupants in a crash is defined by how they can be protected from crash loads by the vehicle structure and restraint systems. Restraint systems and vehicle structures are dependent on each other, i.e., without a stable structural response the restraint systems may not be sufficient to protect occupants against injuries. Vice versa, the detailed structural response may be less important if occupants are not restrained to the vehicle in a crash. Therefore, occupant protection is considered to be based on equivalent shares of structural response and restraint system performance.

In the next layer, the basic model describes vehicle interaction with the crash environment as illustrated in Figure 1. The crash environment is divided into three categories: crash opponent, crash scenario and crash energy, all affecting vehicle response in a crash and defined as follows:

- Crash opponent are all types of objects that a car may collide with, e.g., trees, poles, roadside barriers, animals, trucks, cars or other vehicles.
- Crash scenario is a description of how a car interacts with its crash opponent, e.g., offset frontal crashes, oblique side crashes, etc.
- Crash energy describes, for a situation defined by crash opponent and scenario, the initial velocity and mass of vehicles and objects involved in the crash.

Any combination of these three crash environment parameters will hereafter be referred to as a crash situation.

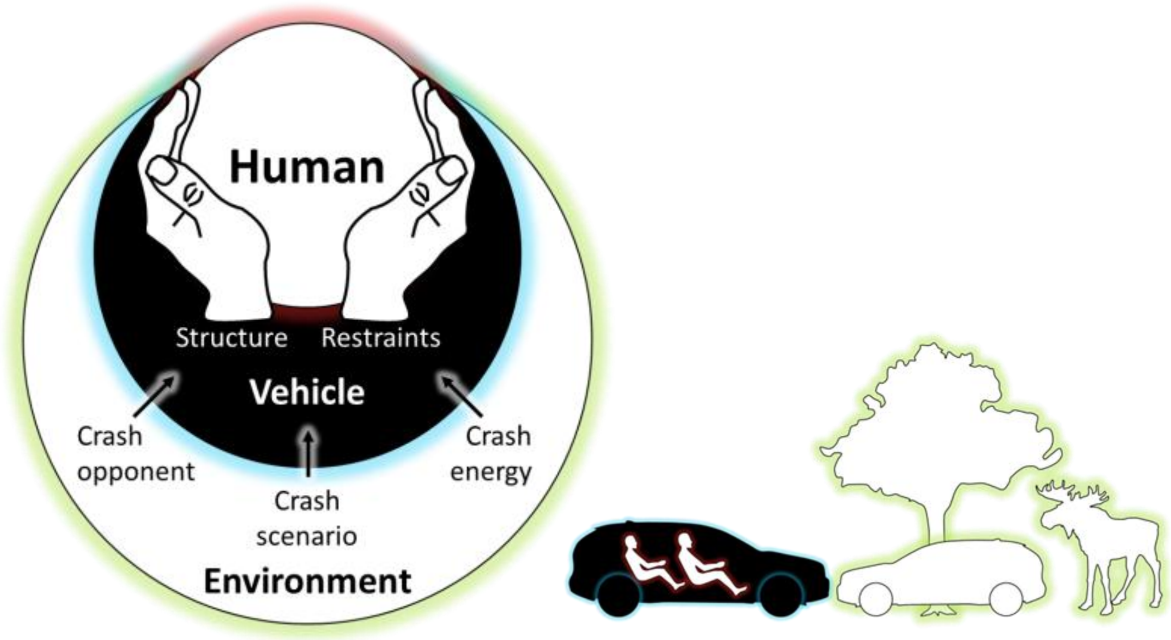


Figure 1. Basic model of how traffic environment affects the vehicle and human occupants in crashes.

The objective when designing passenger cars for safety should be that any crash loads transmitted to the occupant from the traffic environment via the vehicle should be within human tolerance limits. This objective should ideally be realised for all crash situations, i.e. combinations of crash opponent, scenario and energy, which the car may be subjected to during its lifetime. If this cannot be achieved, the most relevant crash situations should be considered based on the real-world occurrence and corresponding injury risk for each type of situation. In order to visualise the described traffic environment parameters, they can be plotted in separate dimensions creating a crashworthiness design volume as illustrated in Figure 2. This design volume can then be considered to contain all the crash situations that a passenger car needs to be designed for in terms of structural response, as well as restraint system performance.

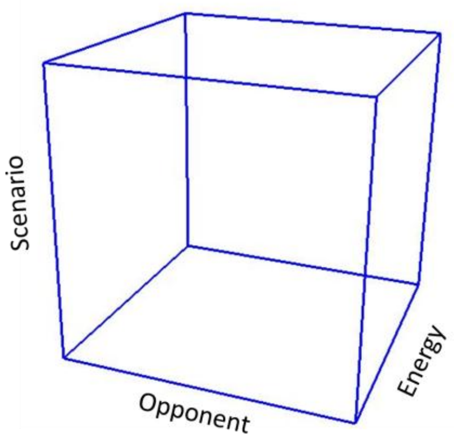


Figure 2. Visualisation of crashworthiness design volume, i.e., three dimensions of traffic environment parameters: opponent, scenario and energy.

The total number of crash situations is inarguably immense and considering every situation when designing a passenger car is simply not feasible. Therefore, a number of regulatory and consumer rating load cases have been developed over the years to represent the most relevant situations. These load cases, or crash setups, are evaluated by means of Computer Aided Engineering (CAE), i.e., crash simulation, in all phases of product development and by crash tests later in the development process. If these load cases are selected in an optimal way, all situations that are relevant in terms of occupant protection for a car's real-world crash performance are covered. Feedback on the actual crash performance will be provided during the car's life cycle, thus showing strengths and weaknesses in terms of real-world crash safety performance years after the car model was first introduced on the market. Any design changes that may be prompted by the real-world crash performance are thus limited to be implemented late in the product's life cycle, or to be passed down to the next generation of vehicles by which time design conditions may have changed.

An alternative approach to awaiting real-world crash data to become available is to use simulation models to predict real-world crash performance. By using crash simulation, which is a faster and less expensive method than crash testing, a large number of situations in the design volume illustrated in Figure 2 can be evaluated. However, this approach relies on model validity, i.e., if the simulation models cannot be shown to be valid for the complete evaluation range, results may be inaccurate. Therefore, increased simulation capability for assessment of real-world crash performance depends largely on improved modelling techniques such as model refinement and advanced material models including failure predictions.

Another advantage of using simulation models is that they are well suited for parameter studies. In this way, individual effects of crash scenario, opponent and energy levels can be studied. Such relationships may be harder to find in real-world crash data since many combinations of crash opponent, scenario and energy occur, and confounding factors are inevitable. Using crash testing to find such relationships is challenging in terms of time and resources. Furthermore, reconstructing and thoroughly analysing a wide range of real-world situations by crash testing often requires extraordinary crash laboratory specifications, including a full range of available crash angles and possibilities to capture film sequences from underneath vehicles.

1.2 General aim and scope

The general aim of this thesis is to contribute to passenger vehicle design methods and solutions towards eliminating road fatalities and serious injuries as described by Vision Zero (Tingvall and Haworth 1999, Johansson 2009). This contribution was attempted by using computer simulation models for use in the design process of passenger cars for improved real-world crashworthiness. As both structural response and restraint system performance represent extensive research areas, this thesis is delimited to the design and response of vehicle structures. Special focus has been directed towards robust structural response, i.e., a vehicle's ability to manage a wide range of crash situations without sudden changes of structural performance.

Furthermore, the studies that comprise this thesis have been delimited to front-to-front car-to-car crashes at initial oblique angles from -45° to +45° relative to the target vehicle heading angle in the horizontal plane as illustrated in Figure 3.

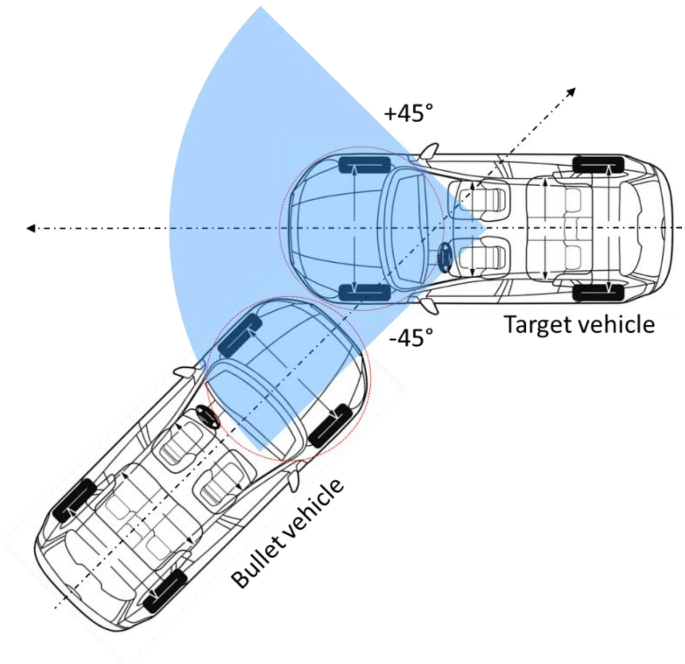


Figure 3. Definition of thesis scope with respect to oblique angle in car-to-car crashes.

The delimitation to structural aspects of passive safety is further illustrated in Table 1, where the scope of this thesis is related to crash scenarios and safety systems.

Part of the aim was to develop methods that are general in their application, i.e., they should be extendable to a wider scenario range such as side and rear crashes as outlined in Table 1. Another part of the objective was that methods should be able to support the development of restraint systems and future decisions on the balance between passive and active safety systems.

Table 1. Overview of crash scenarios and safety systems describing thesis scope limited to methods and solutions for vehicle structures and frontal crashes.

		Frontal crashes	Rear crashes	Side crashes	Rollover, run-off-road	Pedestrian protection	...
Passive safety systems	Structures	Methods Solutions					
	Restraints						
Active safety systems	Mitigation						
	Avoidance						

In order to study and discuss different degrees of horizontal overlap, a definition is needed. Since passenger cars are designed in various ways, connecting degrees of overlap to a percentage of vehicle width may not represent structural engagement. Furthermore, when oblique angles exist, measuring horizontal overlap is subjective and this measurement may vary during a crash sequence. In this thesis, the extent of engaged structures are therefore used to define horizontal overlap as illustrated in Figure 4.

- Small overlap scenarios are defined as when the frontal structures designed for energy absorption, such as longitudinal rails (also called side members) or front subframe, are <u>not</u> engaged.
- Moderate overlap scenarios are defined as when the frontal structures designed for energy absorption on <u>one side</u> of the vehicle are engaged.
- Large overlap scenarios are defined as when the frontal structures designed for energy absorption on <u>both sides</u> of the vehicle are at least partly engaged. If the entire front of the vehicle is engaged, this is referred to as a full-width scenario.

When analysing real-world crashes as well as crash simulations, it may be difficult to draw precise borderlines between the different categories as indicated by the gradient shading in Figure 4.

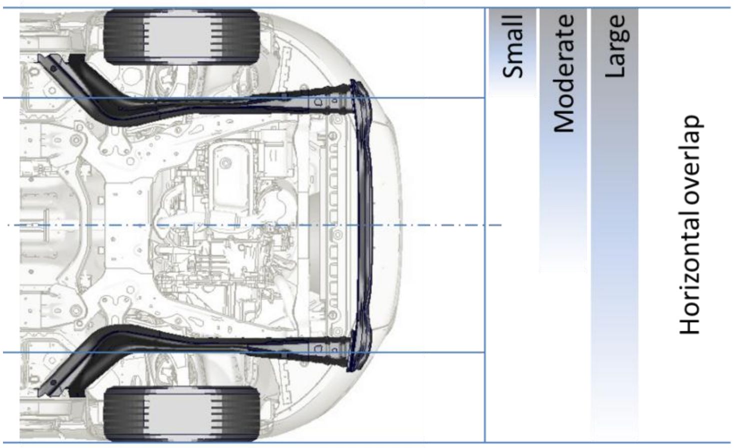


Figure 4. Definitions of small, moderate and large horizontal overlap in car-to-car frontal crashes.

1.3 Thesis outline

This thesis is structured into separate parts as illustrated in Figure 5. The first part serves as an introduction to the research area and describes the delimitations made in order to move from the general area of crash safety to the specific area of simulations of frontal car-to-car crashes. The following part describes a literature review that was conducted in order to provide an overview of previous work in the area of structural design of passenger vehicles focused on frontal car-to-car crashes. The literature review also identifies areas of priority and corresponding research gaps, leading to the objectives of this thesis. In the next part, the five appended research papers are summarised, focusing on methods and results on the specific level of simulations of frontal car-to-car crashes (Paper II applies rigid barriers to estimate energy absorption in frontal crashes). The last part of the thesis discusses the derived methods and results, and relates them to previous work, as well as possible future applications and research questions.

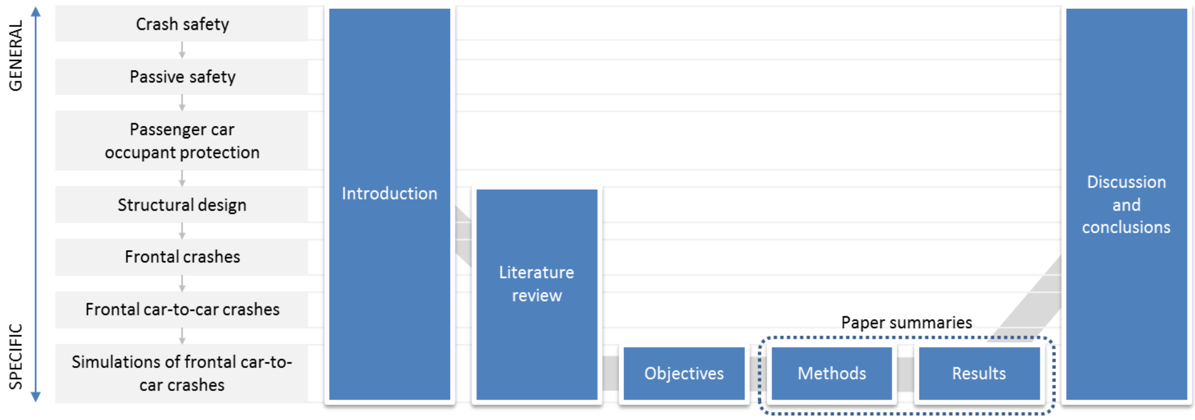


Figure 5. Thesis outline.

2 Literature review

2.1 Current status of passenger car structural design for frontal crashes

Analysing crash statistics in further detail, it is estimated that approximately half of the fatal accidents involving car occupants in Sweden occur in frontal crashes (Lindman 2012). Additional studies show the importance of frontal crashes among situations with severe and fatal injuries. Over the years 1979 to 2007, NHTSA estimated frontal crashes to account for 44% to 51% of US occupant fatalities (NHTSA 2009). The European FP7 project FIMCAR reported that frontal crashes represented 57% of UK occupant fatalities 2008-2010 and 32% of German occupant fatalities 2005-2007 (FIMCAR 2011b).

Regarding frontal crashes, the overall safety level in modern vehicles has been improved since offset deformable barrier (ODB) tests were introduced. Real-world data show that good performance in ODB rating tests correlates with reduced injury risk in traffic accidents (Farmer 2005, Kullgren et al. 2010). However, there are still situations where improvements can be made which will be described in the following sections.

A specific subset of frontal crashes is frontal car-to-car crashes. In these situations, the crash performance of frontal structures is tested in real-world situations versus other designs. European data from frontal car-to-car crashes suggests that the share of occupant fatalities occurring in frontal car-to-car crashes is 12% in Germany and 23% in the UK (FIMCAR 2011b), illustrated in Figure 6. By assuming that approximately 15% of car occupant road fatalities occur in frontal car-to-car crashes, this type of situation is estimated to account for more than 2,000 European road fatalities annually.

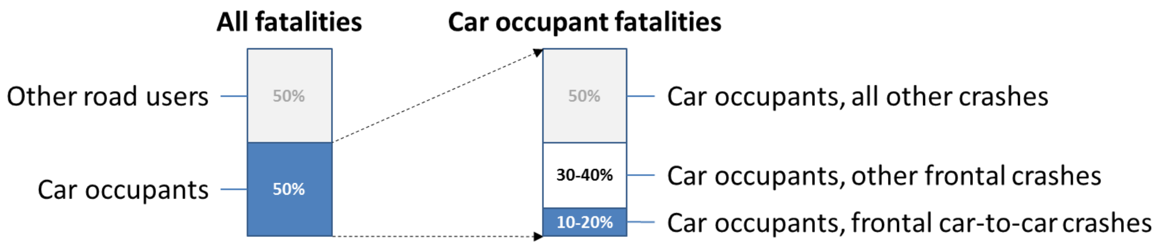


Figure 6. Approximate distribution of EU road fatalities, data from FIMCAR (2011b).

Severe injuries to occupants in frontal crashes, as defined by a Maximum Abbreviated Injury Scale (MAIS) score of three or higher (Sherwood 2009) have been shown to be strongly linked to passenger compartment intrusion. Furthermore, similar correlation with intrusion has been found for Injury Severity Scores (ISS) above 25 (Conroy et al. 2008). These findings support the need for improved car structural integrity, i.e., minimising intrusion into the passenger compartment. When structural integrity is compromised, injury risk for car occupants increase both from direct contact with intruding structures, as well as by influencing the effect of available restraint systems (Brumbelow and Zuby 2009). These findings from real-world situations propose structural integrity as a crucial performance measurement for passive safety of passenger cars.

2.2 Crashworthiness and robustness

In order to ensure that the vehicle structural response will support the occupant restraint systems in all relevant situations in the design volume outlined in Figure 2, some measurement of structural response is needed. Whether it is deceleration, intrusion or another measure, this measurement should be possible to record and visualise over the entire design volume in order to compare design options. Therefore, definitions of optimised design compared to robust design used in this thesis are required.

Consider a system where one output variable describes the system performance as a function of one input variable as illustrated in Figure 7. The objective of the system is to minimise the output variable, e.g., passenger compartment intrusion or deceleration which must be achieved for all input variable settings, e.g., horizontal overlap, within its lower and upper bounds as illustrated in Figure 7. Furthermore, all input settings must yield output below the maximum allowable response indicated by the horizontal line in Figure 7

In this example, the definition of optimal input is the input leading to minimum output. However, if this optima is associated with a small range of input variable settings, a large output variation may be the result of a minor input variation as described in further detail by Lee and Park (2001). Therefore, a more desirable input may be one resulting in a higher minimum performance but lower performance variation.

The definition of a robust solution is a solution where large variations in input result in small variations in output, as illustrated in Figure 7. If the input is considered an adjustable variable, the input should be controlled towards an optimal or robust solution. If the input variable is to be considered a random variable to be handled by the system, the design should be made in such a way that no input variable settings result in unacceptable performance as illustrated in Figure 7. In order to address the problem of robustness and unacceptable performance, a proposal for a modified design is given by the dashed curve in Figure 7. On the one hand, this design provides a higher minimum performance. On the other hand, the modified design provides a more robust system, i.e., smaller variations in the output variable when the input variable is changing. In addition, the modified design does not violate the maximum allowable response limit as indicated in Figure 7.

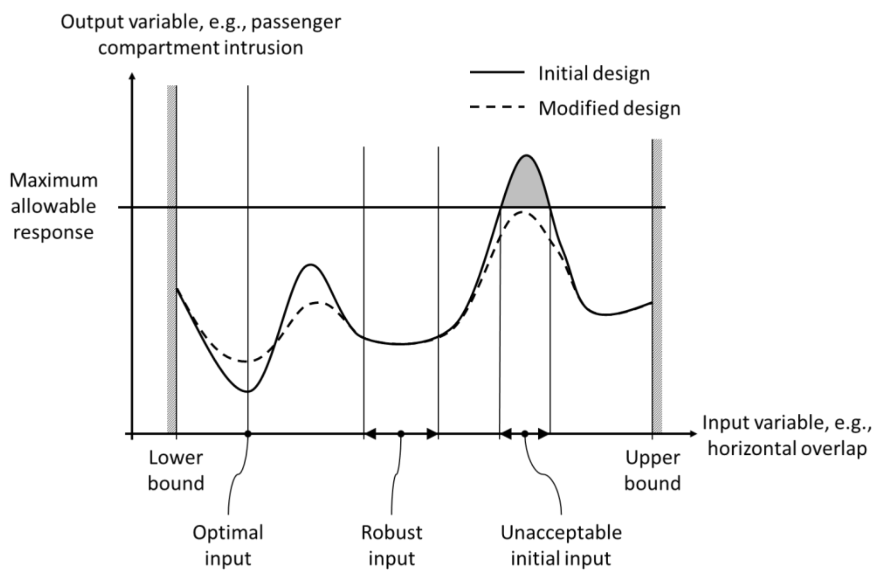


Figure 7. Schematic figure showing principles of optimisation and robustness.

As described in previously published work on this subject (Marklund and Nilsson 2001, Craig et al. 2005, Lönn et al. 2009, Lönn et al. 2010, Lönn et al. 2011), studying both structural optimisation, as well as robustness requires well-defined measurements of performance. The performance is often called response values (one output variable) or response surfaces (two or more output variables). In crashworthiness, injury mechanisms are related to passenger compartment intrusion, intrusion velocity and/or deceleration (Conroy et al. 2008, Sherwood 2009). Therefore optimisation or robustness response surfaces could possibly be based on these measurements. However, since car-to-car frontal crashes can be described as highly non-linear systems, optimisation was not attempted within in the scope of this thesis. The input variables in car-to-car crashes may be regarded as random to some extent; an appropriate objective for structural design could be the robustness of the intrusion and deceleration response of vehicles involved. This means, designing vehicle structures that exhibit small variations in structural response although variations in input as described by parameters such as oblique angle and horizontal overlap may exist.

2.3 Crash compatibility

Aiming at a robust structural response, an important aspect of occupant protection in passenger car frontal car-to-car crashes is the opponent encountered. This topic, called crash compatibility, has been thoroughly studied in various research initiatives over the years. The main findings of this work are presented in the present section, giving an outline of the complexity of the subject.

Crash compatibility is considered a combination of both self and partner protection, i.e., protecting occupants in the own vehicle as well as occupants in opponent vehicles (FIMCAR 2011a). This definition is needed since protection of occupants in one vehicle should not be achieved by reducing occupant protection in opponent vehicles.

Compatibility was considered a prioritised research area in the late 1990s and early 2000s. It was shown that light truck vehicles were over-represented in US car-to-car opponent fatality statistics and the concept of vehicle aggressivity or partner protection was commonly used to describe the problem (Gabler and Hollowell 2000, Austin 2005, Huang et al. 2011). This means that some vehicles cause a disproportionately large number of severe injuries or fatalities in opponent vehicles in car-to-car frontal crashes. Crash incompatibility and aggressivity issues were observed also in UK data (Edwards et al. 2001), French data (Delannoy and Diboine 2001), Canadian data (Fredette et al. 2008) and Japanese data (Mizuno and Kajzer 1999).

Moreover, the compatibility issue was also found in Swedish accident statistics and attempts were made to separate the effects of mass and structure by observing police reported two-car crashes (Kullgren et al. 2001). Kullgren et al. found SUVs to be considerably more aggressive than the average car. Multibody simulation models were developed as an approach to compatibility research for estimating effects of fleet changes. (Buzeman-Jewkes et al. 1999, Buzeman-Jewkes et al. 2000). Similar approaches were later attempted that underlined the usability of such simplified vehicles in order to perform a large number of simulations to provide a basic understanding of the complex system of a large car fleet (Jenefeldt and Thomson 2004, van der Zweep et al. 2005, Watanabe et al. 2005). Other researchers used FE models and crash testing to develop front structure concepts with multiple load paths in order to address compatibility issues (Fujii et al. 2003, Saito et al. 2003). Attempts were also made to make car consumers increasingly aware of the safety issues related to vehicle crashes between vehicles of different size and weight (The Insurance Institute for Highway Safety 2005, The Insurance Institute for Highway Safety 2009)

Starting in 2003, a European consortium consisting of safety research institutes, authorities and vehicle manufacturers joined forces in the project VC-Compat to find solutions to crash compatibility issues. In the final VC-Compat technical report (VC-Compat 2007), it was estimated that approximately 1,000 lives could be saved annually on European roads by improved crash compatibility. It was identified that structural interaction and compartment strength are key issues to be addressed in order to improve crash compatibility. However, no consensus could be reached on standardised test procedures in order to introduce legislative or consumer rating tests for enhanced crash compatibility.

In order to advance further towards better crash compatibility in frontal car-to-car crashes, VC-Compat was followed by the 2010-2012 research programme, Frontal Impact and Compatibility Research (FIMCAR). The starting point of FIMCAR was that crash compatibility consists of both self and partner protection and the consortium finally reached a recommendation to introduce a full-width test procedure for Europe based on priority of structural interaction and restraint system performance (FIMCAR 2011e).

The importance of structural interaction appears to be a common conclusion in many of the studies conducted on crash compatibility. In the United States, a voluntary agreement was initiated by the Alliance of Automobile Manufacturers in 2003 (Barbat 2005). This agreement was based on placing frontal energy absorbing structures in a common interaction zone at 16 to 20 inches above ground level. Later studies from NHTSA (Greenwell 2012) and IIHS (Baker et al. 2008, Teoh and Nolan 2012) all indicate improvements to crash compatibility after the agreement on alignment of vehicle front structures had come into effect.

The results so far from the voluntary agreement suggest that a relatively simple target of a common interaction zone has been effective in addressing crash incompatibility. However, developing test procedures using crash barriers to objectively measure appropriate structural interaction frontal force levels has proved to be difficult.

One approach to measuring structural interaction properties of a vehicle front structure is to measure the height of force (HOF) time history from a full-width barrier. Alternatively, the average height of force (AHOF) has been proposed as an average over a certain evaluation period in time. The applicability of HOF and AHOF were investigated thoroughly by researchers from the United States (Verma et al. 2004, Subramaniam et al. 2007, Nusholtz et al. 2009, Brewer et al. 2011) as well as Japan (Mizuno et al. 2005, Watanabe et al. 2005, Hirayama et al. 2007, Uwai et al. 2007, Yonezawa et al. 2008). Some criticism was raised regarding the reproducibility of AHOF and possible velocity dependence. Regardless of evaluation method, the height at which forces are transmitted appears to be of great importance to car-to-car crash compatibility. Vertical misalignment has been shown to be unbeneficial for energy absorption and deformation modes which may increase injury risk to occupants in both vehicles involved in frontal car-to-car crashes (Baker et al. 2008, Mizuno and Arai 2010).

A second approach suggested is to use the deformation pattern of offset deformable barriers to assess structural interaction. Aiming at increasing the test severity for small vehicles and avoiding bottoming-out of the deformable barrier honeycomb block, the Progressive Deformable Barrier (PDB) was developed for offset crash tests (Delannoy and Diboine 2001, Delannoy et al. 2005, Delannoy et al. 2007). Additional studies were conducted regarding the feasibility of attaching the PDB to a trolley, creating a so-called Moving Deformable Barrier or MPDB (Schram and Versmissen 2007, Versmissen et al. 2007). The MPDB has the potential to further increase the crash severity for light cars which leads to discussions regarding the initial velocity and trolley mass setup for this specific load case. If a MPDB test is always run with a fixed trolley mass and initial velocities as proposed by FIMCAR (FIMCAR 2011d), this test method could potentially decrease the self-protection of heavier vehicles since the crash severity will be reduced for this type of cars. The PDB was further studied by NHTSA (Meyerson et al. 2009) where the PDB was once more proposed as a tool for assessing partner protection. The PDB has been proposed to be introduced into regulation as an update to the UNECE R94 regulation (Chauvel et al. 2011). The crash compatibility assessment potential of the PDB was recognised by FIMCAR, but could not be recommended for regulatory testing since barrier evaluation metrics were identified to require further development and validation (FIMCAR 2011c).

A third approach is a full-width deformable barrier (FWDB) that could potentially replace a full-width rigid barrier (FWRB) in tests for high deceleration response (Edwards et al. 2003a, Edwards et al. 2003b, Edwards et al. 2007, Edwards 2009). By assessing the barrier face deformation, structural interaction may also be assessed. Additional studies were performed by Arai et al. (2007) which supported the FWDB as a tool for assessing structural interaction. Similarly to the PDB, FIMCAR found that although the FWDB could potentially be used for assessment of structural interaction, these assessment metrics needed further development (FIMCAR 2011e). FIMCAR however recommended the FWDB to replace the FWRB in regulatory testing mainly based on indications that the FWDB yields an occupant compartment deceleration response more representative of real-world situations in the initial stage of the crash compared to the FWRB (FIMCAR 2011e).

Once structural interaction is improved, the next level of improved crash compatibility is believed to be stiffness matching, i.e., harmonising frontal force levels for dissimilar vehicle mass categories. One approach to this could be to base frontal force levels on average vehicle mass rather than the actual vehicle mass. Based on FE simulations of frontal car-to-car crashes, it has been suggested that matching the stiffness of a lighter vehicle up to the level of a 30% heavier car may only have minor effects on the crash pulse shape (Volvo Car Corporation et al. 2010). It has been shown however, that when stiffness of a lighter vehicle is matched to a heavier crash opponent, dummy injury values in car-to-barrier tests are increased significantly (Watanabe et al. 2005).

Matching the frontal stiffness of a heavier vehicle down to the level of a lighter vehicle requires extended front structure deformation length to maintain energy absorption for self-protection in the heavier vehicle. Without structure extension, such a design change could reduce intrusion in the opponent car below a certain impact velocity, but increase intrusion at higher impact velocities (Hirayama et al. 2007). Subramaniam et al. (2007) explored the effect of modifying the initial stiffness of a light truck vehicle to match that of a car crash opponent, and increasing the stiffness in the later phases of crush to compensate for the lost energy absorption. The result of this modification was a deteriorated car-to-barrier performance in terms of intrusion as well as deceleration and dummy response. It was recommended that a wide range of crashes should be evaluated for effects on occupant protection before any stiffness matching regulations are implemented.

A concrete proposal on how to implement stiffness matching of frontal structures in passenger vehicles was presented by NHTSA and called KW400 (Patel et al. 2007). This metric was established as an attempt to measure the frontal stiffness during the first 400 mm of deformation in a FWRB crash, and was used for stiffness matching studies (Hirayama et al. 2007, Subramaniam et al. 2007, Nusholtz et al. 2009). One important issue that was raised was the sensitivity of KW400 to the starting time of the signals (also known as time zero). Since KW400 is calculated from the barrier force starting at 25 mm of vehicle displacement, the initial vehicle-to-barrier contact becomes an important parameter, rewarding a low initial barrier force response.

It has been noted that improved passenger compartment integrity does not necessarily mean higher stiffness of the front structure (Lund and Nolan 2003). Overall, the studies performed on stiffness matching underline the balance between self-protection and partner protection. Given the available front deformation distance, it appears inevitable to match frontal force levels of frontal structures without affecting this balance.

Adding to the complexity of the subject, real-world data suggest that incompatibility in frontal crashes can exist even if vehicles are identical. In FIMCAR (2011a), an example of this from the Great Britain Cooperative Crash Injury Study (CCIS) was given. In this case, two vehicles of the same make and model were involved in a head-on crash at approximately 50% overlap. As illustrated in Table 2, the two cars exhibited significantly different structural responses, with up to seven times greater dashboard intrusion in one of the cars which was also reflected in the injuries sustained by the drivers.

Table 2. Overview of the outcome in a head-on crash involving two cars of same model. Adapted from FIMCAR (2011a).

		
Model year	2002	2001
Kerb mass	1,423 kg	1,384 kg
Overlap	51%	50%
Equivalent test speed based on deformation	26 km/h	46 km/h
Dashboard intrusion	190 mm	900 mm
Footwell intrusion	170 mm	1,180 mm
Driver injury level	MAIS 2	MAIS 5

The conclusion from studies of previous work within the research area of crash compatibility between passenger vehicles in frontal crashes is that there is a range of factors that affect structural response and thereby occupant injury risk. Looking solely at real-world crash data appears to be incomprehensive for understanding the mechanisms that affect structural integrity and robustness. Crash simulation could be one tool to isolate unbalance from crash scenario, as attempted using modified public domain models (Thomson et al. 2008).

In line with this, detailed efforts to describe crash incompatibility in the case of dissimilar vehicles will not be attempted in this thesis. Instead the incompatibility that arises from the crash scenario will be explored. In order to protect occupants in frontal crashes, vehicles should be developed to provide a robust and predictable structural response in frontal crashes independent of crash opponent.

2.4 Small overlap situations

Another key issue for the design of vehicle structures for real-world crashworthiness in frontal crashes are situations where structures intended for energy absorption are not engaged. One such situation occurs when the front of a vehicle is loaded outboard of the front structure, often called severe frontal collision with partial overlap (Planath et al. 1993) or simply small overlap crash (Lindquist et al. 2004), as illustrated in Figure 4.

The real-world significance of small overlap crashes is not a new phenomenon. The importance of designing vehicles for this load case was described by Planath et al. (1993) where a test method with 20 to 40% horizontal overlap against a fixed rigid barrier at initial velocities up to 65 km/h was proposed. Accident data from the early 1990s suggested that the percentage of moderately and severely injured drivers was higher in crashes with an overlap below 30% than in crashes with an overlap of more than 30%. (Kullgren and Ydenius 1998). Further indications of the importance of small overlap crashes was presented by Lindquist et al. (2004), where small overlap crashes accounted for 48% of the fatalities to belted occupants in a data set of frontal crashes in Sweden.

