Feasibility study of an adaptive energy absorbing system for passenger vehicles

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Abstract

Many car front crashes happen with an offset between the car and the obstacle (or the second car). In this case, present design of car front crash zone is made as a compromise reconciling requirements of frontal crash tests with and without offset. Using the principle of pyrotechnic detachable connectors, a control of car deceleration is possible. System controller recognizing initial pre-crash parameters (velocity, mass, stiffness and overlap) can choose several levels of dissipated energy for each absorber.
Three methodologies with different levels of simplification have been chosen to calculate absorbed energy and levels of deceleration. A simplified analytical, numerical Lumped Mass – Spring (LMS) model and numerical FE explicit models of car front structures were created. Two crash scenarios were analyzed: full front crash and 50% offset crash for two cases: with an unmodified vehicle and when the absorbing structure was equipped in adaptive system. The objective of the system was to get similar levels of absorbed energy and crushing distance in both impact cases. A feasibility study of adaptive energy absorbing system has been performed based on comparison of crash analysis results.

Keywords: automotive structures, frontal impact, crashworthiness, adaptive systems

1. Introduction

Safety is one of the most important parameters of modern vehicles. Road accident statistics show that around 40 % of the fatalities of car occupants occur in front impacts, moreover it is a frontal impact with car offset [12]. The main parameters influencing occupant safety are vehicle's deceleration pulses and deformation of the occupant compartment [3]. The analysis of real crashes data prove that the severity of occupant injuries due to frontal collisions significantly depends on the impact velocity, the angle and the relative overlap of the crash participants, as well as on the effective stiffness and mass of the crashing vehicles [14].

To evaluate the frontal crash performance of the car and occupants protection level, several crash test modes based on statistical accidents analysis are used. For most common of them, the angle is between 30 to 0 degrees, the offset is 40%, 50% or 100% and the impact velocity varies from 48km/h to 64km/h [4]. The main difference between them is the rigid barrier in the full (100% overlap) impact tests and the deformable barrier for the partial overlap crash tests. In frontal impact tests with partial overlap a deformable barrier is used basing on the statistically proved most common type of accident which is an offset frontal impact of two passenger vehicles. Both vehicles are deformed and dissipate impact energy (during the crash). However in case of collision between a passenger car and a very stiff obstacle like no compatible truck or a rigid part of road environment, the level of safe impact velocity is decreased.

Variety of collision configurations provide plenty of design problems. To minimise the occupant injuries during frontal crashes it is necessary to design a structure capable to satisfy the safety requirements for each realistic crash situation. One of possible solutions to improve car safety can be the use of adaptive systems which are capable to adjust energy dissipation properties to accident conditions.

2. Problem description

The aim of the presented crashworthiness analysis was to design the vehicle front structure which could protect the occupant against injury and demonstrate a similar crash performance in both 100% and 50% overlap frontal collisions cases.

In Fig. 1a two examples of typical occupant compartment deceleration pulses for present passenger cars are shown.

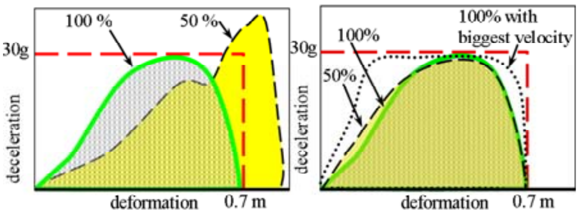


Figure 1. Examples of typical occupant compartment deceleration pulses a) full and 50% offset impact to rigid barrier; b) full and 50% offset with an adaptive system impact to rigid barrier and full impact with an the increased velocity

The area under the curves directly depends on the impact (kinetic) energy and it is almost the same in full collision and 50% ovelap cases. For the full impact case the deceleration curve satisfies the assumed allowable deceleration and deformation limits (Fig. 1a - dashed lines). In the partial overlap crash cases the impact energy dissipates only in one vehicle side and it results in increased crushing distance and decreased deceleration level in the beginning of the impact compared to the full impact characteristics (Fig. 1a). Normally, during the 50% overlap crash to a rigid barrier the frontal deformation zone is not able to absorb enough energy. Then the deformation of the occupant compartment can start and because of suddenly increased stiffness a peak in the deceleration pulse occurs. In such case both safety criteria will be exceeded.

The most important structural elements in the case of frontal collision of cars are the longitudinal members [5]. In the case of partial overlap crash by increasing the stiffness of main energy absorbers it is possible to change characteristic of the deceleration pulse and decrease the crushing distance to the level similar as in the full impact case. However during a full crash the grater stiffness will cause a higher peak in the deceleration pulse. Thus in present solutions a design compromise has to be struck.

Increase of the deceleration pulse fill can increase the safe impact velocity but it is desired only within the limits of the biomechanical limitations and the maximal depth of the deformation**.**

If the deceleration pulse fill could be increased to a safe from the biomechanical point of view level (Fig. 1 b) and the deformation depth could be decreased, then the level of the maximal possible impact velocity would be extended. It can be achieved by increasing the energy absorbing capability for 50% impact case, but additionally keeping the same level of energy dissipation for full impact case.

3. Adaptive system idea

The adaptive system should modify the performance of the frontal body structure according to impact characteristics to provide optimum occupant protection. The principle of the adaptive system is to predict and control the crash stiffness characteristics of the frontal deformation zone to satisfy allowable safety parameters. The main features of the adaptive system are shown in Fig. 2.

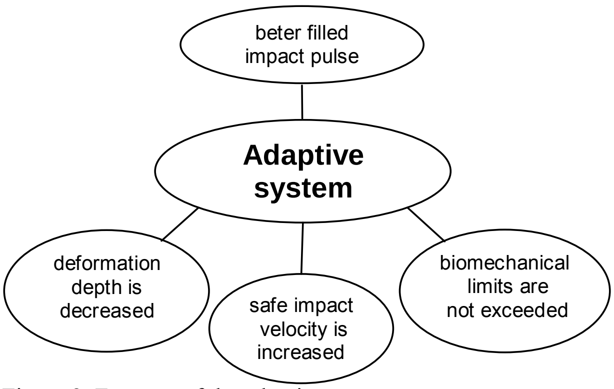


Figure 2. Features of the adaptive system

It is a very difficult task to increase the average crushing force of a typical longitudinal member profile keeping the process under control, however it seems that controlled decreasing of the crushing force should be technically feasible. One of foreseen solutions is presented in the next paragraph.

The idea of adaptive system use in its basic first generation shall be reduced just to increasing the energy absorption ability (or the average crushing force) of the longitudinal member and next to decreasing them if needed. In case of an impact with only one absorber being used (like overlap hit), the increased absorber crushing force shall provide deceleration on a level comparable with the case when two absorbers are being crushed (for ex. full impact) – Fig. 1 b.

