A SITUATION BASED METHOD TO ADAPT THE VEHICLE RESTRAINT SYSTEM IN FRONTAL CRASHES TO THE ACCIDENT SCENARIO

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ABSTRACT

The integration of active and passive safety systems is considered as a significant contribution towards further improvement of traffic safety. The present article describes an approach to integrate these systems. This is done by development of a novel control algorithm where force levels and activation times of an assumed adaptive restraint system are predefined based on the oncoming collision. Reference values for these force levels are generated in order to minimise the acceleration of the occupants.
The method takes into account the actual crash severity by a forecast of the acceleration behaviour of the passenger cell, based on prediction of collision speed, mass and stiffness of opponent and own vehicle. The prediction of mass and collision speed is not part of the present paper and currently under investigation. A forecast of the acceleration pulse is calculated by a simplified multi body model of the impact. The vehicle deformations are considered by non-linear springs with hysteresis. Their characteristics are derived from 53 crash tests published by NHTSA. The occupant of the ego-vehicle is considered also by a simplified multi body model, taking into account its mass and seating position. Optimisation algorithms determine suitable force levels and trigger times of the adaptive restraint components by minimising the acceleration of the occupant while avoiding bottoming-out of the restraint system. Currently, only straight frontal collisions with full overlap are considered. The algorithm is developed in order to provide a real-time application and is verified by detailed off-line crash simulations.
With numerical simulations several configurations with different collision severities and occupant masses were investigated. In almost every configuration significant reductions up to 90 % of the occupant acceleration were observed. The present study forms the basis of future work which includes a real-time application in a vehicle.

INTRODUCTION

Background

Active safety systems and advanced driver assistance systems (ADAS) such as electronic stability control (ESC), emergency brake assist (EBA) and lane keeping system will contribute to avoid and mitigate collisions in future [1, 2, 3].
Passive safety restraint systems are currently activated by electronic control units (ECU) that for example evaluate accelerations, roll rate and door pressure during an accident. The activation is triggered after first contact of the vehicles, in frontal crashes typically after 10 to 30 ms, which wastes ride-down distances and requires fast deployment of airbags. Yet, fast airbag deployment is aggressive to occupants, especially when they are out-of-position, for example after emergency braking or crash avoidance manoeuvres [4].
Moreover, the adaption of passive safety systems to the actual accident is mainly limited to low and high crash severity. Adaption to occupant mass and seating position provides a significant potential for reduction of injuries [5].
Especially, the integration of active and passive safety systems and the adaption of their functionality to the actual collision is considered as a significant step towards improved traffic safety [6, 7].

Objective

The present paper is based on previous work [8, 9] and describes an approach to integrate active and passive safety systems. An algorithm is developed which pre-sets force levels and trigger times of an adaptive frontal restraint system according to parameters of an oncoming collision. Reference values for these force levels are generated in order to minimise the acceleration of the occupant. Important sources for the development of the algorithm are found in [10, 11, 12].

The following section summarises the main approach of the novel approach, further details are published in [8, 9].

METHODOLOGY

The main idea of the algorithm is the prediction of the passenger cell acceleration pulse of the own vehicle (ego-vehicle). This is based on a forecast of collision speed, mass and stiffness of the colliding vehicles, which are the main input parameter of the algorithm.

The method consists of three separate modules, the vehicle model, the collision model and the occupant model. MATLAB/Simulink® software was chosen for the realisation of the complete model, due to its capability for generation of real-time codes that can be run on automotive ECUs. The structure of the Simulink model is depicted in Figure 1, which illustrates the interfaces between the modules.

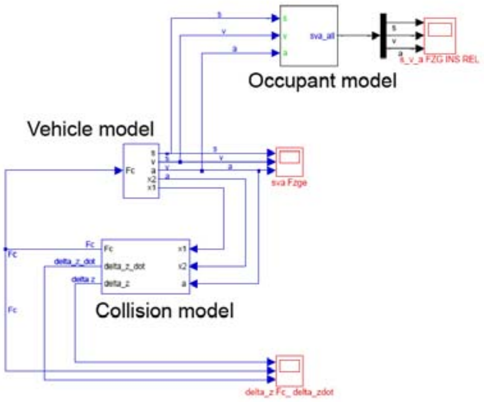

Figure 1: Structure of the Simulink model.