Small overlap situations were given renewed focus when the IIHS highlighted this type of situation in the early 2010s. When observing real-world frontal crashes involving vehicles awarded with good ratings in the IIHS test programme for frontal crash protection (Brumbelow and Zuby 2009), small overlap crashes was one type of scenario where occupants sustained severe injuries. The strong link between occupant compartment intrusion and injury severity in small overlap crashes was presented in a later study by the IIHS (Sherwood 2009). This lead to the development of an IIHS small overlap rigid barrier load case, described in detail by Sherwood et al. (2013). The IIHS also demonstrated that the test method shows acceptable repeatability, indicating that vehicles with poor structural performance in this load case show the largest test-to-test variations in terms of intrusion (Mueller et al. 2013).

NHTSA also suggested that small overlap crashes should be a priority area for future research (Rudd et al. 2009), followed by an additional study where it was demonstrated that small overlap crashes frequently produce oblique kinematics, and the interaction along the side of the struck vehicle increases the risk for injuries from outboard components such as the door and A-pillar (Rudd et al. 2011).

Several studies have thus indicated that these situations are critical for reducing severe and fatal injuries. There are, however, studies suggesting that “the small overlap is at worst a moderately dangerous crash in the overall scheme of frontal crashes” (Scullion et al. 2010). In a later study (Kühn et al. 2013), the German Insurers Accident Research supported the IIHS finding that approximately 25% of frontal crashes can be characterised as small overlap situations. In this German dataset, small overlap crashes were found to represent a small number of fatalities but a large number of serious (AIS2+) injuries to the lower extremities.

Although the representativeness of the IIHS small overlap barrier crash test appears to be a subject for debate, real-world data suggest that small overlap situations should not be neglected when striving for a vision of zero fatalities and serious injuries. Further support for a small overlap barrier load case to represent small overlap car-to-car scenarios was given by Jakobsson et al. (2013a) combined with an overview of vehicle design changes for small overlap situations (Jakobsson et al. 2013b).

2.5 Large overlap situations

In addition to crash compatibility issues and small overlap crashes, there appears to be issues related to the crash pulse or restraint system in frontal crashes with modern cars. In a US study on the types of frontal crashes that cause serious injuries and fatalities to belted front-seat occupants in passenger cars, a considerable portion of serious injuries occur in frontal crashes despite good structural integrity (Brumbelow and Zuby 2009). In that study of cars that had been awarded good ratings in the IIHS frontal moderate (40%) overlap test, it was shown that many belted occupants still sustain severe injuries in frontal crashes without significant vehicle passenger compartment intrusion. This means that even without intrusion, occupants may be injured from contact with restraint systems or car interior, i.e., injury mechanisms are related to the crash pulse rather than intrusion.

Similar findings were presented in the European research project FIMCAR (2011a), where datasets of frontal crashes involving R94-compliant vehicles from Great Britain (Cooperative Crash Injury Study, CCIS) and Germany (German In-Depth Accident Study, GIDAS) were analysed. The study showed that approximately 40% of MAIS 2+ injuries and 30% of fatal injuries suffered by occupants occurred in crashes with more than 75% frontal overlap (compare to Figure 4), and it was suggested that compartment intrusion may not be the direct cause of injury. Besides improving the functionality and robustness of restraint systems, injuries related to the crash pulse can potentially be addressed by the vehicle front structural response.

Current vehicle structures may exhibit stiffer response in frontal crashes compared to older vehicles, especially in the late phases of the crash pulse as a direct effect of improvements in terms of intrusion (Nolan and Lund 2001, Samaha et al. 2010). It is therefore important to balance high-deceleration load cases with offset load cases with significantly different crash pulse shapes and structural loading (FIMCAR 2011a).

2.6 Adaptive structures

Adaptive structures could be one way to optimise the structural response in the wide range of crash scenarios that passenger vehicles encounter. In frontal crashes, it has been suggested that adaptive structures can be used in order to affect the deceleration response. Witteman and Kriens (2001) followed by Witteman (2005) suggested “high-low-high” deceleration pulses to be optimal based on occupant response simulations of crashes with velocity change of 56 km/h or greater. These deceleration pulses were proposed to be accomplished by friction forces applied to steel cables that had the additional benefit of being able to transfer loads from the struck side of a vehicle to the non-struck side.

An alternative approach to achieving “high-low-high” deceleration pulses in a passenger car has been proposed (Motozawa and Kamei 2000, Motozawa et al. 2003), where axial buckling is followed by bending of the main energy absorbing members. A practical solution for the required operational volume for such a system was, however, not presented.

Pipkorn et al. (2005) recommended implementing variable crush force in passenger cars by pressurising vehicle longitudinal frontal members. Since the additional volume required for the pressurised frontal members to function may be minor, this solution may be more efficient in terms of packaging space than the concept proposed by Motozawa et al. (2003). Any solution as to how the Pipkorn et al. proposal would be applied to production vehicles was not presented, although the mass-reducing potential based on increased force levels from pressure in such members was highlighted. In a subsequent study, Pipkorn and Kullgren (2009) again showed that pressurising thin-walled tubular structures can significantly increase the crush force and energy absorption and considerably reduce occupant injury risk as measured by HIII dummy readings. Furthermore, a comprehensive review on adaptive vehicle structures was made by TRL (Thompson et al. 2007), where altering the frontal force level was identified as one of the key principles for adaptive structures.

Other applications of adaptive structures were proposed by Pipkorn et al. (2007) such as attaching an external inflatable airbag to the front structure of an SUV. This was demonstrated to increase structural interaction by creating a lower load path into the sill of a passenger car in side crashes where the SUV is the bullet vehicle. Two types of adaptive front structures, fixed and extendable, were investigated using simplified, two-dimensional, simulation models by Elmarakbi and Zu (2006). The study predicted that improvements, in terms of injury risk, related to both intrusion and deceleration could be accomplished by adaptive front structures.

Pyrotechnically controlling the stiffness of load-carrying front structures of passenger cars in front-to-side crashes were investigated by Ostrowski (2007). In that study, an adaptive structure was shown to decrease the front-end crash stiffness of the bullet vehicle and to extend the crushing distance when needed. Another thorough review on adaptive structures for improved crashworthiness was made by Khattab (2011), where extendable add-on energy absorbers were also suggested. Additionally, Pipkorn et al. (2011) suggested further applications of adaptive structures in passenger cars. In that study, the balance between forward vision and A-pillar load capacity was suggested to be improved by introducing expandable A-pillars.

3 Objectives

The general aim of this thesis is, as previously described, to investigate novel methods based on computer simulation models that can be used in the design process of passenger cars for robust real-world crashworthiness. Supporting Vision Zero through vehicle design requires a holistic view on safety, beyond the load cases defined by legal requirements and consumer rating programmes. Given the level of reliability that crash simulation models have reached through constant development over several decades, these models today represent valuable tools for understanding and proposing countermeasures for the structural challenges seen in real-world crash situations.

One specific objective of this thesis was therefore to develop tools for engineering of safe future passenger vehicles. These tools should be applicable for addressing crash compatibility, small overlap situations and large overlap situations with focus on robust structural response. A second specific objective of this thesis was that the developed tools should be used to explore the applicability of adaptive front structures for crash severity reduction in frontal car-to-car crashes.

4 Paper summaries

The individual papers are linked by their connection to real-world data as illustrated in Figure 8. All five appended papers are summarised in the following section and their relative coherence is explained here.

Two major crashworthiness issues were found in literature regarding real-world frontal car-to-car scenarios where injuries occur in modern vehicles. The first issue concerns small overlap situations, where intrusion appears to be the main cause of injuries. The second concerns large overlap scenarios where intrusion is not necessarily the major cause of injuries but rather the deceleration of the passenger compartment.

Based on these two problems seen in real-world crash data, the work was divided into two paths – one for large overlap situations and one for small overlap situations. Paper I served as an initial study of structural adaptivity in frontal crashes and Paper II followed up on those findings by exploring effects on the crash pulse with a public domain FE model. Paper III laid a foundation for a methodology to be used for structural robustness in frontal crashes regardless of scenario. Paper IV used the methodology from Paper III to examine and classify small overlap situations by validating the FE model against full-scale crash tests. To finally bring both halves of the thesis together, Paper V used parts of the methodology from Paper III to set up a large number of crash simulations to explore a specific concept for structural adaptivity, a detachable front subframe.

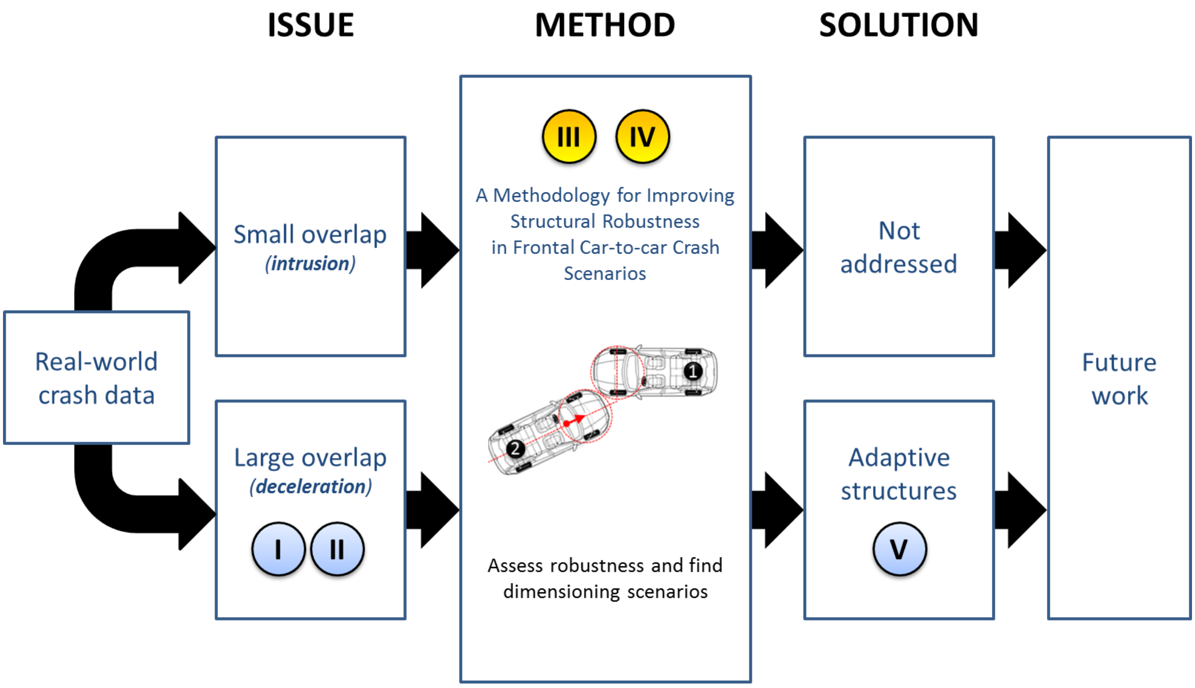


Figure 8. Overview of study coherency with appended papers in roman numerals.

The studies conducted were divided according to the respective issue seen in real-world crash data and their nature, i.e., being a method or a proposed solution. For small overlap scenarios, any specific solutions to address the issues seen in real-world crashes were not proposed. However, for the issues with larger overlap, i.e., deceleration-related injuries, adaptive frontal structures were proposed as a possible solution that could potentially address this type of situation.

All five papers can be related to the crashworthiness design volume in Figure 2. This is illustrated in Figure 9 where each point represents a crash situation evaluated by means of simulation. Figure 9 shows that only Paper I attempts to assess the effect of the crash opponent, whereas Papers II to V are focused on crash scenario or crash energy. Paper II applies rigid barrier simulations for estimation of energy absorption relevant to car-to-car crashes with satisfactory structural interaction.

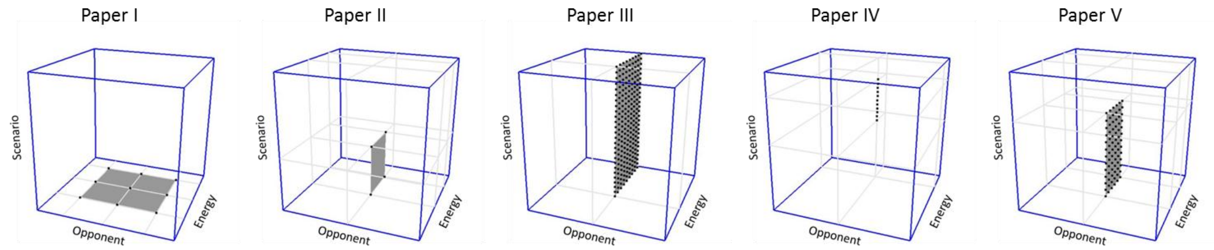


Figure 9. Overview of which crash environment parameters were addressed in the appended papers. Each point represents an evaluated crash situation.

A central concept in Papers III to V is crash pulse severity metrics. These are based on acceleration signals of the sill structures close to the B-pillars. These signals were used to estimate the crash pulse severity based on two simplified models, the Volvo Pulse Index, VPI (ISO 2012) and the Occupant Load Criterion, OLC (Stein et al. 2011). Both models are attempts of generically measuring the restraint forces that the driver is subjected to during a crash, based on deceleration only. Each model uses the occupant displacement relative to the vehicle in the longitudinal direction. Furthermore, both models assume an initial phase of free-flying motion without any occupant deceleration at relative displacement less than 30 mm for VPI and 65 mm for OLC, as illustrated in Figure 10. However, the model responses following the initial slack represent fundamentally different assumptions. VPI assumes a linearly increasing occupant deceleration of 0.25 g/mm of relative displacement, without any limitation on occupant deceleration or relative displacement. The OLC instead assumes a maximum relative displacement of 300 mm and a constant occupant deceleration up to this point. This means that the OLC model assumes a perfectly adaptive restraint system that will always utilise the available interior distance. The VPI model, on the other hand, simulates a non-adaptive restraint system based on chest decelerations measured in crash test dummies in physical tests.

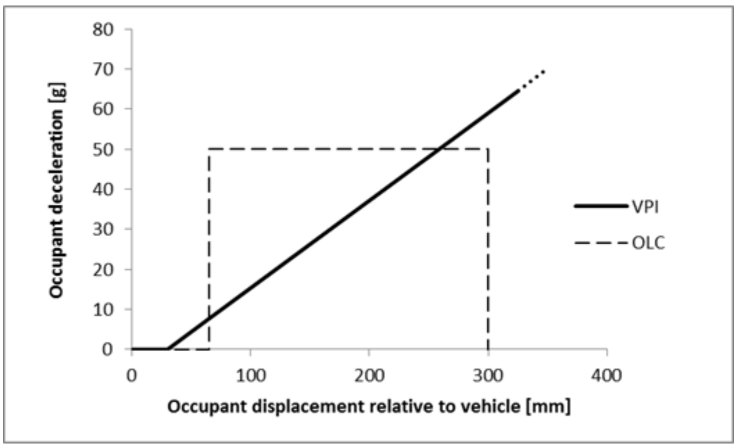
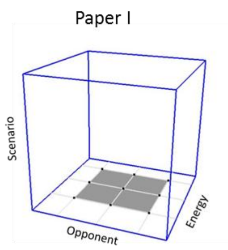


Figure 10. Characteristics of the crash severity indicators VPI and OLC.
Occupant deceleration plotted vs. occupant displacement relative to vehicle.



4.1 Summary of Paper I

4.1.1 Introduction

In-depth studies of crash pulses from real-world frontal crashes have shown a correlation between acceleration levels and injury risk, indicating that high vehicle acceleration during frontal crashes increases the risk of long-term consequences for the occupants. The only way for a vehicle manufacturer to affect the crash pulse is to change the characteristics of the energy absorbing parts of the car. Consumer tests such as Euro-NCAP have prompted car manufacturers to design vehicles with high intrusion resistance. However, in low severity crashes, these structures could potentially subject car occupants to higher accelerations than what would be possible if the stiffness could be adapted to crash severity. Suggestions to design adaptable frontal structures have been found in the literature. This study has sought to find guidelines of how to select deformation characteristics that lead to less harmful acceleration pulses during low-speed crashes while maintaining intrusion resistance at higher crash velocities in frontal crashes.

4.1.2 Method

Car-to-car crashes were simulated using mass-spring models assuming linear frontal stiffness. Typical values for vehicle frontal stiffness were chosen after separating the car fleet into two categories, where one category consisted of cars with a test weight below 2000 kg and the other category consisted of cars with a test weight above 2000 kg. Crash test data was used to approximate frontal stiffness.

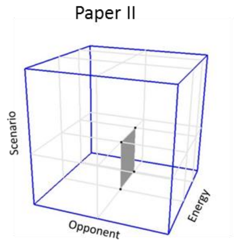
To simulate full-frontal crashes between cars of dissimilar mass or frontal stiffness, three vehicle classes were defined. Using all combinations of car classes and three closing velocities ranging from 40 to 120 km/h resulted in a total of eighteen simulations that were run with the baseline configuration, i.e., constant stiffness. The results of these simulations in terms of maximum deceleration were then compared to the results using an adaptive deformation system, where the frontal stiffness was minimised for each level of kinetic energy.

4.1.3 Results

All peak acceleration values were decreased as a result of the adaptive system. Peak deceleration values were on average reduced by 14% at the highest closing velocity of 120 km/h. When the closing velocity was decreased to 80 km/h, the corresponding reduction was 43% and at 40 km/h, the average peak value was reduced by as much as 73%. It was observed that at the lowest closing velocity, the calculated adaptive stiffness was less than 10% of the original value.

4.1.4 Discussion

Systems including an adaptive frontal stiffness could require pre-crash sensors that can provide information both regarding the state of the own vehicle, as well as the collision object. Structural adaptivity could be accomplished by decreasing or increasing the internal force levels of the deforming vehicle parts. This should be executed in a manner that vehicle occupants are guaranteed an acceptable level of protection in case of undesired system response. It was suggested that a fully-developed adaptive system would have the potential to decrease deceleration levels in low severity accidents, as well as increasing the deformation energy absorption in high velocity crashes.



4.2 Summary of Paper II

4.2.1 Introduction

In passenger-car crashes, frontal crashes are the most frequent accident type. Modern cars are able to maintain structural integrity even in high velocity crashes and thereby reducing the risk of severe and fatal injuries. Indications of safer cars were found in crash statistics in terms of lower injury risk to all body regions, with one alarming exception: injuries to the neck. Studies of crash pulse recorder data have suggested mean deceleration as a candidate crash-severity measure for AIS1 neck injuries.

It has been suggested that greater passenger compartment integrity may have made vehicles less forgiving in terms of deceleration-related injuries such as long-term neck injuries. To reduce harmful crash pulses caused by stiff frontal structures, it is suggested that the optimal frontal structure stiffness should be adapted to the crash situation. The logical approach to achieve lower deceleration is to weaken the engaged structural members. By altering the structural deformation characteristics in suitable locations of a passenger car front, the response could be made sufficiently supple for a low-velocity crash while the front-end stiffness is maintained or increased in a high-velocity crash.

4.2.2 Method

A public domain full-vehicle LS-DYNA FE model of the passenger car model Geo Metro was used to simulate full-width crash and offset crash with a rigid barrier that covered 40% of the maximum vehicle width. In four preliminary simulations, the longitudinal rails were identified to represent the largest portion of absorbed internal energy for impact velocities of both 32 km/h and 56 km/h.

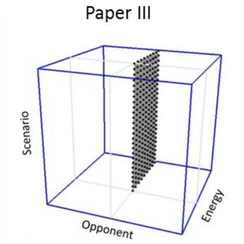
To study the effect on crash pulse characteristics and the possible implications of structures with alternating structural strength, the material of the longitudinal rails was given three levels in subsequent simulations at a range of initial velocities chosen from 16, 32, 48 and 64 km/h. The passenger compartment was modelled in two variants, a rigid compartment and the original deformable compartment.

4.2.3 Results

Minimising passenger compartment intrusion, assessed by a rigid compartment model, will potentially reduce the crash pulse duration and thus increase mean deceleration. The effect of front longitudinal members crush strength on the crash pulse depends largely on the geometric constraints in the engine compartment and the kinetic energy to be absorbed in a crash. Reducing the yield stress of longitudinal members can only reduce the peak deceleration if this peak is associated with structural deformation and not engine contact with a stiff passenger compartment.

4.2.4 Discussion

To design adaptive frontal structures that have a significant effect on the crash pulse, strategies for affecting the global load paths must be investigated. This study suggests that changing the strength of the most significant energy absorbing structural members would only affect the crash pulse to a limited extent.



4.3 Summary of Paper III

4.3.1 Introduction

From data collected in real-world frontal crashes involving new vehicles, it has been suggested that consumer rating crash tests have encouraged improvements to passenger vehicle structural crashworthiness. However, there still seems to be room for further improvement in so-called small overlap conditions, i.e., where vehicles are involved in frontal crashes without engaging the main frontal crash absorbing structures.

It has also been shown that the available coding standards for crush damage in passenger vehicle crashes do not always capture differences in crash configuration, making detailed parameter analyses of car-to-car crash configurations a difficult task. Additional approaches such as computer simulation are therefore needed for understanding how different crash scenarios are linked to vehicle structural performance and possibly occupant injury risk.

Based on these findings from previous studies and real-world crash data, it is suggested that improved crash simulation techniques should be developed in order to better understand the structural mechanisms that lead to large passenger compartment intrusion or deceleration in frontal car-to-car crash scenarios. The aim of this particular study was therefore to develop a methodology for identifying dimensioning frontal car-to-car crash scenarios by assessing crash configuration parameters that influence structural response.

4.3.2 Method

A full-vehicle FE model was validated in terms of intrusion and deceleration response in frontal crashes and used to establish a car-to-car crash simulation model with two identical passenger cars. The car-to-car FE model was employed for a parameter study including 378 simulations on how the crash setup affects passenger compartment intrusion and deceleration. Based on this output, a set of scenarios with outstanding properties in terms of structural response were defined and considered candidates for crash scenarios that should be used for dimensioning of car structures.

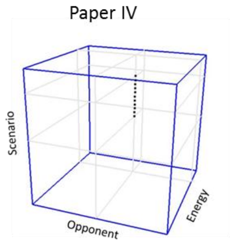
Since the FE model used for the parameter study cannot be validated by physical testing in all of the scenarios defined by the simulation matrix, a few of the most noteworthy crash setups have been selected for further validation work as they fell outside the scope of the study. By improving numerical robustness and validity of FE models, the methodology would be suitable for comprising part of a comprehensive toolbox for ensuring robust response of vehicle structures.

4.3.3 Results

The intrusion area that displayed the best correlation with other intrusion areas was the central A-pillar intrusion which was therefore suggested to represent the overall intrusion levels in each car. At 15° oblique angle and 1,200 mm lateral offset scenarios, this intrusion was more than three times greater in one of the identical cars. The greatest crash pulse severity was found at scenarios around 300 mm lateral offset and 0 or 5° oblique angle.

4.3.4 Discussion

The purpose of the methodology presented in this study was to establish a tool for structural robustness in the development process of passenger vehicles. The proposed compatibility domain lends itself to visualising structural differences in car-to-car crashes regardless of the vehicles involved. In an extended application, this methodology could be used to compare the relative importance of different aspects of incompatibility, e.g., studying when a structural advantage is cancelled by an unbeneficial car-to-car crash scenario.



4.4 Summary of Paper IV

4.4.1 Introduction

It is estimated that approximately half of the European Union road transportation fatalities are occupants in passenger cars, and that about half of these occur in head-on crashes. Consumer rating programmes using ODBs performed in both Europe and the United States have encouraged car designs with improved passenger compartment integrity in order to increase self-protection. However, fatalities and severe injuries still occur and it is therefore important to study these types of crashes. Data from Europe as well as the United States suggest situations were defined as situations where front structures designed for energy absorption are not engaged as major load paths, often called small overlap frontal crashes.

In addition to designing vehicles for self-protection, consideration should also be made for partner protection. Incompatibility in two-vehicle frontal crashes is characterised by differences in injury risk to occupants in one vehicle compared to the other vehicle. These differences can be caused by both occupant and vehicle dissimilarities, but are normally discussed in terms of vehicle mass, geometry and stiffness. Incompatibility has typically not been used to describe crash situations in cases where both vehicles and occupants are identical. In Paper III, scenarios with large differences in terms of passenger compartment intrusion in one car compared to the other were found around 15° oblique impact angle combined with small overlap of car front ends. These results needed to be validated and it was therefore decided that model validation by comparing to physical crash tests was required.

4.4.2 Method

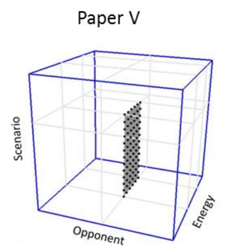
Two full-scale crash tests were performed in order to validate the crash simulation model, a 15° oblique angle car-to-car setup, and a rigid barrier test overlapping 25% of the car. During the analysis of these two crash tests in comparison to simulation results, three major areas were identified where modelling improvements were required for increased model validity: front subframe bushings, rim failure and tyre separation. A parameter study was then performed in order to describe oblique small overlap car-to-car crashes.

4.4.3 Results

Five separate crash categories were distinguished in the simulations. Category A was defined by at least one of the frontal structures being deformed and thus contributing to energy absorption in the crash. Category B refers to the set of crashes where the wheels overlap each other, creating a locking phenomenon which leads to substantial deformations of one of the identical car models. Categories C and D included situations where the wheel of one car becomes detached and is pushed along the sill of the same car, leading to un-robust response in terms of intrusion. Category E comprised sideswipe situations, with a minimal influence on the struck vehicle front structure and A-pillar. The combination of large intrusions and large lateral velocity change was only found for the Category B crashes.

4.4.4 Discussion

Predicting rim failure proved to be difficult since a substantial degree of variation is involved in the physical tests and the borderlines between crash categories do therefore not represent exact limits. If detachment of the front wheels cannot be achieved in a predictable way, intrusion may vary substantially. Previous studies have identified that severe injuries often occur in situations without significant intrusion, suggesting that further development of restraint systems in combination with improving the deformation modes of frontal structures may be needed.



4.5 Summary of Paper V

4.5.1 Introduction

Studies in Europe as well as the United States into the real-world performance of vehicles have shown that many belted occupants still sustain severe injuries in frontal crashes without significant vehicle passenger compartment intrusion. Besides improving the functionality and robustness of restraint systems, injuries related to the crash pulse can potentially be addressed by the vehicle front structural response. It has been suggested that adaptive structures can be used in order to affect the deceleration response in frontal crashes. The front engine subframe has been identified as a major load path in frontal crashes, important both for self-protection and crash compatibility. Equipping vehicles with an adaptive detachable front subframe has the potential to benefit the overall real-world crash performance. The aim of this study was therefore to quantify the effect an adaptive detachable front subframe has on occupant loading in car-to-car frontal crashes in a range of lateral offset distances and closing velocity levels.

4.5.2 Method

A full vehicle model was simplified by removing the majority of the structural components from a plane rear of the A-pillars in order to perform a large number of simulations. It was shown that the deceleration response of the simplified model was similar to the original simulation model at both full overlap and approximately 50% overlap with identical vehicles. The simplified crash model was also shown to represent the crash pulse shape of vehicles with similar stopping distance in physical crash tests. A simulation matrix was established to study the effect of the vehicle deceleration pulse on two simplified crash severity indicators called Volvo Pulse Index (VPI) and Occupant Load Criterion (OLC).

4.5.3 Results

A high level of correlation was found between longitudinal velocity change and both crash severity indicators, VPI and OLC, for lateral offset up to and including 1,000 mm. Using this relationship, an equivalent velocity change reduction (EVCR) was calculated, indicating the required velocity change reduction in order to achieve the same reduction in crash severity indicators in the base, passive model, as was achieved by actively detaching the subframe. The greatest reduction in crash severity gained by releasing the subframe was predicted to be equivalent to a 44% ΔV_x reduction for the VPI model, and 31% for the OLC model. As an average of the considered crash scenarios, the results based on the VPI model suggest 28% relative EVCR compared to 18% for the OLC model.

4.5.4 Discussion

The simplification of the vehicle model resulted in a small influence on the crash pulse shape and it was therefore considered an adequate substitute for the original model. If the subframe cannot be detached by both vehicles as assumed in this study, the effect on the stopping distance will be reduced for both vehicles. The VPI and OLC models represent fundamentally different assumptions on the restraint system characteristics. OLC assumes a perfectly adaptive restraint system whereas VPI assumes a linearly increasing deceleration without any limitation on relative displacement. A typical restraint system may therefore be considered as a combination of the two models and the real-world effect of an adaptive subframe is most likely within the range given by the VPI and OLC models. Using the front subframe for structural adaptivity was demonstrated to have a considerable effect on the full-vehicle crash pulse shape. Benefits over previously proposed solutions were seen, since additional packaging space or modifications to the frontal longitudinal members would not be required and the concept may be implementable in an already existing vehicle structure.

5 General discussion

Although passenger car crashworthiness has improved tremendously over the years, there are still issues that should be considered in order to preserve the positive development in the reduction of the number of fatal and severe injuries in passenger car crashes. Improved CAE modelling techniques bring new possibilities for predicting the real-world crashworthiness before vehicles are produced. The methodology presented in this thesis is one such approach towards vehicle designs that can constitute one of the enablers for reaching a common vision of zero fatalities and severe injuries in road traffic. The methods and solutions in this thesis are not expected to be directly applicable to all aspects of robust crashworthiness in passenger cars. Nevertheless, the suggested approaches adopted in the present thesis are intended as inspiration to utilise the power of state-of-the art simulation technology to chart future safety strategies.

Model validity is an issue that always needs to be addressed when findings based on FE simulation results are discussed. During development of the methodology presented in Paper III, it became obvious that model updates were needed in order to obtain realistic model response in terms of separation of non-structural parts, such as wheels and wheel suspension. Judging by crash tests in small overlap situations with rigid barriers or other cars, the structural response can differ remarkably compared to standardised barrier crash tests with deformable honeycomb elements or full-width rigid barriers. Therefore a first step towards better simulation models was taken and presented in Paper IV. Additional modelling improvements are expected to be required in order to advance with the proposed methodology. This is in line with the general trend that can be seen in CAE, since shorter development time and fewer physical tests during the design phase of passenger cars require greater confidence in simulation models. Improved reliability of CAE models must, however, be combined with an appropriate set of load cases that guarantees robust structural behaviour in real-world situations.

5.1 Discussion of results with respect to crash opponent

The compatibility domain established in Paper III was used to provide an overview of how different crash scenarios compare to each other in terms of intrusion or deceleration in two identical passenger cars, as illustrated in Figure 11. It is suggested that the usage of the compatibility domain could be extended to assess factors such as vehicle mass, size, stiffness in relation to crash configuration factors, such as oblique angle and horizontal or vertical misalignment. In Paper III, a specific area was used to represent the overall passenger compartment intrusion levels. For other vehicle designs, such a representative intrusion area may be more difficult to establish, which would require additional intrusion areas to be monitored in order to compare intrusion response in a set of crash situations. This will introduce a more complex comparison of crash situations and may lead to conflicting results, i.e., one situation leading to large intrusions in one measurement area but small intrusions in another area, and vice versa for a different situation.

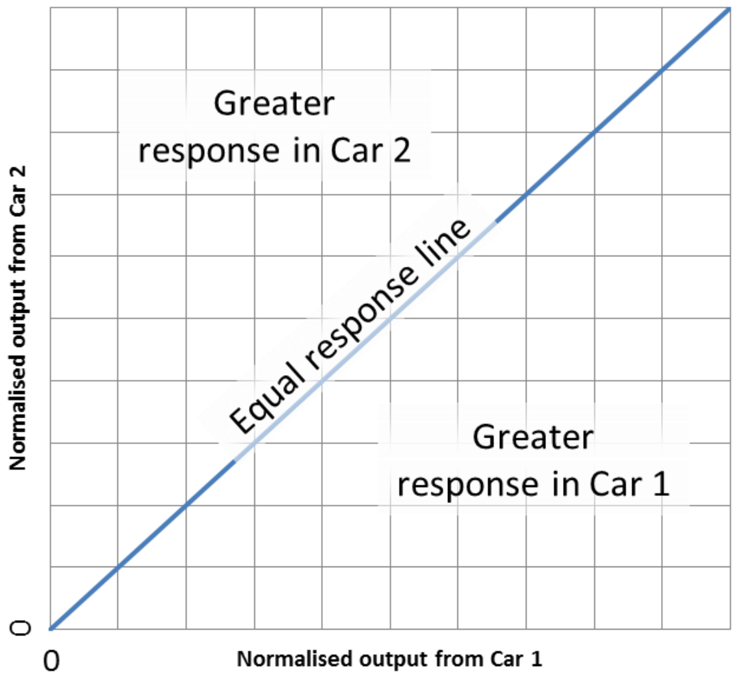


Figure 11. Compatibility domain described in Paper III.

