

Design of the Impact Attenuator for a Formula Student Racing Car: Numerical Simulation of the Impact Crash Test

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Abstract

Vehicle safety is one of the major research areas in automotive engineering. The car industry is developing new passive and active safety systems and techniques to increase the safety of vehicle occupants. To reduce the development and testing costs of a new safety design, it is recommendable to use computational crash simulations for early evaluation of safety behavior under vehicle impact test. A device which is intended to reduce the damage done to vehicle structures and the possible injury to passengers resulting from a collision is impact attenuator. Numerical simulation by using finite element methods and design of the impact attenuator of a Formula student car body, is presented in this paper. It was developed for an auto-cross racing car, with a small formula style, which was later manufactured, prepared, tested and demonstrated on the race competition. After initial study of the behaviour of impact attenuator and car body as independent structures, we have added links, which are connecting all parts in one assembly. Final results show the comparison between attenuator device, as the independent structure and two solutions of the complete assembly with the car body. All initial requirements were set in accordance with the 2008 Formula SAE rules. These requirements were completely satisfied with the final configuration of the impact attenuator. The main aspects of the research were oriented to get optimal values of kinetic energy evolution during the frontal impact, with the lowest initial deceleration.

Keywords: Impact attenuator, car body, finite element simulation

1. Introduction

Driver safety in case of accident is a main problem when developing a new vehicle design, and this is of particular importance when dealing with racing car (EuroNCAP 2008). This is due to the higher velocity that generally characterizes the operation of a racing car with respect to a urban use vehicle. Higher velocity means higher (square power) kinetic energy to be dissipated in case of impact.

Safety in case of accident is a particular concern not only for formula 1 racing car but also for formula SAE student racing car and for go-kart (Belingardi and Obradovic 2009, Boria 2008).

This paper is presenting the design and numerical simulation of impact event for the frontal safety structure of car body for the formula SAE vehicle developed by the Politecnico di Torino team. Figure 1 shows the final solution of the racing car, which was produced in the collaboration with partner enterprises in Turin and successfully demonstrated in several racing competitions in Europe.



Fig. 1. Final product of the racing car

After initial analysis of the response of the designed impact attenuator considered independent from the car body structure, links, that connect all parts in one assembly, have been added. The aim of the project to make good links, i.e. links that have a good response in case of vehicle front impact, reveals to be not a secondary subject in the development of the vehicle structure. Also, the design of the initial impact attenuator was changed in order to get the best response of the assembly (Zhang et al. 2007, Xiong Zhang et. Al. 2008) .

The performed numerical simulations put in evidence the needs for the design of a good energy absorbing structure and also the requirements for a good design of the links.

Complete geometry was done by means of the software for 3D modeling Catia. The finite element model of the structure was developed by means of the software code Hyper Mesh. Finally the crash event simulation has been developed by means of the Radioss code.

The main aspects of the research were oriented toward the dissipation of the kinetic energy during the front impact that should be as progressive as possible and toward the evaluation of the initial deceleration which has to be as little as possible (Belingardi and Chiandussi 2008, Peroni et al. 2002, Peroni et al. 2009). It is well known that the optimal solution in absorbing energy for this type of application is to obtain a nearly flat diagram of the impact force, that means a nearly constant value of the deceleration (Jones Norman 2009, Karagiozova 2004, Hosseinipour et al. 2003). All initial requirements, which gave good results at the end of simulation, were done regarding the 2008 Formula SAE rules. These constraints are mainly oriented to dimensions of the attenuator, mounting of the parts, material selection, construction of attachments and final results requirements.

2. Geometry of the model

This work is based on the analysis and improvement of the design of the impact attenuator and after on the design and improvement of the behavior of the links of the energy absorbing structure to the car body.

One of the most important SAE constraints is about the construction solution of attachments between impact attenuator and bulkhead: “Attached securely and directly to the Front Bulkhead and not by being part of non-structural bodywork. The attachment of the Impact Attenuator must be constructed to provide an adequate load path for transverse and vertical loads in the event of off-centre and off-axis impacts. If not integral with the frame, i.e. welded, a minimum of four (4) 8 mm Grade 8.8 (5/16 inch Grade 5) bolts must attach the Impact Attenuator to the Front Bulkhead” (2008 Formula SAE Rules).

An initial solution for the linkage was taken in consideration and then discarded because of its big inconvenience from the point of view of its practical implementation, as we can see in the image of Fig 2.

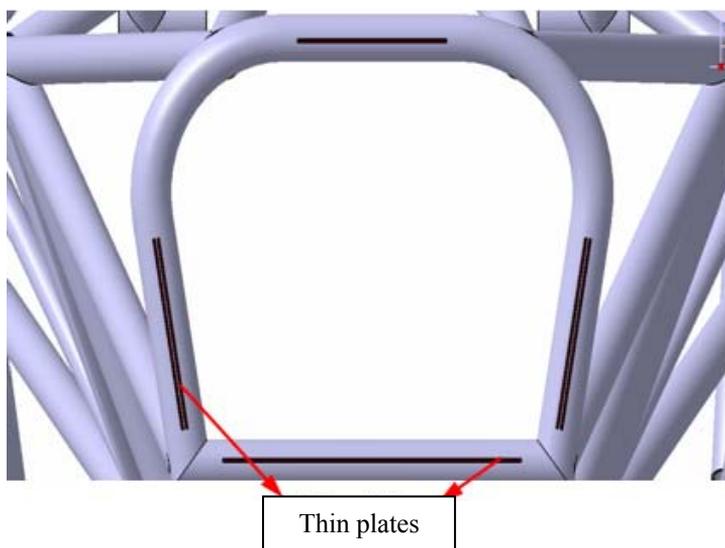


Fig. 2. Unacceptable solution for links between car body and impact attenuator

After several consultations with judges of SAE 2008 from Germany, the final solution for the design of links was chosen, which gave good results from the point of view both of practical implementation and, finally, of good impact behavior, i.e. desired amount of absorbed energy and suitable level of deceleration (Otubushin 1998).