Prior to the impact, the vehicle model is exchanging data (state vector of ego-vehicle and collision opponent) with the collision model which predicts the acceleration of the passenger compartment. The collision model is exchanging data (predicted acceleration pulse) with the occupant model where an adaptive restraint system is minimising the occupant acceleration. Outputs of the model are force levels and activation times for an adaptive restraint system.

Vehicle Model

The vehicle model serves to predict the impact energy and delivers the input parameters (mass and collision speed) for the collision model. The module is based on state-of-the-art longitudinal vehicle dynamics modelling and includes driving resistances (air, climbing and rolling resistance), for the basic theory refer to [13].

Additionally, an Antilock Braking System (ABS) and an EBA of the ego-vehicle are integrated to include their influences on the mitigation of oncoming collisions. Therefore, rotational degrees of freedom for the wheels are modelled.

The control algorithms of the driver assistance systems were taken from literature [14, 15]. Nevertheless, serial ABS or EBA applications can be integrated any time during full vehicle system integration.

One component of the vehicle model is the collision prediction block which uses the input of an environment recognition system (such as radar, video or laser scanner) to predict the state vector (position and velocity) and mass of the collision opponent prior to collision. This block is not yet implemented and part of ongoing work. The Simulink structure of the current vehicle model is depicted in Figure 2. A detailed description of the model can be found in [8].

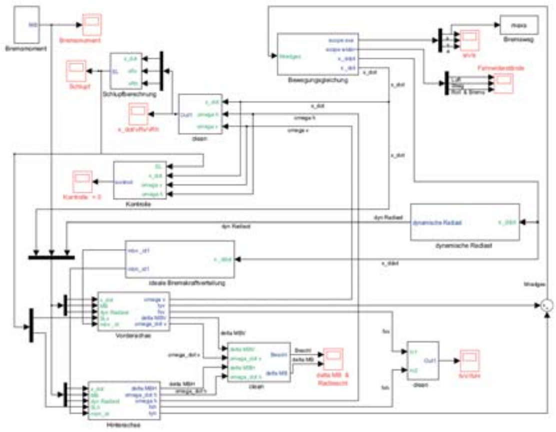

Figure 2: Simulink structure of vehicle model.

Collision Model

The collision model serves to predict the acceleration of the passenger compartment of the ego-vehicle. It consists of two rigid bodes with a single degree of freedom for each in longitudinal direction (x_{opp} and x_{ego}), see Figure 3. They represent the opponent vehicle (mass m_{opp}) and the ego vehicle (mass m_{ego}). The rigid bodies are linked together by force elements (nonlinear springs with hysteresis, F_C).

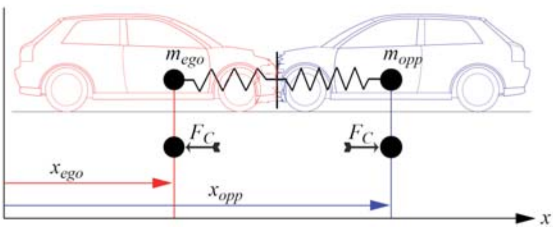

Figure 3: Scheme of collision model.

By numerical integration of the equations of motion of this model, Eq. (1), the acceleration of the passenger compartment of the ego-vehicle is calculated.

$$
\begin{aligned}
m_{opp} \cdot \ddot{x}_{opp} + F_C &= 0 \\
m_{ego} \cdot \ddot{x}_{ego} + F_C &= 0 \\
F_C &= c_{def} \cdot (x_{opp} - x_{ego}) \\
c_{def} &= f(x_{ego}, x_{opp})
\end{aligned} \qquad (1).
$$

The spring characteristics of the force elements, which describe the deformation behaviour of the vehicles in a full overlapped frontal crash, are derived from crash tests published by NHTSA [10]. A total number of 53 vehicles were analysed, for an example see Figure 4.

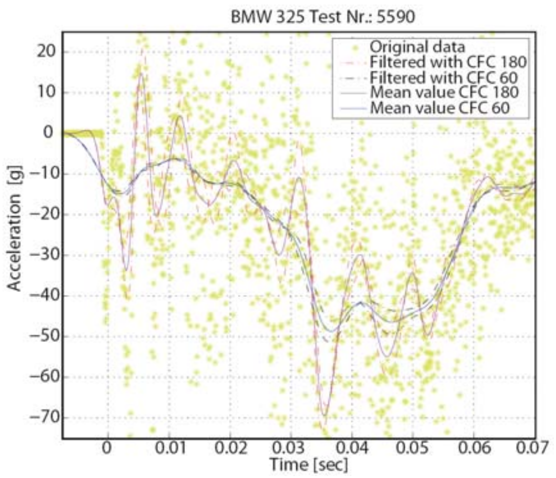

Figure 4: Example of passenger compartment acceleration [8].