The preliminary findings based on the current state of FE models support previous work on the significance of small overlap situations (Eichberger et al. 2007, Brumbelow and Zuby 2009). In relation to all the crash scenarios evaluated in Paper III, small overlap situations stand out as extreme in terms of passenger compartment intrusion. Whether the same tendencies would be found for a different car design is unknown, however, the methodology should be readily applicable for this type of studies. Furthermore, there are many combinations of car designs that could be investigated in terms of car-to-car crash response. The wheels and wheel suspension components have been identified as important load paths for small overlap crashes. The design and strength of these components are therefore likely to have a considerable effect on the structural response and consequently the categorisation into sub-types of small overlap crashes described in Paper IV.

5.2 Discussion of results with respect to crash scenario

As previously identified, robust structural behaviour constitutes a foundation for restraint system functionality. Further, optimising passenger car structures towards a limited set of crash load cases may introduce sub-optimisation in a wider perspective on crashworthiness. The concepts behind robust structural response should therefore be applicable to other crash scenarios, such as side and rear crashes. The compatibility domain suggested in Paper III may not be directly applicable in front-to-side crashes since injury risk is normally significantly higher in the struck vehicle than the striking vehicle in a front-to-side crash (Summers et al. 2003). For front-to-rear crashes, passenger compartment intrusion in the striking vehicle could be plotted versus relevant intrusion measurements in the struck vehicle. In this way, the compatibility domain may be employed to compare different structural concepts in order to ultimately balance injury risk in both the striking and the struck vehicle.

For the car-to-car simulations performed in Paper III, a rough probability distribution of oblique angles from Eichberger et al. (2007) were taken into account when setting up a simulation matrix. This indicated that most crashes occur at an oblique angle below 10° although the number of cases considered in that study was limited to twenty cases. Therefore, in order to obtain a comprehensive view on frontal crashes, it was decided to cover the complete range of oblique angles between 0 and 45°.

All studies in this thesis were focused on horizontal alignment in frontal car-to-car crashes; vertical misalignment was not addressed at all. Based on previous work, there appears to be agreement that vertical misalignment increases the risk of under riding and increased intrusion (Baker et al. 2008, Mizuno and Arai 2010). This is obviously an important factor to consider when striving for improved compatibility in car-to-car crashes. In terms of methodology development however, it was decided not to attempt to incorporate vertical misalignment as well.

When considering dimensioning crash scenarios for car structural design, road infrastructure design also plays an important role. An example of this is lane separation actions as pointed out by Eugensson et al. (2011). This type of road safety countermeasure can be used in order to avoid head-on collisions on roads with speed limits above a certain level and thereby reducing the crashworthiness design volume described in Figure 2.

Small overlap scenarios stand out as particularly demanding for car structures in terms of passenger compartment intrusion. Crash scenarios combining small horizontal structural overlap and oblique angles between 10° and 20° are suggested as dimensioning for intrusion in frontal car-to-car crashes. This finding should be confirmed with alternative vehicle designs using the methodology described in Paper III. Further, structural countermeasures for small overlap situations were not suggested or within the scope of this thesis. However, to prevent opposing front wheels to lock, it has been proposed to actively turn the front wheel toe-in in order to create a sliding plane from which the crash opponent could be diverted (Winkler et al. 2001). This action may require a substantial wheel rotation angle before achieving a positive effect. Such action must also be well balanced with the risk of a second impact in cases where turning of the wheels is successful in avoiding a first impact.

5.3 Discussion of results with respect to crash energy

For the studies conducted in this thesis, energy levels were estimated and probability distributions were not taken into account. For all studied scenarios, exposing the vehicle structure to increased energy resulted in either higher accelerations, larger intrusions, or both. From a car designer's perspective, the crash energy should be regarded as a framework for which structural integrity must be ensured. Again, road infrastructure can contribute significantly to Vision Zero, by setting speed limits to appropriate levels in order to keep crash energy levels within the crashworthiness design volume.

During the process of active safety systems becoming increasingly efficient in terms of avoiding and mitigating crashes, virtual tools will be essential in order to ensure that occupants will be protected in any crashes that still occur. These active safety systems will thus reduce the crashworthiness design volume as illustrated in Figure 12 which means reducing the energy level that passive safety systems need to be designed for. In a future without crashes, passive safety systems would consequently become obsolete.

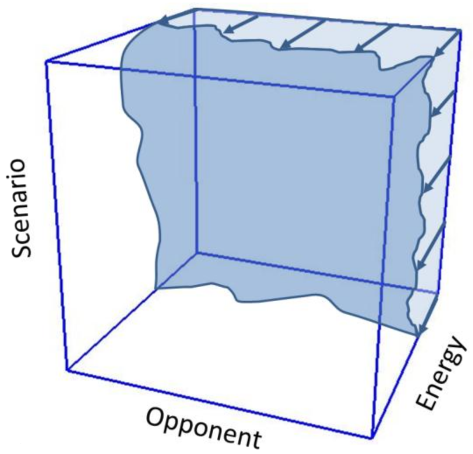


Figure 12. Visualisation of crashworthiness design volume, and how the number of design situations for passive safety can be reduced by active safety systems.

When investigating an adaptive front subframe in Paper V, it was found that such a solution could not be recommended at a closing velocity above 100 km/h for identical vehicles. This recommendation was based on intrusions being increased above the level of the base model with a non-releasing subframe. Furthermore, this result indicates that an adaptively detaching subframe may not be able to reduce fatalities and severe injuries at closing velocities above 100 km/h. An alternative approach to address these high-speed large overlap situations may be to increase the frontal structural force, e.g., by pressurised structures as proposed by Pipkorn and Håland (2005). In this way, it would be possible to change a stiff response in the end of the crash from the integrity of the passenger compartment to a more even deceleration over the same distance. However, there is undoubtedly an upper limit of velocity change that can be tolerated by the average car occupant without sustaining severe or fatal injuries. It was concluded in another study by Pipkorn et al. (2005) that occupant protection in frontal crashes up to 80 km/h is feasible but may require extended distances available for absorption of occupant kinetic energy.

The initial studies reported in Paper I and II were directed towards the applicability of adaptive front structures for reducing the risk of AIS1 neck injuries in frontal crashes. Although indications of harmful crash pulse characteristics were found in literature (Kullgren et al. 2000, Jakobsson 2004), no clear guidelines towards safer crash pulses with regards to AIS1 neck injuries were found. Therefore, no further studies were conducted within the scope of this thesis in order to use adaptive structures for this specific purpose. It is expected, however, that when crash pulse shape recommendations for reduced AIS1 neck injury risk can be given, adaptive structures may be one way to realise such crash pulses.

6 Conclusions

The following conclusions were drawn:

I. Structural adaptivity can reduce deceleration levels significantly in frontal crashes. This, however, requires that the available front structure crush length is not exceeded.

II. In order to achieve efficient structural adaptivity, global load paths need to be modified. When the material strength of the most significant energy-absorbing frontal structures were changed by ±50%, only moderate changes to the passenger compartment deceleration were observed.

III. A methodology for assessing structural robustness based on FE simulation models was proposed. This methodology points out large overlap situations as extreme in terms of occupant deceleration loading, whereas oblique small overlap situations were predicted to be extreme in terms of passenger compartment intrusion.

IV. The results obtained with an updated model based on validation against physical crash tests exhibited large variations in intrusion response as a function of the input variables lateral offset and oblique angle. Car-to-car front crash situations where the front wheels lock up were suggested as critical for occupant safety since this type of situation combined large intrusion with high crash longitudinal crash pulse severity and large lateral velocity change.

V. In the last study, the methodology in Paper III was used as a starting point for addressing the possibility of structural adaptivity to reduce crash pulse severity. It was shown that a detachable front subframe could be used at closing velocities below 100 km/h. Detaching the front subframe was suggested to reduce the crash severity equivalent to a velocity change reduction of up to 44%.

6.1 Lessons learned

Some experiences related to the number of crash simulations performed with the standard models in Paper III as opposed to the updated, more detailed models in Paper IV were gained. When searching for dimensioning scenarios, it would have been beneficial to use a fixed set of initial velocities to save simulation effort in the first stage. The car-to-car model validation work conducted in Paper IV provided several important insights into how details of the models (Centeno G. 2009, Dharwadkar 2011) affect the structural response. Therefore, a larger number of simulations could have been based on the updated Paper IV models to scan the crash configuration domain, also denoted crash scenario matrix.

The detail level of the FE model used for Paper II (16,000 elements) appears to be a limiting factor when compared to the model used for Paper III and IV (>2,000,000 elements). Element size has a direct effect on a model's ability to capture buckling modes of structural components; a coarse model can simply not capture some deformation modes that a finer model can. Nevertheless, usage of simplified or coarse models should not be disregarded since using these can be an effective way of sorting between high-level decisions. For instance, the studies on adaptive front longitudinal rails, or side members, in Paper II indicated that significant load paths would need to be affected in order to considerably affect the vehicle crash pulse. This was later supported when the concept of an adaptive subframe, which is a major load path in frontal crashes, was demonstrated in Paper V.

Important strategic decisions should be made regarding on how to spend simulation resources. On the one hand, crash models can be made increasingly detailed in order to predict crash performance with higher precision in a constant number of load cases, corresponding to moving right on the horizontal axis in Figure 13. On the other hand, as was demonstrated in Paper III, the number of load cases can be increased with a constant or even reduced model detail level, corresponding to moving up on the vertical axis in Figure 13. Based on studies conducted for this thesis, it is not obvious which choice is more effective in terms of designing for real-world crashworthiness. A trade-off is needed between increasing the model detail level and the number of situations to evaluate. When a new computer system generation becomes available, as illustrated in Figure 13, which enables a larger number of computations to be performed, this trade-off needs to be considered. It is expected that this balance will be affected by which part of the development process that is considered, i.e., at earlier phases of development a lower model detail level may be preferable while exploring a larger number of situations.

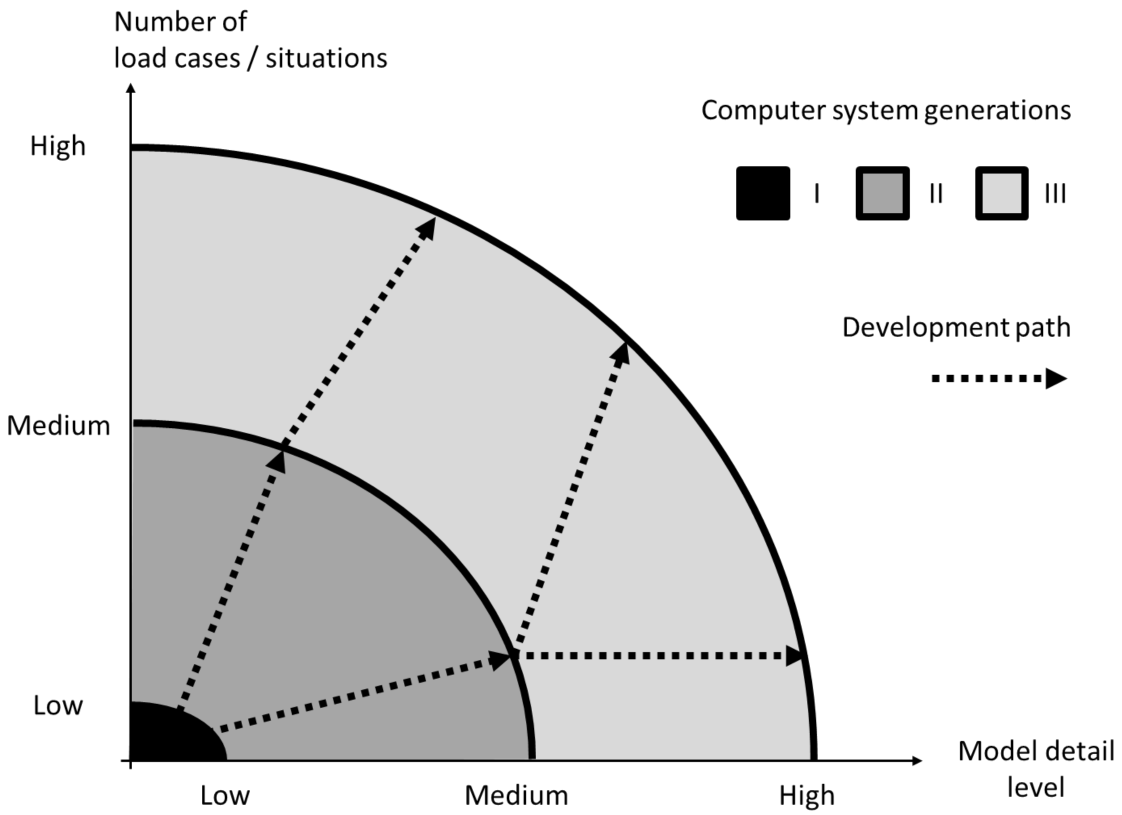


Figure 13. Schematic view on trade-off between model detail level and number of situations to evaluate during design process of passenger cars. Alternative development paths marked by dotted arrows.

6.2 Suggestions for future studies and development work

Based on the research performed for this thesis, there appears to be three obvious paths for continued work. Table 1 was therefore updated with an outline of these paths in Table 3.

The first path would be useful for going into further details of passive safety issues and vehicle structures in frontal crashes as illustrated in Table 3. Issues such as vertical misalignment and robustness in car-to-car crashes with dissimilar vehicles that were not covered by the present studies could be addressed. Furthermore, connecting crash simulation setups more closely to accident data including the probability of crash scenarios occurring may be one way of advancing towards robust structural design. A further application could be single vehicle frontal crashes with fixed objects or large animals.

A second path deals with the generalisation of the methodology to encompass other crash scenarios such as rear and side crashes. Using a set of bullet vehicle models, a structural robustness map could be established in a similar fashion as was made for frontal crashes.

A third, and perhaps the most innovative, way forward is looking into integrated safety for frontal crashes, i.e., the interaction between active and passive safety systems aiming to reduce accidents and injuries. The first step in this work is to combine the methodology for structural robustness described in this thesis with detailed occupant models. This could possibly quantify injury risks related to intrusions and decelerations on a considerably more detailed level than can be achieved using intrusion measurements and crash severity indicators such as VPI (ISO 2012) or OLC (Stein et al. 2011).

Table 3. Overview of how methods and solutions could be expanded to include additional crash scenarios and safety systems. Numbers refer to alternative future work paths.

		Frontal crashes	Rear crashes	Side crashes	Rollover, run-off-road	Pedestrian protection	...
Passive safety systems	Structures	1	Methodology generalisation				2
	Restraints						
Active safety systems	Mitigation	Integration					
	Avoidance	3					

As continued focus is being placed on energy efficiency, reducing vehicle mass is one of the options for producing cars that use less fuel. The safety consequences of a future lighter vehicle fleet in the United States has been thoroughly analysed by NHTSA (2013). If lighter materials such as ultra-high strength steel, aluminium or composites are implemented, this must be done with robust structural response in mind. This kind of assessments during the development phase requires both detailed models showing the structural response and broad assessments of crash situations.

Another potential opportunity to increase energy efficiency fuel is electric vehicles, an area where several research projects have been initiated, e.g., ELVA (2013) and EVERSAFE (2013). The transition to alternative powertrains may introduce challenges, as well as opportunities in terms of the structural response when future vehicles are involved in crashes. Furthermore, safety design experience built around conventional vehicles may not be directly applicable to vehicles incorporating alternative powertrains. Therefore, it is proposed that it would be beneficial to apply the methodology presented in Paper III when designing electric vehicles offering robust structural response.

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Paper I

"Structural adaptivity for acceleration level reduction in passenger car frontal collisions"

Wågström, L., Thomson, R., Pipkorn, B. (2004)

International Journal of Crashworthiness 9 (2), 121-127.

Structural adaptivity for acceleration level reduction in passenger car frontal collisions

doi:10.1533/ijcr.2004.0285

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Abstract: A mathematical model was developed to explore and demonstrate the injury reducing potential of an adaptable frontal stiffness system for full frontal collisions. The model was validated by means of crash tests and was found to predict the peak accelerations of the crash test vehicles well, whereas correlation concerning mean acceleration or residual crush was not found. Vehicles were divided into three mass classes, and a test matrix was established in order to evaluate different combinations of vehicles involved in frontal crash at three closing velocities. In a baseline simulation setup, constant stiffness values were used and the results were compared to the corresponding simulations using adaptable frontal stiffness. Results show promising acceleration peak reductions at low speeds, implying that injury risk reductions are possible.

Key words: Frontal collisions, energy absorption, front stiffness, adaptivity

NOTATION

m	Mass
K, k	Stiffness coefficient, force per displacement
v	Velocity
x	Displacement axis
L	Light car, simulation weight 1200 kg
M	Medium car, simulation weight 1600 kg
H	Heavy car, simulation weight 2000 kg
v_c	Closing velocity, relative velocity
E_k	Kinetic energy
E_d	Deformation energy
$x_{\max}$	Maximum spring deformation
$d_{\max}$	Maximum vehicle front deformation
d_0 , d_1	Vehicle front deformation
$F_{\max}$	Maximum vehicle front total force
F_1	Vehicle front total force

INTRODUCTION

As passenger cars are becoming safer and protecting occupants more efficiently against serious injuries or death, more attention must be directed to the less serious, long-term injuries that occur even in low-speed accidents. Studies have shown that neck injuries, which are frequent in low-speed collisions, are the most common car occupant injury type leading to medical impairment (von Koch [1]) and are thus creating societal costs that could be avoided with novel design strategies. In-depth studies of crash pulses from real-world frontal collisions have shown a correlation between acceleration levels and injury risk (Kullgren [2]), indicating that high vehicle accelerations during frontal impact increase the risk of long-term consequences to the neck. In the cases where long-term consequences where recorded, the maximum of the average acceleration pulses was more than 70% higher than in the cases where none or only short-term consequences were found. This data implies that reduction of the peak acceleration during crash would decrease the risk of occupant long-term consequences to the neck substantially.

Since the crash pulse of a given vehicle depends on the initial velocity, collision object and deformation mode of the engaged structures, the only way for a vehicle manufacturer to affect this crash pulse, and hence the peak acceleration, is to change the characteristics of the energy absorbing parts of the car body. In severe collisions, most of the front structure is deformed and the passenger compartment must withstand high forces in order to protect occupants from injuries related to intrusion. Consumer tests such as Euro-NCAP, where test vehicles are run into an offset deformable barrier at 64 km/h, have promoted car manufacturers to design vehicles with high intrusion resistance. This means that for this type of situation, modern vehicle frontal structures are often effectively designed to resist compartment intrusion. However, in low severity collisions, these structures might be too stiff

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and hence subject the car occupants to high accelerations. In other words, the structure will deform less than what would be possible if the stiffness could be altered with collision severity, e.g. the closing velocity of colliding objects.

Suggestions to design adaptable frontal structures have been found in the literature. Solutions including hydraulic systems with adaptive parts have been proposed by Witteman [3] and Jawad [4]. In contrast to these studies, the present research project does not constrain the adaptive frontal structure to current technologies, neither regarding stiffness alteration capabilities nor current practice of building cars. Such limitations would inhibit the design of a new crashworthiness concept. Instead, the project aims at finding guidelines about how to choose deformation characteristics that lead to less harmful acceleration pulses during low-speed collisions while maintaining or even improving intrusion resistance at higher crash velocities in frontal collisions.

OBJECTIVE

The aim of this work was to assess the occupant injury-reducing potential of adaptive energy absorbing front structures of vehicles engaged in full-frontal collisions, with special reference to long-term consequences to the neck. At this stage of the research project, the total peak acceleration of the crash pulse is used as indicator of injury risk. Part of the objective was also to identify feasible steps in future modelling and refinement in acceleration pulse injury indicators. By creating a dynamic model of car-to-car collisions, the possible peak acceleration reduction was to be investigated.

METHOD

Each one of two cars in a simulated crash was assumed to have a known mass, denoted m_1 and m_2 respectively, as indicated in Figure 1. The collision impact velocities are known to be v_1 and v_2 and the displacements of the two masses were assumed to occur in one direction only (denoted x). Similarly, both cars were assumed to have a linear frontal stiffness coefficient denoted K_1 and K_2 (force/displacement). The reader should be aware of that in the model, no distinction has been made between elastic and plastic deformation; the term "stiffness" is usually associated with linear elastic deformation, however since the model describes a linear elastic-plastic behaviour where the relative proportions of elastic and plastic deformation are irrelevant, "stiffness" is used to describe the slope of total crash force vs. deformation (elastic or plastic).

The combined stiffness of the springs is:

$$K = \frac{K_1 \cdot K_2}{K_1 + K_2} \qquad [1]$$

Typical values for vehicle frontal stiffness were chosen after a categorization of the car fleet was made with a separation at a test weight of 2000 kg. NHTSA crash test data from Summers [5] suggest that the initial stiffness of cars can be approximated to

$$K = 1000 \text{ kN/m, Test weight} < 2000 \text{ kg} \qquad [2a]$$

$$K = 2000 \text{ kN/m, Test weight} \geq 2000 \text{ kg} \qquad [2b]$$

It should be pointed out that these values only represent the initial slope of the force-deformation curves, and by assuming linear springs the forces are highly overestimated for larger measures of deflection (which is calculated from the second integral of acceleration). Furthermore, large vehicles such as sport utility vehicles, vans and trucks were modelled with the greater stiffness even if their test mass was below 2000 kg. The solid lines in Figure 2 show how the force was assumed to depend on deflection, the dashed lines represent a typical behaviour in real crashes.

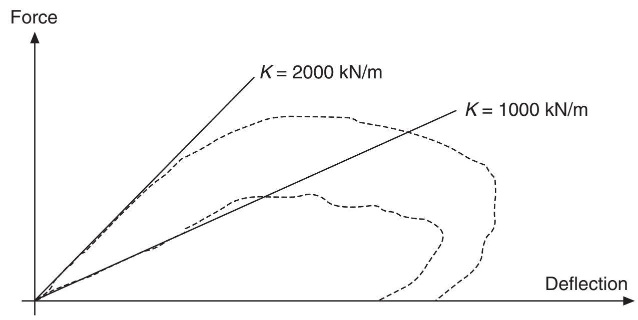

Figure 2 Stiffness coefficients.

MODEL VALIDATION

In order to verify that the model produces deformations and accelerations comparable to the ones found in real vehicles, calculations of single car to rigid barrier collisions were compared to full frontal, rigid barrier crash tests in the NHTSA Vehicle Crash Test Database [6]. For this application only one spring was employed, and the mass-spring system appeared as shown in Figure 3.

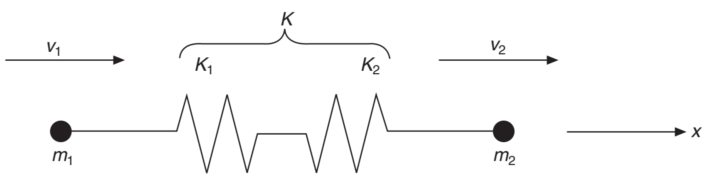

Figure 1 Mass-spring model after contact.

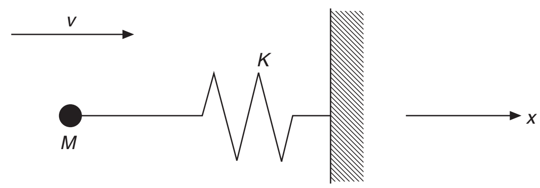

Figure 3 Rigid wall model.

doi:10.1533/ijcr.2004.0285

A Newmark time integration scheme described by Geradin [7] was applied to simulate the transient behaviour of the system. As the spring is deformed to a maximum, i.e. when the mass has zero velocity, the mass will rebound from the wall. Using a linear elastic spring would mean that the spring eventually would act in tension. This was prevented by setting the contact stiffness K to zero when the velocity of the vehicle changes sign, i.e. assuming no structural restitution. It should be noted that the assumption of zero restitution does not affect the vehicle peak acceleration in the model, as this maximum will occur exactly when the velocity is equal to zero.

To validate the model, a small number of vehicles from crash tests were chosen to span a range of passenger vehicle types, test weights and velocities according to Table 1 below. All vehicles were tested against a rigid barrier with 100% overlap.

RESULTS OF MODEL VALIDATION

Referring to Figure 4, it is obvious that the model does not match the test results in terms of deformation, although a correct trend is predicted. It must be noted that due to the model characteristics, the maximum deformation in the model is compared to the average residual crush of the vehicles in the crash tests.

As indicated above, the linear spring model generally overestimates the maximum deformation. The overall error is 26% and since the sample size is very small this error estimation has a large uncertainty.

Comparing the maximum accelerations of tests and simulations as indicated in Figure 5, a better correspondence is found compared to deformations in Figure 4. Again, the model is a simplification of realistic behaviour, which means that acceleration histories are quite different in crash tests and simulations.

On account of the linear properties of the model, the peak acceleration occurs in the end of the crash phase, which is after some 50–60 ms, according to Figure 6. This is about the same time as the maximum is reached in a real crash pulse, however the real crash pulse will generally last longer, since a larger velocity change is obtained in the model before this point in time, as indicated by the area under the graph in Figure 6. Another consequence of using a linear model is the overestimation of mean acceleration, which makes the model unsuitable for predicting vehicle mean acceleration.

All acceleration signals from crash tests were filtered according to SAE J211 using channel frequency class 60. There was no clear trend when comparing test data to the

Table 1 **Selected crash tests for model validation**

NHTSA Crash test No.	1705	2198	2327	3150	3537	3902	4196	
Vehicle	Cadillac Seville	Saab 900	Volvo 850	Ford Taurus	Toyota Yaris	Ford F150	Isuzu Rodeo	Unit
Model Year	1992	1995	1996	2000	2001	2001	2002	
Test weight	1855	1601	1634	1727	1138	2292	1909	kg
Test velocity	56.6	56.5	47.6	48.1	56.5	47.7	38.9	km/h
Modeled frontal stiffness	1000	1000	1000	1000	1000	2000	2000	kN/m

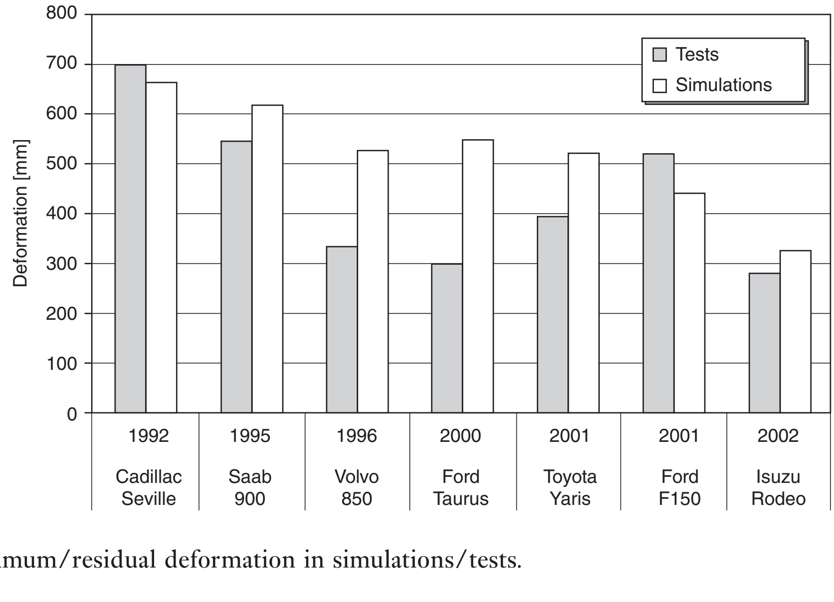

Figure 4 Maximum/residual deformation in simulations/tests.

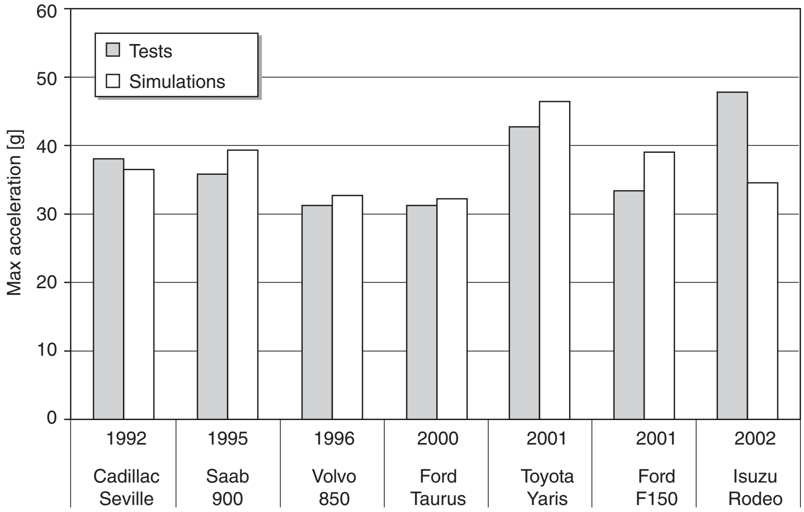

Figure 5 Maximum acceleration in simulations and tests.

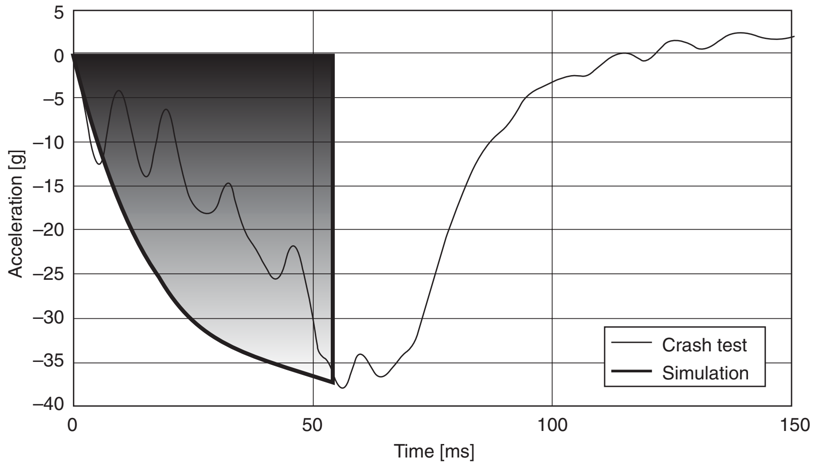

Figure 6 Crash pulse shape comparison, simulation vs. crash test.

model, i.e. the model will overestimate the maximum acceleration in some cases but underestimate it in about the same number of cases. This is also reflected in the fact that the overall error is about 2%. Again, this result must be put in relation to the small sample size.

The conclusion of the validation is that the model predicts the peak accelerations at the chosen crash velocities with an error that is small in relation to the difference in peak accelerations found by Kullgren [2] that separate long-term and short-term injury risks. This suggests that the level of uncertainty in the model is much smaller than the difference in acceleration that was desired. Thus the confidence level of the model is suitable for the analysis.

ADAPTIVITY ANALYSIS

To simulate full-frontal collisions between cars of dissimilar mass or frontal stiffness, three vehicle classes were defined according to Table 2. Note that stiffness values are assumed to be equal for all cars lighter than 2000 kg. Presumably, there is in reality a gradually increasing stiffness as the mass is raised from 1600 to 2000 kg, however this is not accounted for in the current car-to-car collision analysis.

Table 2 **Vehicle classes used for simulations**

Name	Annotation	Mass [kg]	Front stiffness [kN/m]
Light	L	1200	1000
Medium	M	1600	1000
Heavy	H	2000	2000

Using all combinations of the three car classes, six simulation settings are possible for each closing velocity. To limit the number of calculations, three typical closing velocities are chosen, $vc_1 = 120$ km/h, $vc_2 = 80$ km/h and $vc_3 = 40$ km/h.

Six combinations of cars involved in accidents at three closing velocities gives a total of eighteen simulations that are run with the baseline configuration, i.e. constant stiffness. The results of these simulations will then be

 doi:10.1533/ijcr.2004.0285

compared to the results using an adaptive deformation system.

Adaptivity algorithm

The adaptable vehicle stiffness strategy was based on a target deformation length of 700 mm, a feasible crush distance for most types of present-day vehicles. From this, an applicable front stiffness is calculated upon the assumption that both cars have an identical frontal stiffness at the specific time of impact (adapted to the crash severity measure, which in this case is the closing velocity).

Before the crash, the system of two cars has the total kinetic energy E_k of:

$$E_k = \frac{m_1 v_1^2}{2} + \frac{m_2 v_2^2}{2} \text{ where } v_1 = v_2 = \frac{v_{ci}}{2} \; i = 1, 2, 3 \qquad [3]$$

If the maximum deformation in the springs is denoted $x_{\max}$ (since the front stiffness is the same for both vehicles, both vehicles will have the same maximum deformation, here 700 mm), the total deformation energy E_d is:

$$E_d = \frac{k_1 x_{\max}^2}{2} + \frac{k_2 x_{\max}^2}{2} = k x_{\max}^2 \qquad [4]$$

Assuming that all the kinetic energy E_k [kJ] is transformed into deformation energy E_d [kJ], the stiffness k is simply:

$$k = \frac{E_d}{x_{\max}^2} = \frac{E_k}{x_{\max}^2} = \frac{E_k}{0.49} \text{ [kN/m]} \qquad [5]$$

In practice, this means that the frontal stiffness of each collision partner is altered according to Figure 7.

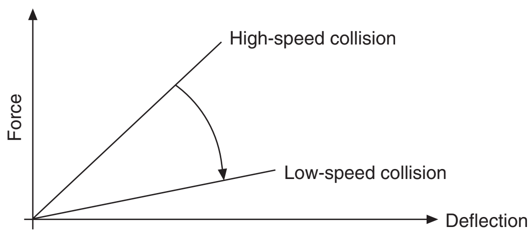

Figure 7 Stiffness adaptation in simulations.