Active or adaptive system behaviour would be utilized in situations when both absorbers are parallel crushed and occupant compartment deceleration can exceed the limits of the biomechanical criteria. Then a detonation of explosives would reduce the average crushing force to a suitable level providing a lower deceleration pulse. For future system development, a multiscenario selection of energy absorbing ability, being controlled using real time crash parameters can be foreseen as possible.

4. Crushing stiffness control using pyrotechnics devices

Researchers and engineers are trying to find a suitable solution. Cable systems [13], semi active systems in Audi construcion and kinematical systems [10] are developed.

The idea of an adaptive system described in the previous paragraph can be used with control of crushing forces by pyrotechnic detachable connectors integrated in vehicle structure. A control of crash forces level is possible, and therefore a proper deceleration pulse variant can be selected. Pyrotechnic detachable joints or stiffeners can be used to gain decrease the car deceleration level. System controller, based on signals from pre-crash sensors [7, 9], can select a proper crushing stiffness of frontal longitudinal members depending on crash conditions. However, the problems of sensing and system control were not considered in this study.

Using an adaptive system, pyrotechnic devices could be launched if needed and then the average crushing force of the dissipating structure can be changed.

Excluding at this stage all possible difficulties like sensors and controls problems, it is possible to consider two variant cases. The first of them is a front crash of two cars with similar weights and the second one (a compatibility problem) is a situation when both cars have significant difference in gross weights. Possibility to change front absorbers stiffness of the heavier car could be helpful to reduce accelerations of the lighter car.

A typical front crash zone of a passenger car consist of two dissipating mechanisms inside: a front absorber, which is crushed axially and the “s” frame, which works in deep plastic bending as a thin-walled structure. The level of the average crushing force of a longitudinal member can be modified in several different ways. One of possible control techniques can be the use of additional stiffening steel plates connected to the longitudinal member by pyrotechnically detachable connectors (Fig. 3) or a pyrotechnic profile perforating.

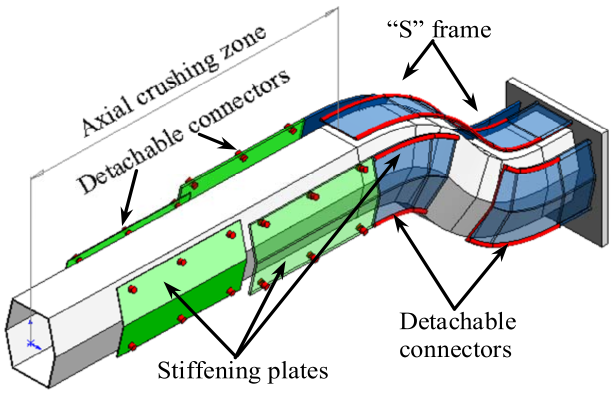


Figure 3. Model of longitudinal member with detachable connectors illustrating the idea of crushing stiffness control.

5. Feasibility study

Three methodologies with different levels of simplification have been chosen to describe the behaviour of the adaptive system. A scheme of the presented feasibility study is shown in Fig. 4.

For all analyses two variants of crash configuration were checked: full (100% overlap) front crash and 50% overlap crash to a rigid barrier. Next, the same crash situations have been analyzed for a vehicle equipped with the adaptive system.

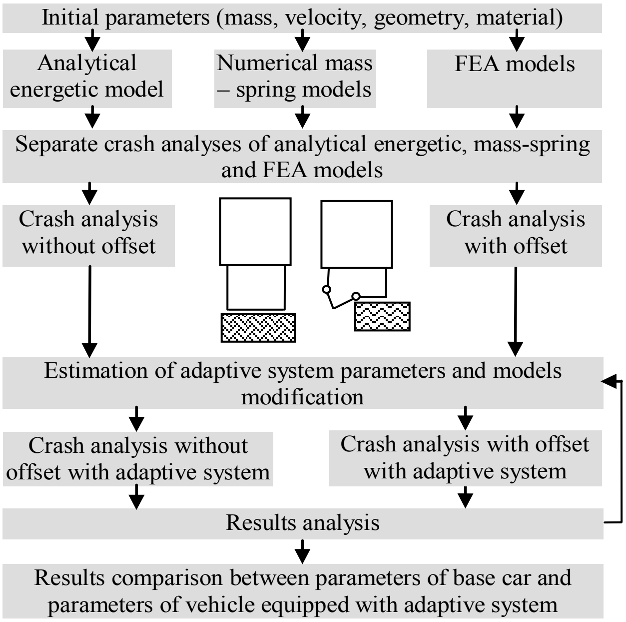


Figure 4. Scheme of the feasibility study

The objective of the system was to get a similar level of the absorbed energy, a better filled deceleration pulse and a similar crushing distance in both impact cases.

6. Simplified analytical energetic approach

Vehicle passive safety criteria are based on the allowable deceleration pulse and structural parts intrusion into the occupants' compartment. Last criteria is directly related to the amount of dissipated energy in the frontal deformation zone.

According to the basic idea to get similar crash characteristics in both 100% and 50% frontal collision cases a simplified analytical energetic model has been used. The aim was to evaluate the additional stiffness of the longitudinal members sufficient for absorption of the impact energy per assumed crash distance in case of 50% overlap crash. The scheme of the analytical approach is shown in Fig. 5.

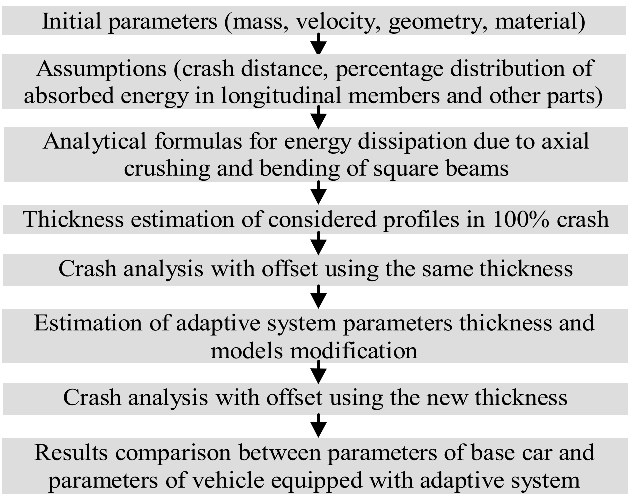


Figure 5. Scheme of the analytical approach

The longitudinal members were chosen as the main adaptive parts to control the energy absorption. Simple cross sections of the longitudinal members were assumed. In the simplified energetic analytical model the stiffness changes and the control of the dissipated energy were performed by changing the thickness of the longitudinal members.