The filtered data forms the basis for further analysis. In case of a frontal collision with full overlap, a method [8] was developed to combine the individual stiffness's of the opponents to one single spring F_c with discrete non-linear force-deflection characteristics c_{def}, Figure 5 and Eq. (1).
The solid (red) and dashed (blue) line correspond to two different vehicles, the dot and dashed line (green) represents the combination of them.

Occupant Model

The main approach for the occupant model is the following assumption: The injury risk and severity in a vehicle accident are reduced when maximum and mean accelerations are reduced. This is especially true in low to medium crash severity where the integrity of the passenger compartment prevents intrusion-induced injuries. Modern cars are designed to withstand collision severities in frontal crash of up to 56 kph against a rigid barrier or 64 kph against a deformable honeycomb barrier with minimal intrusion into the footwell.
In the present occupant model, the occupant of the ego-vehicle is considered by a simple rigid body model, comparable to the collision model. It takes into account mass and seating position of the occupant (see Figure 6).

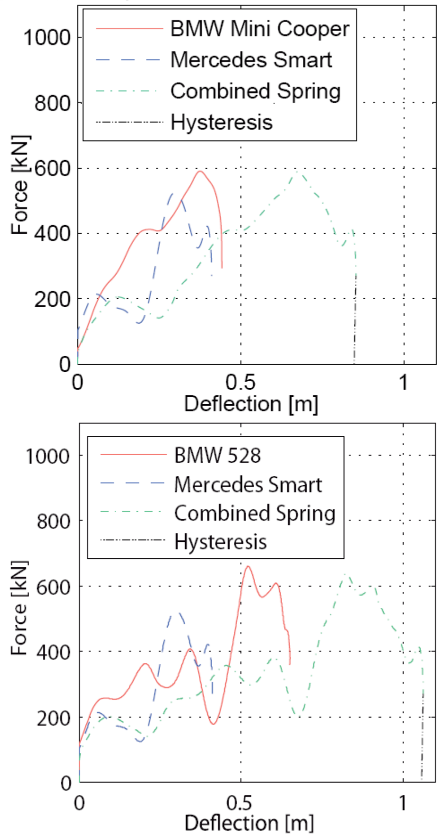

Figure 5: Two examples of the combined deformation spring [8].

In a later vehicle application these parameters have to be provided by occupant sensing systems. The equation of motion for the occupant model is:

$$
m_{occ} \cdot \ddot{x}_{occ} = F_{Air,Steer} + F_{Belt} + F_{Seat} \qquad (2).
$$

m_{occ} denotes the mass of the occupant, x_{occ} the position of the occupant, $F_{Air,Steer}$ (airbag in combination with steering column), F_{Belt} (seatbelt), and F_{Seat} (seat) the forces of the restraint system acting on the occupant, see Figure 6.
Optimisation algorithms determine suitable force levels and trigger times of the adaptive restraint system, based on the criterion of minimising the maximum and mean acceleration of the occupant (represented by the single rigid body m_{occ}). A secondary condition is the avoidance of bottoming-out of the restraint sytem, by limiting the forward motion of the occupant rigid body, $x_{occ}>0$. Within this study, genetic as well as gradient based optimisation algorithms were investigated [8].
The optimisation of the force levels and activation times of the restraint system will be the key issue for a real-time vehicle application. At the moment, putting the results of a large amount of optimisation runs into a database (characteristic diagram) is the most promising solution for real-time performance.