This algorithm implies that the frontal deformation characteristics would have to be altered prior to impact, i.e. there is no stiffness adaptation during the crash.

RESULTS OF ADAPTIVITY ANALYSIS

Running the simulations with adaptive frontal stiffness values k gives lower values for maximum acceleration according to Figure 8 (the baseline simulations were run with the frontal stiffness values given in Table 2) The following acronyms are used: A collision between two light vehicles is denoted LL, a light-to-medium car collision is denoted LM and a collision between two heavy cars is denoted HH etc.

As shown in Figure 8, all peak acceleration values are decreased as a result of the adaptive system. Fairly small reductions of peak values are found at the highest impact velocity: on average 14% lower peak acceleration values at a closing velocity of 120 km/h. When the closing velocity is decreased to 80 km/h, the level of maximum acceleration values is also decreased: the highest acceleration peak is roughly 42 g for the lighter car in the light-to-medium car collision, compared to some 62 g in the 120 km/h simulation. Figure 8 also shows how the structural adaptation leads to acceleration levels lower than the smallest acceleration peak in the baseline setup and an overall reduction of 43% when the closing velocity is set to 80 km/h. Reducing the impact energy even further by setting the closing velocity to 40 km/h, suggests that the average peak value could be reduced by as much as 73%.

As can be observed above, the greatest relative acceleration reduction is found at the lowest closing velocity (73% compared to 43% and 14%). It must be observed that at such a low collision speed, the calculated adaptive stiffness is less than 100 kN/m, which is only some 10% of the original value.

DISCUSSION

Systems including an adaptive frontal stiffness studied in this analysis would require pre-crash sensors that can provide control units with information both regarding the state of the own vehicle as well as the collision object. When that is possible, measures can be taken to avoid the collision in the first place, but also prepare the vehicle for a probable crash. However, the development of such sensors lies beyond the scope of the research project presented in this paper.

A central question to consider in structural adaptivity is whether the optimal way of changing the frontal stiffness is by decreasing or increasing the internal force levels of the deforming vehicle parts. In this preliminary study it was assumed that adaptable vehicles would be weakened as they encounter a collision partner or object at low velocity. The advantage of this approach is that if the system fails to activate during a collision, the occupant will have sufficient protection from intrusion and the passenger compartment will resist collapse. On the other hand, if the system has been activated and there is a sensor inaccuracy, this might prove devastating if the car is colliding at a substantially higher closing velocity. To avoid this scenario, the opposite situation is possible. This means that the structure is less stiff in an original state and that it is shifted into a stiffer mode as a severe situation approaches. For such an approach to be feasible, the less stiff mode must provide enough protection not to result in collapse of the compartment and cause dangerous intrusions. In the case of system malfunction, i.e. a crash at high velocity with a low stiffness, the occupant would have to be protected against intrusion but will be subjected

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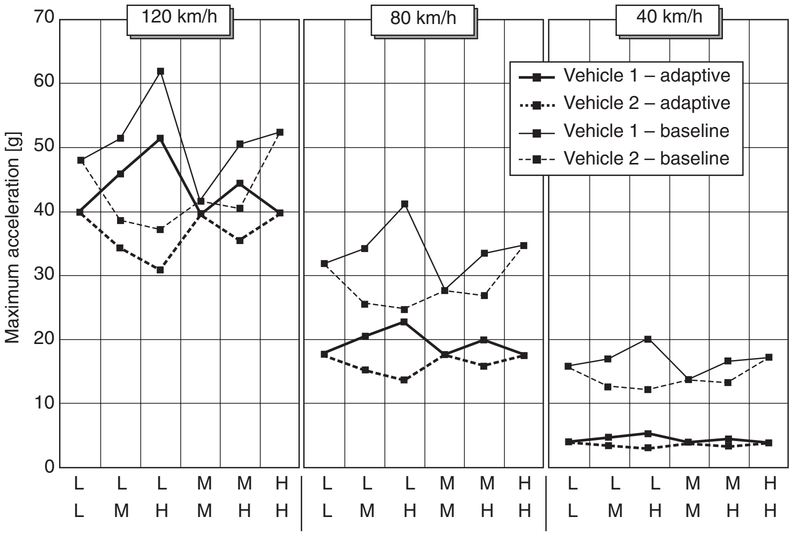

Figure 8 Maximum accelerations for closing velocities 120, 80 and 40 km/h.

to high acceleration. It seems that the latter approach is preferable, since vehicle occupants must be guaranteed a lowest level of protection in case of undesired system response.

As illustrated in Figure 9, a fully developed adaptive system could have two extreme levels, low and high. The two series represent a desirable force-deformation behaviour for the same kinetic impact energy in these two adaptivity modes, respectively. By using the lower stiffness level, a larger deformation (d_{max} in Figure 9) would be required to accommodate for the current kinetic energy (i.e. the area under the graph). However, if the higher stiffness level is engaged, only a smaller deformation would be needed (d_1 in Figure 9), leaving further capacity for extremely high impact energies. If it is assumed that present-day vehicles have a force-deformation curve somewhere in between the high and low level in Figure 9, an adaptive system like the one proposed in the current work would benefit the occupant twofold by (1) lowering acceleration levels in low severity accidents and (2) increasing the deformation energy absorption in high-velocity impacts. The adaptive system is proposed to be active only in a first deformation zone, indicated by d_0 in Figure 9, which presumably is a shorter distance than the proposed deformation length of 700 mm in the adaptivity algorithm. As a consequence of this, the difference in energy absorption between a high and a low stiffness level is reduced compared to the results found in this study. Furthermore, the force levels F_1 and F_{max} would have to be designed with compatibility aspects in mind, i.e. ensuring that large vehicles do not produce force levels leading to passenger compartment collapse in smaller car-to-car collision partners.

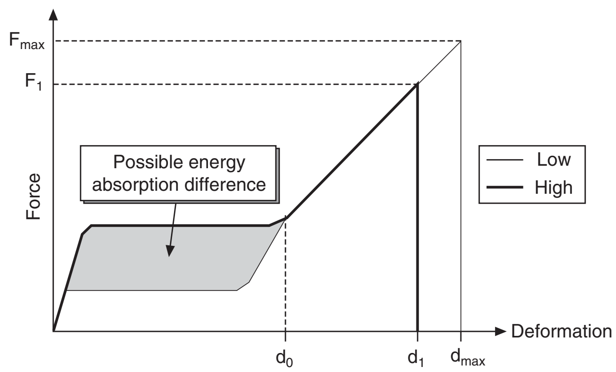

Figure 9 Force-deformation outline for adaptive frontal stiffness.

Finally, the level of stiffness reduction proposed by the simple linear spring model is most likely difficult to achieve in a real structure. A reduction of more than 90% means a dramatic change of deformation properties and presents a challenge in terms of designing energy absorbing structures.

RECOMMENDATIONS FOR FUTURE RESEARCH

An important issue in this study is of course that in real life, accidents do not occur in the idealized way the model assumes. Aspects such as vehicle impact orientations must be taken into consideration. Moreover, future models with more detail must consider vehicle incompatibility, and ideas for solving this growing problem for vehicle manufacturers should be incorporated in a final adaptivity concept.

In the model, one of the assumptions is total kinetic energy transformation into deformation energy which was done in order to simplify the adaptivity algorithm. New research models must be able to describe structural adaptivity not only as a function of vehicle mass and velocity, but also depending on the front geometry and stiffness of crash partners.

Furthermore, the shape of the acceleration pulse will be of interest for future studies. Improved models will need to include ways of not only controlling the vehicle peak acceleration, but also where this peak occurs and the magnitude of the mean acceleration in different stages of the crash pulse. When analyzing the pulse shape, the inertia effects of the engine is necessary to include: the current design of passenger cars makes the engine inertia a problem when acceleration pulses are to be optimized.

All of the issues above could be addressed by using finite element models of passenger cars.

ACKNOWLEDGEMENTS

This study was supported by Autoliv Research.

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Paper II

”Structural adaptivity in frontal collisions: implications on crash pulse characteristics”

Wågström, L., Thomson, R., Pipkorn, B. (2005)

International Journal of Crashworthiness 10 (4), 371-378.

Structural adaptivity in frontal collisions: implications on crash pulse characteristics

doi:10.1533/ijcr.2005.0353

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Abstract: Today's passenger cars protect occupants better than ever against most injury types in passenger car frontal collisions. There is, however, one notable exception: neck injuries. Studies have shown that high mean vehicle deceleration is likely to lead to a greater risk of sustaining neck injuries. In order to design future cars that minimize occupant injury risk, it is suggested that the response of the front structure should be adapted to impact severity. A finite element model was used to predict the implications on acceleration time history by yield-strength variation of the longitudinal rails. Results indicate that lower mean deceleration can be attained by lower-yield-stress material, but caution must be taken to avoid stiff engine- firewall contact as this can create high mean decelerations. Furthermore, results indicate that for an adaptable frontal structure to reduce mean acceleration and neck-injury risk, global load paths must be controlled in frontal impacts.

Key words: Frontal collisions, energy absorption, front stiffness, structural adaptivity.

NOTATION

x	Vehicle longitudinal direction
y	Vehicle lateral direction
z	Vehicle vertical direction
Peak deceleration	Maximum value of filtered deceleration time history
Mean deceleration	Average value of filtered deceleration time history

INTRODUCTION

In real-life passenger-car crashes where injuries are recorded, frontal collision is the most frequent accident type. Statistics show that about 50% of real-world crashes are frontal impacts [1]. Since many of these accidents involve head-on impact with an approaching vehicle, the closing velocity between the involved vehicles is often large compared to side or rear impacts. Because of this, the design of crashworthy frontal structures of vehicles has been a major concern to vehicle manufacturers for a long time. Mandatory crash tests as well as consumer information crash tests has also contributed to modern cars are able to maintain structural integrity even in high-velocity crashes; thereby reducing the risk of severe and fatal injuries. The evolution of design has made cars safer.

Evidence of safer cars can be found in crash statistics, when injuries sustained by passengers in cars introduced in 1995-99 are compared to those of passengers in cars from 1980-84 [2]. These data suggest that the relative disability and fatality risks for various parts of the body have decreased substantially from 1980 to 1995. This applies to all body regions, with one alarming exception: injuries to the neck, where the vast majority of injuries are classified as AIS1 injuries [3]. Although AIS1 neck injury risk has been observed to be greater in rear impacts than in frontal impacts, a great number of neck injuries are sustained in frontal impacts because of the large number of frontal impacts [4]. What, then, could explain this trend in AIS1 neck injury risk?

If there was a clear understanding of the neck-injury mechanism in frontal impacts, the answer to this question would be easier to find. However, there is still no universally accepted explanation as to why these injuries occur. Variables such as occupant stature, weight and gender, combined with the characteristics of restraint systems, have been found to affect AIS1 neck injury risk [4]. Furthermore, data obtained from crash-pulse recorders fixed to the passenger compartment in real-life crashes show connections between the crash-pulse characteristics and the risk of AIS1 neck injuries. Kullgren et al. [5] found that the shape of the crash pulse is a possible risk factor; a large difference between the 2nd and 3rd part of

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a 100 ms pulse seems to increase the risk of long-term neck injury. The same study also showed that the mean deceleration of the average crash pulses was more than 45% greater in cases where long-term disability to the neck was found, compared to cases where none or short-term neck injury was recorded. After sorting crash pulses in groups of different velocity changes, Jakobsson [6] did not find significant differences in the average mean deceleration between cases with no AIS1 neck injuries compared to those with initial AIS1 neck injuries; however, data for the evaluation of crash pulses causing long-term neck injuries were not available in Jakobsson's data set. This suggests that the effect of a high mean deceleration on short-term neck injuries is small compared to the corresponding effect on long-term neck injuries. Nonetheless, both crash-pulse-recorder studies [5, 6] suggest mean deceleration as a candidate crash-severity measure regarding AIS1 neck injuries: If there is a relation between high mean deceleration and risk of neck injury, what could explain that the increased neck-injury risk in modern cars?

Since the deceleration of the passenger compartment in frontal crashes depends on the deformation mode of frontal structures, one explanation for the increased neck injury risk is that frontal structures have become stiffer, i.e. exhibit greater total force levels for a given deformation distance. Studies of both North American and European crash test data point to the probability of such a trend. The first study analyzes full-frontal-crash-test data from the United States New Car Assessment Program (US-NCAP) from 1982 to 2001 [7]. It was found that the dynamic stiffness, i.e. the force-deflection relationship found from the double integral of the vehicle deceleration, has increased gradually in the studied time period. The second study was done on European vehicles tested in offset frontal crash in the European New Car Assessment Programme (Euro-NCAP) [8], where a trend of increasing vehicle front stiffness was suggested.

The results presented above suggest that the frontal stiffness of cars has increased and that the mean and/or peak deceleration of current vehicles may be larger than the decelerations found in older vehicles. Since neck-injury risk has increased within the same time frame, it is probable that one way of decreasing this injury risk in future cars is by lowering decelerations in frontal impacts compared to today. For a given crash velocity, lower mean deceleration can be achieved only in one way: making the deformation distance longer. For obvious reasons, the deformation space is limited in passenger cars and hence there is a theoretically lowest possible mean deceleration. Furthermore, even if this deformation distance is utilized optimally, i.e. minimizing mean deceleration for one crash configuration, the front structures will not be optimal with regard to mean acceleration for all impact velocities.

As described above, greater passenger compartment integrity appears to have made vehicles less forgiving in terms of deceleration-related injuries such as long-term neck injuries. To reduce harmful crash pulses caused by stiff frontal structures, it is probable that the optimal frontal structure should have a stiffness adapted to the crash situation. The logical approach to achieve lower deceleration is to weaken the engaged structural members. By altering the structural deformation characteristics in suitable locations of a passenger car front, the response could be made sufficiently supple for a low-velocity crash while the front-end stiffness is maintained or increased in a high-velocity crash.

AIM

The objective of this study was to identify the most important energy absorbing structural components of a small car in a frontal barrier crash and observe the effects on the crash pulse when the strength, i.e. yield stress, of these components were decreased or increased. The material data were kept constant throughout the simulation: no transient material properties were studied.

METHOD AND PRELIMINARY RESULTS

In order to simulate a range of impact configurations, a full-vehicle LS-DYNA finite-element model of a Geo Metro was acquired from a public model archive at the National Crash Analysis Center [9, 10]. Selected for computational efficiency, a reduced version of the full model was used containing 19,000 nodes and 16,000 elements with a total mass of 808 kg (see Figure 1).

Barrier impacts were modeled by prescribing an initial translational velocity (in x-direction according to the coordinate system indicated in Figure 1) for the vehicle and adding a rigid barrier. Two barrier configurations were simulated: Full-width impact, where the plane defining the barrier was infinite; and offset impact where a finite surface was used to define a rigid barrier that covered 40% of the maximum vehicle width. To simulate internal contact between the different parts of the vehicle, an automatic, penalty-based, contact algorithm in LS-DYNA was employed [9].

Three accelerometers were placed in the vehicle model according to Figure 2, and accelerometer data was filtered

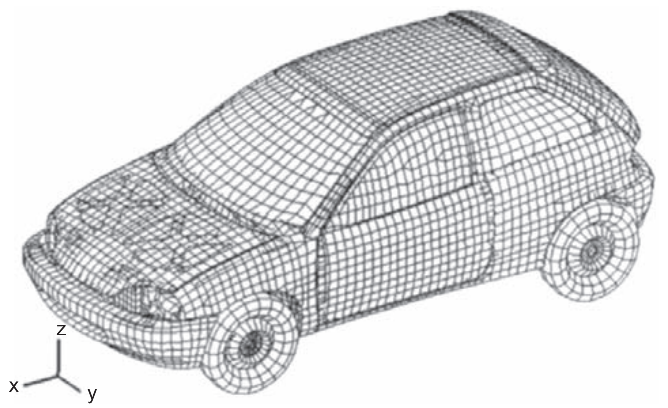

Figure 1 NCAC finite element model of Geo Metro.

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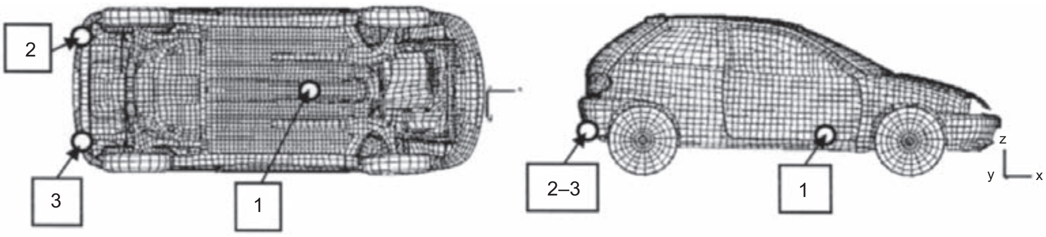

Figure 2 Bottom and side view of Geo Metro showing accelerometer locations.

according to SAE J211 using Channel Frequency Class 60 [11]. The first accelerometer (Indicated by '1' in Figure 2) was placed at the vehicle center of gravity and the second and third accelerometers ('2' and '3' in Figure 2) were placed on the left and right side of the rear bumper where no deformation was expected. Ideally, accelerometers should be placed in stiff locations, i.e. on structures with high resonance frequencies compared to the frequency content of the structural response. Unfortunately, the center-of-gravity accelerometer could not be placed near a stiff structure and therefore exhibited vibrations that could not be filtered out using a 60 Hz filter. For this reason the rear accelerometers were used when comparing acceleration histories between simulations.

When analyzing simulation results, the mean deceleration was defined as the average deceleration from the first barrier contact to the time when the filtered deceleration signal was zero. This time was found by inspection of the deceleration graphs, thus it is a subjective measure of the deceleration end time.

Since a prerequisite of the study was to identify those parts that affect the crash pulse the most, the first task was to divide the front structure into components significant to the response in frontal impacts. Figure 3 illustrates the chosen components and their location in the vehicle front.

In four preliminary simulations, a full-frontal rigid-barrier crash and a 40% offset-barrier crash at 32 km/h and 56 km/h, the relative energy absorbed by the selected parts were studied (see Table 1).

As shown in Table 1, all preliminary simulations showed that the longitudinal rails represented the largest portion of absorbed internal energy for impact velocities of both 32 km/h and 56 km/h against a rigid barrier. To study

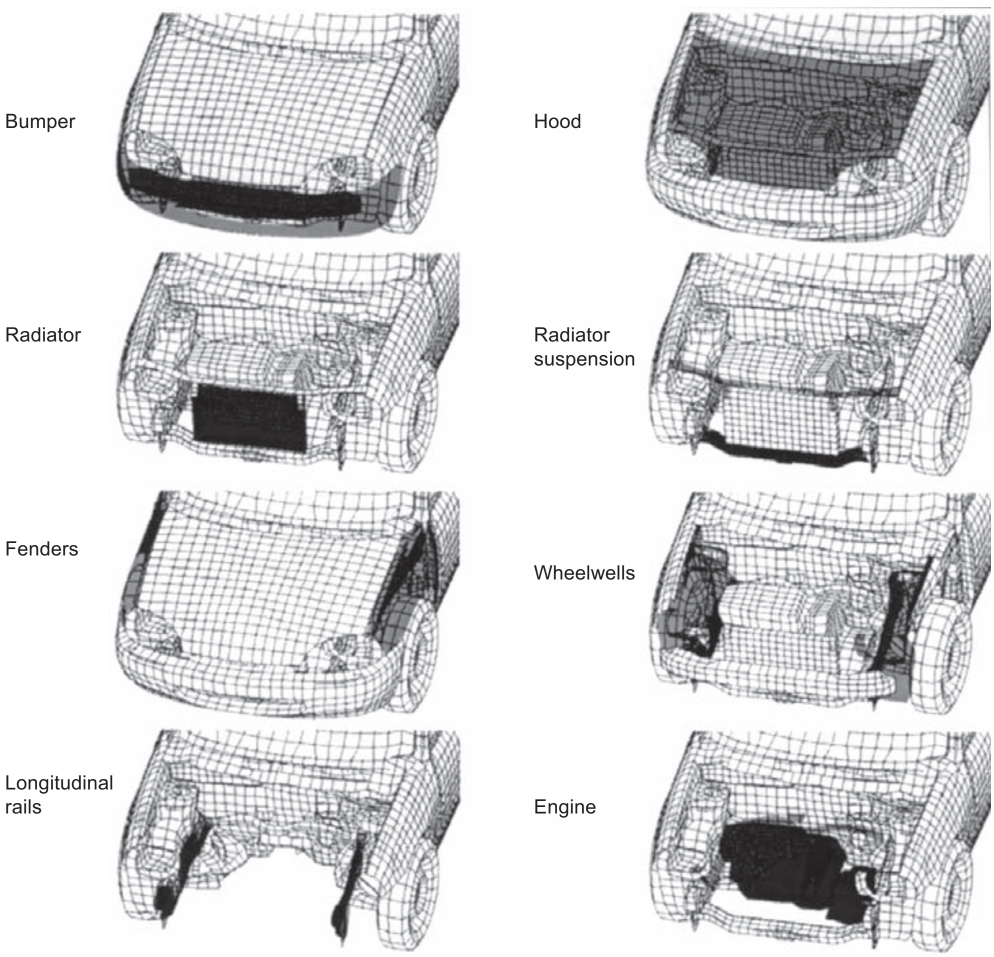

Figure 3 Structural components of vehicle front.

Table 1 **Relative energy absorption of front structural components in preliminary simulations**

Rigid barrier type Impact velocity	Full-frontal		40% offset	
	32 km/h	56 km/h	32 km/h	56 km/h
Bumper	29%	24%	29%	26%
Hood	3%	4%	7%	8%
Radiator	1%	3%	3%	5%
Radiator suspension	2%	2%	4%	4%
Fenders	1%	2%	2%	3%
Wheelwells	16%	16%	15%	18%
Longitudinal rails	44%	45%	36%	28%
Engine	3%	3%	3%	9%

the effect on crash pulse characteristics and the possible implications of structures with alternating structural strength, the material of the longitudinal rails was given three levels in subsequent simulations:

Level 1: Annotation 100% Original material
Level 2: Annotation 150% Strength of material increased by 50%
Level 3: Annotation 50% Strength of material reduced by 50%

To model the material in the longitudinal rails a piecewise linear plasticity material was used. This material model requires a modulus of elasticity, a yield stress and a load curve describing the relationship between effective stress and effective plastic strain to simulate post-yielding behavior. To reduce or increase the strength, the affected materials was given a scale factor for the yield stress and the same scale factor for the load curves (see Figure 4; the longitudinal rails consisted of several parts with different material parameters, Figure 4 shows an example). The modulus of elasticity was not changed since this could affect the time step of the explicit time integration. Furthermore, the relationship between effective stress and effective plastic strain was assumed to be constant with loading velocity, i.e. no strain rate dependency was included in the model.

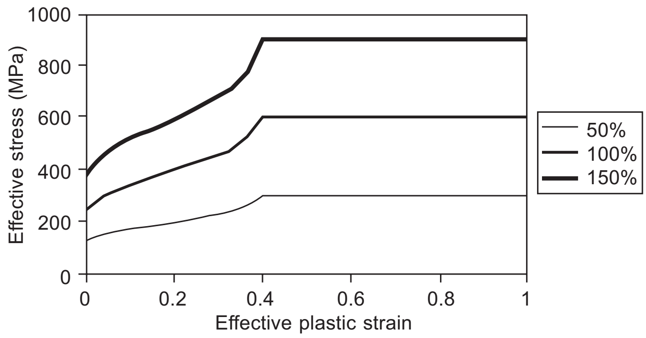

Figure 4 Example of plastic behavior of material in longitudinal rails.

To limit the number of simulations where the concept of structural adaptivity was to be applied, a range of crash velocities was chosen: 16, 32, 48 and 64 km/h. To compare the effect of a rigid compartment to the original deformable compartment, all parts aft of (and including) the firewall were defined as rigid in a second simulation series. Using these two compartment definitions with two barrier configurations and three rail material levels as described above, a total of 48 simulations were carried out (see Table 2).

Table 2 **Simulation plan: Numbers refer to strength-of-rail-material levels**

	Deformable compartment		Rigid compartment	
Impact velocity	Full-frontal crash	40% offset crash	Full-frontal crash	40% offset crash
	50%	50%	50%	50%
16 km/h	100%	100%	100%	100%
	150%	150%	150%	150%
	50%	50%	50%	50%
32 km/h	100%	100%	100%	100%
	150%	150%	150%	150%
	50%	50%	50%	50%
48 km/h	100%	100%	100%	100%
	150%	150%	150%	150%
	50%	50%	50%	50%
64 km/h	100%	100%	100%	100%
	150%	150%	150%	150%

 doi:10.1533/ijcr.2005.0353

RESULTS

The results section is divided in three parts focusing on different characteristics of the crash pulse: peak deceleration, mean deceleration and crash pulse shape. In deceleration comparisons, the rear left side accelerometer was used (see Figure 2). The reason for this is that the left side is considered to be the driver side of the vehicle and consequently the offset barrier is struck there.

Peak deceleration

Higher impact velocities resulted in an increased peak (maximum) value of the filtered acceleration history (see Figure 5). Furthermore, the peak deceleration was significantly larger in the case of a rigid compartment compared to a deformable compartment for a given crash scenario. Especially in the highest of the simulated impact velocities, this was observed to be caused by a very stiff engine-tofirewall contact (lower graphs in Figure 5: note scale difference compared to upper graphs).

A clear overall effect of the different yield stress levels could not be observed. In the 16 km/h crashes, the crash pulse displayed significant oscillations and the time of peak deceleration was difficult to predict; in the higher impact-velocity simulations, the peak deceleration was clearly associated with engine-to-firewall contact, i.e. the bottoming-out of the front structure between the firewall and the barrier (see example in Figure 6). This effect was more pronounced in the case of the rigid compartment, where very high decelerations could be observed. It is also interesting to note that the mid-level yield stress (denoted 100%) resulted in the lowest peak deceleration in some cases, implying that the global response is not governed solely by the properties of the most energy-absorbing structural members, but rather by the global response of the vehicle front structure.

Mean deceleration

As in the case of peak deceleration, an overall increase of mean deceleration with impact velocity was found (see Figure 7). Again, a clear relation between the material properties of the rails and the vehicle response could not be observed. In spite of this, it is interesting to note how the material levels affected the mean deceleration in the case of full-frontal crash simulation with the rigid compartment (lower left in Figure 7). As can be seen in Figure 7, a lower yield stress of the rail material (50% of the reference strength) implied lower mean deceleration at 16 km/h and 32 km/h but on the contrary resulted in a higher mean deceleration in the 48 km/h and 64 km/h simulations. Similar to the peak-acceleration results, it is interesting to note that the mid-level yield stress (denoted 100%) resulted in the lowest mean deceleration in some cases.

Crash pulse shape

In order to study the effect of rail material on the shape of the crash pulse, the first 100 ms of the crash pulse were divided into three equal parts similarly to Kullgren et al.

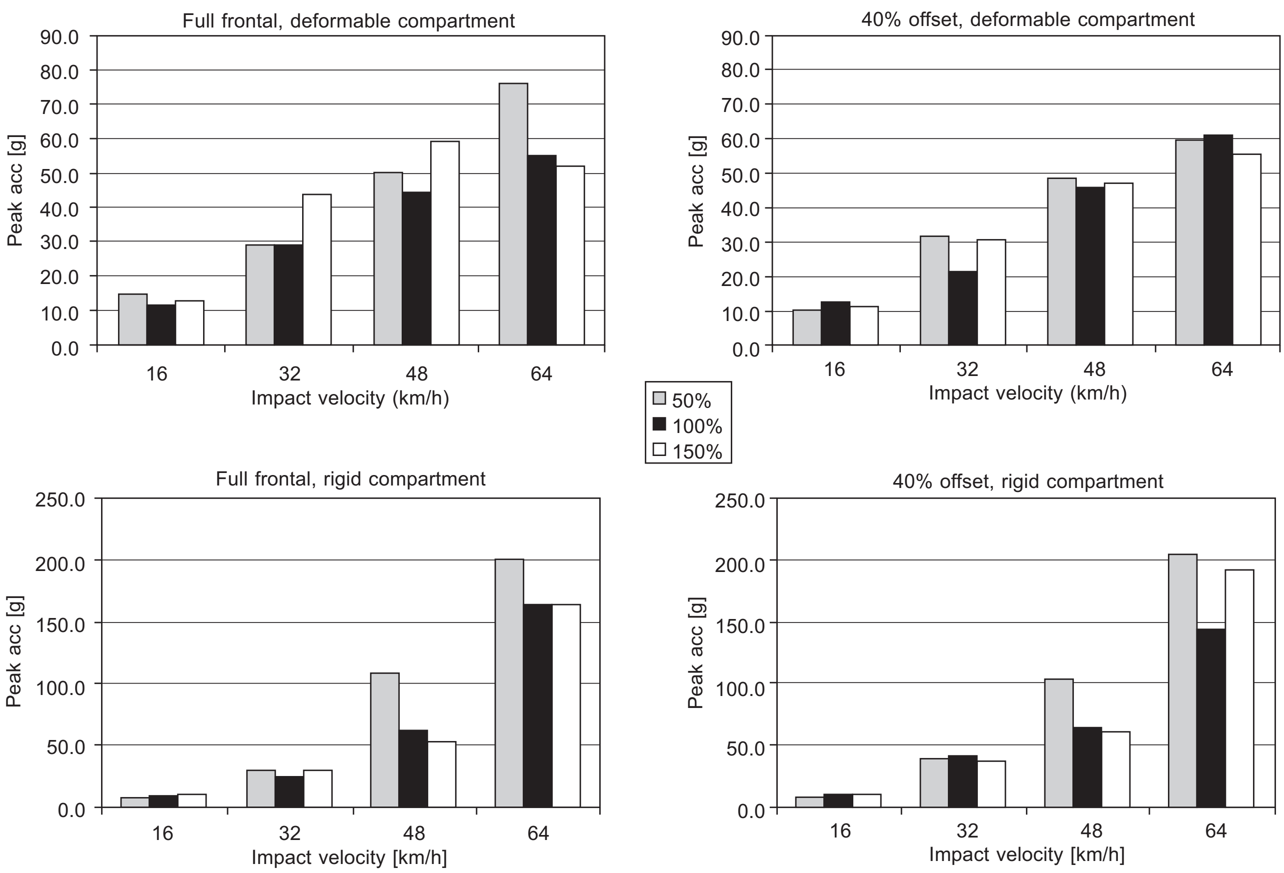

Figure 5 Peak decelerations of rear left side accelerometer.

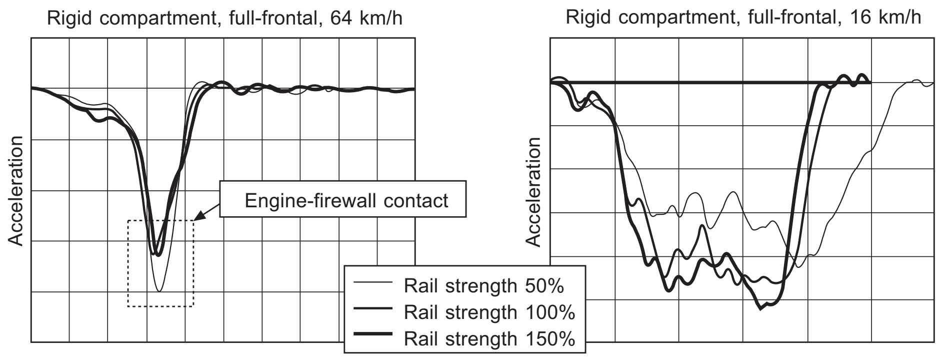

Figure 6 Example of acceleration time histories and the effect of engine-firewall contact.

[5]. For each part of the pulse, mean decelerations were calculated and their relative magnitudes are presented in Figure 8. Note that in some simulations, the total does not add up to 100%. This situation is caused by a mean acceleration (i.e. velocity increase) in the last third of the crash pulse. Furthermore, for a crash pulse longer than 100 ms, the acceleration after 100 ms was not included in the comparison.

From these graphs, it can be seen that the rigid compartment significantly increases the proportion of mean deceleration of the first third, i.e. shortening the duration of the pulse and thus increasing the mean deceleration for a given crash scenario. A lower material yield stress decreases the mean deceleration portion of the first third of the crash pulse at impact speeds up to 48 km/h.

The results can be analyzed by connecting to the findings of Kullgren et al. [5] in which a drop of mean deceleration between the 2nd and 3rd portion of the pulse was found to increase injury risk. It can be seen in Figure 8 that higher strength materials will generally increase this drop: A stronger material reduces the mean acceleration of the 3rd portion of the crash pulse compared to the mean acceleration of the 2nd portion.

