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Material and shape crash-box influence on the evaluation of the impact energy absorption capacity during a vehicle collision

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Abstract

During a light collision between two vehicles or a vehicle and an obstacle, the internal structure of the car is irrecoverably damaged. In order to limit as much as possible the body in white defacement the new vehicles are equipped with so called crash boxes situated in front and rear of the car. The purpose of our study is to analyze the influence of the crash box material in order to reduce the side member permanent deformations, since once damaged, this component, even repaired, will never assure a good safety in case of major collision. Although the speed necessary for of a car homologation is 16 km/h, a second purpose of this paper is to identify the influence of the velocity upon the crash box behaviour. Consequently, a numerical study has been undertaken to investigate axial crushing responses, energy absorption performance of empty & foam filled steel and aluminium concentric tubes with different density of foam.

Keywords: crash-box; side member; collision; energy absorption; impact.

1. Introduction

Nowadays, aluminium in its various product forms (sheet, extrusions, die castings) is an established automotive lightweight material offering excellent weight saving potential, including crashworthiness applications for passive vehicle safety. Crashworthiness is the ability of a structure to protect its occupants in the event of a crash. Frontal impact cars is one of the most often crash types [1]. Modern ductile aluminium alloys have an outstanding ability to absorb impact energy in case of accidents. The main idea of this paper was to evaluate the influence of the crash-box material for reducing the side member plastic deformation.

The vehicle used for the analysis has been developed by The National Crash Analysis Center (NCAC) of The George Washington University under a contract with the FHWA and NHTSA of the US DOT [2].

For comparison, different materials have been taken into account for the simulations of the crash box: a) automotive steel H360, with E = 200 GPa and yield limit of 420 MPa; b) aluminum extruded 6008T7, with E = 70 GPa and yield limit 250 MPa;

Concerning the filling material, low-density polymeric (non-metallic) foams have been widely used in several engineering applications.

The density of polyurethane foam was considered as 95 kg/m^3; this type of foam is widespread in the energy absorption and impact applications due to its excellent energy absorbing capability. The chemical reaction can also take place in moulds, leading to, for example, a car bumper. According to the new regulations in the Federal Motor Vehicle Safety Standards it is required the use of polymeric foam materials inside motor vehicles in order to protect the occupants during severe collisions [3].

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The crash box was modeled in two different configurations: a) rectangular caisson shape (height h = 90 mm, width b = 80 mm and thickness t = 2 mm); and b) hexagonal shape (diagonal of 90 mm and thickness t = 2 mm).

Crash boxes are manufactured from automotive steel – an advanced high strength steel with carefully selected chemical compositions and multiphase microstructures resulting from precisely controlled heating and cooling processes.

Crash box, with which a vehicle is equipped at the front end of its front side frame, is one of the most important automotive parts for crash energy absorption.

In case of frontal crash accident, crash box is estimated to be collapsed with absorbing crash energy prior to the other car components such that the damage of the cabin frame is minimized the vehicle occupants having the possibility to save their lives. According to the new regulations, it has been strictly required to satisfy both reduction of body weight and improvement of crash worthiness and consequently, regarding the crash box, it is absolutely necessary to ensure high energy absorption using cross sections with minimum thickness.

Sometimes, it happens that these boxes do not work as designed when a thin sheet is applied, being difficult to acquire sufficient energy absorption only through the thin walls.

In general, the use of foam filler is to reinforce and stabilize the crush response of thin-walled tubes when subjected to impact loads [4].

In several studies [5] attention is focused upon finding an optimum cross sectional shape of a crash box to ensure high capability for energy absorption without crash bead.

2. Numerical configurations models used in the analysis

2.1. Crash test configuration

In Fig. 1 are depicted the three views of a car before collision with an obstacle (barrier).

In reality the barrier is a very stiff wall (made of hard steel material). The case depicted above represents in fact the most unfavorable situation, which, in reality, is rarely encountered. The centering of the barrier is considered to be 40% of the total car width.

For a better understanding of the car subassemblies subjected to large deformations during the collision, in Fig. 2 the authors present the body in white (the whole car structure) highlighting these components.