The initial parameters of the analytical model were: the mass of vehicle m, impact velocity v_0, geometrical and material properties of the longitudinal members. Geometrical parameters: distance between the longitudinal members $w_0 = 0.9\,m$, height of the quadratic cross-section $b = 100\,mm$, allowable crushing distance $\delta = 0.7\,m$ (Fig. 6) and material properties evaluated by mean plastic stress $\sigma_0 = \dfrac{\sigma_Y + \sigma_U}{2}$.

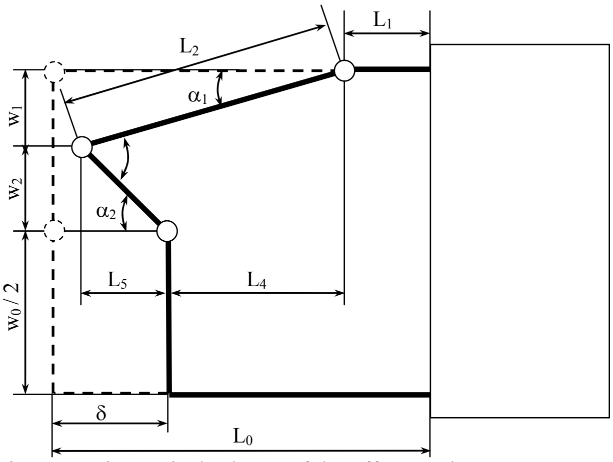


Figure 6. Kinematical scheme of the offset crash

According to the FE results we assumed that the energy dissipated during the plastic deformation is equal to the 90% of the kinetic energy

$$E_{total} = 0.9 \cdot E_k = 0.9 \cdot \frac{m \cdot v_0^2}{2} \tag{1}$$

In the analytical model for 100% overlap crash the energy dissipation was calculated in two axially crushed longitudinal members and in the flattening of the front beam.

Energy dissipation in the quadratic axially crushed beams [2]:

$$E_{Lax} = \frac{\sigma_0 \cdot t^2}{4} (52.22 \cdot \sqrt[3]{\frac{2b}{t}}) \cdot \delta \tag{2}$$

The energy absorbed in the quadratic front beam during the flattening (Fig. 7) was calculated as a sum of the dissipation energy in two 180° plastic hinges and four 90° plastic hinges in the edges of beam [2]:

$$E_{Bfl} = \frac{\sigma_{0b} \cdot t_b^2}{4} \cdot w_0 \cdot \left(2 \cdot \pi + 4 \cdot \frac{\pi}{2} \right) \tag{3}$$

here σ_{0b} and t_b are the mean plastic stress and the thickness of the front beam

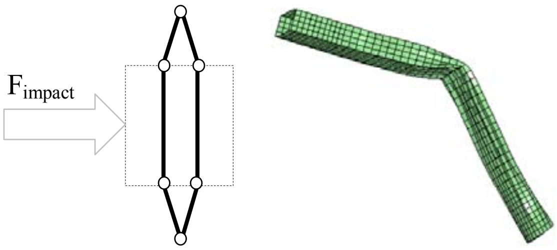


Figure 7. Flattening of the front beam a) scheme to calculate the energy dissipation during the flattening; b) example of flattening from the analysed FE model

The bending moment in the square thin walled elements for relatively small bending angle [2]:

$$M_b = 2\cdot 1.17\left(3\pi\frac{\sigma_0 t^2}{4}\sqrt[3]{\frac{b}{t}}\left(0.576+\frac{1}{2\cdot\alpha_1(\delta)}\right)\cdot b\right) \quad (4)$$

The energy dissipation in the longitudinal member and the front beam during the bending:

$$E_b = \int_0^{\alpha} M_b(\alpha)d\alpha \quad (5)$$

The energy dissipated in other front structural parts E_{other} is evaluated by coefficient $\eta = E_{other}/E_k$ which shows the percentage difference between the energy dissipated in the longitudinal members and in the other parts. Here E_k is the kinetic energy which should be absorbed by frontal deformation zone. The ratio η was taken from FE calculations.

The total absorbed energy in 100% overlap crash was evaluated as:

$$E_{total,100} = 2\cdot E_{Lax}(t) + E_{Bfl} + E_{other,100} \quad (6)$$

In the case of 50% offset crash the energy dissipation during the axially crushing of one of the longitudinal members, the flattening of the half of the front beam, the bending collapse of a longitudinal member and of a frontal beam were calculated and summed. The influence of the other front structural parts evaluated by the coefficient $\eta_{50} = 0.8\cdot\eta$. Factor 0.8 was chosen from FE calculations and shows that in 50% offset crash some parts of the frontal structure absorb less energy then in a full impact.

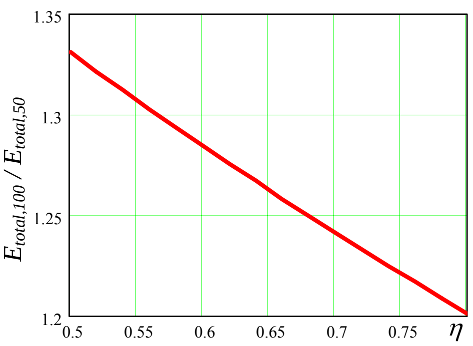


Figure 8. Influence of coefficient η on the diference between 100% crash and 50% overlap crash

The total absorbed energy in 50% offset crash:

$$E_{total,50} = E_{Lax}(t) + E_{Lb}(t) + 0.5\cdot E_{Bfl} + E_{Bb} + E_{other,50} \quad (7)$$

In 50% crash case the quantity of the energy absorbed is less compared to 100% overlap crash (Fig. 10). The difference decreases (Fig. 8) with increasing coefficient η.

Increasing the thickness of the longitudinal members we get the frontal deformation zone stiffer and capable of absorbing in 50% offset crash the same quantity of energy as in the full impact case with the same initial thickness. However for 100% overlap crash the frontal deformation zone will be too stiff and could initiate a higher deceleration pulse. To avoid it the stiffness should be decreased according to the impact velocity. The corresponding dependence of the ratio between thicknesses of stiffener and standard longitudinal member on the type of the front body structure represented by the coefficient η is shown in Fig. 9.