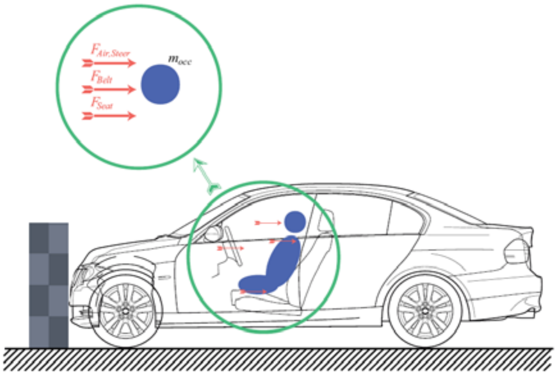

Figure 6: Simplified rigid body model of the occupant.
$F_{Air,Steer}$ describes the forces by the frontal airbag in combination with the steering column, F_{Belt} is the resulting force in lateral direction of shoulder and lap belt and F_{Seat} stands for the frictional force of the seat, [8].

The next section explains the verification of the model with numerical simulation since experimental verification of the model would require several cost-intensive full-scale crash tests.

VERIFICATION

The verification of the presented model was carried out by a detailed off-line simulation. This off-line simulation was based on a Finite-Element-Method (FEM) model of a full frontal car to car crash. A FEM model of the FORD Taurus [16] was used for simulation of the crush behaviour of ego- and opponent vehicle, Figure 7.

Figure 7: FEM model of investigated vehicle [16].
LS-Dyna FEM model of the 2001 FORD Taurus, occupant and restraint system were added.

Since this model did neither include an occupant nor a restraint system, a Hybrid-III 50 percentile male crash test dummy, a seat with 3 point belt system and a frontal driver airbag was added [17], see Figure 8.
The first step for verification was to simulate a 56 kph frontal crash against a rigid barrier with full overlap (US-NCAP crash test), see Figure 9. The validation of the model was published in [18] and is accurate within the requirements of the present study.
The acceleration, measured in a similar location as in the NHTSA data [10] was derived from the FEM simulation. Using this simulation, the acceleration and crush behaviour of the vehicle was derived in a similar manner than in the real tests published by NHTSA, see chapter methodology. This simulation forms the basis for further verifications.

Figure 8: Occupant and restraint system in the ego-vehicle [17].

Nevertheless, the car body of the FORD Taurus FEM model was reinforced in order to demonstrate a stiffer car structure which withstands EURO-NCAP frontal crash requirements with minimal intrusion to the passenger compartment in the footwell area. The reason for that was the approach of the algorithm which minimises acceleration induced injuries and not intrusion induced injuries. Also, steering wheel displacements, which decreases the available ride-down is not implemented in the collision model.

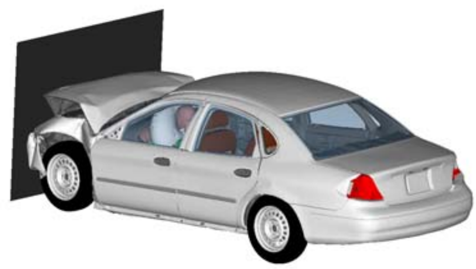

Figure 9: USNCAP frontal crash.
56 kph impact speed, full overlap, rigid barrier, 50th percentile H-III dummy.

As depicted in Figure 10, the accuracy of the predicted passenger compartment acceleration (red solid line) is remarkably high for the first 40 ms, the 70 g peak at 45 ms is filtered out through the prediction algorithm. Nevertheless the integral of this curve (velocity during the collision) fit accurately, Figure 11. For the desired application, namely pre-setting an adaptive restraint system, the accuracy is sufficient.

The next step in the verification was to perform car to car crash simulations.

The FORD Taurus was impacted at different impact speed against another FORD Taurus FEM model, (for test set-up, see Figure 12 and Table 1).

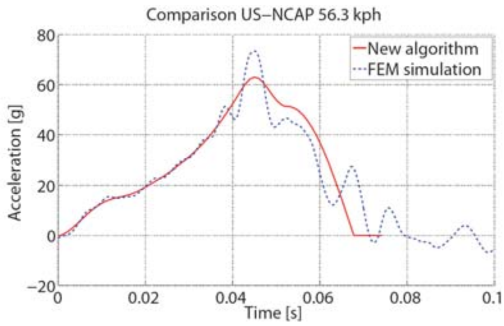

Figure 10: Comparison of passenger compartment acceleration (US-NCAP).