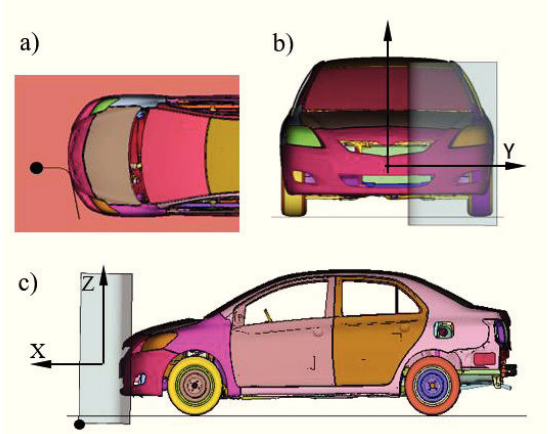

Fig. 1. Crash t est configuration: a) top view (X = 46.25 mm), b) front view (Y = 169,5 mm), and c) left side view.

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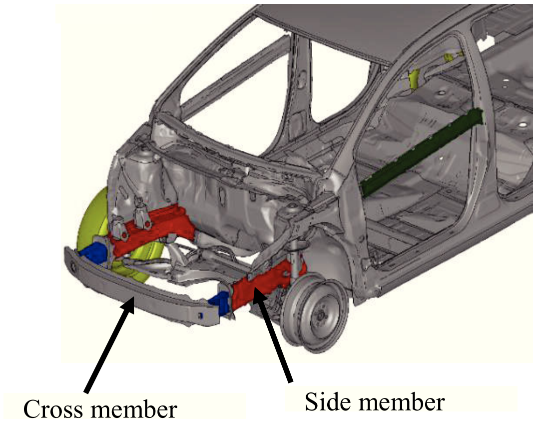

Fig. 2. Car components subjected to large deformations after impact.

2.2. Crash boxes modeling

The authors considered in their study four variants of the caisson shape: a) the original caisson, already existent on the vehicle; this shape was considered as reference during the simulations; b) a rectangular caisson, with a thickness t = 2 mm, c) a hexagonal caisson, with a same thickness; and d) an optimized shape, with a same thickness. All variants presented above are depicted in Fig. 3.

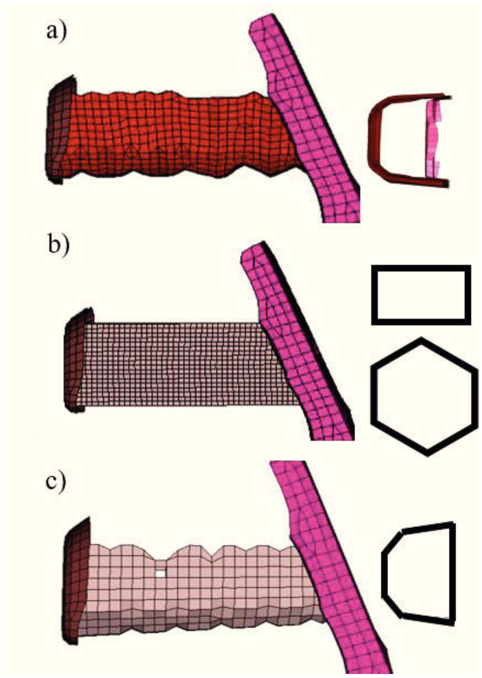

Fig. 3. Variants of the caisson cross-section shape: a) initial variant (original caisson), b) rectangular/hexagonal shape, and c) optimized shape.

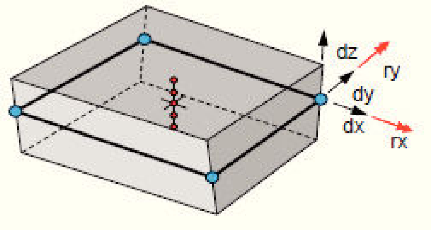

Fig. 4. Belytschko-Lin-Tsay shell formulation.