DISCUSSION

To this date, studies on the real-world effect of crash-pulse characteristics on neck-injury risk are limited in number; however mean acceleration is suggested as a probable impact-severity measure for neck injury risk [5, 6]. The primary observation from the simulations in this study was the importance of avoiding stiff engine-to-firewall contact in order to prevent high deceleration peaks and short pulses that lead to high mean decelerations. In general, less stiff frontal members would imply lower deceleration if sufficient deformation space were available. However,

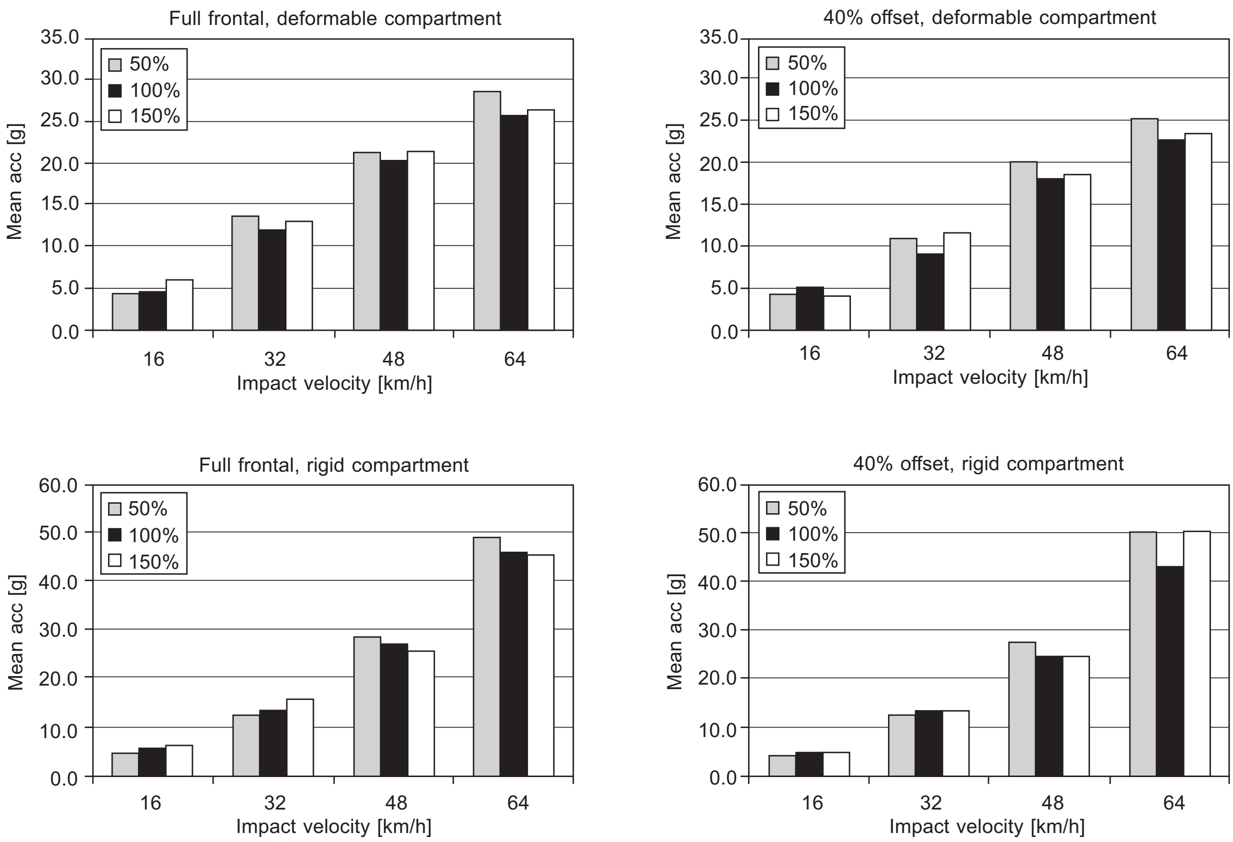

Figure 7 Mean decelerations of rear left side accelerometer.

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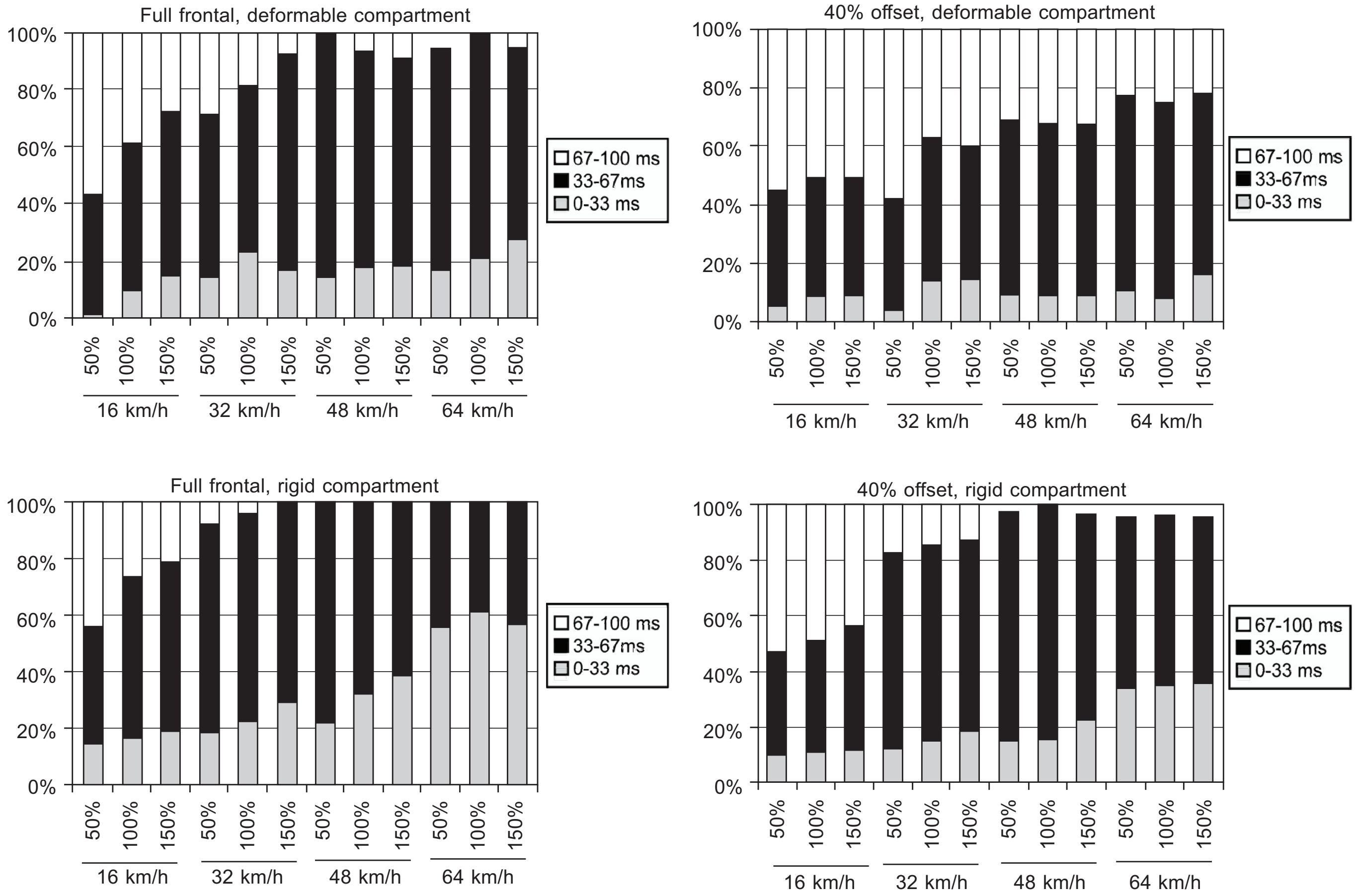

Figure 8 Mean deceleration distribution, rear left side accelerometer.

due to the geometry of the engine compartment in modern cars, only a short deformation length is possible if an intact compartment is to be maintained: For a given firewall-to-bumper-leading-edge distance where the engine requires a certain space, only a limited deformation distance is available before the engine is pressed up against the firewall. So, by trying to reduce the peak or mean deceleration in a frontal crash by reducing the strength of frontal members, the result could be the opposite: A low mean deceleration during deformation of the less stiff structural members until a stiff engine-to-firewall contact occurs will result in higher peak and mean deceleration compared to the reference scenario.

Results from the simulations with a rigid compartment appear to grossly overestimate the vehicle deceleration of a real-world vehicle. However, as modern vehicle crashworthiness design is based on a non-deformed passenger compartment that supply a sufficiently large survival space in most crash situations, a rigid firewall model might not be unrealistic. Particularly in the lower firewall area, where lower extremity contact is probable, there is a small distance between the occupant and the intruding structure where minimized deformation is desirable from an injury-reducing point of view. With this in mind, the effects on the crash pulse of a rigid firewall should not be neglected. There is another aspect to consider when comparing crash pulses of a vehicle with a rigid passenger compartment to a vehicle with a deformable passenger compartment; a frontal structure which is too weak might lead to significant intrusion of the firewall and dashboard into the passenger compartment in the case of a deformable passenger compartment. This might mean that the mean deceleration is lower for a lower material strength, but the high intrusions render low decelerations irrelevant for occupant injuries.

Initial simulations exhibited unacceptably high levels of hourglass energy, i.e. unphysical energy caused by the suppressing of zero-energy deformation modes in under-integrated shell elements. To decrease hourglass energy, a stiffness form of hourglass control option in LS-DYNA was used. However, since this made the acceleration response extremely stiff, the model had to be modified to include fully integrated shell elements that do not create hourglass energy and thus makes hourglass control redundant.

An important aspect to consider when using simulation results in crashworthiness strategies is that until a model has been validated by means of crash tests, it can only be used to make qualified predictions. Since the model in this study was not validated against crash tests in all the crash configurations considered, the findings are only valid as input to further tests and simulations. Another issue to recognize is that a rigid barrier crash is not equivalent to a car-to-car crash. In a real-world collision between two vehicles, the geometry of the impact plays an important role. For instance, when a certain amount of impact energy is designed to be absorbed by the longitudinal members, this assumes that the member is properly triggered, i.e. that an intended folding mechanism is initiated and that the energy absorption is not reduced by e.g. bending. If in a car-to-car collision a stiff part of a vehicle, such as a longitudinal member, hits a much softer structure in the other car, the deformation and the crash pulse could have very different appearances compared to a rigid-barrier simulation.

To design adaptive frontal structures that have a significant effect on the crash pulse, strategies for affecting the global load paths must be investigated. This study suggests that changing only the strength of some significant members could affect the crash pulse only into a limited extent. In this context, it should be noted that the levels of yield stress used in the simulations were chosen arbitrarily as attainable levels; there could be ways of changing the yield stress (or in effect, the transmitted forces) to more extreme levels.

An efficient way to increase the crash duration, and thus decrease the mean deceleration, would be to use inherent properties of physics: Compared to a constant deceleration, a higher deceleration in the initial stage of the crash pulse would decrease the velocity faster and thus enable a lower deceleration over the remaining deformation length.

CONCLUSIONS

From the analysis of kinetic energy absorbed by structural deformation in simulations of vehicle-to-barrier collisions in this study, the following conclusion was drawn:

– The front longitudinal rails absorbed most of the impact energy in both offset and full-frontal collisions with a rigid barrier at both 32 km/h and 56 km/h.

From the simulations where the yield stress of the front longitudinal rails was altered at different impact velocities and barrier configurations, the following conclusions were drawn:

– A rigid compartment is likely to reduce the duration of the crash pulse and thus increase the mean deceleration.
– The yield stress of the material in the front longitudinal members has an effect on the crash pulse in both full-frontal and offset collisions. The nature of this effect depends largely on the geometric constraints in the engine compartment and the kinetic energy of the crash, i.e. the impact velocity of a given vehicle.
– Reducing the yield stress of longitudinal members can only reduce the peak deceleration if this peak is associated with structural deformation and not engine contact.

To reduce mean deceleration and possibly the risk of long-term neck injuries it is suggested that, compared to the rest of the pulse, the structural force should be higher in the initial stages of a crash which requires sophisticated systems for crash sensing and structural control.

RECOMMENDATIONS FOR FUTURE RESEARCH

In order to attain optimized crash pulses for all crash velocities using adaptive vehicle frontal stiffness, the structural deformation of future cars must be globally controllable. This study suggests that changing the strength of isolated components such as the front rails is not sufficient to drastically affect the crash pulse shape: Therefore, strategies for global force adaptivity should be investigated.

Future research within this project will be focused on neck injuries; however attention must be directed to all types of injuries in order to avoid sub-optimization of occupant protection systems.

ACKNOWLEDGEMENTS

This study was supported by Autoliv Research.

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Paper III

"A methodology for improving structural robustness in frontal car-to-car crash scenarios"

Wågström, L., Kling, A., Norin, H., Fagerlind, H. (2013)

International Journal of Crashworthiness, 18 (4), 385-396.

International Journal of Crashworthiness, 2013
Vol. 18, No. 4, 385–396, http://dx.doi.org/10.1080/13588265.2013.801292

A methodology for improving structural robustness in frontal car-to-car crash scenarios

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(*Received 4 December 2012; final version received 29 April 2013*)

There has been significant development in passenger car crashworthiness over the last few decades. However, real-world crashes often occur in scenarios dissimilar to laboratory barrier crash set-ups. Further knowledge is required on how different impact scenarios affect vehicle structural response and occupant injury risk in real-world scenarios. This study introduces a methodology for assessing crash configuration parameters that influence the structural response in car-to-car frontal collisions by using finite element models of two identical vehicles. The crash configuration parameters included in this study were initial velocities, oblique angle and lateral offset distance. An evaluation was made in terms of passenger compartment intrusion and crash pulse severity. Special focus was directed towards investigating whether these input parameters can be used to define incompatible scenarios, i.e. where the structural response in one vehicle is significantly different compared to the other vehicle. Results indicate that collision scenarios with large overlap as extreme in terms of crash pulse severity, and incompatible car-to-car crash scenarios were found at small overlap and an oblique angle of 15°. An outlook for future model and method validation work is described.

Keywords: vehicle design; oblique frontal collisions; structural robustness; compatibility; small overlap

1. Introduction

From the data collected in real-world frontal crashes involving new vehicles, it has been shown that the vehicles receiving high scores in consumer rating tests possess the greater prospect of protecting their occupants than the vehicles receiving low scores [3,6]. The studies on both the US data [3] and the European data [6] suggest that the introduction of consumer rating crash tests using offset deformable barriers (ODBs) has encouraged improvements in passenger vehicle structural crashworthiness. However, there still seems to be room for further improvement. The Insurance Institute for Highway Safety (IIHS) conducted a study on frontal crashes resulting in severe injuries to belted occupants (AIS 3+) occurring in vehicles awarded with high occupant protection scores in the IIHS rating tests [1]. An interesting finding of that study was that a considerable portion of crashes occurred in the so-called small overlap conditions, i.e. where vehicles were involved in frontal crashes without engaging the main frontal crash absorbing structures. In other words, the general crashworthiness improvements driven by ODB-based consumer ratings may not have had the same effect in small overlap situations. The IIHS [1] therefore suggested that it may be necessary to introduce additional dimensioning load cases that cover oblique and/or offset crashes into consumer rating test setups to further improve crashworthiness.

The IIHS findings [1] correspond well with Swedish data. In a study on fatal frontal crashes in Sweden 2000–2001, Lindquist et al. found that small overlap scenarios accounted for almost half of the fatalities involving belted occupants [8]. Some of the vehicles included in that study were designed prior to the introduction of ODB tests and may not include the structural improvements encouraged by this test method. Still, the large proportion of small overlap impact configurations in fatal frontal impacts is remarkable.

Another important point mentioned in the Swedish study [8] was the lack of detailed data collection methods for the investigations on real-world crashes. It was shown that the available coding standards for crush damage in passenger vehicle crashes do not always capture differences in impact configuration [8]. Hence, while real-world data can be used for sequence and outcome analysis in crashes, it is not sufficiently extensive to be used for detailed parameter analyses of car-to-car collision configurations. In other words, when all parameters are considered, it is usually difficult to draw precise conclusions regarding a specific parameter since numerous confounding factors (e.g. differences in crash participating vehicles) are usually present [17,15]. Additional approaches for understanding how different crash scenarios are linked to vehicle structural performance and possibly occupant injury risk are therefore needed. One such approach is computer simulation, which was previously employed by Eichberger et al. [2] in small overlap situations. The Eichberger et al. study pointed out the 'rim-locking effect' as a critical cause of

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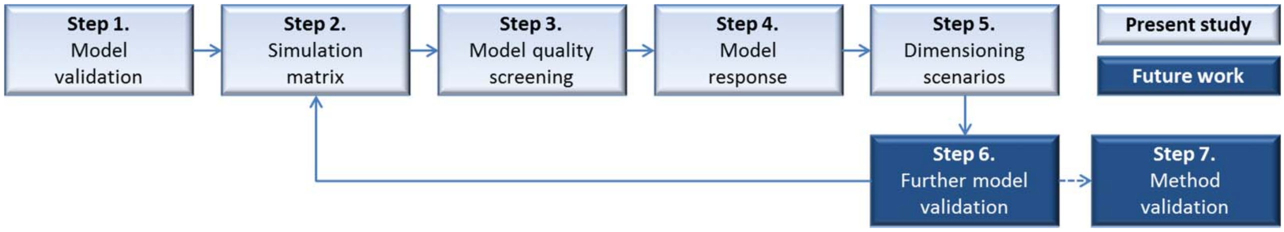

Figure 1. Overview of methodology and future work needed for validation.

passenger compartment intrusions. This intrusion mechanism was proposed to occur when impacted wheels are locked up with each other during the crash and pushed into the passenger compartment in small overlap car-to-car collisions. However, neither details on the employed computer models nor any attempts to use finite element (FE) models for parametric studies on angled car-to-car-collisions were presented.

Besides the situations where occupant injuries can be directly linked to vehicle intrusion, studies from both the European FP7 research project FIMCAR [4] and the IIHS [1] have identified deceleration without significant intrusion as a considerable cause of injuries to occupants of modern passenger cars. These situations were found when the available restraint system was not able to prevent occupant motion leading to contact with interior structures or when occupants sustained injuries from the restraint system itself. Vehicle design for future crashworthiness must therefore address structural integrity combined with appropriate vehicle deceleration levels to support the functionality of restraint systems in frontal collisions.

In a previous study by Thomson et al. [17], large overlap, small overlap as well as oblique car-to-car crash scenarios were approached by crash simulation. In that study, the effect of structural interaction on vehicle crash response on a modified public domain FE model was studied. A simplification of the FE model was implemented by introducing a beam to restrict the upward rotation of the longitudinal-side members. This major modification to the simulation model was carried out in order to make the crash response more representative of the European vehicles. Results underlined the large importance of structural alignment and deformation sequence for the intrusion and deceleration response in car-to-car collisions.

Based on these findings from previous studies and real-world crash data, it is suggested that improved crash simulation techniques should be developed in order to better understand the structural mechanisms that lead to large passenger compartment intrusion or deceleration in frontal car-to-car crash scenarios. Therefore, the aim of this study is to develop a methodology for identifying dimensioning frontal car-to-car crash scenarios by assessing crash configuration parameters that influence structural response.

2. Method

The methodology is divided into several steps, where each step builds on a previous step as indicated in Figure 1. Step 1 includes description and validation of the single-vehicle FE model used in the methodology. The FE model was validated in terms of intrusion and deceleration response in frontal impact before being used in the proposed methodology. This FE model is used to establish a car-to-car crash simulation model with two identical passenger cars. In a general application, the two vehicles do not need to be identical, although the methodology has been developed for this specific scenario. Following Step 1, the car-to-car FE model is employed for a parameter study described in Step 2 of the methodology, the description of how a simulation matrix is established. To ensure the quality of the simulations in Step 2, a model quality screening is performed in Step 3 where the numerical quality is assessed. To measure the model response in all the scenarios defined by the simulation matrix, Step 4 describes structural output generated from the FE model. Based on this output, a set of scenarios with outstanding properties in terms of structural response are defined in Step 5. These are considered candidates for crash scenarios that represent extreme loading situations and may therefore constitute load cases that should be used for dimensioning of car structures. Since the FE model used for the parameter study cannot be validated by physical testing in all of the scenarios defined by the simulation matrix, a few of the most noteworthy collision set-ups are selected for further validation work in Step 6, which is outside the scope of the present study. After the completion of Step 6, further FE model validation, the FE model should be updated with this new information and a refined simulation matrix can be defined and simulations performed (Step 6 back to Step 2). After model quality screening in Step 3, similar or updated output based on the knowledge gained from model validation is generated in Step 4. In this way, a re-examination of the dimensioning crash scenarios can be made in Step 5, and the loop can continue until the results of the methodology can be shown to represent the physical behaviour of the studied vehicles (Step 7). By improving numerical robustness and validity of FE models, the methodology can thereby become a comprehensive tool for ensuring robust response of vehicle structures.

Table 1. Properties of LS-DYNA single-vehicle base model.

Property (unit)	Setting
LS-DYNA solver	mpp971s R5.1.1
Precision	Single
Time step (ms)	6.3E-04
Typical shell element sizes (mm)	
Front structure / A-pillars (Type 16)	5.5
Rest of model (Type 2)	10.0
Number of elements	2,174,341
Number of nodes	2,018,366
Total model mass (kg)	1900
Crash test dummies, front seats	Two simplified Hybrid III dummies [10]
Gravity loading	Yes

2.1. *Step 1: model validation*

To develop the methodology in this study, a full vehicle model was selected. The model has previously been used for in-house simulation in product development of a mid-size passenger vehicle currently in production (2012) and was developed in Livermore Software-describes dynamic (LS-DYNA) [9]. A brief description of the most important properties of the FE model is given in Table 1.

Mass scaling was used in LS-DYNA in order to maintain the minimum time step in Table 1. Simplified crash test dummies [10] were included in the model for an accurate representation of the mass distribution during crash.

The model has been validated during previous development work in terms of passenger compartment intrusion and deceleration in car-to-barrier load cases. As an example of this, from full-width rigid barrier crash at 56 km/h, the deceleration vs. displacement data from accelerometers located in the left-hand side (LHS) and right-hand side (RHS) sill at the B-pillar base are given in Figure 2. As illustrated in this comparison, the model qualitatively represents the physical response, both in terms of stopping distance (<2% error) and the shape of the deceleration vs. displacement graphs.

The model was also validated in terms of intrusion by comparing simulation results to the scanned post-crash data from a physical test with the same car model against a 40% overlap deformable barrier at 64 km/h. The maximum dynamic intrusion relative to the undeformed B-pillar plane in the model was compared to the surface obtained by three-dimensional scanning of the tested car after the crash test was performed, see Figure 3.

For the offset deformable barrier crash at 64 km/h, the deceleration vs. displacement data from accelerometers located in the LHS sill (struck side) at the B-pillar base in test and simulation were compared, see Figure 4. As illustrated in this comparison, the model qualitatively represents the physical response, both in terms of stopping distance and the shape of the deceleration vs. displacement graphs.

2.2. *Step 2: simulation matrix*

A two-vehicle model was set up from the base car FE model described in Step 1 above. For the base car model (Car 1) as shown in Figure 5(a), a rotation circle was defined by inscribing the front end within a circle in such a way that no part of the car crosses the rotation circle as seen from a horizontal projection, while minimising the rotation circle radius. The next step was to copy and rotate the base car 180° around a vertical axis directly in front of Car 1 as described in Figure 5(b), where Car 1 is defined as the car on the right, and Car 2 on the left.

The centre point of the rotation circle on Car 1 was subsequently used as the rotation centre for oblique angle between the vehicles. The oblique angle was applied by keeping the position of Car 1 fixed and rotating Car 2 around a vertical axis passing through the rotation centre on Car 1, as shown in Figure 5(c). Car 2 was thereafter shifted laterally by an offset distance as illustrated in Figure 5(d). The gap between the vehicles was reduced to minimise simulation time before the first point of contact, as shown in Figure 5(e). For each vehicle, initial velocity was applied

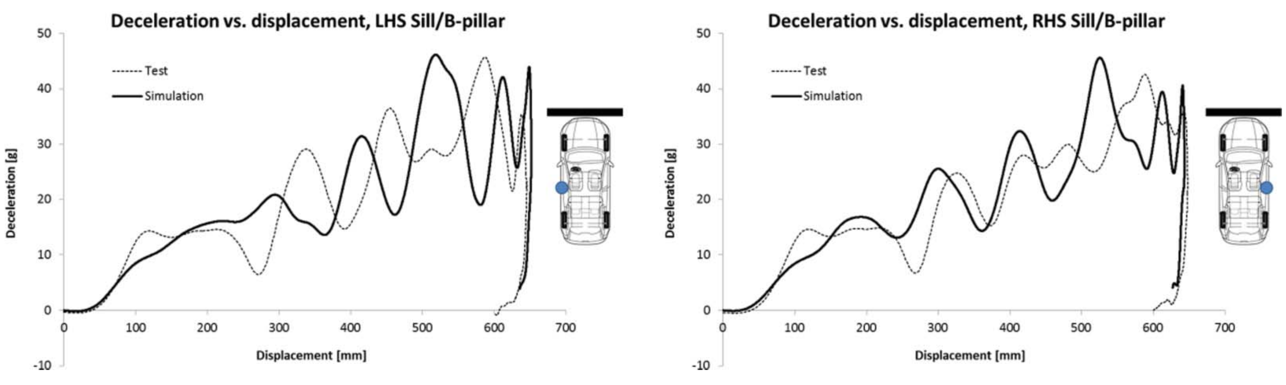

Figure 2. Validation of deceleration response of single-vehicle base model, rigid barrier. Deceleration plotted vs. displacement at left and right sills in vehicle model. Data from full-width rigid barrier test and corresponding simulation at 56 km/h.

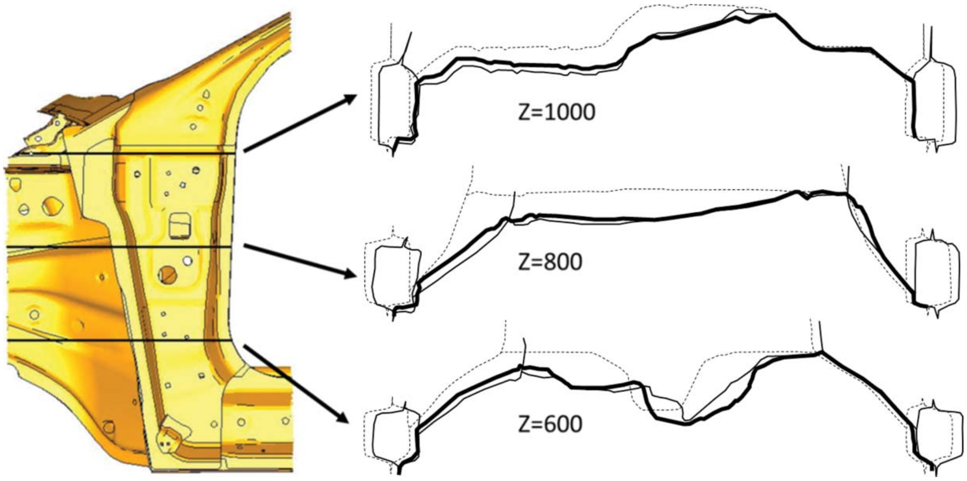

Figure 3. Validation of intrusion response of single-vehicle base model, offset deformable barrier. Passenger compartment intrusion from simulation (thin solid lines) and test (thick solid lines) vs. undeformed geometry (dashed lines). Cross-section through three levels of the A-pillars and dashboard panel. Data from 40% overlap deformable barrier test and corresponding simulation at 64 km/h.

in the longitudinal direction only. Furthermore, no friction with ground was assumed while the car-to-car friction coefficient was set to 0.2.

A range of oblique angles, lateral offset distances and combinations of initial velocities were selected in order to study the structural response of the car-to-car crash simulation model. Initial velocities of 30, 50 or 70 km/h were selected for both vehicles. Three levels of initial velocity per car yields nine different initial velocity combinations for each collision set-up, see Appendix section A. Oblique angles were selected at six levels from 0° to 45°, with smaller increments applied at low angles, to approximately represent the oblique angle frequency distribution suggested by Eichberger et al. [2]. Simulations were made at 0°, 5°, 10°, 15°, 30° and 45°, see Figure 6 for a graphical representation. Offset distances were selected at seven levels from

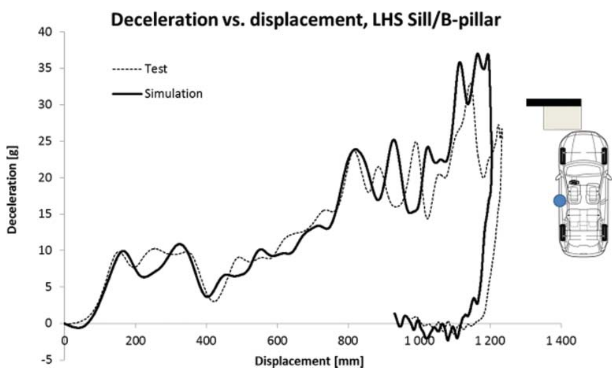

Figure 4. Validation of deceleration response of single-vehicle base model, offset deformable barrier. Deceleration plotted versus displacement at left sill (struck side) in vehicle model. Data from 40% overlap deformable barrier test and corresponding simulation at 64 km/h.

0 to 1800 mm by an increment of 300 mm, which means going from full overlap to no overlap in the non-angled scenario as illustrated in Figure 6. In total, this resulted in 378 simulations, all with a termination time of 150 ms.

2.3. *Step 3: model quality screening*

All 378 simulations were assessed for numerical stability in order to minimise the occurrence of non-physical phenomena. Total energy over time was not allowed to change by more than 10% compared to its initial value, since this indicates creation or destruction of energy during simulation. Since all simulations were performed using an explicit numerical integration method with a constant time step, the model was subjected to an added mass in order to maintain this time step [9]. All simulations were excluded where more than 1% of the initial model mass was added during simulation. Furthermore, screening was performed in order to exclude simulations where the total absorbed energy did not reach a steady state at the end of the simulation. The criterion for steady state was based on the difference between the minimum and maximum values for total absorbed energy during the last 10 ms of simulation. A steady state was defined when this difference in total absorbed energy did not exceed 3% of the initial total kinetic energy.

A criterion for non-collisions was established. The exclusion of non-collisions was made when the maximum longitudinal velocity change of the non-struck side of Car 1 was below 5% of the initial velocity for Car 1. To focus on frontal impacts, all simulations were disregarded when the maximum lateral velocity was larger than the maximum longitudinal velocity change of the non-struck side of Car 1.

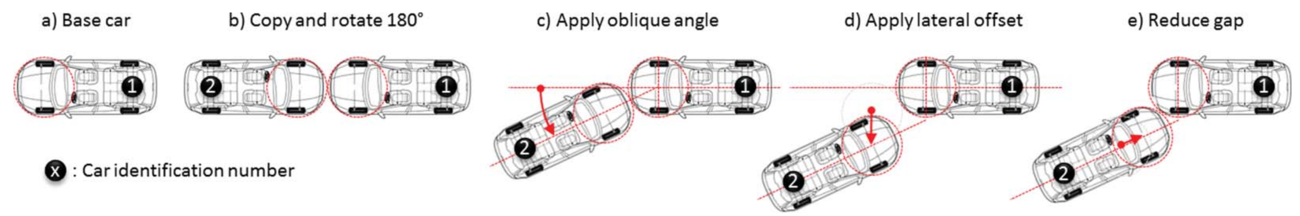

Figure 5. Car-to-car positioning setup for crash simulations.

2.4. Step 4: model response

The structural response of the vehicles was given as output. Passenger compartment intrusion was measured in seven areas according to Figure 7. Intrusion for each area was measured as the maximum total dynamic intrusion, i.e. the combined intrusion in all directions as measured relative to an undeformed coordinate system rear of the B-pillar plane. This was performed in order to ensure that an undeformed reference coordinate system was used even when there was deformation to the B-pillar.

To minimise the number of intrusion areas used for comparison between car-to-car simulations, a correlation study between different intrusion areas was performed by comparing the Pearson product-moment correlation coefficient [13]. In this approach, a perfect positive correlation corresponds to a correlation coefficient of +1, i.e. all available data show that the first variable will increase linearly with the second variable. If the correlation coefficient equals −1, there is a perfect negative correlation and a value of zero corresponds to no correlation. This means looking at the intrusions in one area of the vehicle and comparing to other intrusion areas, and assessing the degree of correlation. If there are areas with high levels of correlation with other intrusion areas, these key intrusion areas are indicative of passenger compartment intrusion in general and may be used for comparison of different crash scenarios.

Since simplified dummy models were used, no detailed information on dummy response was extracted for this study. To evaluate the vehicle longitudinal deceleration for the two cars, accelerometers located where the B-pillar meets the sill (see Figure 7) on both sides of the vehicles were used. For each of these signals, a crash pulse severity metric called volvo pulse index (VPI) is calculated [5]. The VPI is calculated by a simplified model of the occupant chest deceleration while interacting with a restraint system connected to the vehicle. The model includes an initial slack of 30 mm relative displacement where no deceleration takes place, followed by a linearly increasing deceleration of 0.25 g/mm. The VPI is defined as the maximum chest deceleration of this simplified occupant model.