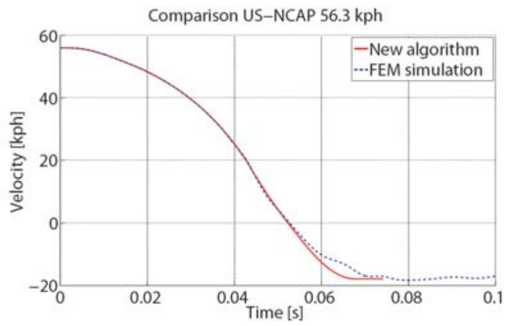

Figure 11: Comparison of passenger compartment velocity (US-NCAP).

Figure 12: car to car frontal collision (FORD Taurus vs. FORD Taurus).

The occupant and the restraint model was not verified in detail, which is part of future work. Results regarding the dummy responses only can be treated to tell qualitative trends.

Impact speeds of the ego-vehicle and the opponent car were chosen based on a hypothetical impact scenario of a frontal collision on a wet road with limited vision due to fog. The kinematics of this hypothetical collision was simulated using PC-Crash® accident reconstruction software, Figure 13. The results were forwarded as an input to the vehicle model, which calculated the impact conditions listed in Table 1. Different settings for driver assistance systems were taking into account, [17].

Figure 13: Hypothetical crash scenario simulated with PC-Crash®.
The blue vehicle on the right side is overtaking. Due to fog, the approaching red vehicle on the left side is recognized late.

Table 1.
Car to car crash simulation matrix

	Ego-vehicle	Opponent	Driver assistance
Make and model	FORD Taurus	FORD Taurus	
Impact speed Simulation 1 [kph]	52	52	None
Impact speed Simulation 2 [kph]	48	48	ABS
Impact speed Simulation 3 [kph]	31	31	ABS and EBA
Overlap [%]	100	100	
Impact angle [deg]	0	0	

Exemplarily the result from simulation 3 is depicted in Figure 14. The acceleration of the passenger compartment predicted by the collision model (red solid line) is sufficiently accurate for adaption of adaptive restraints. Single peaks as calculated by the FEM model (blue dashed line) are not predicted, since during calculation of the combined "collision" spring the input data is filtered (passenger cell acceleration).

Figure 15 illustrates the result of the velocity change in the same load case. At time 53 ms, the velocity passes the zero line. The predicted rebound velocity is higher than in the detailed crash simulation. This behaviour can be modified by the hysteresis model of the algorithm. A fine-tuning of the hysteresis parameter was not yet performed.

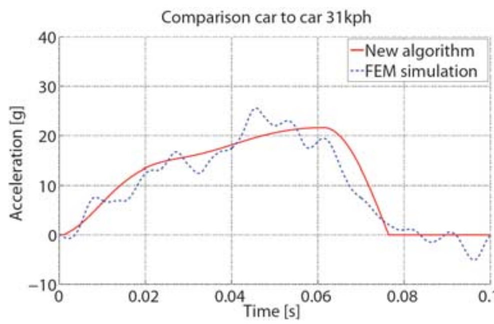

Figure 14: Comparison of FEM simulation and algorithm of collision model (acceleration).

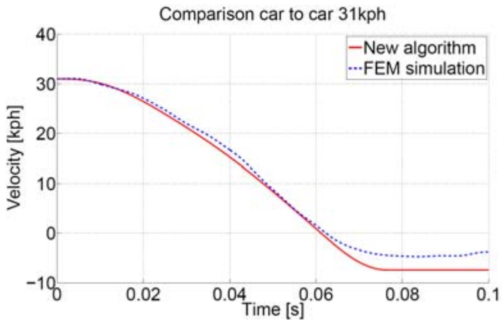

Figure 15: Comparison of FEM simulation and algorithm of collision model (velocity).

RESULTS AND DISCUSSION

The potential of the presented model was investigated in a parameter study. Different configurations with different collision severities and occupant masses were investigated. The parameters are listed in Table 2.

Table 2.
Input for parameter study

parameter	from	to
collision speed [kph]	20	54
mass opponent vehicle [kg]	800	3000
occupant weight [kg]	30	125

The collision severity ranges from low speed (20 kph) to high speed 56 kph for each vehicle. At lower speeds an adaptive restraint system will not make sense, due to low injury risk and no-fire requirements for active restraint systems. Higher speeds are not feasible for the algorithm because there is no data available for high speed crush behaviour. Additionally, at high speeds the passenger compartment will start to collapse, which would cause intrusion-induced injuries, which is not covered by the present approach.
For the mass of the opponent vehicle a range from A-segment vehicles (800 kg) up to luxury class cars (3000 kg) were investigated, which covers the majority of the passenger car fleet.
For the passenger weight, a range from 30 kg up to 125 kg was chosen in order to cover most occupants on the driver and passenger seating position.