The finite element used for the modeling of the crash box structure was Belytschko-Lin-Tsay shell, based on Reissner-Mindlin kinematic assumption with five points of integration in thickness, as in Fig. 4.

3. Results and discussions

Nowadays, in parallel with the continuous evolution of digital computing systems have appeared and created specialized software developed to simulate and to analyze the structures behavior during a collision. These numerical codes, used by almost all vehicle manufacturers in the world, were developed to simulate the collision tests or other types of passive safety predictions. An initial simulation was performed on the reference model, made from H360 steel (Fig. 3a).

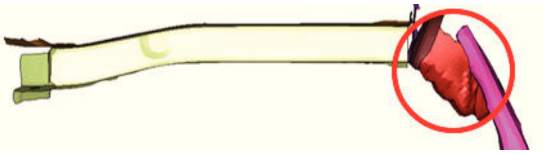

Fig. 5. The deformed shape of the steel reference model.

As in Fig. 5, the reference crash-box has a partial compression (Δ_x = 78 mm) and a bending along *z* axis (perpendicular to the ground). The absorbed energy was in this case equal to 4500 J. The plastic deformation occurred on the side member was 1.4%, as in Fig.6.

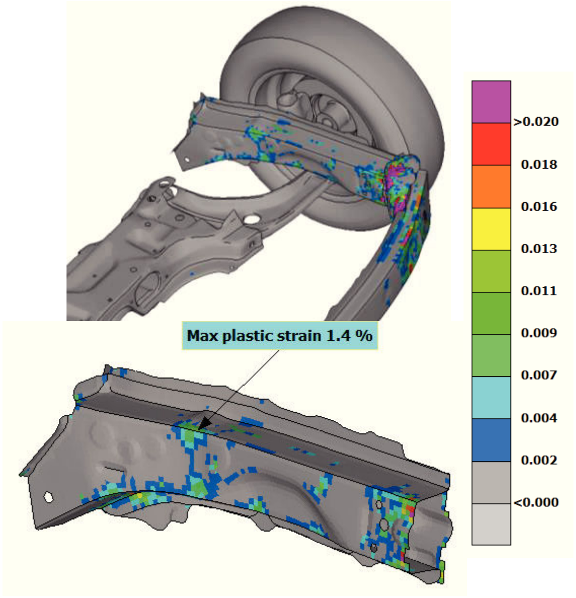

Fig. 6. Deformation map along the side member.

A second simulation was performed on a H360 steel rectangular caisson shape, as one can see in Fig.7.

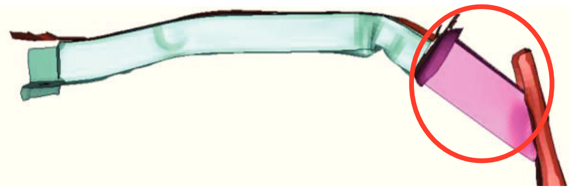

Fig.7. The deformed shape of the steel rectangular caisson.

In this case, the deformation on x axis is quite small, (Δ_x = 46 mm), there is no significant bending along z axis. Due to the high stiffness of the crash box, the bending is transmitted to the side member, inducing large deformations (77% in the vicinity of the crash box, Fig. 8). The absorbed energy by the crash box was in this case equal to 1140 J.

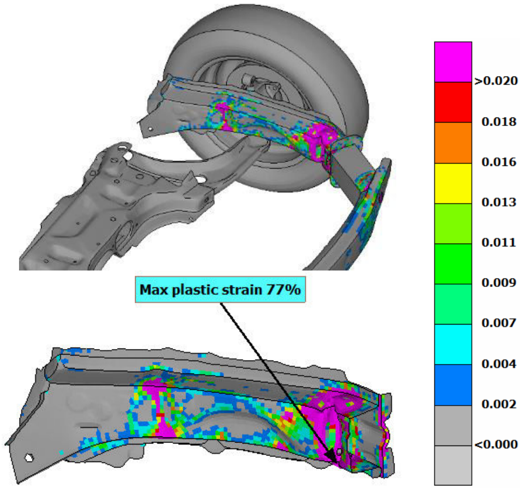

Fig. 8. Deformation map along the side member for the rectangular caisson shape.