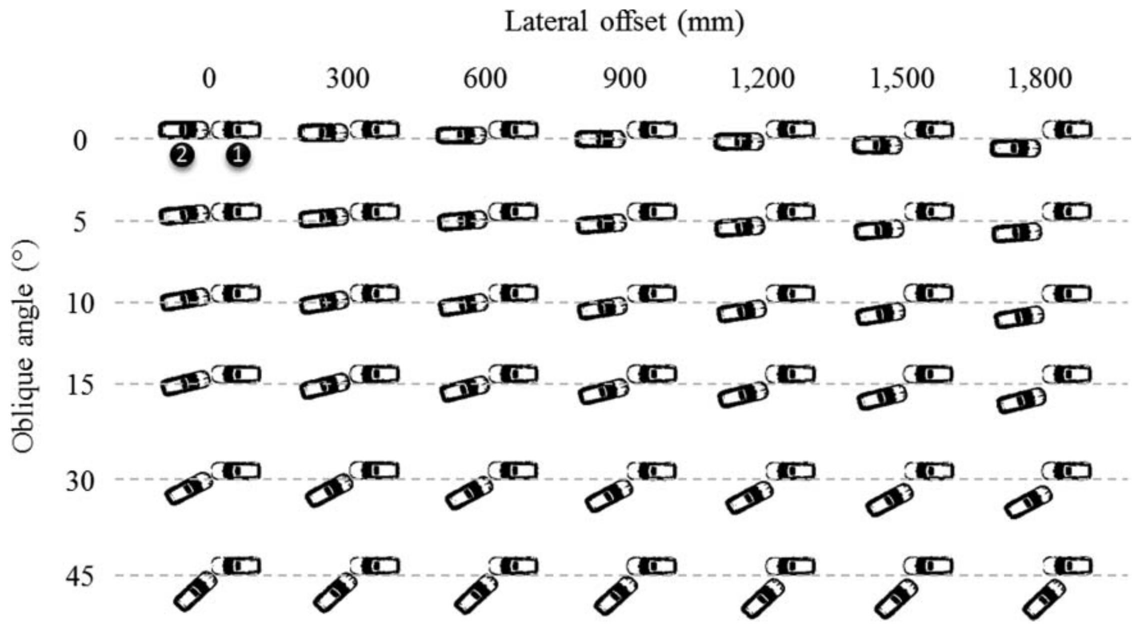

Figure 6. Overview of crash configuration domain showing simulated car-to-car crash scenarios. Each collision setup contains nine different initial velocity combinations, $N = 378$.

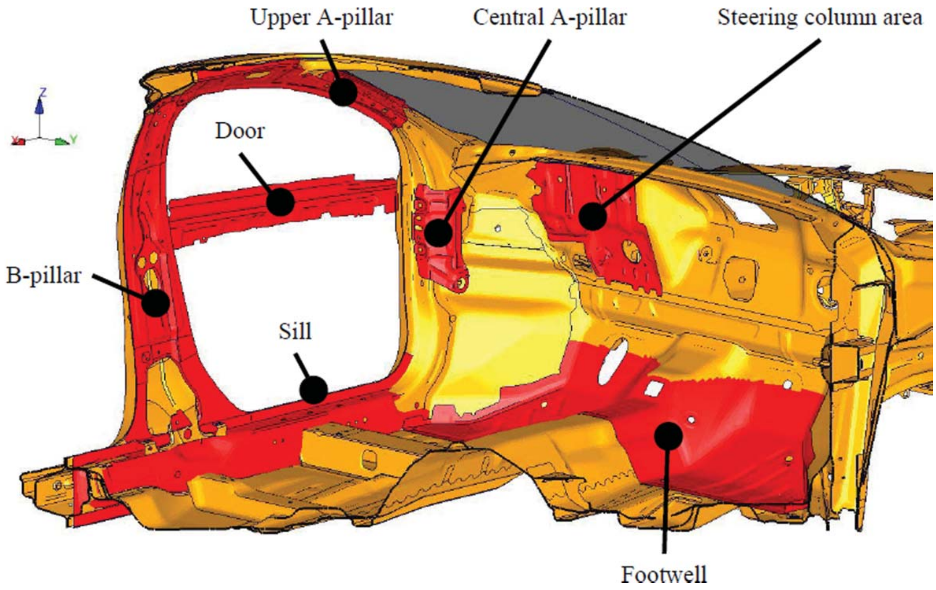

Figure 7. Intrusion areas used for intrusion output from simulations. For each area, the maximum total dynamic intrusion was recorded, i.e. the combination of directions *x* (longitudinal), *y* (lateral) and *z* (vertical).

2.5. *Step 5: dimensioning scenarios*

To compare the response between the two vehicles, a compatibility domain was established, see Figure 8. By plotting the same output variable for both vehicles in the same graph with equally scaled linear axes, an overview can be given on which car-to-car crash scenarios that represent high or low response are compared to each other. In addition, the relative response between the two colliding vehicles can be directly assessed. This is visualised by plotting one data point for each simulation in the compatibility domain. Scenario data points along the equal response line represent collisions where the responses in the two cars are identical. Scenario data points that are not on the equal response line are consequently unbalanced in terms of the structural response property currently studied. The inclination of a line passing through the origin and a specific data point gives information on the degree of unbalance that the scenario represents. This means that scenarios in the bottom-right half of the compatibility domain represent greater response in Car 1, whereas scenarios in the upper-left half represent greater response in Car 2. When scenarios with outstanding structural response were identified in the compatibility domain illustrated in Figure 8, the crash configuration domain illustrated in Figure 6 was used to visualise the corresponding collision set-ups.

3. Results

All 378 simulations were assessed for numerical stability. After the completion of this screening, 248 simulations were remained as shown in Appendix section B. The exclusion of non-collisions left 224 simulations, see Appendix section C. Visual inspection of animations from these scenarios confirmed that none or very little contact was made between the cars. After exclusion of side impacts, 204 simulations were remained for further analysis, see Appendix section D.

The intrusion data from all the 204 screened simulations presented in Tables 2 and 3 generally show a high level of correlation between intrusion areas in each car.

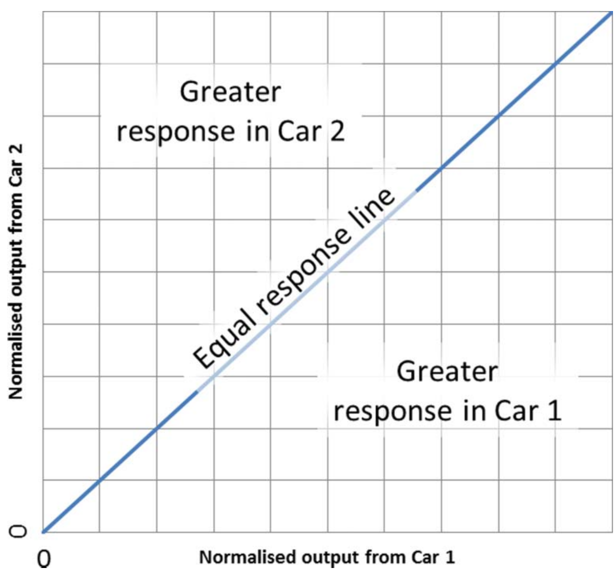

Figure 8. Definition of compatibility domain, where structural response in Car 2 is plotted vs. the same response in Car 1 using linear, equally scaled axes.

Table 2. Correlation coefficient for intrusion areas in Car 1, $N = 204$.

Intrusion areas in Car 1	Upper A-pillar	B-pillar	Central A-pillar	Door	Footwell	Steering column area	Sill
Upper A-pillar	1.000						
B-pillar	0.979	1.000					
Central A-pillar	0.959	0.944	1.000				
Door	0.968	0.959	0.978	1.000			
Footwell	0.910	0.905	0.976	0.938	1.000		
Steering column area	0.961	0.941	0.919	0.938	0.851	1.000	
Sill	0.898	0.898	0.958	0.918	0.976	0.799	1.000

The intrusion area that displays the best average correlation coefficient is the central A-pillar intrusion (highlighted in Tables 2 and 3). The central A-pillar intrusion is therefore suggested to be monitored in order to show one value per car and simulation to represent the overall intrusion levels.

To compare passenger compartment intrusion, the central A-pillar intrusion for Car 1 and Car 2 was plotted in Figure 9. As expected, in non-angled scenarios, both vehicles sustain approximately the same intrusions along the equal response line in Figure 9. There is a clear tendency towards increased intrusion in small overlap scenarios compared to large overlap scenarios. For example, at 70 vs. 70 km/h, the 1200 mm offset scenarios at 0° or 5° (diamond) were compared with the 300 mm offset scenario at 0° (square) as highlighted in Figure 9. Here it is clear that the lack of structural interaction in small overlap scenarios may result in significantly higher intrusions in both vehicles compared to the large overlap scenarios.

Analysing the central A-pillar intrusion in Figure 9, a few scenarios with large intrusions stand out as particularly asymmetric. These are the 15° and 1200 mm offset scenarios (pyramid). In these scenarios, intrusions are more than three times greater in Car 1 than in Car 2, implying a considerable difference in response in the two identical vehicles. For Car 1, the central A-pillar intrusion in this crash configuration is larger than at 0° (large filled circle) as illustrated in Figure 9.

In Figure 10, the crash pulse severity as represented by the VPI is visualised in the compatibility domain. To isolate the effect of impact scenario on impact severity, all simulations where both cars had an initial velocity of 50 km/h were studied in terms of VPI. For reference, non-oblique impacts at 100% overlap (offset = 0 mm, square in Figure 10) and at 50% overlap (offset = 900 mm, diamond in Figure 6) were compared with all the simulations in this dataset ($N = 30$). A full overlap car-to-car collision resembles a full-width rigid barrier crash since the complete front structure of the vehicle is engaged, and all deforming parts meet matching counterparts if there is no significant structural asymmetry in the vehicle or powertrain design. A 50% overlap car-to-car scenario resembles approximately an offset deformable barrier set-up since only a portion of the vehicle front structure is involved. As illustrated in Figure 10, there are only a few impact scenarios resulting in higher crash pulse severity than zero offset in combination with zero oblique angle. These scenarios are concentrated at 300 mm offset and 0° or 5° oblique angle (scenarios marked with a pyramid in Figure 10) and show approximately 10% higher VPI than full overlap.

4. Discussion

The purpose of the methodology presented in this study was to establish a tool for structural robustness in the development process of passenger vehicles. By finding factors that influence the vehicle response in car-to-car collisions, robust structural integrity and energy absorption can be obtained through vehicle design before physical prototypes

Table 3. Correlation coefficient for intrusion areas in Car 2, $N = 204$.

Intrusion areas in Car 2	Upper A-pillar	B-pillar	Central A-pillar	Door	Footwell	Steering column area	Sill
Upper A-pillar	1.000						
B-pillar	0.983	1.000					
Central A-pillar	0.980	0.967	1.000				
Door	0.986	0.968	0.983	1.000			
Footwell	0.908	0.913	0.955	0.914	1.000		
Steering column area	0.931	0.928	0.905	0.929	0.843	1.000	
Sill	0.910	0.918	0.948	0.903	0.957	0.770	1.000

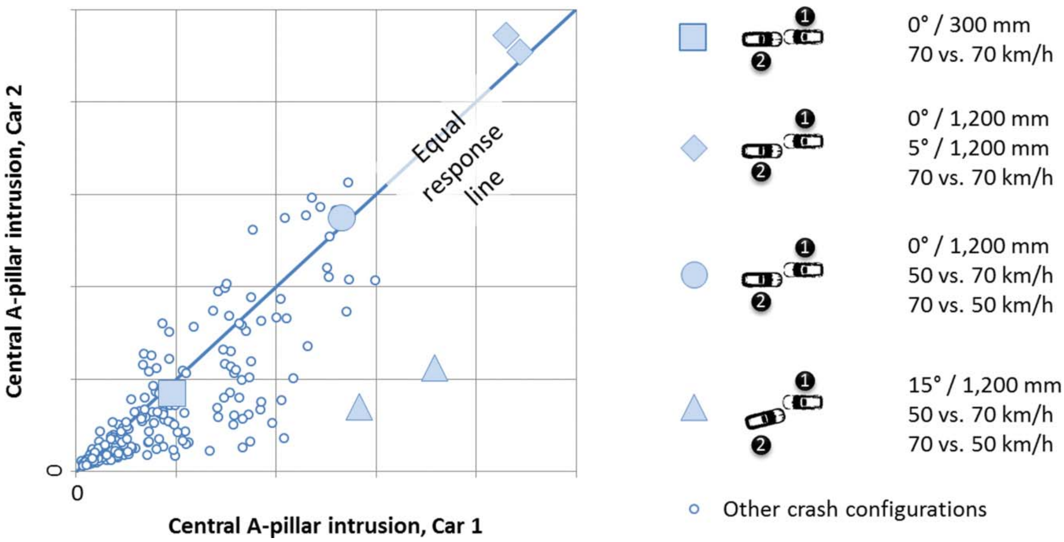

Figure 9. Compatibility domain showing central A-pillar intrusion in Cars 1 and 2 with four different crash scenarios highlighted, $N = 204$. Linear, equally scaled axes without numerical values for relative comparison.

are built. The preliminary results from this study indicate that the most critical crash scenarios in terms of intrusion are angled small overlap situations. Results also point out scenarios with large, but not full, overlap as extreme in terms of crash pulse severity, suggesting that full-frontal impact may not be the most severe crash pulse configuration for dimensioning restraint systems. By further addressing model validity, it is predicted that the methodology can be improved and thereby support increased structural robustness combined with approaches described in literature [7,11] when future vehicle designs are to be developed.

A scenario where the response in one vehicle compared to the other leads to a significant difference in injury risk to occupants, is often called an incompatible crash scenario [16]. However, while crash incompatibility normally is defined in terms of unbalanced vehicle mass, geometry or stiffness, in this study any unbalance was explored based on oblique angle and lateral offset distance. Since the term incompatibility previously has not been used to characterise frontal crashes where both vehicles and occupants are identical, this represents a novel approach to the issue of car-to-car crash compatibility. With further studies based on the presented methodology, these incompatible frontal

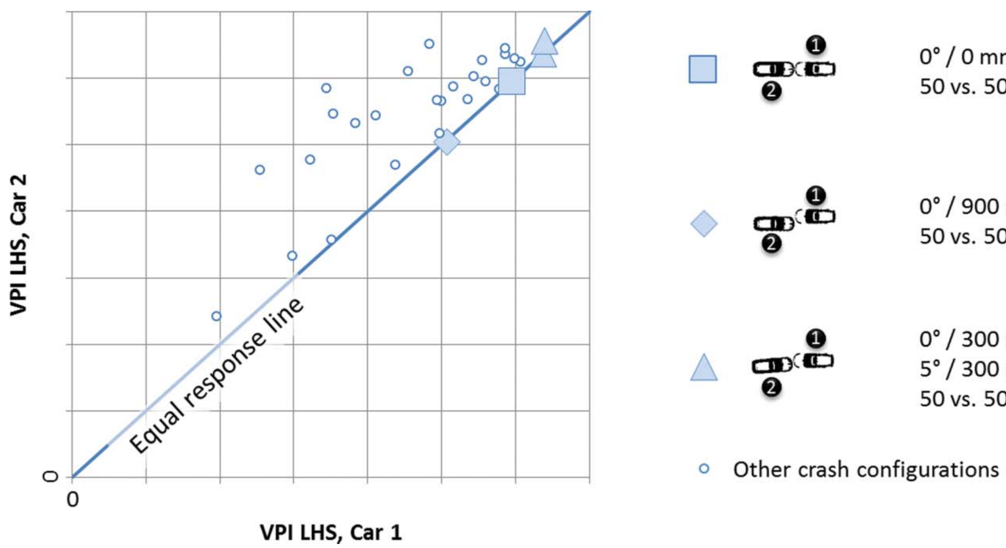

Figure 10. Compatibility domain showing VPI for left-hand side (LHS) in Cars 1 and 2 at 50 vs. 50 km/h with three different crash scenarios highlighted, $N = 30$. Linear, equally scaled axes without numerical values for relative comparison.

collisions can be verified and countermeasures can be made to avoid them.

The comparisons of the base FE model with test data indicate the capability of the base model to capture passenger compartment intrusions and decelerations in barrier load cases. However, there is always a limited set of validation data available and a completely validated FE model is difficult, if not impossible, to obtain. Moreover, once a physical test is available for model validation, there are always uncertainties regarding the representativeness of that specific physical test in the dispersion of test results. Due to the high costs related to crash testing, tests are rarely repeated in order to evaluate the statistical distribution of results. In the comparison of passenger compartment intrusion in Figure 3, the maximum dynamic intrusion from simulation is compared with the residual intrusion from the crashed car. In this way, the spring-back is not accounted for in the simulation model, which would bring simulation results closer to the scanned data in the struck side of the vehicle.

The choice of offset distances that was made in this study should be considered as a suggestion and a preliminary scan of the crash configuration domain as shown in Figure 6. A better way of covering the design domain may very well exist in terms of maximising the useful information generated from a limited number of simulations. A suitable approach for future studies could be to run fewer initial simulations in order to get a rough understanding of the system response, performing validation activities according to Figure 1, and returning to a more detailed investigation of the areas of interest.

Alternative ways of identifying non-collisions may be needed besides studying animations of the corresponding crash simulations or measuring contact force between vehicles. The approach of comparing change of velocity to initial velocity for a specific vehicle is obviously not appropriate, if one of the vehicles does not have an initial velocity. The currently used criterion for non-collision becomes infinite for all such simulations. Another approach could be to consider the change of velocity in relation to the initial relative velocity. This approach would be straightforward for non-angled scenarios, while the treatment of angled scenarios would be less obvious.

The intrusion areas in Figure 7 were chosen based on the structural characteristics of the specific car model used in this study. This choice should not be considered as a recommendation for other vehicles used with this methodology. It is however recommended that the intrusion areas should be structurally significant, i.e. they may introduce direct contact with occupants or influence the support for restraint systems. In the present study, the total intrusion was used in order to account for combinations of longitudinal, lateral and vertical intrusion. The rationale for this is that occupant injuries may be caused by intrusion in any of these directions and the methodology should not exclude any particular intrusion direction.

In this study, the longitudinal deceleration alone was used for crash pulse severity. As the oblique angle between the vehicles increases, the lateral deceleration becomes increasingly important. It is therefore suggested to include lateral deceleration in further analyses to assess the risk of lateral occupant motion in frontal collisions. This aspect may prove especially important in scenarios where passenger compartment intrusion directly reduces the amount of available interior distance for restraint systems to operate within. The purpose of calculating VPI is to provide a crash pulse severity indicator that takes crash pulse shape, and not only velocity change, into account. In this study, VPI was used, but other similar metrics such as the occupant load criterion (OLC) [14] exist. Both VPI and OLC have the disadvantage, however, of not capturing the combination of longitudinal and lateral deceleration. This could be addressed in future work by combining the structural methodology presented in this study with detailed occupant models for predictions of occupant kinematics and injury risk as measured by, e.g. the abbreviated injury scale (AIS).

The screening of simulations with respect to numerical stability, as shown in section B in Appendix, gives information on which scenarios that represent high or low numerical robustness. To enhance simulation output, it is recommended that improvements are made to the FE model in order to minimise the number of simulations that are taken out of the analysis dataset due to numerical instabilities. Ideally, none of the simulations should be discarded due to numerical stability. The limits for added mass and energy ratio used for disqualifying simulations due to poor numerical quality are to be considered as guidelines and not absolute requirements. The values used for the development of the methodology in this study are based on previous in-house simulation experience but may be subject to change as further loops are performed according to Figure 1. Moreover, the simulation termination time may need adjustment when the methodology is further employed. Preferably, simulations should be terminated when a post-crash steady state has been detected, however further development needs to be carried out in order to define such a function.

Comparison of crash configurations at 70 vs. 70 km/h could only be done to a limited extent since there were only six simulations at these initial velocity levels that passed the numerical stability assessment. This calls for further model development if FE models are to be used for parameter studies at these high initial velocities. The 50 vs. 70 km/h and 70 vs. 50 km/h scenarios, however, indicate large differences between the vehicles in terms of central A-pillar intrusion at 15° oblique angle and 1200 mm offset as illustrated in Figure 9. An obvious question is how this crash configuration compares to the non-angled scenario in terms of intrusions at the same initial velocity level and overlap. This analysis found that for Car 1, the maximum intrusion was greater at 15° compared to 0°. This finding suggests that for small overlap, the 15° scenario may represent larger intrusions than the non-angled scenario.

The methodology and results of the current study were compared to the few similar previous studies using crash simulation that were found in literature. Compared to the Thomson et al. study [17], a more detailed base FE model validated by crash test data was used. In general, the findings of the present study agree with the Thomson et al. findings that crash configuration clearly affects structural response. The main difference, besides model validity, between the present study and the Thomson et al. study is the systematic approach to analysis and visualisation of output data from a larger set of simulations that was proposed in the present study. It is however predicted that FE models need further development before detailed predictions can be performed in oblique small overlap scenarios. The rim-locking effect identified by Eichberger et al. [2] was observed in several simulations in the present study. The wheels and wheel suspension component models used in the present study were often observed to be highly stressed, leading to questions regarding the probability of the corresponding physical components to withstand these loads without material failure. Therefore, to be able to assess the influence of the rim-locking effect in a range of crash configurations with the presented methodology, it is proposed that the wheels and wheel suspension components should be modelled with further detail. This refined modelling technique should be based on model validation against physical crash tests of components or full vehicles before further parameter studies can be made according to Figure 1. After the completion of such model validation, the FE models can be used to analyse the rim-locking effect and reassess to which extent incompatible frontal collision scenarios can occur, see Wågström et. al [18].

When considering the structural challenges for passenger car designers identified in real-world data regarding small overlap and deceleration without major intrusion [1,4], the proposed methodology is predicted to supply a suitable tool to assess robustness of vehicle designs before physical prototypes are built. By expanding the crash opponent to vehicles of other size, weight and type, improvements can be realised for future crashworthiness. By studying identical vehicles, differences in crash configuration are isolated. In real-world situations, however, collisions between identical vehicles are rare. Therefore, the methodology presented here should be further employed in order to seek factors that influence structural response also in collisions between vehicles of dissimilar type or to study the effect of vertical offset. Adding vertical offset as a parameter is expected to cause further unbalances between identical vehicles as demonstrated by Mizuno et al. [12]. The compatibility domain lends itself to visualising structural differences in car-to-car collisions regardless of the vehicles involved. In an extended application, this methodology could be used to compare the relative importance of different aspects of incompatibility, e.g. studying when a structural advantage is cancelled by an unbeneficial car-to-car crash scenario.

5. Conclusions

A methodology intended to support development of robust vehicle structural response in car-to-car collisions was presented. Initial results were used to demonstrate the use of the methodology and indicate candidates for car-to-car scenarios that are extreme in terms of passenger compartment intrusion and crash pulse severity. These scenarios are proposed for further studies through model validation by correlation to crash tests.

The concept of incompatible frontal collisions was introduced. This is defined by frontal car-to-car crash scenarios where the structural response is remarkably unbalanced although the two vehicles are identical. Such scenarios were found at small overlap and an oblique angle of 15°.

Acknowledgements

This work has been carried out in association with SAFER – Vehicle and Traffic Safety Centre at Chalmers, Sweden. The authors would like to thank Dr. Karin Brolin for valuable advice regarding the written presentation.

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Appendix

A. Original simulation matrix

Number of simulations per crash scenario		Offset [mm] 0	300	600	900	1,200	1,500	1,800
Angle [°]	0°	9	9	9	9	9	9	9
	5°	9	9	9	9	9	9	9
	10°	9	9	9	9	9	9	9
	15°	9	9	9	9	9	9	9
	30°	9	9	9	9	9	9	9
	45°	9	9	9	9	9	9	9
							N =	378

Number of simulations per initial velocity combination		Car 1 30 km/h	50 km/h	70 km/h
Car 2	30 km/h	42	42	42
	50 km/h	42	42	42
	70 km/h	42	42	42
			N =	378

B. After numerical stability assessment

Number of simulations per crash scenario		Offset [mm] 0	300	600	900	1,200	1,500	1,800
Angle [°]	0°	8	9	6	6	9	3	7
	5°	7	4	5	7	6	5	6
	10°	6	4	7	7	6	5	5
	15°	7	7	6	8	5	4	3
	30°	8	8	7	8	5	2	4
	45°	8	7	7	7	4	3	2
							N =	248

Number of simulations per initial velocity combination		Car 1 30 km/h	50 km/h	70 km/h
Car 2	30 km/h	33	32	32
	50 km/h	34	37	28
	70 km/h	19	27	6
			N =	248

Remaining: 66%

Legend: More than 40% removed; 20-40% removed; Less than 20% removed

C. After removal of non-collisions

Number of simulations per crash scenario		Offset [mm] 0	300	600	900	1,200	1,500	1,800
Angle [°]	0°	8	9	6	6	9	3	0
	5°	7	4	5	7	6	5	0
	10°	6	4	7	7	6	5	0
	15°	7	7	6	8	5	4	0
	30°	8	8	7	8	5	2	1
	45°	8	7	7	7	4	3	2
							N =	224

Number of simulations per initial velocity combination		Car 1 30 km/h	50 km/h	70 km/h
Car 2	30 km/h	28	27	27
	50 km/h	31	34	27
	70 km/h	18	26	6
			N =	224

Remaining: 59%

Legend: All removed; Partially removed; None removed

D. After removal of side impacts

Number of simulations per crash scenario		Offset [mm] 0	300	600	900	1,200	1,500	1,800
Angle [°]	0°	8	9	6	6	9	3	0
	5°	7	4	5	7	6	5	0
	10°	6	4	7	7	6	5	0
	15°	7	7	6	8	5	3	0
	30°	8	8	7	8	1	0	0
	45°	8	7	7	4	0	0	0
							N =	204

Number of simulations per initial velocity combination		Car 1 30 km/h	50 km/h	70 km/h
Car 2	30 km/h	26	25	26
	50 km/h	28	30	24
	70 km/h	16	23	6
			N =	204

Remaining: 54%

Legend: All removed; Partially removed; None removed

Paper IV

“A Correlation Study for Oblique Frontal Impacts with Focus on Small Overlap Situations”

Wågström, L., Kling, A., Norin, H., Fagerlind, H. (2012)

Proceedings of the International Crashworthiness Conference (ICRASH), Milan, Italy.

A Correlation Study for Oblique Frontal Impacts with Focus on Small Overlap Situations

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Abstract – Road transportation account for tens of thousands of fatalities annually in the EU, a considerable amount of these are occupants of passenger cars involved in head-on collisions. It has been shown that both passenger compartment intrusion and deceleration increase occupant injury risk, emphasising the need for car structural integrity to reduce intrusion-related injuries and to support restraint system functionality. Small overlap situations is a collision type frequently linked to occupant injury.

The aim of this study is to describe small overlap car-to-car situations in terms of intrusion and deceleration metrics by means of a validated finite element model. A number of collision categories were established based on the car structural response in crash simulations.

Results indicate a range of incompatible collision scenarios with large intrusions in one of the involved cars combined with substantial lateral velocity change. Intrusion levels display considerable variations for collision types involving wheel interaction and a more robust response when front structures are engaged.

Keywords: frontal crash; oblique collisions; small overlap; crash simulation

NOTATION

ODB	Offset Deformable Barrier
FIMCAR	Frontal Impact and Compatibility Assessment Research
STATS19	National Accident Statistics for Great Britain, police attended
CCIS	Co-operative Crash Injury Study, Great Britain
GIDAS	German In-Depth Accident Study
PENDANT	Pan-European Co-ordinated Accident and Injury Databases
FP7	European Commission Seventh Framework Programme
IIHS	Insurance Institute for Highway Safety
ATD	Anthropometric Test Device, crash test dummy
LCA	Lower Control Arm
VPI	Volvo Pulse Index
FWRB	Full-Width Rigid Barrier

INTRODUCTION

Road transportation continues to claim more than 35,000 lives annually in the European Union, approximately half of these are occupants in passenger cars [1]. Looking at accident scenarios, it is estimated that about half of the fatal accidents involving car occupants occur in head-on collisions [2]. Furthermore, studies indicating a strong correlation between passenger compartment intrusion and occupant injury risk in front-to-front collisions [3,4] support the need for car structural integrity. When structural integrity is compromised, injury risk for car occupants increase both from direct contact with intruding structures as well as by influencing the effect of available restraint systems [5].

Consumer rating programmes using ODBs performed in both Europe and the United States have encouraged car designs with improved passenger compartment integrity in order to increase self-protection. High scores in such rating tests have been shown to correlate with decreased risk of fatal and severe injuries in real-world collisions [6,7]. However, fatalities and severe injuries still occur and it is therefore important to study the types of collisions involving modern cars, resulting in occupant injuries. Data from real-world collisions involving cars compliant with the current ECE-R94 regulations have been collected from Great Britain (STATS19, CCIS), Germany (GIDAS), and the EU (PENDANT), and analysed in the FP7 research project FIMCAR [8]. The FIMCAR analysis highlighted small overlap frontal collisions, defined as situations where front structures designed for energy absorption are not engaged as major load paths. The study suggested that small overlap situations may represent 15-20% of occupant MAIS2+ injuries related to intrusion in crashes involving modern cars. This finding is in line with previous research from the IIHS based on US accident data from cars rated as good for frontal crash protection, indicating that small overlap situations could cause approximately 25% of severe (AIS3+) injuries in frontal collisions [5].

Besides situations where occupant injuries can be directly linked to intrusion, both the FIMCAR and the IIHS studies [5,8] have identified deceleration without significant intrusion as a considerable cause of injuries to occupants of modern passenger cars. These situations were found when the available restraint system was not able to prevent occupant motion leading to contact with interior structures or when occupants sustained injuries from the restraint system itself. Vehicle design for future crashworthiness should therefore promote structural integrity combined with appropriate vehicle deceleration levels to support the functionality of restraint systems in frontal collisions.

In addition to designing vehicles for self protection, considerations should also be made for partner protection. Incompatibility in two-vehicle frontal collisions is characterised by differences in injury risk to occupants in one vehicle compared to the other [9]. These differences can be caused by both occupant and vehicle dissimilarities, but are normally discussed in terms of vehicle mass, geometry and stiffness [8]. Incompatibility has typically not been used to describe crash situations where both vehicles and occupants are identical. In a yet unpublished study [10], scenarios with large differences in terms of passenger compartment intrusion in one car compared to the other was defined as incompatible frontal collisions. These incompatible situations were found around 15° oblique impact angle combined with small overlap of car front ends. However, the increments at which the previous study was made left detailed analysis of these impact scenarios to be completed. Furthermore, the accuracy of results for the identified collision scenarios was uncertain and it was therefore decided that validation of the finite element model by means of physical crash tests was required.

The aim of this study is to describe oblique small overlap car-to-car collisions in a range of impact configurations. This will be achieved by validating a finite element model and by using this model for classifying collisions into a number of categories based on structural response in crash simulations.

METHOD

Model validation

A series of in-house crash tests were performed in order to increase the level of knowledge on small overlap collisions. To establish a database for comparison to finite element simulation models, various car models were tested in a range of initial velocities and impact configurations. Two examples of the crash tests used for finite element model validation of a single car model are presented below.

The first crash test was performed using a 15° oblique angle car-to-car setup, see Figure 1 where car identification numbers 1 and 2 are defined for use throughout this study. The horizontal alignment was chosen based on findings from the previous crash tests and corresponding simulations where the structural asymmetry could be expected to generate substantially different response in the two cars involved [10]. Both cars were given an initial velocity of 35 mph (56 km/h) and were equipped with identical front wheels with aluminium rims. Each car had a total weight of 1,762 kg, including ATDs and test equipment. The setup for the car-to-car crash test was chosen in order to avoid significant glance off, i.e. without interaction between frontal structures or wheels, in the following denoted sideswipe situations. An excessively large offset would increase the risk of minimal car interaction giving limited information on wheel response in small overlap collisions. Initial velocities were chosen to enable comparison to previously performed car-to-car crash tests. Effort was made to test cars as identical as possible, including choice of engine, gearbox as well as height and crash weight.

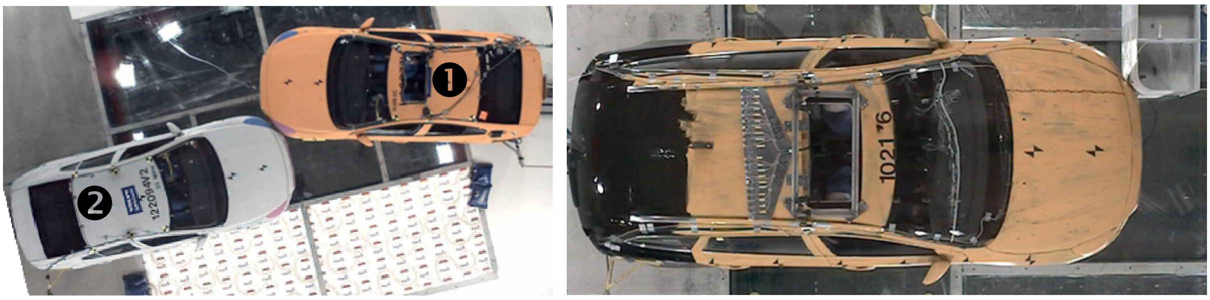

Figure 1. Crash test setup - #122094 car-to-car, and #102176 car-to-barrier. Numbers 1 and 2 are car identification numbers for car-to-car tests or simulations.