A standard restraint system optimised for FMVSS 208 requirements forms the reference for the parameter study. As a working hypothesis, the force levels of the reference restraint system were determined with the optimisation algorithm as used for the adaptive restraint system (see section Methodology). The following load cases were used to define the reference restraint system

- "48 kph frontal collision with an unbelted 75 kg occupant"
- "56 kph frontal collision with a belted 75 kg occupant"

The fire time of the reference restraint system was set to 10 ms.

The results for the parameter study are depicted in Figure 16. In almost every configuration significant improvements up to 90 % were observed. For collisions close to the FMVSS 208 standard requirements (e.g. 54 kph closing speed and 75 kg occupant mass) the improvements are small because the non adaptive reference restraint system is already optimised for that configuration. The main improvements occur at lower severity and especially occupant masses outside of the 50th percentile (75 kg), which demonstrates the effectiveness of the integrated safety approach in real traffic conditions.

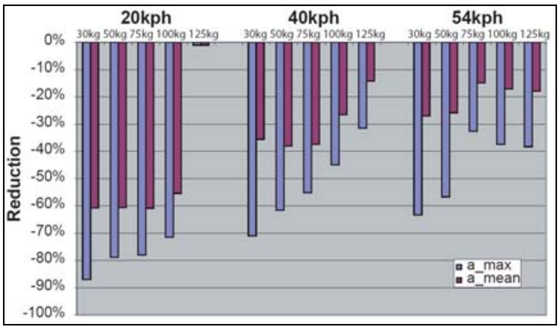

Figure 16: Reduction of the occupant maximum (a_max) and mean acceleration (a_mean) with respect to different collision speeds and occupant masses.

Since the present study assumes that make and model of the collision opponent are known, the influences of the accuracy of mass and stiffness of the opponent vehicle were analysed in a further parameter study:
Even when estimating the mass of the collision opponent based on data from a video recognition system, there is a lack of knowledge about the actual payload.
According to statistics of vehicle registrations in Austria [12], the NHTSA database [10] was searched for most likely collision opponents. 53 vehicles were investigated for that purpose. But through questionable acceleration data it was not possible to retrieve valid data for all 53 vehicles. After all, 39 vehicles with plausible acceleration data were analysed and grouped into six mass classes, as Table 3 shows.

Table 3.
Investigated Cars and Defined Mass Classes

Class	Make	Model	Mass
1	Mercedes	Smart	963 kg
1	VW	Polo	1100 kg
2	Toyota	Yaris	1245 kg
2	Kia	Rio	1352 kg
2	Mini	Cooper	1371 kg
2	Dodge	Neon	1379 kg
2	Toyota	Corolla	1379 kg
2	Ford	Focus	1394 kg
2	Honda	Civic	1394 kg
2	Ford	Focus	1398 kg
3	Toyota	Prius	1515 kg
3	Subaru	Impreza	1585 kg
3	BMW	Z4 Roadster	1630 kg
3	Honda	Accord	1673 kg
3	Saab	9-3	1705 kg
3	Subaru	Forester	1708 kg
3	VW	Jetta	1719 kg
3	Nissan	350Z	1729 kg
3	Volvo	S60	1732 kg
4	VW	Passat	1765 kg
4	Ford	Taurus	1785 kg
4	BMW	325l	1806 kg
4	Audi	A4	1820 kg
4	Volvo	S80	1820 kg
4	Saturn	Aura	1828 kg
4	Nissan	350Z	1855 kg
4	Mercedes	C300	1864 kg
4	Chrysler	Sebring	1915 kg
4	BMW	528l	1924 kg
5	Dodge	Journey	2136 kg
5	Volvo	XC90	2389 kg
5	Hummer	H3	2404 kg
5	Mercedes	ML350	2431 kg
5	BMW	X5	2458 kg
6	Audi	Q7	2582 kg
6	VW	Touareg	2600 kg
6	Toyota	Sequoia	2816 kg
6	Toyota	Tundra	2884 kg
6	Ford	F250 Pickup	3054 kg