A third simulation was performed on a H360 steel hexagonal caisson shape, as one can see in Figures 9 and 10.

Fig. 9. The deformed shape of the steel hexagonal caisson.

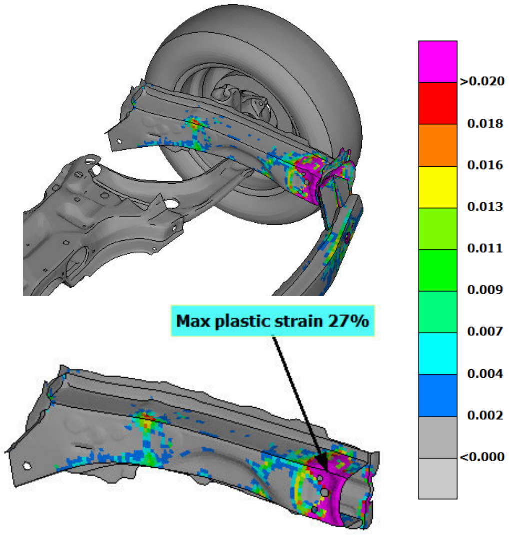

Fig. 10. Deformation map along the side member for the hexagonal caisson shape.

In this case, the deformation on x axis is higher than in the previous cases, (Δ_x = 80 mm), with a significant bending along z axis. It is also observed a small bending of the side member and a significant deformation (27%) in the vicinity of the crash box. The absorbed energy was in this case equal to 1800 J.

A fourth case of our analysis was performed on an optimized caisson (Fig. 3c).

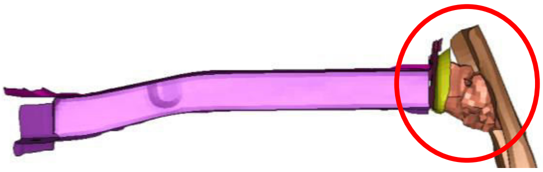

Fig. 11. The deformed shape of the optimized steel caisson.

In this case, the deformation on x axis is higher than all the previous cases, (Δ_x = 102 mm), with an insignificant bending along z axis. It is also observed in Fig. 12 a very small deformation (0.9%) of the side member at the mid span distance with respect to the crash box.

The internal energy was in this case equal to 5770 J, this caisson type being the most efficient regarding the energy absorption. A general form of optimization including geometrical parameters (necking radius) as well as specified limits on other responses (constraints – rotations on z axis) has been performed in the analysis. LS-OPT is a standalone Design Optimization and Probabilistic Analysis package with an interface to LS-DYNA.

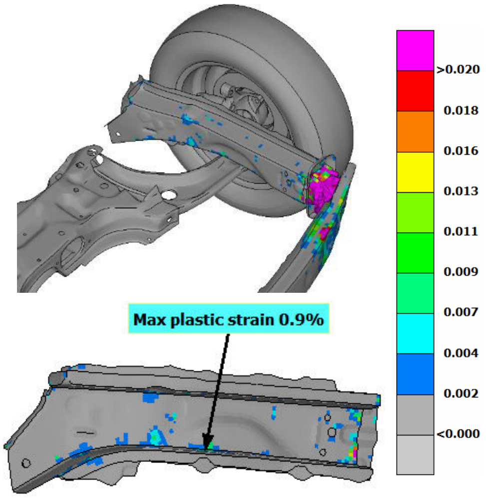

Fig. 12. Deformation map along the side member for the optimized caisson shape made from steel.

For the last version (the optimized variant), in order to check the influence on the material upon the impact behavior, a fifth variant has been performed, where, the same caisson shape has been analyzed replacing the

H360 steel, with an yield limit of 420 MPa with an extruded aluminium 6008T7, having a corresponding yield limit of 250 MPa. In Fig.13 is presented the deformed shape for this case, whereas in Fig.14 is plotted the deformation map on the side member.

Fig.13.The deformed shape of the optimized Aluminium caisson.