The second crash test used for model validation was a rigid barrier test [11] overlapping 25% of the car maximum width excluding external rear-view mirrors, see Figure 1. The initial velocity was set to 40 mph (64 km/h) and the car total weight was 1,916 kg, including ATDs and test equipment. Sill acceleration was measured on both sides of the car in the same positions as in the car-to-car test. Further, under view cameras were used in order to study the response of the wheels and wheel suspension during the collision, and the same type of wheel rim as used in the car-to-car test.

During the analysis of these two crash tests in comparison to simulation results, three major areas were identified where modelling improvements were required for increased model validity.

Front subframe bushings

The rubber bushings linking front suspension components and connecting the front subframe to the car body have been shown to have a significant influence on the initial stiffness and deformation modes of frontal structures. All the rubber bushings involved in frontal impacts were therefore modelled using solid elements in combination with the rubber material model previously developed by Centeno [12]. The initial phenomenon in both car-to-barrier and car-to-car crash tests was the separation of the LCA at the front-most bushing. This was included through a material failure model in the bolt connecting the bushing centre to the subframe. Also, material failure was introduced in the subframe sheet metal for capability of simulating edge failure around bolt holes.

Rim failure

In small overlap crash situations, there are considerable impact loads being transferred through the wheels. Occasionally, these loads exceed the design loads from normal driving conditions, leading to failure of the rim. By studying the crash tests, the ability for rim failure was judged to be critical for realistic results. Therefore, a modelling technique originally developed for engine suspension components [13] was applied to the aluminium rims. This model uses solid and shell elements in combination to represent the rim with elements eroding when a failure criterion is reached during crash.

Tyre separation

With the ability for the wheel rim to fail in a similar manner as in crash tests, the possibility for the tyre to be wrenched from the rim was identified as the next model improvement needed for correct geometrical interaction conditions. This feature required a tyre airbag model including leakage in proportion to a separation area to be developed for the purpose of this study. When the tyre is subject to lateral forces inducing shearing of the tyre relative to the rim, the separation area is initiated and enlarged. This, in turn, increases ventilation until the pressure inside the wheel is equal to the ambient air pressure.

The validated car simulation model was developed based on close studies of the response of structural components during the crash through film analysis and by post-crash inspection of phenomena such as bending, shearing, cracking and separation. Further, different car acceleration signals were compared until sufficiently correlated, see Appendix.

Parameter study

Based on the validated simulation model described above, a parameter study was set up for car-to-car collisions. Using the same setup logic as used in a previous study [10], impact angles between 10° and 20° were chosen in combination with offset distances between 900 and 1,500 mm, see Figure 2. The shaded area in Figure 2 represents this range of situations and should not be interpreted as a comprehensive set of small overlap scenarios. A uniform Latin hypercube approach [14] was used in order to populate the design domain (covering combinations in impact configuration). In total, 40 simulations were used for further analysis after passing a model quality check. Simulations were discarded in this check if more than 10% change in energy ratio was observed (total energy in relation to initial energy) or if more than 1% of the initial model mass was added during simulation.

Animations of the crash simulations from top, bottom and sectional views were studied in order to categorise distinct types of crash scenarios based on the structural interaction between the cars. These categories were then presented in terms of passenger compartment intrusions and pulse severity metrics. Passenger compartment intrusion was measured as the total intrusion in car longitudinal, lateral and vertical directions and the maximum value during each collision was stored for the A- and B-pillars, sill, front floor and dashboard panels. Passenger compartment longitudinal acceleration was evaluated by means of VPI [15]. This is a pulse severity indicator that represents the maximum occupant chest deceleration using a simplified mass-spring model with an initial slack. Since there are substantial differences in terms of longitudinal deceleration comparing the left and the right side of the car in a small overlap situation, only the struck side (left-hand side) sill accelerometer signal was evaluated for VPI. For the lateral deceleration, a pulse severity index is not established and therefore the maximum struck-side lateral velocity change was recorded for an indication of the risk of the driver being thrown outboard towards the door possibly sliding off the driver airbag during crash.

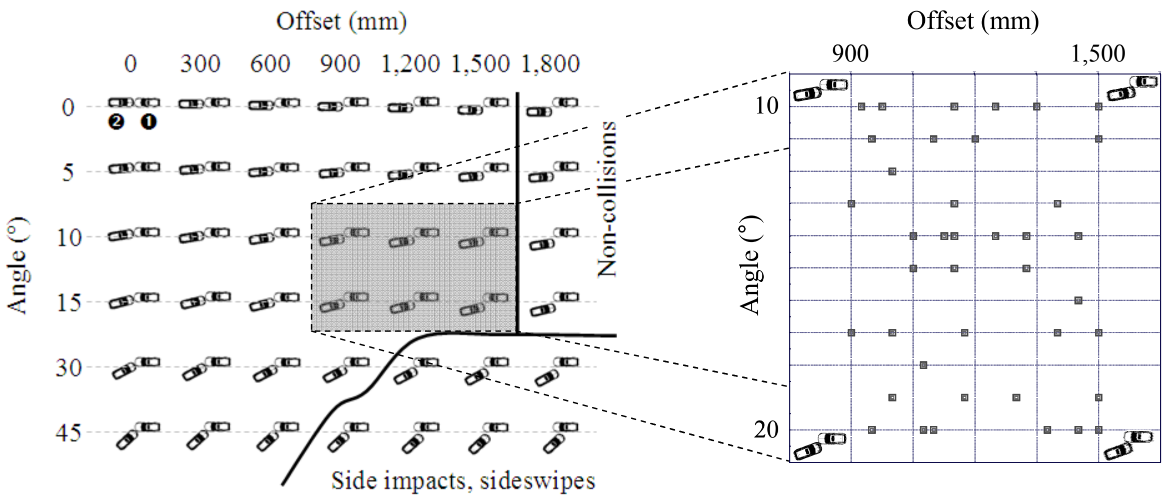

Figure 2. Overview of impact scenarios showing definition of offset (left, adapted from [10]) and detailed area studied with validated finite element model (right, each square indicating a scenario evaluated, N = 40). Car identification numbers are defined at top left corner. Shaded area indicates situations included in this study.

RESULTS

Categorisation of collisions

Five separate collision categories were distinguished in the simulations as seen in Figure 3. The categories were defined by the following criteria:

Category A was defined by at least one of the frontal structures being deformed and thus contributing to energy absorption in the collision. Frontal structures are defined by bumper beam, longitudinal sidemembers or front subframe.

Category B refers to the set of incompatible collisions as defined previously. In such situations, the wheels overlap each other, creating a locking phenomenon which leads to substantial deformations of the sill and A-pillar in Car 1, while there was sufficient strength in the sill of Car 2 to withstand this kind of loading.

Category C includes situations where the wheel of Car 2 becomes detached and is pushed along the sill of Car 2, in some cases deforming the sill structures during displacement.

Category D is similar to Category C, but with the distinct difference that the wheel of Car 2 is not pushed completely outboard of the sill. The wheel of Car 2 displays a greater rotation around the vertical axis than in Category C and it is compressed between the sill of Car 2 and different components in Car 1. All collisions in Categories B, C and D fall under the small overlap definition, i.e. where front structures designed for energy absorption are not engaged as major load paths.

Category E comprises sideswipe situations, where the wheels pass each other with a minimal influence on the A-pillar and deformation possibly occurring further rearward causing intrusion in the door region.

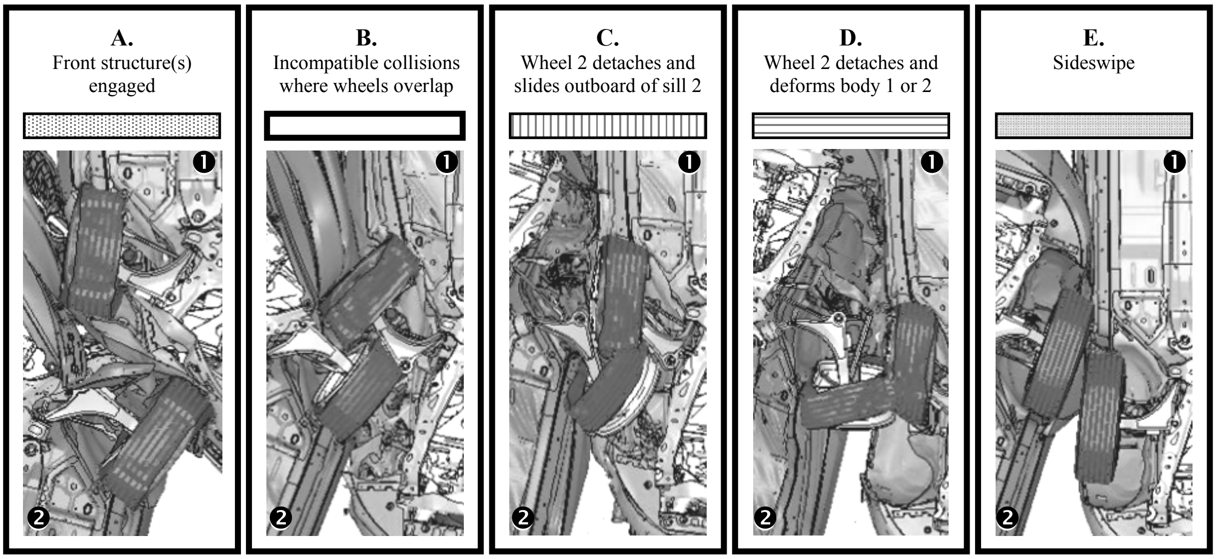

Figure 3. Categories A-E defined according to interaction response.
Bottom view of cars at maximum deformation including car identification numbers.
Categories B, C and D comprise small overlap situations.

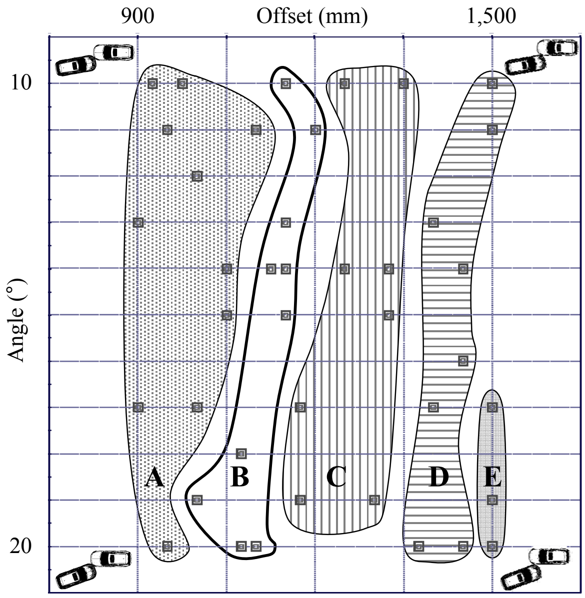

Figure 4. Evaluated impact scenarios grouped in Categories A-E, N = 40.

The collision categories can be visualised in the parameter study design domain, i.e. different combinations of offset distance and oblique angles as shown in Figure 4. This shows how the parameter offset distance appears to have a greater effect on the collision characteristics and consequently which collision category each evaluated impact scenario belongs to. The oblique angle appears to affect the collision scenario to a minor extent within this domain. However, there are situations that displays sensitivity to the oblique angle, e.g. between Categories A and B at oblique angles close to 20°.

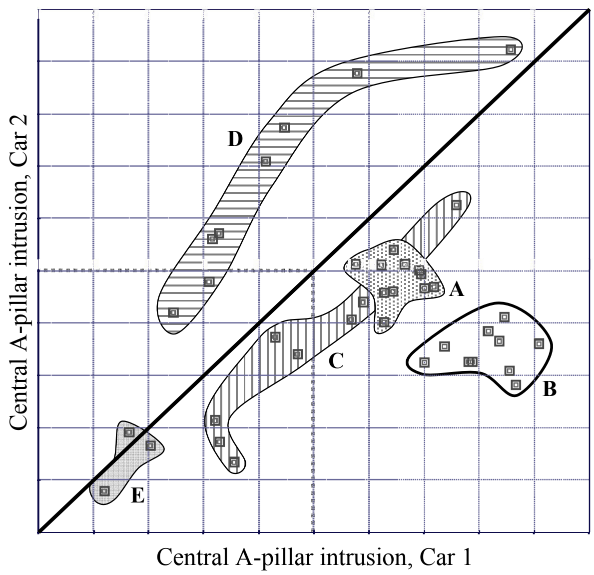

Figure 5. Central A-pillar intrusion in Car 2 vs. Car 1 grouped in Categories A-E, N = 40. The dotted line indicates intrusion recorded for both car-to-car 50% overlap collinear at 35 mph and car-to-ODB 40% overlap at 40 mph.

Intrusion

In accordance with the previously conducted study [10], intrusion of the A-pillar measured halfway between the sill and the roof was found to correlate strongly to intrusion in the sill, floor and dashboard panel areas and was therefore selected for further analysis. All simulation results were compared in terms of central A-pillar intrusion and grouped into the previously defined collision categories, as shown in Figure 5. The solid diagonal line represents equal intrusion in both cars and separates the domain into the lower-right area containing cases comprising greater intrusion in Car 1 than Car 2 and less intrusion in Car 1 than Car 2 in the top-left area.

The different collision categories were clearly distinguished in the intrusion domain shown in Figure 5. The majority of simulations showed greater intrusion in Car 1 than in Car 2, except situations in Category D where the wheel of Car 2 is caught between the cars, resulting in greater intrusion in Car 2. As expected, the sideswipe situations (E) give limited intrusion balanced in approximately equal proportions between the two cars. Collisions in Category C were widely and consistently distributed within the lower-right area of the diagram, i.e. Car 1 sustaining greater intrusion. Some of the greatest intrusions were found in Category B, which also represent the largest relative difference between the cars. In terms of intrusion, both Category A (frontal structures engaged) and Category B displays relatively low variation compared to Categories C and D.

As can be seen in Figure 5, only ten of the evaluated impact scenarios fall within the intrusion that can be expected in a 50% overlap car-to-car collinear collision, i.e. at 0°, at 35 mph or car-to-ODB at 40% overlap and 40 mph initial velocity (indicated by the dotted line).

Deceleration

All simulations included evaluations of VPI calculated from the struck (left-hand) side acceleration. This crash pulse metric approximating driver chest deceleration based on a simplified model of a restraint system connected to the car longitudinal deceleration is shown for Car 2 vs. Car 1 in Figure 6. The VPI distribution suggests the highest occupant deceleration loading in the Category A situations involving the front structures, followed by the incompatible collisions in Category B. In both Categories C and D, the wheel of Car 2 is detached, resulting in a considerably lower deceleration in both cars, illustrated in Figure 6.

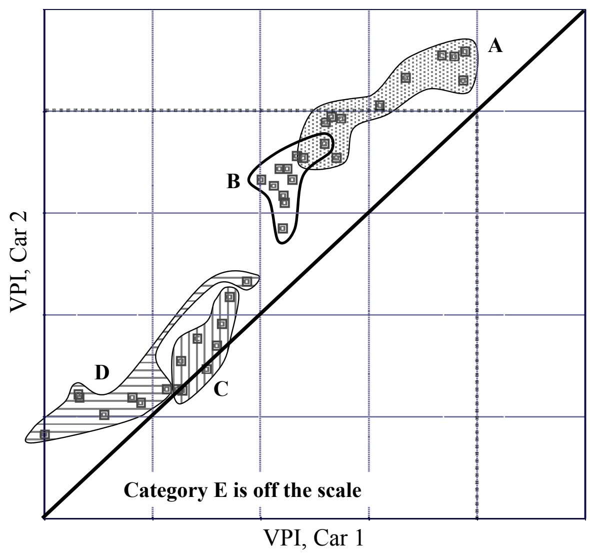

Figure 6. Volvo Pulse Index for Car 2 plotted vs. Car 1 grouped in Categories A-E, N = 40. The dotted line indicates VPI recorded for both car-to-car 100% overlap 0° at 35 mph and car-to-FWRB 100% overlap at 35 mph.

Sideswipe situations in Category E represented low longitudinal deceleration and were therefore not included in the diagram in Figure 6.

Most evaluated scenarios represent crash pulses comprising lower severity than the reference scenarios, 100% overlap 0° car-to-car collision or car-to-FWRB, both at 35 mph. However, there are six scenarios where the VPI for Car 2 is predicted to be higher than the reference scenarios.

Lateral velocity change

The maximum velocity change measured in the lateral direction was recorded and is shown in Figure 7. In all but one simulation, a greater maximum lateral velocity change was recorded in Car 1 than in Car 2. Categories A and B overlap in this domain, representing situations comprising a greater lateral velocity change for Car 1 in combination with a lower lateral velocity change for Car 2. Category C and D situations resulted in a more similar velocity change for Cars 1 and 2. Sideswipe situations in Category E were left out of the scale in Figure 7 as the lateral velocity change in Car 1 is considerably greater than in Categories A-D due to the car being struck from the side, see Figure 3.

Comparing the lateral velocity changes recorded in these oblique situations to the reference situation of 50% overlap collinear car-to-car collision at 35 mph suggests a considerably greater lateral velocity change in Car 1 and, for the majority of cases, less for Car 2.

Summary of results

The results presented in Figures 5-7 were collected and compared in terms of the different collision categories and their relative magnitudes in Table 1. As can be seen in Table 1, Categories A, B and D represent the greatest intrusions and VPI. The combination of large intrusions and large lateral velocity change was only found for the incompatible collisions in Category B.

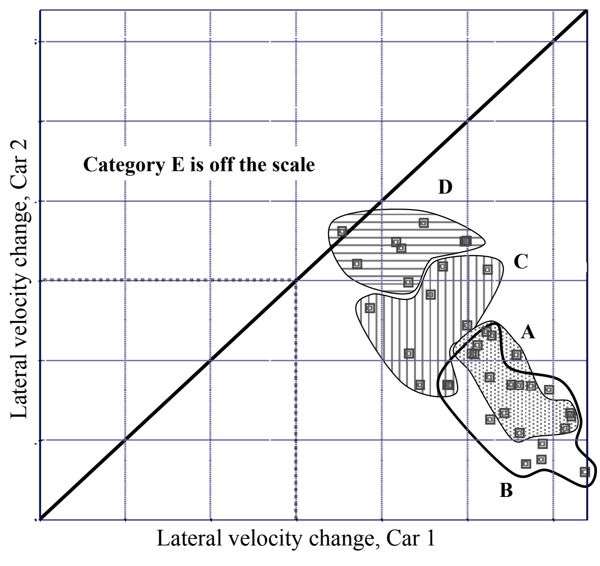

Figure 7. Maximum lateral velocity change in Car 2 plotted vs. Car 1 grouped in Categories A-E, N = 40. The dotted line indicates lateral velocity change recorded for car-to-car 50% overlap 0° at 35 mph.

Table 1. Summary of results grouped by collision categories.

		Categories A Front structure(s) engaged	B Incompatible collisions wheels overlap	C Wheel 2 detaches and slides outboard of sill 2	D Wheel 2 detaches and deforms body 2	E Sideswipe
Car 1	Intrusions	Medium	Large	Medium	Large	Small
	VPI	Large	Medium	Small	Small	Small
	Lateral velocity change	Large	Large	Medium	Medium	Large
Car 2	Intrusions	Medium	Medium	Medium	Large	Small
	VPI	Large	Medium	Small	Small	Small
	Lateral velocity change	Small	Small	Medium	Medium	Small

DISCUSSION

Structural safety design of passenger cars is based on legal and internal requirements. These requirements should be based on knowledge from real-world collisions combined with additional knowledge gained from controlled environments like laboratory crash testing and computer simulation. Crash testing can however only be performed to a limited extent due to the substantial costs involved, making parameter studies difficult to conduct. Furthermore, there is always some variation in crash test results caused by car and crash test variations. Therefore, finite element simulation is better suited for parameter studies but may have disadvantages in terms of model validity. This implies that a predictive finite element model should only be used within the boundaries for which it has been validated. In order to expand these boundaries and to support continuous improvement of structural integrity and crash energy absorption in passenger cars, finite element modelling techniques must therefore be improved regularly.

While developing a well correlated base model on which to build the detailed parameter study, several difficulties were encountered. The rotation around a vertical axis of the front wheel of Car 1 (striking car) was consistently greater in the crash test compared to the simulation model. Possible sources for discrepancies may be the friction model between the tyre and the surrounding components. Furthermore, predicting rim failure proved to be difficult since a substantial degree of variation is involved in the physical tests.

The parameter study boundaries were selected on the basis of the previously conducted methodology study [10]. In that study, it was found that within these limits, both a threshold zone for sideswipe scenarios as well as highly incompatible impact scenarios could be found. It should be noted that the borderlines between collision categories seen in Figure 4 are drawn to include all the evaluated scenarios within each category and do not represent an exact limit. The borderlines could possibly be refined if extended simulation data was available.

Situations where at least one of the front structures was engaged in the deformation zone (Category A) represent a robust impact configuration in this dataset. This means that even if there is relatively large variation in the impact oblique angle and/or offset distance as shown in Figure 4, it only affects the intrusion to a minor extent, see Figure 5. In terms of VPI, Category A collisions represent some of the most severe deceleration levels as was expected when increased structural interaction was applied. For Category A, the VPI distribution in Figure 6 displays a greater spread compared to intrusion in Figure 5, which reflects the VPI sensitivity to crash pulse shape.

The objective for passenger cars to maintain robust structural response is to minimise the incidence of major intrusions, thereby allowing the occupant restraint systems to operate in a controlled environment. For both cars in Figure 5, the dotted line is given as an example of an acceptable level of intrusion. For Category A, where the frontal structures of the cars are engaged in energy absorption, the robustness in terms of structural response is already at an acceptable level within the limits of this parameter study, therefore further robustness efforts appear not to be needed.

The unequal distribution of the intrusion in Category B appears to be caused by the wheels overlapping each other. Due to the above, the wheels and wheel suspension lock and the opportunity for these components to deform or being displaced is very slim. This in turn often leads to greater intrusion in the car possessing the most disadvantaged loading into the car body (Car 1 in this case). To prevent front wheels to lock, it has been proposed to actively turn the front wheel toe-in in order to create a sliding plane from which the crash opponent would be diverted [16]. This action would seemingly require a substantial wheel rotation angle before achieving a positive effect. Such action must also be well balanced with the risk of a secondary collision in cases where turning of the wheels is successful in avoiding a collision.

For Categories C and D, where the wheel of Car 2 detaches, the issue appears to be relating to robustness. As shown in Figure 5, a large spread in central A-pillar intrusion is apparent and the severity level is depending on the structural response before and after the wheel has become detached. In terms of intrusion and VPI, it is suggested that wheel detachment is preferable compared to wheels locking as seen in Category B. However, if detachment cannot be achieved in a predictable way, intrusion may vary substantially.

The last category of collisions, sideswipe situations (E), has not been studied in further detail. In terms of intrusion into the passenger compartment, it is mainly the door or B-pillar region which is affected. It is, however, worth noting that considerable lateral deceleration may occur in similar situations and this should be taken into consideration when occupant restraint systems are designed.

The situations described above may be addressed by structural countermeasures, but may also be addressed by future active safety systems helping drivers of passenger cars avoid collisions. Depending on the cars involved in the collision, certain scenarios to avoid could be defined based on which combinations of braking and steering are employed with the aim of seeking the least harmful impact scenario.

The cars used in this study were identical; including crash opponents of different size and mass could affect the findings on separation effects. Further to body structures being dimensioned for greater crash loads, larger and heavier cars will be dimensioned for greater dynamic loads during normal driving, thus making separation forces higher. In situations where the wheel became detached from the wheel suspension in this study, a different outcome may have been the case if a larger car was involved in the collision. Furthermore, larger cars equipped with larger wheels would affect the interaction response between the cars and could also lead to other outcomes. To bring a wider perspective to the subject matter in this study it would be advantageous to consider impacts with fixed objects such as trees, poles and roadside objects as well as other car types such as heavy goods vehicles.

Both the FIMCAR project and the IIHS have identified that severe injuries often occur in situations without significant intrusion. This suggests that further development of restraint systems in combination with improving the deformation modes of frontal structures may be needed in order to decrease the number of acceleration-related injuries. However, structural integrity must not be compromised in order to obtain lower deceleration crash pulses. The front structure should provide the needed deformation length supported by a compartment that has sufficient strength to provide structural integrity for situations where the front structure cannot fully be utilised.

CONCLUSIONS

The following conclusions were made:

- A set of oblique car-to-car collision types were identified and described. Results show substantial variations in terms of passenger compartment intrusion for collision types involving wheel interaction and more robust response when front structures are engaged.

- A clearly distinguishable transition zone where small overlap situations occur was described between situations with engagement of front structures and sideswipe situations.

ACKNOWLEDGEMENTS

This work has been carried out in association with SAFER – Vehicle and Traffic Safety Centre at Chalmers, Sweden.

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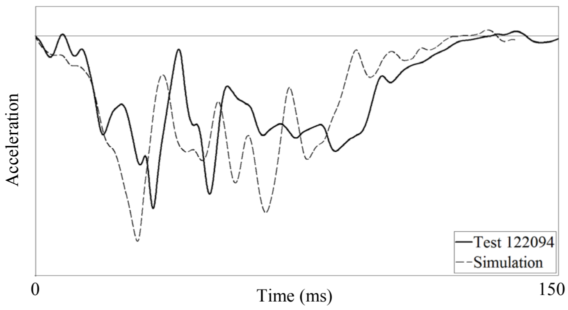
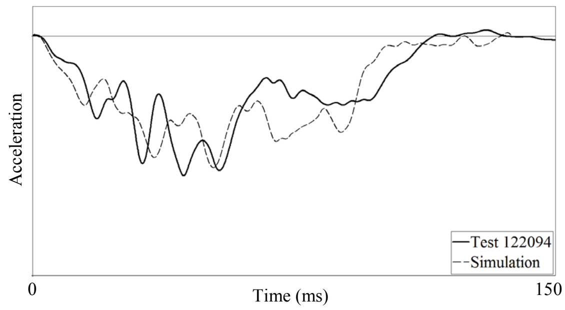
Appendix

Comparison of car longitudinal acceleration for car-to-car test 122094

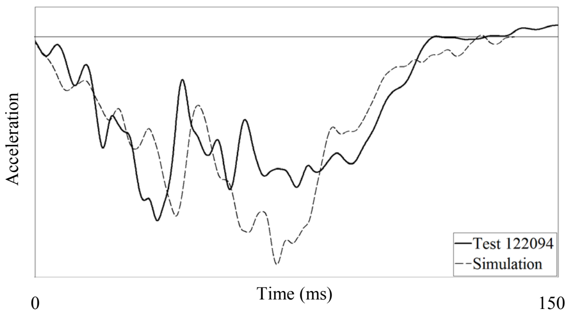
Car 1 left sill



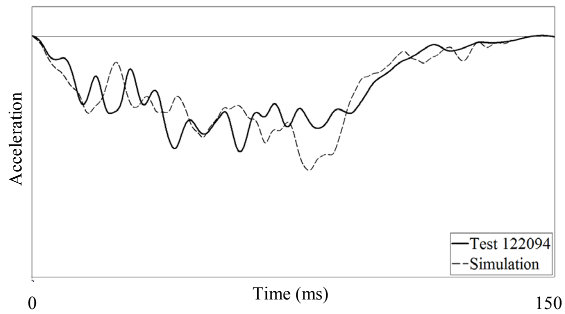
Car 1 right sill



Car 2 left sill

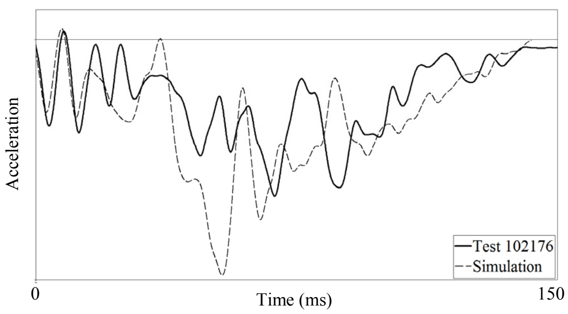


Car 2 right sill

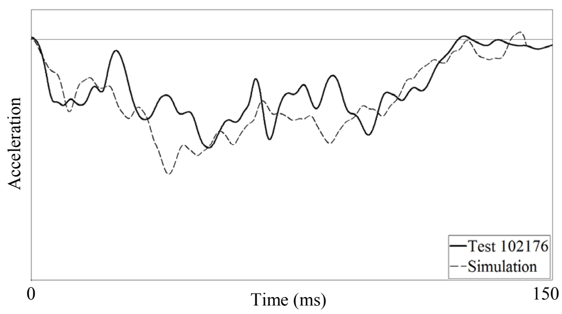


Comparison of car longitudinal acceleration for car-to-barrier test 102176

Car 1 left sill



Car 1 right sill



Paper V

"Adaptive structure concept for reduced crash pulse severity in frontal collisions"

Wågström, L., Kling, A., Berge, S., Norin, H., Fagerlind, H. (2013)

International Journal of Crashworthiness (published online).

International Journal of Crashworthiness, 2013
http://dx.doi.org/10.1080/13588265.2013.833394

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Adaptive structure concept for reduced crash pulse severity in frontal collisions

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(*Received 2 April 2013; accepted 29 July 2013*)

From accident statistics in real-world frontal collisions, it has been shown that a considerable portion of injuries occur in situations without major passenger compartment intrusions and that these injuries can be attributed to the occupant interaction with the restraint systems. To address these types of injuries, a novel front structure concept is proposed. This structure includes a partly detachable front subframe that can actively be released from the passenger compartment and thereby reduce the crash forces and related decelerations. The aim of this study was to quantify the effect of an adaptive front structure on occupant loading in a modern passenger car in frontal crash situations. A simplified finite element model was derived from a full vehicle model in order to run a large simulation matrix spanning from full overlap to small overlap situations. Occupant loading was estimated by using two simplified occupant chest deceleration models, the Volvo Pulse Index and the Occupant Load Criterion. Results suggest that modifying the crash pulse shape can be equivalent to reducing the velocity change in a crash by up to 44%. In relation to scenarios without subframe release, this indicates a considerably lower force required to be applied to the occupant from the restraint system.

Keywords: frontal crash; adaptive structures; crash pulse severity; crash simulation

1. Introduction

In frontal crashes involving modern passenger cars, compartment intrusion, as well as deceleration can lead to occupant injuries. Studies in the USA into the real-world performance of cars, which have been awarded good ratings in the Insurance Institute for Highway Safety (IIHS) frontal moderate overlap test, have shown that many belted occupants still sustain severe injuries in frontal crashes without significant vehicle passenger compartment intrusion [1]. In such cases, the injury mechanisms are related to the crash pulse and occupant interaction with the restraint system or car interior. Similar findings were presented in the European Commission Seventh Framework Programme project FIMCAR (frontal impact and compatibility assessment research) [2], where datasets of frontal crashes involving R94-compliant vehicles from Great Britain (Cooperative Crash Injury Study) and Germany (German In-Depth Accident Study) were analysed. The study showed that approximately 40% of Maximum Abbreviated Injury Scale (MAIS) 2+ injuries and 30% of fatal injuries suffered by occupants occurred in crashes with more than 75% frontal overlap, and it was suggested that compartment intrusion may not be the direct cause of injury. Besides improving the functionality and robustness of restraint systems, injuries related to the crash pulse can potentially be addressed by the vehicle front structural response.

It has been suggested that adaptive structures can be used in order to affect the deceleration response in frontal impacts. Witteman suggested 'high–low–high' deceleration pulses to be optimal based on occupant response simulations [15]. These deceleration pulses were proposed to be accomplished by friction forces applied to steel cables that had the additional benefit of being able to transfer loads from the struck side of a vehicle to the non-struck side. A different approach to achieve 'high–low–high' deceleration pulses in a passenger car was proposed by Motozawa et al. [5], where axial buckling was followed by bending of the main energy absorbing members. A practical solution for the required operational volume for such a system was, however, not presented. Furthermore, Pipkorn et al. [8] recommended implementing variable crush force in passenger cars by pressurising vehicle longitudinal frontal members. Since additional volume is not required for the pressurised frontal members to function, this solution may be more efficient in terms of packaging space than the concept proposed by Motozawa et al. [5]. Any solution to how the Pipkorn et al. [8] proposal could be applied to production vehicles was not presented, although the mass-reducing potential based on increased force levels from pressure in such members was highlighted.

Wågström et al. [13] demonstrated in 2004 that adaptive structures could greatly affect the vehicle deceleration in

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car-to-car (C2C) crashes by using simplified mathematical models. In a subsequent study, Wågström et al. [14] used finite element models to study how a considerable effect on the crash pulse shape could be achieved by adaptive structures. The conclusion from that study was that the structural deformation of future cars should be globally controllable, i.e. the main load paths should be adaptive in order to affect the crash pulse significantly. This finding was supported by a review on adaptive vehicle structures made by UK's Transport Research Laboratory (TRL) in 2008 [11]. The TRL study identified altering the frontal force level as one of the key principles for adaptive structures.

The primary function of the front engine subframe is to support loads from the powertrain, as well as road loads, during normal driving conditions, through powertrain supports and the front wheel suspension. Moreover, the front engine subframe has been identified as a major load path in frontal collisions, important both for self-protection and crash compatibility [7]. From crash test data, it can be seen that several car models already use detachment of the front subframe as a means to reduce the longitudinal deceleration in full-frontal crashes [6]. Equipping cars with an adaptive detachable front subframe may have the potential to benefit the overall real-world crash performance. This way, benefits for self-protection regarding passenger compartment deceleration could be actively achieved without compromising energy absorption needed for reducing compartment intrusion in other scenarios.