Table 4 lists the results of the investigation of the mass influence for two different vehicles (FORD Taurus and MERCEDES C300). An occupant with 75 kg is taken into account. The closing speed is 106 kph. The parameters $a_{max,occ}$ and $a_{mean,occ}$ present the resulting loading to the occupant. Respectively, $s_{disp,occ}$ is the relative displacement, $a_{max,occ}$ the maximum acceleration and $a_{mean,occ}$ the mean acceleration of the occupant. The average acceleration of an occupant sitting in a MERCEDES C class is almost doubled when it is impacted from an 3000 kg vehicle compared to a 800 kg vehicle. When increasing the mass of the vehicle in steps of 200 kg, the average acceleration increases by approximately 1.5 g. The typical payload of a passenger car is around 300 kg [2], so the accuracy of the results are in the range of about 2 g, which has to be taken into account by the algorithm.

There where no results in the simulation of the first investigated vehicle (FORD Taurus) when colliding against a vehicle with a mass higher than 2600kg (marked with "-" in Table 2). The reason for that is that the maximum force levels of the restraint system are limited. At high impact energy levels, the occupant strikes through the airbag and contacts the dashboard.

Table 4.
Influence of Collision Opponent Mass on Occupant Loading

	Ford Taurus			Mercedes C300		
m_2	$s_{disp,occ}$ [mm]	$a_{max,occ}$ [g]	$a_{mean,occ}$ [g]	$s_{disp,occ}$ [mm]	$a_{max,occ}$ [g]	$a_{mean,occ}$ [g]
800 kg	24.1	23.17	14.90	30.3	29.43	16.70
1000 kg	40.7	23.54	16.00	52.1	30.74	18.33
1200 kg	63.2	29.13	17.22	70.0	38.02	20.25
1400 kg	74.1	32.93	18.46	87.4	45.14	21.67
1600 kg	87.5	32.93	19.37	105.5	52.49	23.61
1800 kg	87.5	32.93	20.30	116.5	57.00	24.67
2000 kg	130.3	32.93	21.12	125.8	57.40	25.93
2200 kg	158.0	32.93	21.62	133.8	57.40	27.30
2400 kg	187.8	32.93	21.98	141.4	57.40	28.02
2600 kg	221.8	32.93	22.28	149.4	57.40	28.86
2800 kg	-	-	-	161.4	57.40	29.55
3000 kg	-	-	-	173.3	57.40	30.22

For investigation of the influence of the crush zone stiffness the following approach has been chosen: All studied vehicles were classified according to their mass (Table 3).

Next the stiffness springs of these vehicles were compared and combined to an average force-deflection curve (thick line in Figure 17).

It can be seen, that according to the mass classes, the deformation characteristics of different vehicles are similar. The reason for that is that vehicles structures are designed to fulfil requirements of standard laboratory crash tests. There, only the vehicle mass has an influence on the energy that has to be absorbed by the crush zone. Restraint systems are designed to meet injury criteria of dummy responses in these specific tests.

To evaluate the influence of the stiffness a certain crash scenario (fully overlapped car to car frontal collision, masses of each vehicle 1785 kg, collision speed 108 kph, mass of occupant 75 kg) is calculated using the algorithm. The only investigated parameter is a variation of the stiffness of the crush zone according to the six classes described above. The average acceleration of the occupant scatters by approx 1 g (mean value 20.4 g). The results are shown in Table 5.