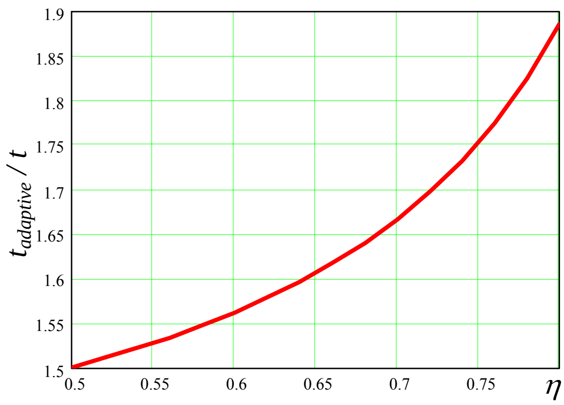


Figure 9. Influence of coefficient η on the additional thickness of the longitudinal members with the quadratic cross-section

The dependence of the energy absorbed in full impact and in 50% offset crash with initial and stiffened longitudinal members on the crushing distance is presented in Fig. 10.

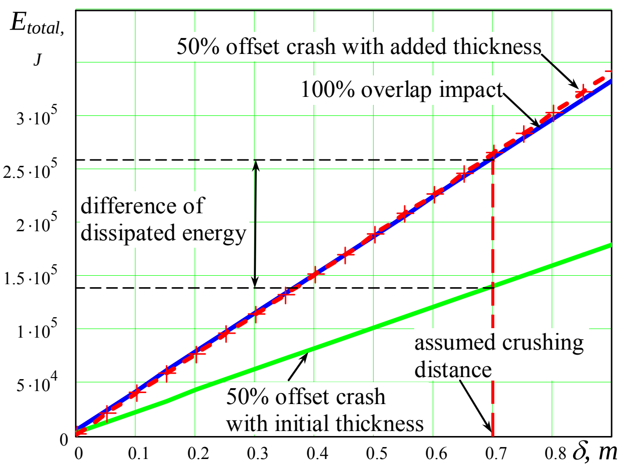


Figure 10. Comparison of calculated energy absorption in full impact, 50% offset impact with initial and stiffened longitudinal members

The distribution of energy dissipation in three analysed cases is shown in Fig. 11. In the presented graph the ratio of the energy dissipated in other parts compared to all plastically absorbed energy was taken from the FE results and was equal to $\eta = 0.7$.

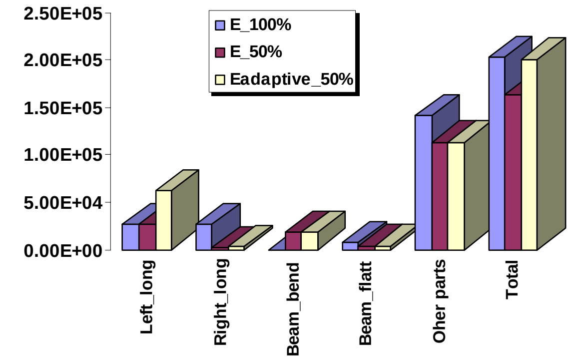


Figure 11. Distribution of dissipated energy got from simplified analytical model

The total dissipated energy in 100% overlap impact with initial thicknesses and in 50% offset crash with increased thickness was the same.

According to the average force acted on the front body of the structure during the crushing process the total dissipated energy could be evaluated:

$$E = F_{av} \cdot \delta \tag{8}$$

It was assumed that the level of the average deceleration of he occupants' compartment should be limited to $a_{av} = 30 \cdot g$ level. Then the expression of the average force is:

$$F_{av} = m \cdot a_{av} \tag{9}$$

From the last formulas we obtain the expression to calculate the minimal crushing distance for the assumed impact velocity.

$$\delta = \frac{v^2}{2 \cdot a_{av}} \tag{10}$$

This minimal crash distance is significantly lower than the real crash distance and it depends on the impact velocity assumed in the design stage.

The simplified analytical energetic model based on the amount of energy and the crushing distance taken from the full crash situation allows to estimate the additional stiffness of the longitudinal member for offset crash case and shows the differences of the energy absorbed with depending on the frontal body construction type (coefficient η). However, using the simplified analytical energetic model we cannot evaluate the various local impacts in contacts between different inner parts and the engine. The influence of these local impacts to the deceleration pulse of the compartment is very significant and could be evaluated using the simplified Lumped Mass-Spring model.

7. Simplified Lumped Mass–Spring (LMS) modelling

For dynamic response research simplified vehicle energy absorbing structure model has been developed. Low complexity level significantly decrease computing time and allow to fast parameters change, additionally gives possibility of model sensitivity study. Model scheme is shown above on Fig. 12.

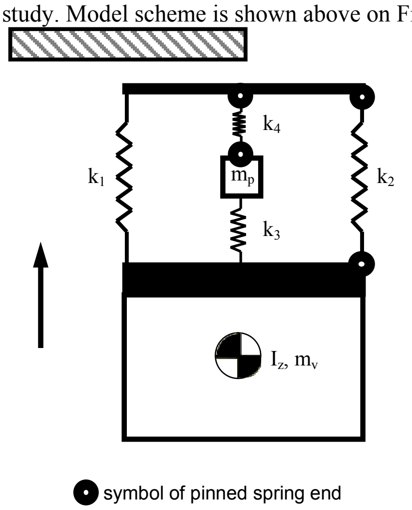


Figure 12. Simplified LMS model

Basing on assumptions and conclusions comes from energetic model research simplificated passenger car energy absorbing zone has been introduced. Numerical model has been built of four non linear springs with constitutive law describing force-displacement characteristics with use "plastic" spring behaviour. Two nodal (point) masses (one of them simulates powertrain module mass and second one simulates vehicle body mass) and one rotary mass moment of inertia (vehicle moment of inertia around vertical – Z axis) were used. Front part of structure simulating behaviour of front bumper beam was replaced by 16 plastic beam elements with cross-section defined as solid round bar. Rigid ideally plastic material model with yield stress on 500 MPa level has been used. For calculations Abaqus/Explicit solver has been used. Definition of plastic springs was implemented with "connector" type elements. A SAE 60 class filter has been used.

A force – displacement dependencies of nonlinear springs were adjusted in suitable order to simulate contacts between powertrain module to front bumper beam and to passenger compartment firewall (Fig. 13c, d). Also rigidity and fracture of powertrain block mountings and rigidity of firewall structure were added to spring characteristics. For front rails members, a constant crushing force characteristics were added (Fig. 13a, b).