The aim of this study was to quantify the effect an adaptive detachable front subframe has on occupant loading in C2C frontal collisions in a range of lateral offset distances and closing velocity levels.

2. Method

2.1. Finite element model validation

A full vehicle LS-DYNA [4] model used for in-house product development, as described in further detail by Wågström et al. [12], was selected. To perform a large number of simulations, the original model illustrated in Figure 1(a) was simplified by removing the majority of the structural components from a plane rear of the A-pillars, with a few exceptions, as illustrated in Figure 1(b). The bonnet and interior components such as seats, as well as crash test dummy models were removed. To compensate for the removed mass in this simplification process, a rigid body was established with centre of gravity and inertial properties representing those of the removed components. Since the deceleration response in crashes has to be studied, mass was added to match the total displacement in full-width rigid barrier (FWRB) crash at 56 km/h. While the total mass of the original model was 2010 kg, the total mass of the simplified model was 1845 kg. The bulkhead between the engine bay and the passenger compartment was not modified and thus had the potential to be deformed as in the original model.

To verify that the dynamic structural response was preserved after the simplification, the simplified vehicle model was compared with the original model in terms of passenger compartment deceleration. The two models were therefore crashed into a FWRB at 56 km/h and into an identical vehicle in a C2C crash, each with an initial velocity of 56 km/h and at full lateral overlap. According to Figure 2, this comparison showed relatively small differences between the two models and crash scenarios in terms of average left-hand side (LHS) and right-hand side (RHS) sill deceleration. Hence, the simplified model was considered to be acceptable for use in further analyses of crash pulse effects. The simplified model displays lower initial deceleration, which is attributed to the absence of crash test dummies and car interior components. Since the mass of the removed parts were incorporated into a rigid body, a larger portion of the total vehicle mass was directly coupled to the body structure to which the accelerometers were rigidly attached.

The simplified model was also compared to 457 vehicles crash tested with FWRB between the years 2000 and 2010 in the US NHTSA (National Highway Traffic Safety Administration) new car assessment program (US-NCAP) at 56 km/h [6]. To make a fair comparison of the structural response of vehicles in this database, a normalised stopping distance was calculated. This normalisation was made by subtracting the displacement when the average sill deceleration first exceeds 5 g from the total displacement. As illustrated in Figure 2, the displacement for the

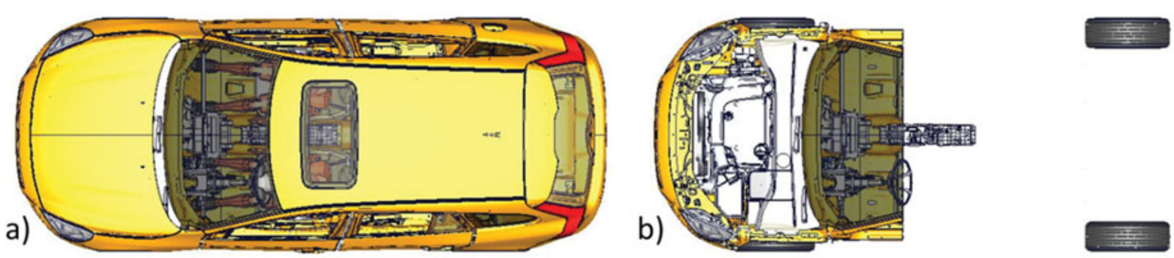

Figure 1. Model simplification, top view. Original model to the left (a) and simplified model to the right (b).

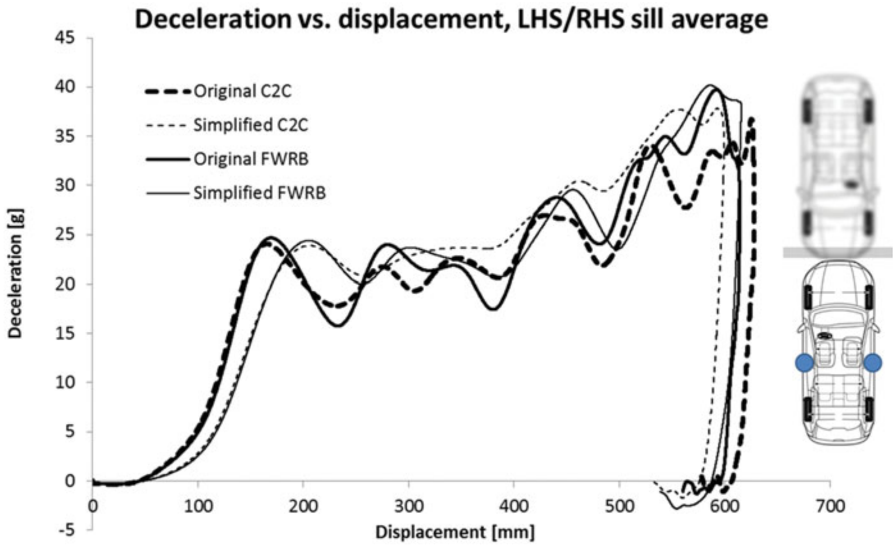

Figure 2. Average sill crash pulses from original and simplified models in FWRB crash at 56 km/h and C2C crash at 56 km/h in both of the identical vehicles, at full overlap. Filled dots indicate accelerometer locations.

simplified model when deceleration exceeds 5 g is approximately 100 mm. The corresponding maximum displacement is approximately 600 mm, making the normalised stopping distance of the simplified simulation model approximately 500 mm. The normalised stopping distance of all the 457 studied vehicles from US-NCAP is illustrated as a histogram in Figure 3. This overview shows that the simplified simulation model has a shorter normalised stopping distance than the majority of vehicles included in the comparison.

To examine how representative the crash pulse shape of the simplified vehicle model is, all vehicles with a normalised stopping distance between 450 and 550 mm were

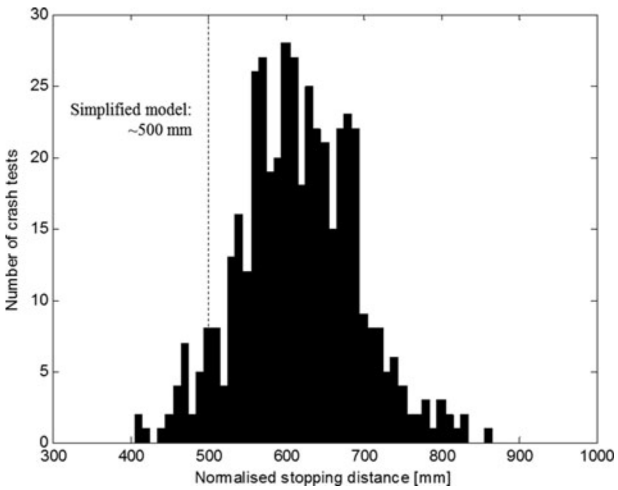

Figure 3. Normalised stopping distance of simplified vehicle model (dashed line) compared to distribution of vehicles tested in the NHTSA, NCAP, $N = 457$.

selected for comparison. Out of the total 457 vehicles, 74 vehicles fell into this category. An average crash pulse was calculated from the deceleration vs. time signals, after time-shifting all crash pulses to zero when crossing 5 g. The average crash pulse was complemented by adding and subtracting one sample standard deviation, indicated by the shaded areas in Figure 4. The normalised stopping distance for the average crash pulse shape was calculated by considering the velocity change in the average crash pulse shape as it exceeds 5 g. As illustrated in Figure 4, the simplified simulation model response lies within ± 1 standard deviation of the average crash pulse with a few exceptions late in the crash pulse. When plotted versus time in Figure 4(a), these exceptions are clearly visible. However, when the model response is plotted vs. normalised stopping distance in Figure 4(b), the difference between the simulation model and the crash tests of US-NCAP vehicles with similar normalised stopping distance appears less substantial.

Since this study was aimed at simulating a range of lateral offset scenarios, the simplified model was also compared to the original model at 900 mm lateral offset in terms of deceleration vs. displacement response for the LHS and RHS sill, as illustrated in Figure 5. As previously observed, the simplified model shows lower initial deceleration. Overall, the simplified model response was, however, comparable to the original model response and the simplified model was therefore considered acceptable for studies on crash pulse effects at lateral offset distances up to around 900 mm as well.

To estimate the effect of subframe detachment on the vehicle crash pulse, the front subframe rear connection points on the car body, marked with circles in Figure 6, were modified. Connection beams with failure at a specified time were implemented using *MAT_SPOTWELD in LS-DYNA [4].

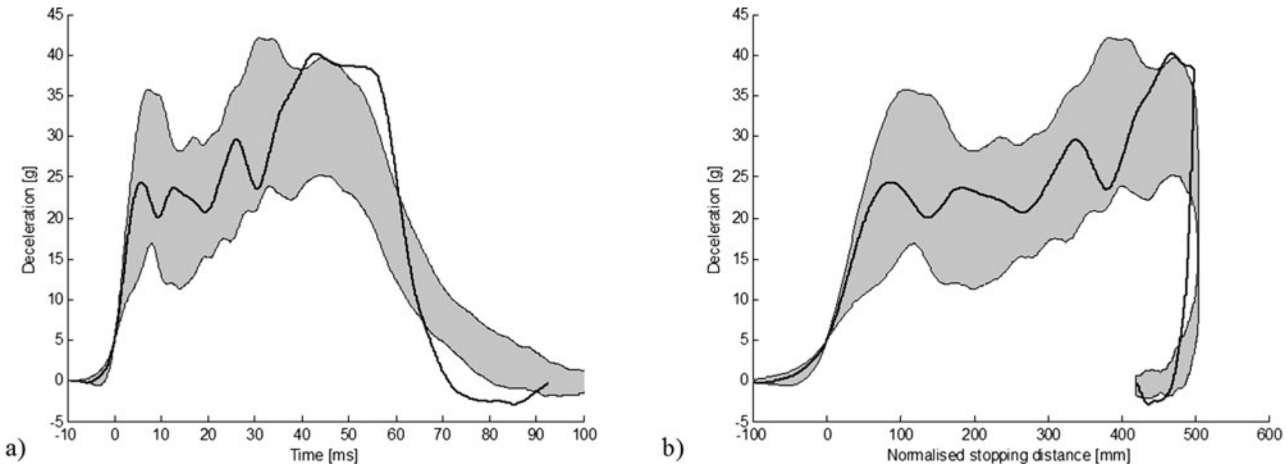

Figure 4. Crash pulse shape of simplified simulation model (thick solid lines) compared to ± 1 sample standard deviation (shaded areas) of pulses from US-NCAP with 450 to 550 mm normalised stopping distance ($N = 74$).

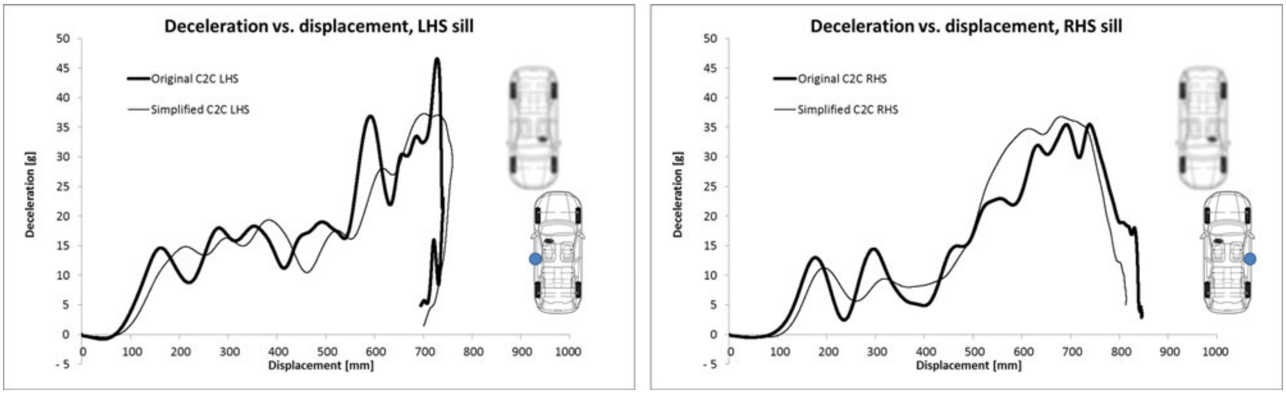

Figure 5. LHS and RHS sill crash pulses from original and simplified model in C2C crash with identical vehicles at approximately 50% (900 mm) lateral offset. Filled dots indicate accelerometer locations.

By applying this method, it was possible to control instantaneous subframe release by an input parameter. The connections at the front were not modified, i.e. these connections were non-detachable, marked by squares in Figure 6.

2.2. *Crash severity indicators*

The acceleration signals of the sill structures close to the B-pillars were defined for time history output. These signals were used to estimate the crash pulse severity based on two

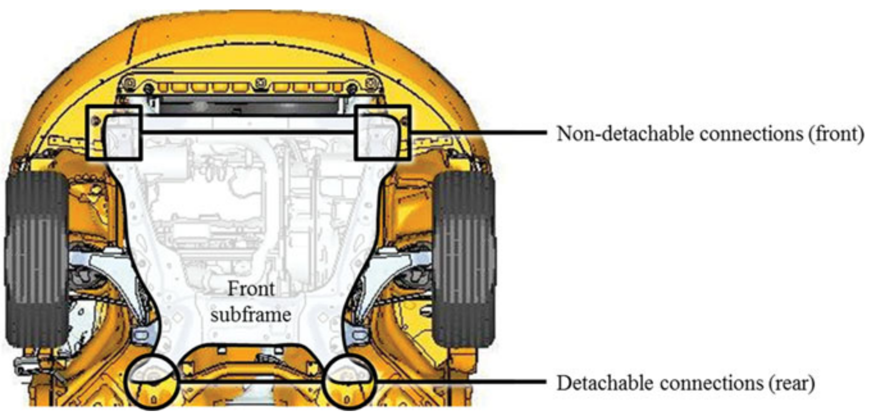

Figure 6. Bottom view of front subframe with connections to body structure. Detachable connections marked with circles, non-detachable connections marked with squares.

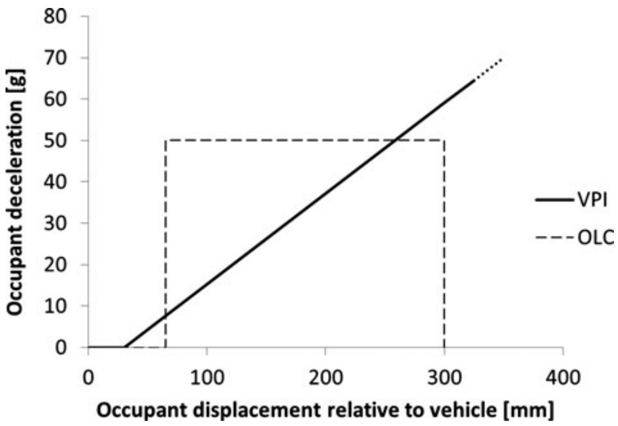

Figure 7. Characteristics of the crash severity indicators VPI and OLC. Occupant deceleration plotted vs. occupant displacement relative to vehicle.

simplified models, the Volvo Pulse Index (VPI [3]) and the Occupant Load Criterion (OLC [10]). Both models are attempts of generically measuring the restraint forces that the driver is subjected to during a crash, based on deceleration only. Each model uses the occupant displacement relative to the vehicle in the longitudinal direction, hereafter referred to as relative displacement. Furthermore, both models assume an initial phase of free-flying motion, 'slack', without any occupant deceleration at relative displacement less than 30 mm for VPI and 65 mm for OLC, as illustrated in Figure 7. However, the model responses following the initial slack represent fundamentally different assumptions. The VPI assumes a linearly increasing occupant deceleration of 0.25 g/mm of relative displacement, without any limitation on occupant deceleration or relative displacement. The OLC instead assumes a maximum relative displacement of 300 mm and a constant occupant deceleration up to this point. This means that the OLC model assumes a perfectly adaptive restraint system that will always utilise the available interior distance. The VPI model, on the other hand, simulates a non-adaptive restraint system based on chest decelerations measured in crash test dummies in physical tests.

2.3. Simulation matrix

The simplified C2C model was used to simulate crashes between identical vehicles at lateral overlaps from 0 to 1200 mm, with increments of 100 mm and initial velocities from 30 to 55 km/h with increments of 5 km/h, as illustrated in Figure 8. No oblique angle or vertical offset was assumed and any friction with ground was disregarded. The coefficient of friction between the vehicles was set to 0.2. Identical initial velocity was applied to both vehicles, i.e. the closing velocity was twice the initial velocity of each vehicle. Based on previous studies [12] and test runs, a sufficient termination time was determined to 100 ms for offsets below 1000 mm and 120 ms for offsets above and including 1000 mm. In total, 780 simulations were performed.

2.4. Numerical quality assurance

All simulations were controlled with respect to numerical quality. Total energy over time was restricted not to change by more than 10% in relation to its initial value, as indicated by the energy ratio in LS-DYNA [4]. Since all

	Min	Max	Increment	Number of levels
Horisontal offset	0 mm	1,200 mm	100 mm	13
Initial velocity	30 km/h	55 km/h	5 km/h	6
Subframe release time	0 ms	90 ms (no release)	10 ms	10

Figure 8. Simulation matrix used to evaluate effect on crash pulse of detachable subframe, $N = 780$. Same initial velocity applied to both vehicles.

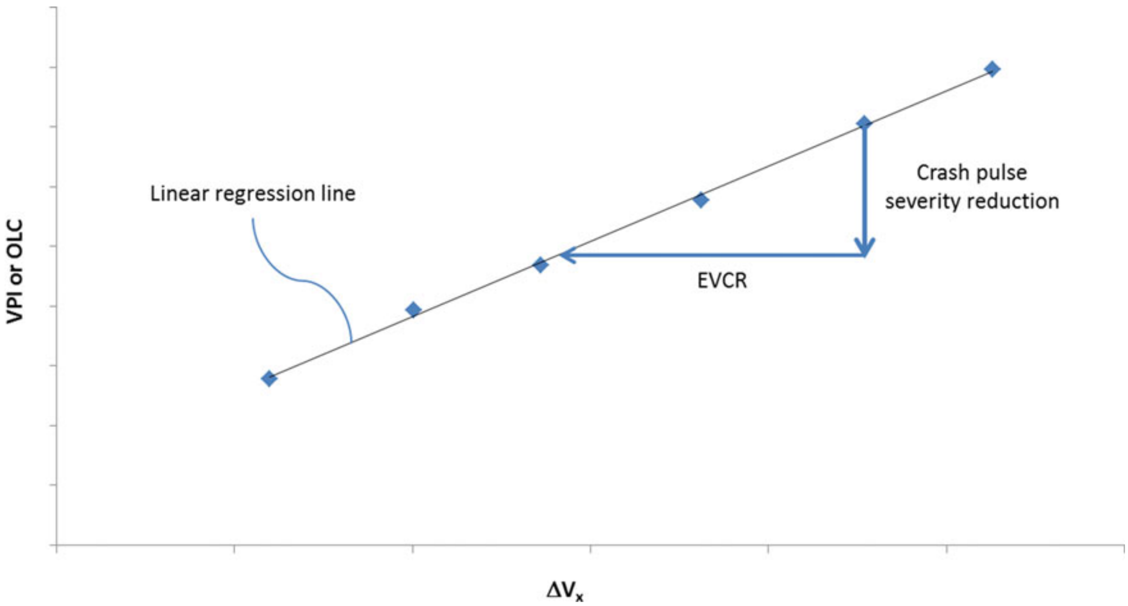

Figure 9. Definition of EVCR based on relationship between maximum longitudinal velocity change (ΔV_x) and VPI or OLC. Example given from one crash simulation scenario.

simulations were performed using an explicit numerical integration method with a constant time step, the model was subjected to added mass in order to maintain the selected time step. All simulations where more than 1% of the initial model mass was added during simulation were excluded.

2.5. *Equivalent velocity change reduction*

The results from the VPI and OLC models were used to estimate the equivalent velocity change reduction (EVCR) using the simplified crash simulation model. That is, how much lower the change of velocity would have to be in order to arrive at the same reduction in VPI or OLC that was accomplished by changing the crash pulse shape with subframe release. For each crash scenario, the EVCR was calculated by establishing the relationship between the maximum longitudinal velocity change (ΔV_x) and OLC or VPI, as illustrated in Figure 9. A linear regression was performed by comparing the Pearson product-moment correlation coefficient R [9]. Based on the regression line, an estimate for the EVCR was made based on the reduction seen from adaptively detaching the subframe in terms of VPI or OLC. It was decided not to calculate EVCR for a specific lateral offset if R was found to be less than 0.97 for either the VPI or the OLC models, indicating a considerable nonlinearity between ΔV_x and OLC or VPI.

3. Results

After controlling the numerical quality in the 780 original simulations, 770 simulations (98.7%) were included for further analysis. The remaining simulations were rejected due to the failing of the energy-ratio criterion.

The effect of subframe release is increased stopping distance, and whether this is associated with lower VPI or OLC depends on the crash pulse shape. As an example, the influence on the crash pulse shape from releasing the subframe at 30 ms is illustrated in Figure 10. In this example, with 100 mm lateral offset and initial velocity of 35 km/h in each vehicle, the increased stopping distance is clearly associated with lower deceleration in the late phase of the crash pulse. Such reduced deceleration is beneficial for occupant loading as measured by the VPI model as well as the OLC model. However, if the initial velocities are increased, the extended stopping distance could potentially be associated with higher deceleration, which may result in higher occupant deceleration compared to the case without subframe release.

As illustrated in the example given in Figure 9, data from all simulations without subframe release were collected to establish the correlation between longitudinal velocity change ΔV_x and VPI or OLC values, respectively. The results of the linear regression are presented in Table 1 and indicate strong correlation between ΔV_x and occupant deceleration as measured by both VPI and OLC models. The correlation weakened as the lateral offset increased above 800 mm, which is connected to a nonlinear intrusion response for the large offset scenarios as the initial velocity increases.

For all the crash scenarios defined by the simulation matrix in Figure 8, the minimum value for VPI and OLC was compared to the corresponding values without

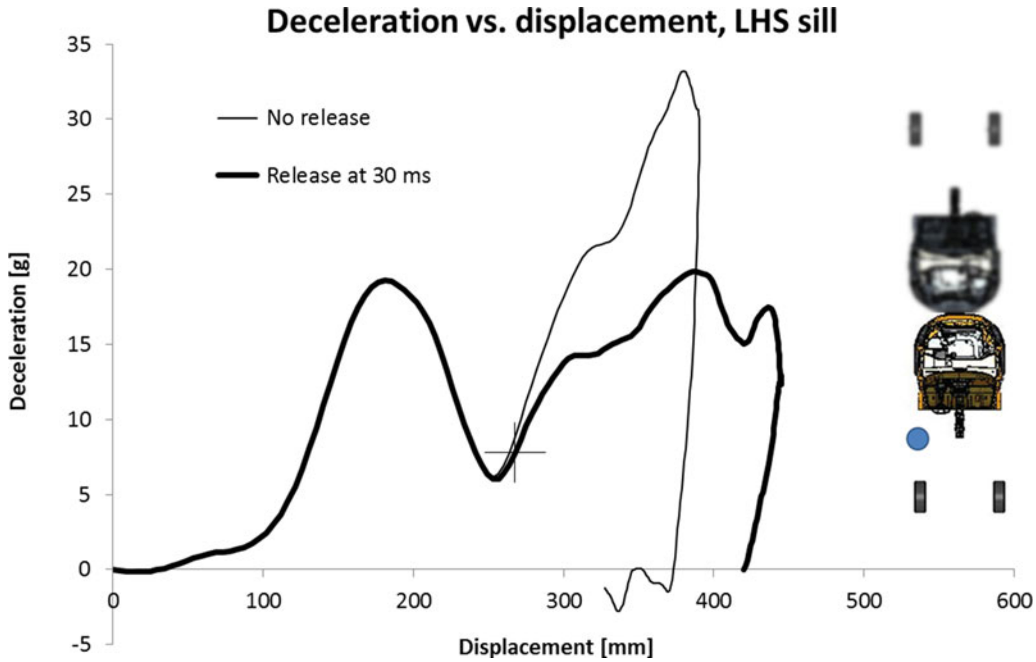

Figure 10. LHS crash pulses from simplified model without release compared to release at 30 ms (marked by cross). Load case: 35 km/h initial velocity in each vehicle and 100 mm lateral offset. Filled dots indicate accelerometer location.

subframe release. Each of the minimum values was associated with a certain time to release the subframe. To compare the crash severity reducing effect of releasing the subframe, the EVCR based on the linear regression model presented in Table 1 was calculated for each crash scenario and presented in Table 2 and 3. When analysing the results, it was observed that releasing the subframe at initial velocities above 45 km/h should not be recommended, since some intrusion measurements increased above the level recorded in the model without subframe release. Results for offset distances greater than 1000 mm are not presented for the VPI model, since the correlation coefficient R between ΔV_x and the VPI model was less than 0.97, as seen in Table 1. For the OLC model, results for offset distances greater than 1100 mm are not presented for the same reason. By comparing Table 2 and 3, it is apparent that the reduction in EVCR is predicted to be greater based on the VPI model than on the OLC model for all considered

Table 1. Pearson product-moment correlation coefficient R between longitudinal velocity change ΔV_x and crash pulse severity models VPI and OLC categorised by lateral offset.

Correlation coefficient R		Offset [mm]												
		0	100	200	300	400	500	600	700	800	900	1000	1100	1200
Model	VPI	0.993	0.991	0.994	0.995	0.997	0.999	0.998	0.997	0.983	0.982	0.976	0.964	0.864
	OLC	0.996	0.995	0.995	0.995	0.998	0.997	0.995	0.992	0.994	0.993	0.995	0.981	0.930

Table 2. EVCR, km/h, per offset and initial velocity based on VPI model.

EVCR [km/h] VPI		Offset [mm]												
		0	100	200	300	400	500	600	700	800	900	1,000	1,100	1,200
Velocity [km/h]	30	12.9	9.0	8.9	9.8	11.0	11.3	13.0	13.4	13.1	13.6	9.5	N/A	N/A
	35	13.5	14.0	12.5	12.4	12.3	12.9	13.5	16.7	18.7	14.7	7.4	N/A	N/A
	40	14.6	13.5	13.1	13.0	12.4	11.3	12.2	13.4	12.8	14.8	8.7	N/A	N/A
	45	13.1	11.5	11.5	12.9	10.2	12.1	14.5	12.8	11.7	9.9	11.7	N/A	N/A
	50	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A
	55	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A

Legend:
≤6 km/h
6-12 km/h
≥12 km/h

Table 3. EVCR, km/h, per offset and initial velocity based on OLC model.

EVCR [km/h] OLC		Offset [mm]													
Velocity [km/h]		0	100	200	300	400	500	600	700	800	900	1,000	1,100	1,200	Legend:
	30	7.8	6.1	7.1	7.7	9.0	9.2	10.8	10.5	10.0	10.2	5.0	4.4	N/A	≤6 km/h
	35	7.1	8.5	8.1	7.9	8.5	9.0	9.9	11.9	13.2	10.4	3.8	5.9	N/A	6-12 km/h
	40	8.3	8.4	9.0	9.2	8.5	8.1	8.5	9.0	8.9	9.8	3.8	5.0	N/A	≥12 km/h
	45	7.1	6.7	6.8	7.5	6.4	7.6	8.7	8.5	7.7	6.3	5.9	4.1	N/A	
	50	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	
	55	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	

Table 4. Relative EVCR based on the VPI model. EVCR compared to velocity change ΔV_x without subframe release.

EVCR rel VPI		Offset [mm]													
Velocity [km/h]		0	100	200	300	400	500	600	700	800	900	1,000	1,100	1,200	Legend:
	30	36%	25%	24%	26%	29%	30%	35%	36%	37%	38%	27%	N/A	N/A	≤15%
	35	33%	34%	30%	29%	28%	30%	31%	39%	44%	35%	18%	N/A	N/A	15-30%
	40	31%	28%	27%	27%	26%	23%	25%	28%	27%	31%	18%	N/A	N/A	≥30%
	45	26%	22%	22%	24%	19%	22%	26%	23%	21%	18%	21%	N/A	N/A	
	50	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	
	55	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	

crash scenarios. Both models predict the greatest EVCR in 800 mm offset at 35 km/h initial velocity for each vehicle.

The absolute values for EVCR presented in Table 2 and 3 do not reveal how great this reduction was in relation to the velocity change that the vehicle underwent without subframe release. The relative EVCR was calculated and is presented in Table 4 and 5 by dividing the EVCR by the velocity change, without subframe release for each crash scenario. Furthermore, the relative EVCR was predicted to be greater when based on the VPI model than the OLC model. The greatest reduction was seen at 800 mm offset at 35 km/h initial velocity for each vehicle. In this crash scenario, the reduction in crash severity based on a released subframe is predicted to be equivalent to 44% (VPI model) or 31% (OLC model) reduction of ΔV_x, if subframe release is not available. Taken as an average of the considered crash scenarios in Table 4 and 5, the results based on the VPI model suggest 28% relative EVCR compared to 18% for the OLC model.

4. Discussion

The reduced mass of the simplified model and the removal of the crash test dummies constitute modifications to the inertial properties of the simulation model. After tuning the lumped mass for an unchanged total displacement in FWRB crash, a small influence on the crash pulse shape was observed and the simplified model was considered as an adequate substitute for the original model. The absolute values for intrusion could be expected to be lower for the simplified model since the stiffness of the front structure was not modified, while the total mass of the vehicle was modified. However, since increased intrusion was defined relative to the intrusion levels observed without subframe detachment, this simplification was considered acceptable.

If the subframe cannot be detached by both vehicles as assumed in this study, the effect on the stopping distance will be reduced for both vehicles. Furthermore, if the mass of the vehicles involved in a C2C crash is considerably different, the effect of detaching the subframe will be affected.

Table 5. Relative EVCR based on the OLC model. EVCR compared to velocity change ΔV_x without subframe release.

EVCR rel OLC		Offset [mm]													
Velocity [km/h]		0	100	200	300	400	500	600	700	800	900	1,000	1,100	1,200	Legend:
	30	22%	17%	19%	21%	24%	24%	29%	28%	28%	29%	14%	12%	N/A	≤15%
	35	17%	20%	19%	18%	20%	21%	23%	28%	31%	25%	9%	13%	N/A	15-30%
	40	18%	18%	19%	19%	18%	17%	17%	19%	18%	20%	8%	10%	N/A	≥30%
	45	14%	13%	13%	14%	12%	14%	16%	15%	14%	12%	11%	7%	N/A	
	50	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	
	55	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	

The decision on releasing the subframe is dependent on the mass ratio of the two vehicles involved. If pre-crash sensing systems or vehicle-to-vehicle communication can be further developed and implemented, this information can be utilised in order to support the actual timing of release.

In Section 3, the response was studied in one of the vehicles only since the models were identical. Because the observed response was considered to be very similar in both vehicles, a decision to omit results for the other vehicle was made. When results were presented, increased intrusion was deemed unacceptable. If this requirement is not respected, a released subframe could be used in even higher velocities. However, it is not recommended to obtain an improved crash pulse shape by deteriorating intrusion.

Some notes should be made on the similarities and differences between the VPI and OLC models. The OLC assumes a perfectly adaptive restraint system that is not yet feasible in reality since the total velocity change during a crash is unknown in advance of the actual crash and, furthermore the velocity change depends on the crash opponent. The VPI, on the other hand, assumes a linearly increasing deceleration without any limitation on relative displacement, which may also be regarded as unrealistic. A typical restraint system may therefore be considered as a combination of the two models and the real-world effect of an adaptive subframe is most likely within the range given by the VPI and OLC models.

The EVCR is consistently greater based on the VPI model compared to the OLC model. One reason may be that the OLC model is perfectly adaptive, i.e. always utilising the 300 mm of assumed available interior ride-down distance. This means that the response without subframe release is already associated with a maximised ride-down distance. For the VPI model, however, the maximum occupant deceleration may be associated with a peak vehicle deceleration late in the crash pulse that can be attenuated or removed when the subframe is released.

This study proposes using the front subframe for structural adaptivity. This approach was demonstrated to have a considerable effect on the full-vehicle crash pulse shape as suggested in a previous study by Wågström et al. [14]. An adaptive front subframe has further benefits over the proposed solution from Motozawa et al. [5], since additional packaging space would not be required. Compared to the proposal by Pipkorn et al. [8], the solution in this study does not require any modifications of the frontal longitudinal members and may be implementable in an already existing vehicle structure.

5. Conclusions

The effect of releasing the front subframe in frontal collisions in a controlled manner was explored by comparing two generic occupant restraint models for estimation of crash pulse severity. It is suggested that the reduction in crash severity that can be achieved by releasing the subframe in frontal collisions is equivalent to reducing the velocity change by up to 44%. Releasing the front subframe is not recommended at initial velocities above 45 km/h since this could potentially increase passenger compartment intrusion.

Acknowledgements

This work was carried out in association with SAFER – Vehicle and Traffic Safety Centre at Chalmers, Sweden.

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