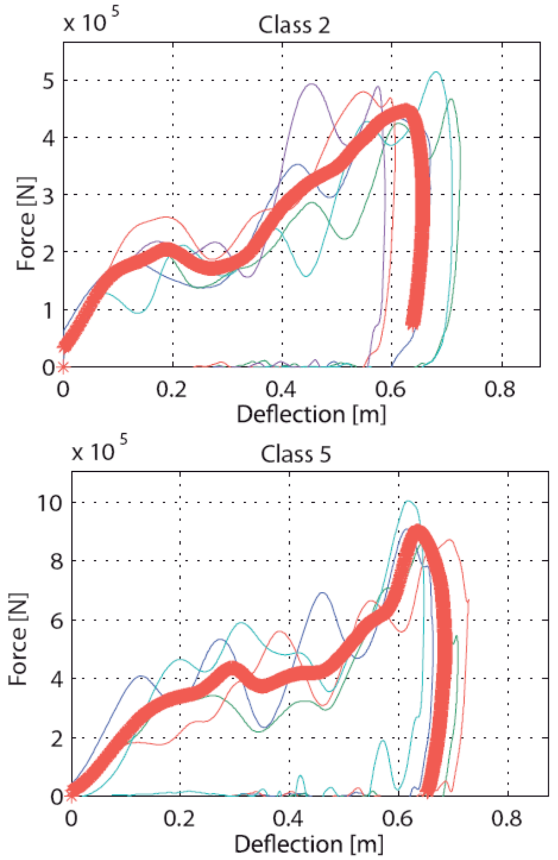

Figure 17: Examples for deformation springs with respect to different mass classes. Mass class 2 represents vehicles from 1200 to 1400 kg, mass class 5 from 2000 to 2500 kg. The thick line is the combination of the different deformation springs.

$s_{disp,veh}$ denotes the displacement of the vehicle, which is the deformation of the vehicle front in this load case. Analogue to Table 4, $a_{max,veh}$ and $a_{mean,veh}$ represent the maximum and mean acceleration of the vehicle under consideration.

Table 5.
Influence of Crush Zone Stiffness

	Occupant			Vehicle		
	$s_{disp,occ}$ [mm]	$a_{max,occ}$ [g]	$a_{mean,occ}$ [g]	$s_{disp,veh}$ [mm]	$a_{max,veh}$ [g]	$a_{mean,veh}$ [g]
Class 1	162.2	32.93	21.28	579.30	40.06	21.59
Class 2	100.4	32.93	20.23	665.50	33.64	19.88
Class 3	101.6	32.93	20.16	663.40	34.50	19.85
Class 4	108.7	32.93	20.41	653.90	39.24	20.00
Class 5	94.6	32.93	19.87	677.70	38.25	19.56
Class 6	120.4	32.93	20.52	649.90	44.74	20.22
Class 1-6	103.5	32.93	20.21	662.00	37.64	19.89

The small influence of the stiffness shows that it is sufficient to estimate roughly the mass of the opponent vehicle and to use the stiffness characteristics derived in the corresponding mass class introduced in this paper.

LIMITATIONS

As a first step, only straight frontal collisions with full overlap are considered, but basically the method can be enhanced for other impact scenarios such as rear-end collision, lateral or oblique impact. Another shortcoming is the simplification of the model in order to achieve real-time performance for a full vehicle application. Especially, it is assumed that a minimisation of occupant acceleration lowers the injury risk and severity. Detailed injury responses such as the Head Injury Criterion (HIC) cannot be assessed. The application in a vehicle, verification of the real time performance and functionality of the algorithm is part of future work.

CONCLUSIONS

An algorithm for the integration of active and passive safety systems was prepared.
The main idea is the generation of reference values for an adaptive restraint system by calculating force levels and trigger times of the different restraint components, such as belt and airbag. These were optimised with respect to maximum and mean acceleration of the occupant. The presented method consists of three separate models (vehicle model, collision model and occupant model), which are interacting. Simplified models were used in order to maintain a future real-time application.
As a first step, the model was verified with detailed FEM crash simulation models. The prediction of the passenger compartment acceleration showed sufficient accuracy for the present application.
The potential of the algorithm was demonstrated in simulations of fully overlapped frontal collision considering different input parameters:
a) Mass of colliding vehicles
b) Crush zone stiffness of colliding vehicles
c) Collision speed
d) Occupant mass of ego-vehicle
e) Seating position of occupant
These input parameters were supposed to be known, since the development of the vehicle model and the collision prediction module is still under progress.
Significant improvements up to 90 % with respect to maximum and average acceleration of the occupant could be demonstrated in different crash scenarios. The influence of mass and stiffness were investigated in order to derive requirements for the environment recognition system.

OUTLOOK

The next step will be the completion of the collision prediction module and the integration into the vehicle model. Intensive verification with driving tests and an environment recognition system will be necessary.

Further crash simulations with different FEM models and varying accident severity will be performed to verify the collision model on a broader basis and adjust some of the model parameters.
Additionally the occupant model will be verified in more detail by comparison with a validated detailed occupant simulation model and performing parameter studies.
Also applications in other load cases such as rear and side impact will be part of future work.
Next, the model will be enhanced for application a real vehicle to demonstrate a real-time application.
Finally the prediction of the opponent vehicle mass using video recognition will close the last missing link and complete the method for full vehicle integration.

ACKNOWLEDGEMENT

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