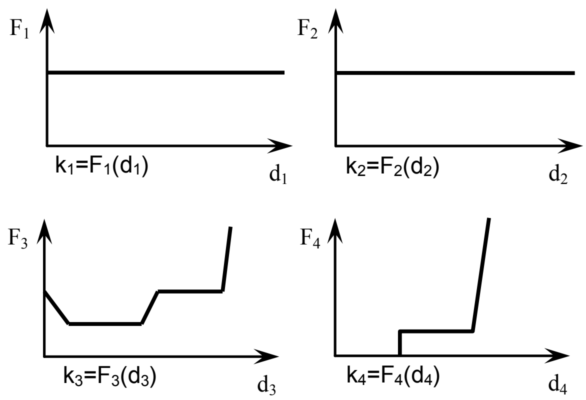


Figure 13. Nonlinear springs characteristics applied by different force-deflection curves: a) and b) springs characteristics for longitudinal members applied using average force; c) spring characteristics between powertrain module and front bumper beam; d) spring characteristics between powertrain module and firewall

Effect of described above simulation was a model which was "tuned" to simulate behaviour of real car. A 300 m/s^2 deceleration pulse was assumed as optimal from biomechanical reasons.

First from simulation series was rigid wall impact without offset with initial velocity 15 m/s. Calculation was replaced with various parameters to reach desired model response like value of maximal deformation and shape of deceleration curve (Fig. 14b).

Second analysis has been performed for model parameters obtained in full impact hit simulation, but half of obstacle width has been removed, so 50% overlap hit with 15 m/s and 18 m/s initial velocities was simulated (Fig. 14c). Comparing to full impact simulation a lower deceleration pulse fill and longer impact time was observed. Both models described above were treated as a "non active system equipped" – with typical car properties.

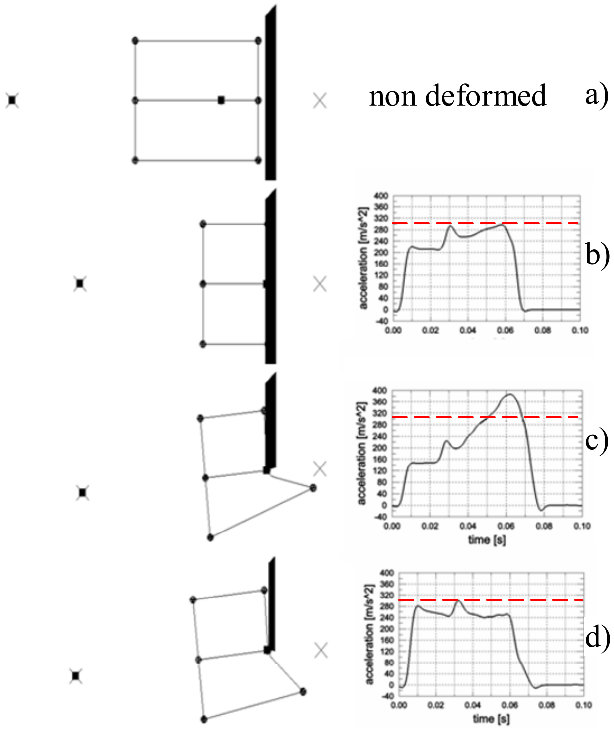


Figure 14. Deformed shapes and deceleration pulses obtained using simplified LMS model a) non deformed shape b) 100% hit normal stiffness; c) 50% hit normal stiffness; d) 50% hit left longitudinal member stiffness increased to 50% of original;

In third simulation a level of longitudinal member averaged crushing force was increased to 150% of genuine. Other model parameters were left unchanged relating to second model. Results were shown on Fig. 14d. Obtained deceleration pulse shape (Fig. 14 d) is close to desired (authors assumption) rectangular pulse.

Group plot of resultant deceleration curves archived in described simulations was shown on Fig. 15. A better deceleration pulse fills and impact time can be observed.

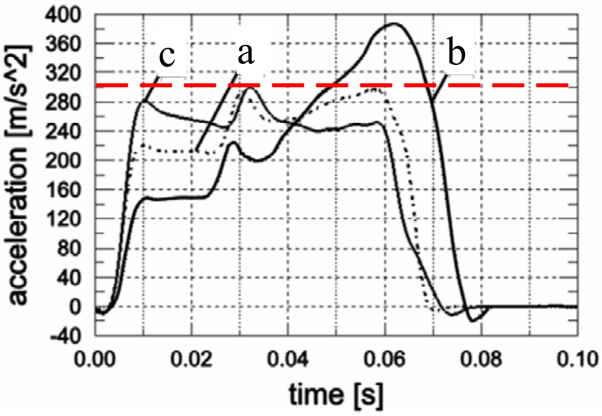


Figure 15. Group plot of resultant deceleration curves with 15 m/s impact: a) full impact; b) 50% offset impact; c) 50% offset impact with "active system"

To visualize increase of safety improvement impact simulations with increased initial velocity from 15 m/s to 18 m/s have been performed.

Deceleration curves (Fig. 15) and maximal vehicle c.g. longitudinal displacement values shows difference (Fig. 16).

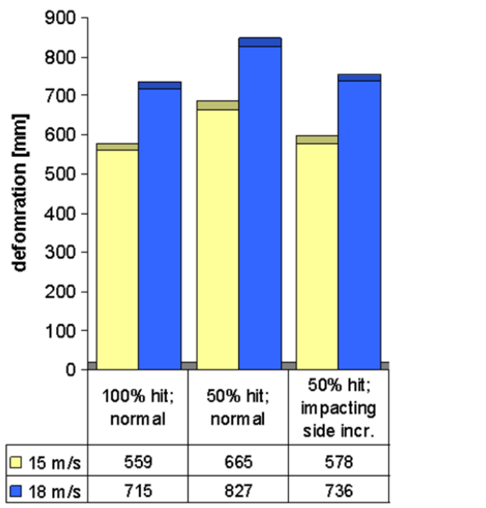


Figure 16. Comparison of deformations obtained with analyses models

8. Numerical simulation of simplified crash model

To prove the basic idea of the adaptive system the FE simulations were performed on a simplified vehicle crash model. The numerical simulations of frontal crash were done using the non-linear explicit Finite element analysis solver LS-DYNA v.960.

The vehicle's crash model was created using the shape of FE example taken from "Livermore Software Technology Corp." ftp server. The front body structure (Fig. 17) of the downloaded FE model was fundamentally changed.

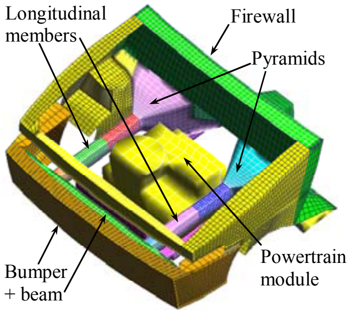


Figure 17. Main parts of front body structure in FE crash model

The modified FE model consists of about 18,000 nodes. Basically shell elements were used to model vehicle crash model. The edge length of the shell elements in the front body structure was in the range of 10 mm to 25 mm. The FE model of vehicle thin walled structures contains lower order, four nodes, flat Belytschko-Tsay shell elements, with Mindlin-Reissner plate theory formulation [6]. In the created model changes of elements thickness during the strain were not evaluated. The shell elements had one integration point in the elements‘ plane and three integration points through the thickness. In LS-DYNA FE model the material properties were described by *MAT_PLASTIC_KINEMATIC model with isotropic hardening and strain rate insensitive.