In this case, the deformation on x axis is the maximum of all the previous studies, (Δ_x = 178 mm), with an insignificant bending along z axis. It is also observed a very significant deformation (35%) of the side member at the mid span distance with respect to the crash box. The internal energy relatively low in this case, with a value of 2000 J, this caisson type being one of the most inefficient regarding the energy absorption. Two last variants were to fill as well the optimized steel caisson, as the optimized aluminum caisson with polyurethane foam, with a density of 95 kg/m^3.

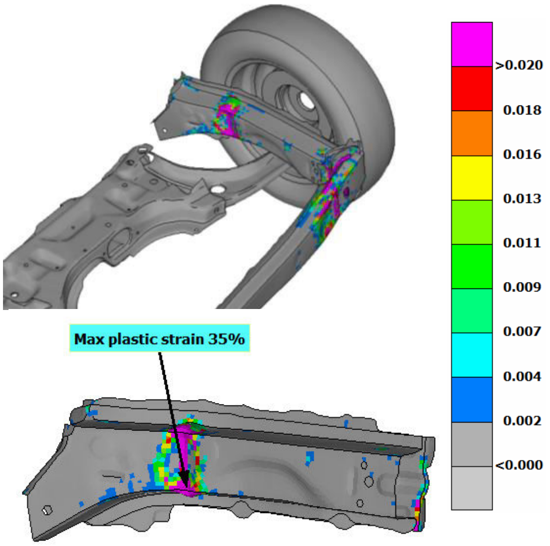

Fig. 14. Deformation map along the side member for the optimized caisson shape made from Aluminum.

As it is known, this type of foam is widespread in the energy absorption and impact applications due to its excellent energy absorbing capability. Since during the modeling of the structure presented in Fig. 11 were observed small deformations of the crash box after filling with PU foam, the authors considered that a substantial diminishment of the caisson wall thickness width (from 2 mm to 0.2 mm) is more appropriate in the final evaluation of energy absorption capacity. In Fig. 15 is depicted the deformed shape for this case, whereas in Fig.16 is plotted the deformation map on the side member.

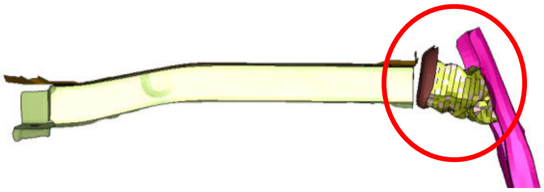

Fig. 15. The deformed shape of the optimized steel caisson filled with PU foam, with a wall thickness diminishment

In this case, the deformation on x axis of the crash box was Δ_x = 72 mm, with an insignificant bending along z axis. It is also observed in the side member an acceptable deformation of 1.6% (less than the imposed limit of 2%). The internal energy is relatively high in this case, with a value of 4900 J, this caisson type being one of the most efficient regarding the energy absorption.

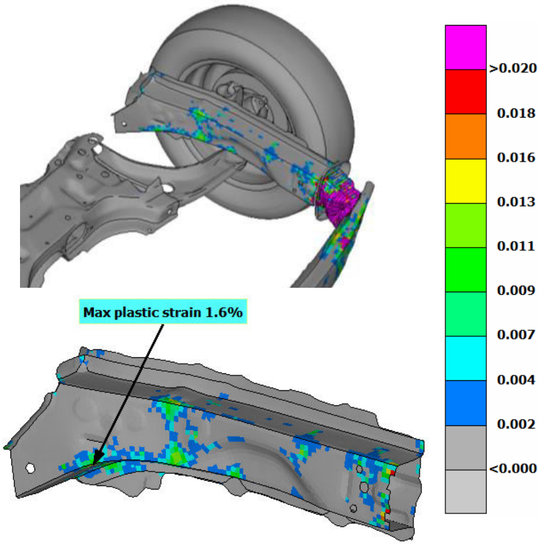

Fig. 16. Deformation map for the optimized caisson shape made from steel filled with PU foam.