Before analysing the effect of the adaptive system it has to be made sure that the FE model is valid for the simulations. First, the performance of the vehicle body and resulting crash pulses were evaluated using USNCAP full frontal crash test mode. The deceleration pulse was measured on seat/floor panel using SAE 60 class filter. From various reports published in International Iron&Steel Institute webpage [15], the material properties and geometrical parameters of main front structural parts were selected and modified to obtain the suitable crash characteristics.

Later, using the determinate characteristics of front structural elements the simulation of 50% offset crash with the same conditions (rigid barrier and 15m/s impact velocity) was performed. Comparing the curves (Fig. 25) of the absorbed energy versus time we get the value of the necessary additional stiffness which we should apply to the longitudinal members.

Subsequently, a macro element model [1] of longitudinal member cross-section was analyzed in macroelement CrashCAD software. The aim was to obtain in shorter time proper geometrical characteristics of longitudinal members capable to absorb additional quantity of impact energy. The influence of thickness on the average force for the analysed cross–sections of the longitudinal members is shown in Fig. 18. Having this curve it is easy to evaluate the value of additional thicknesses according to the necessary additional amount of energy.

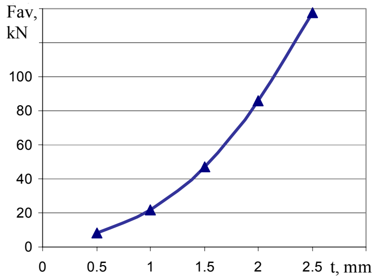


Figure 18. Dependence of the average force and thickness for the selected profiles of the longitudinal members

The poor analytical models show that by increasing the stiffness of the longitudinal member it is possible to absorb enough energy in 50% offset crash and to obtain similar crash characteristics as in full impact case. However, adding the thickness or making stiffer longitudinal members we are increasing the probability to obtain Euler buckling and to lost a significant value of not absorbed energy. It is also known that the type of the crushing (symmetrical, diamond or global bending) of axially compressed thin walled elements [11] depends not only on its geometrical and material properties but also on crushing velocity [8]. In the beginning of impact when the velocity is great enough the axial crushing phenomena occurs. However, later when the velocity decreases for the stiffer longitudinal member a global bending is possible. During a global bending just one plastic hinge occurs, which means that the dissipation of energy related to the crushing distance is not big enough. To avoid losing of the global stability we should decrease the stiffness of the compressed elements. Decreasing of stiffness should be performed during the crash, otherwise the smallest stiffness will initiate in the beginning of impact a non-consecutive crushing.

Usually the peaks in typical deceleration curves of frontal impact are in the middle part of the crushing time (Fig. 24) when the contact between the engine and other parts occurs (Fig. 22-b). The deceleration value directly depends on forces acting on the compartment. The longitudinal members and the subframe are the main parts which transfer the axial forces to the compartment. To decrease the peak of deceleration we should decrease the level of axial forces acting in the longitudinal members in the moment when the contact with the engine occurs.

The value of the forces in the axially crushed thin walled elements can be decreased using triggers (Fig. 19a). However, in the same time we should have the longitudinal members stiff enough to withstand lower speed impacts without initiating a damage. To analyse and to compare behaviours of the longitudinal members with and without the triggers a FE model (Fig. 19b) with two different longitudinal members was created. In the model the nodes of the firewall were fixed and rigid plates with different masses were impacting the longitudinal members with an applied initial velocity.

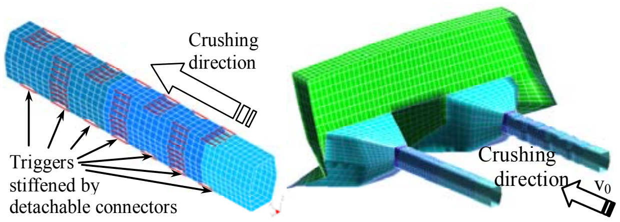


Figure 19. FE model to analyse the behaviour of axially crushed longitudinal members a) view of longitudinal member with triggers and detachable connectors; b) simplified FE model

Adding the stiffeners with detachable connectors (modelled as rigid spot welds in the FE model) we have the construction stiff enough in the beginning and consecutively detaching them in the time when we can decrease the stiffness. Controlling the time of consecutive detachings of the stiffeners we can significantly decrease axial forces (Fig 20) which influence the deceleration pulse and eventually decrease the energy absorption for the analysed cross-section up to 20%. If the stiffeners would be detached consecutively with delays, we can obtain the same level the of absorbed energy as in the straight longitudinal members and at the same time we avoid the global bending.

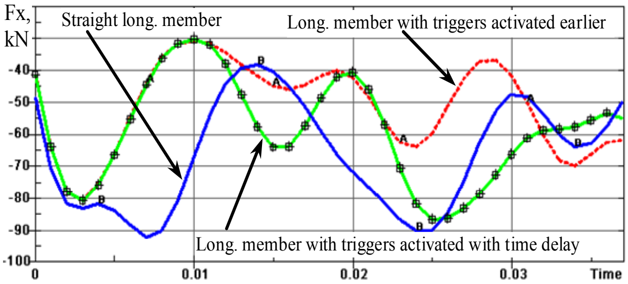


Figure 20. Axiall forces in longitudinal members (used SAE 60 class filter)

The moment to decrease the stiffness depends on such impact characteristics as impact velocity, percentage of overlap and the properties of the frontal deformation zone (stiffness). Assuming that the change of the system kinetic energy during the crushing is equal to the energy dissipated in the plastic deformations, we can calculate the time to detach the stiffeners by the following equation:

$$0.5m\left(v_0^2 - v^2\right) = \int_0^x F(x)dx \tag{11}$$

Assuming that the present stiffness of the part being crushed is known we can predict the increase of the compression forces and, if necessary, decrease the stiffness using the detachable connectors in the time which we calculate from the presented Eqn. (11).

The shapes deformed in frontal crash simulation in both analysed modes are presented in Fig. 21 and Fig. 22.