A last variant of the optimized caisson was the one made of aluminum, filled with PU foam. For this last variant the authors maintained the initial wall thickness of 2 mm. In Fig. 17 is plotted the deformed shape of this case, whereas in Fig. 18, the deformation map on the side member. In this last case, the deformation on x axis of the crash box was Δ_x = 157 mm, with an significant bending along z axis. It is also observed in the side member an acceptable deformation of 0,7% (less than the imposed limit of 2%).

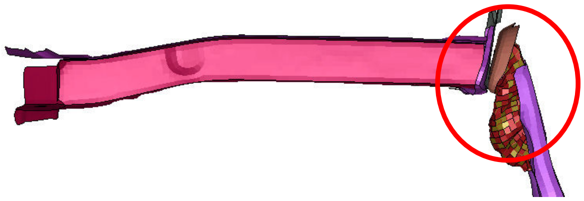

Fig. 17. The deformed shape of the optimized Aluminium caisson filled with PU foam, without a wall thickness diminishment.

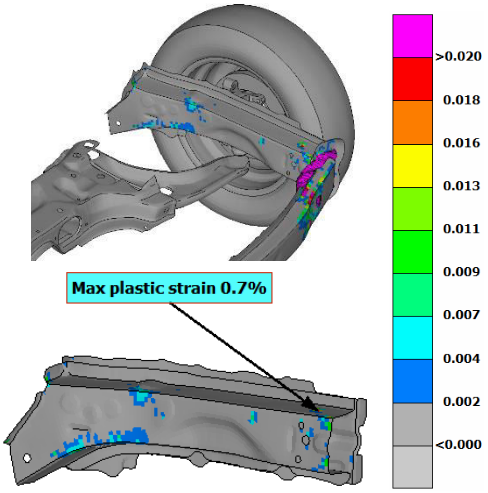

Fig. 18. Deformation map for the optimized caisson shape made from Aluminum filled with PU foam.

The internal energy is also relatively high in this case, with a value of 4000 J, this caisson type being as well one of the most efficient regarding the energy absorption.

In Fig. 19, is depicted the variation in time of the side member absorbed energy.

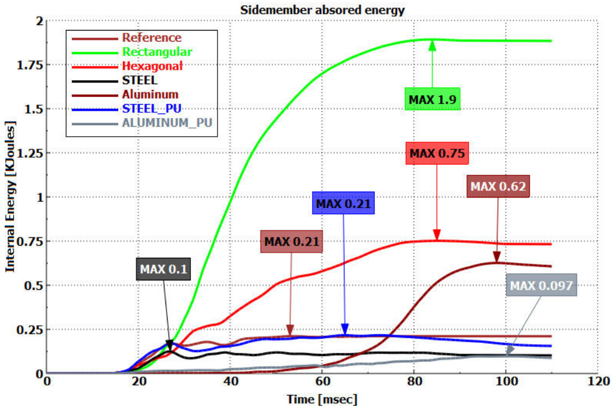

Fig. 19. Variation in time of the side member energy.

The shock consumption during impact in all cases has been evaluated at 110 msec. As one can see, there is a large discrepancy between the behavior at impact for the steel rectangular caisson and the aluminum one, filled with PU foam regarding the impact absorbed energy by the side-member. In order to reduce the damage of the vehicle during the collision, this energy should be as lower as possible (below 500 J, ranging from one vehicle to another).

4. Conclusions

Crash box is introduced to vehicle design to improve the impact performance and reduce the damage of vehicle body at impact speed. Repair cost at collision accident can be cut down by use of this box.

In this paper several numerical simulations have been performed on different crash-boxes models in order to obtain an optimized shape necessary to reduce the absorbed energy by the side-member. The impact velocity considered in these simulations was 16 km/h, according to ENCAP standardization.

The best behavior in case of low speed impact is obtained by the optimized shape of the caisson made from Aluminum, filled with PU form. The most unfavorable caisson shape is the rectangular one due to the fact that this type of section induces large deformations in the side member structure which represents a total damage of the vehicle structure.

The optimized shape made from steel should be another option, with a difference of 0.4% deformation of the side member.

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