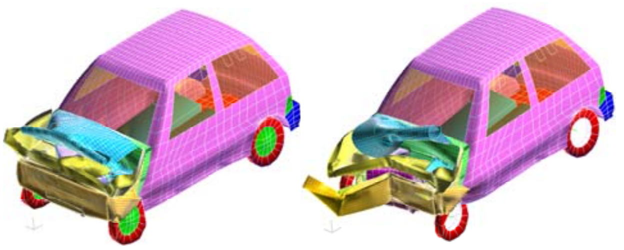
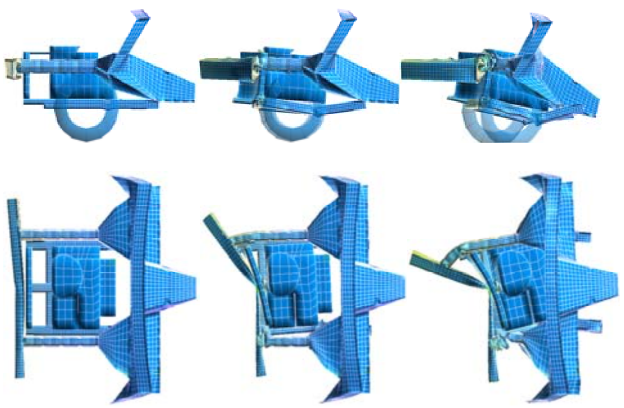


Figure 21. Deformed shapes of FE vehicle crash models a) full impact into a rigid barrier with velocity 15m/s; b) 50% offset impact with the same conditions



a) b) c)

Figure 22. Deformed shapes of front structural parts during 50% offset crash into a rigid barrier with velocity 15m/s a) $t = 10\,\text{ms}$, b) $t = 30\,\text{ms}$, c) $t = 80\,\text{ms}$

To decrease the deceleration pulses initiated by engine contacts, a trigger in the subframe part is added to obtain bending collapse (in plastic hinge it absorbs additional amount of energy and decrease the axial forces acted to the occupant's compartment). To increase a bending moment in a selected place of the subframe, an additional rectangular thin-walled tube between the top edge of the radiator and the engine is added.

Additionally, to decrease the deceleration peak at time 37 ms the joints between the subframe and the occupant's compartment are detached. The influence of the mentioned changes is visible on the deceleration curve for 100% impact case (Fig. 24).

To get the same crash performance characteristics in 50% offset crash as in the full impact case, the thickness of the longitudinal members was increased 1.8 times. The effect of the increased thickness to the dissipated energy in two longitudinal members is shown in Fig. 23. In both cases of 50% offset crash (with equipped adaptive system and without) the left longitudinal members absorbed approximately the same amount of energy about 93%.

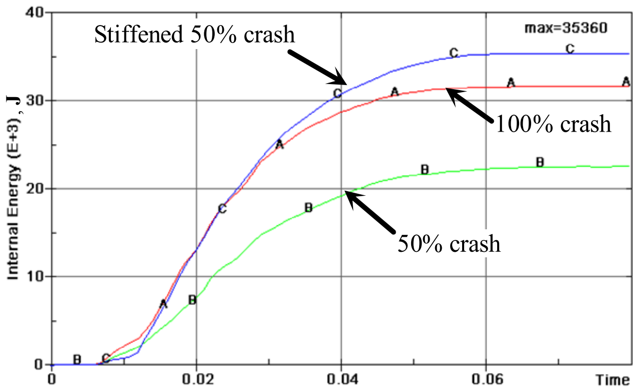


Figure 23. Energy dissipation in longitudinal members during the three simulated crash cases

The influence of the stiffened longitudinal members on the deceleration pulse is shown in Fig. 24.

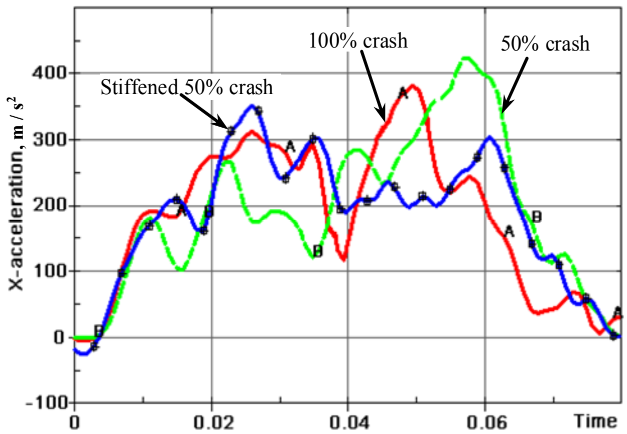


Figure 24. Deceleration pulses (used SAE 60 class filter) of full and 50% offset frontal impact simulations

In case of 50% offset crash with stiffened longitudinal members the peak in the end of the impact decreased compared to 50% offset impact without stiffening. This because of higher amount of the energy dissipated in the beginning of the impact.

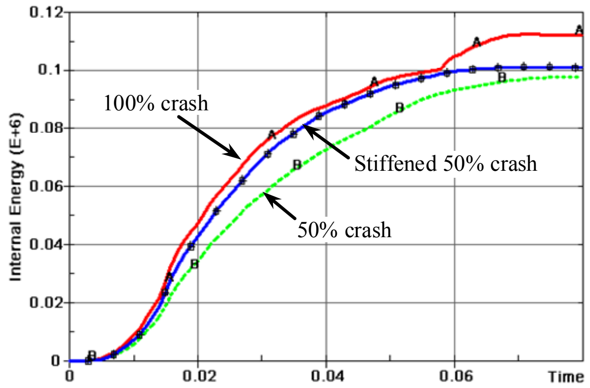


Figure 25. Energy absorption during the three simulated crash cases

The effect of the increased stiffness on the absorbed energy is shown in Fig. 25. The energies dissipated in full impact and in 50% overlap crash with the adaptive system were close to similar. The difference at the end of the impact occurs because in 50% overlap crash more impact energy are transformed to kinetic energy.

9. Proving of idea using validated FE crash model

Finally to check differences between reality and simplified models a behaviour of an explicit FE model of a real car has been simulated. A validated in full impact test Audi A8 crash finite element model was taken from Oak Ridge National Laboratory web resources [16] with ORNL permission for use. Next the model has been translated from LS-DYNA to Abaqus/Explicit code and simplified for the considered cases. Car body was cut on the driver's seat level, rear part was deleted and replaced with suitable nodal mass and rotary inertia to keep proper mechanical properties of all car assembly.

All nodes on cut line were connected with mass node by rigid body structure. Some parts were remeshed to improve mesh quality and reduce minimal allowed time step. Four and three node shell elements with reduced integration were defined for body mesh. For non heavy plastic deformable parts three integration points on element thickness were used, however important body areas were simulated with five integration points on the thickness. Vehicle hood, some small parts like engine equipment and occupant compartment doors were removed. Rods with material failure definition were used to attach powertrain module to car cassis. Vertical gravity field was not applied. Variable mass scaling has been used. Maximal 6% of additional stabilizing mass was added during simulations and artificially added hourglassing energy was on less then 3% level comparing with total model energy. Enhanced hourglass control was turned on. For acceleration curves SAE 60 class filter was used.

Three simulations were performed for following situations:

- 100% impact into rigid wall (impact velocity 15 m/s)
- 50% impact into rigid wall (impact velocity 15 m/s)
- 50% impact into rigid wall stiffer left (impacting) longitudinal member (impact velocity 15 m/s)

For first two cases genuine car properties from ORNL model were used. In last simulation a connector element with nonlinear spring properties was added into left (impacting) longitudinal member structure. Aim of such element use was to increase crushing stiffness of longitudinal member profile. Spring force as a displacement function was constant and similar to average crushing force of longitudinal member tubular absorber.

Configuration of simulated car during case of 50% impact with genuine parameters were shown on Fig. 26. Detailed pictures provided on Fig. 27 show Iso view of model at the beginning and the end of simulation.

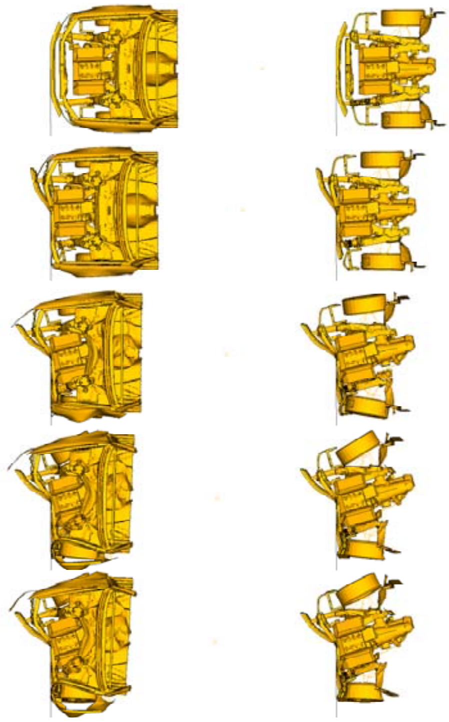


Figure 26. Top views of Audi A8 front end model with genuine parameters, during 50% overlapped impact into rigid wall from 15 m/s (from top ordered: 0ms, 20ms, 40ms, 60ms, 80ms time shoots)

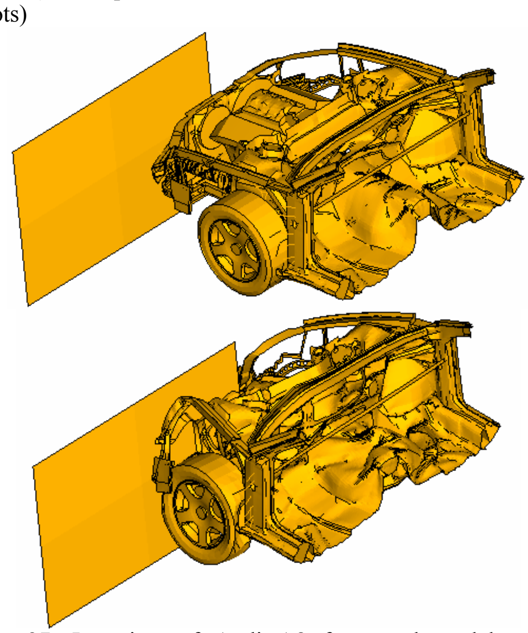


Figure 27. Iso view of Audi A8 front end model on the beginning and the end of 50% overlapped impact simulation (from 15 m/s)

On Fig. 28 deceleration curves taken from rear mass node simulating mass and inertia of rest of car for all three simulations. Pulse fill for version with additional connector element and 50% hit is increased comparing to version without system in 50% impact case.

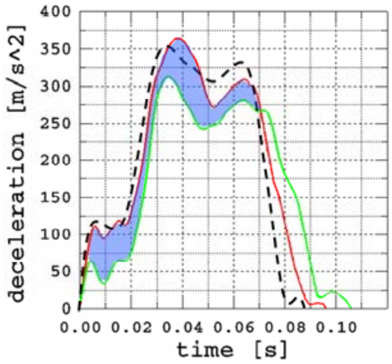


Figure 28. Deceleration of rear mass node versus time curves

Character of described curve is close to shape of curve for case of 100% impact. Depths of deformation measured in car longitudinal direction from undeformed shape were shown on Fig. 29.

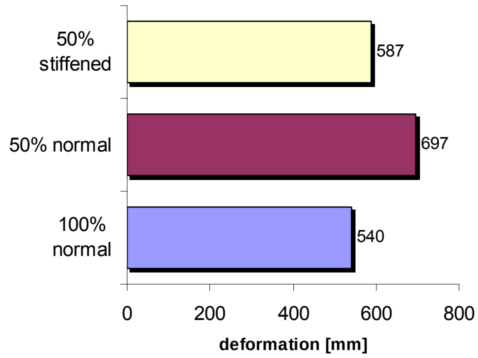


Figure 29. Depth of deformations of vehicle structure for all simulated cases

In examined model equipped into adaptive system response in acceleration domain for 50% overlap hit, were similar to model response for 100% overlap crash. Thus, assumed target was achieved, however Audi A8 structure is equipped into “mechanical” system improving car behaviour during offseted impacts, so difference is not very large.

Due to high computing time of each version (over 30 hours on double processor 3 GHz computer with 4GB of RAM), other studies of modified ORNL model than shown above were not placed in present article.

10. Conclusions

Use of an adaptive system for automotive purposes in a way shown in this paper can be advisable, however the level of improvement strictly depends on the ratio of energy dissipation in longitudinal members and the whole structure. A specially designed structure seems to be the best way to achieve a good result. Applicability of the system in real-world automotive structures depends also on economic reasons.

The presented results demonstrated that a significant safety improvement in front impact situations can be achieved with actively adaptive systems.

The proposed additional stiffeners allow to increase energy dissipation in the case of 50% offset impact to the same level as in the case of full collision.

Detaching the stiffeners in the full frontal impact case allows to dissipate a predesigned amount of impact energy.

Additionally, reduction of the maximum deceleration level in the case of full impact can be achieved by a proper selection of the time instant to detach the stiffeners.

Also, the effect of increase of the maximum allowable impact velocity (in the case of full impact) can be achieved with additional stiffeners.

11. References& Acknowledges

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