These final links are C – shaped. They are independent parts of the system. They require bolts for the connection between car body and impact attenuator. In the image of the next figure 3(a), we can see the front impact attenuator structure and the links assembled to that structure.

In Fig. 3(a) some holes are well visible in all the four side walls of the impact attenuator structure. They are operating as triggers, that means have both the scope to decrease the weight of the structure itself and to obtain a better crush behavior by decreasing the first peak of the collapse force (Feraboli et al. 2007, Montanini et al. 1997, Chung Kim Yuen et al. 2008, Yong – Bum Cho et al. 2008, Avalor and Belingardi 2004).

The response of the assembly during the impact was not as good as the response of the independent front impact device. Therefore the design of the impact attenuator has been changed and the comparison of the old and improved solution is shown in Figs. 3(a) and 3(b). By putting holes on different positions, removing thin plates on both sides of the attenuator and adding new holes on fillet surfaces we got the solution which has smaller mass and, at the same time, has better impact performance, in particular smaller initial deceleration pick during the crash test.

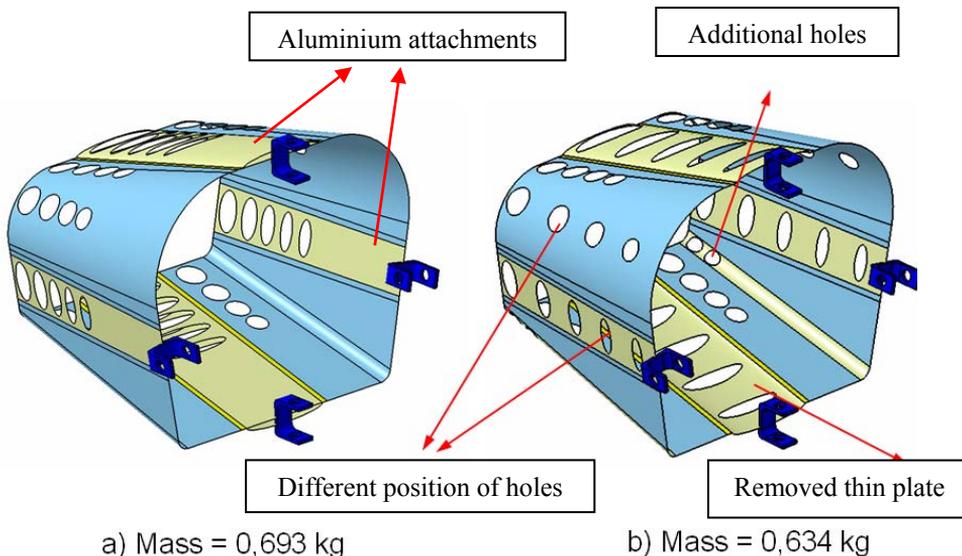


Fig. 3. Final solution for links with impact attenuator. (a) First solution; (b) Improved solution

Figure 4 shows the complete structure; it consists of three different parts: car body frame, impact attenuator and links.

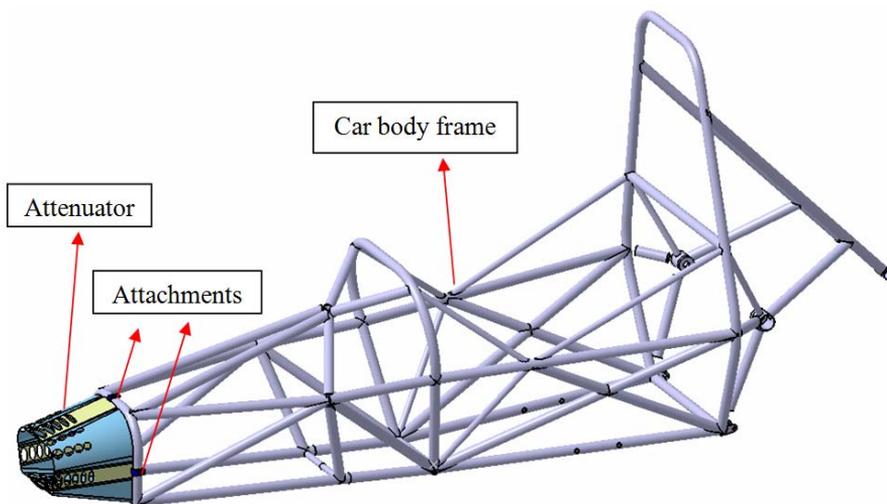


Fig. 4. Complete structure

3. Finite element model

Complete simulation was done using finite element model developed in Hyper Mesh software. It is very important to notice that it was not used all car body structure, but only the frame that is the most important part from the point of view of the vehicle front impact and driver safety. Unnecessary parts were cut off before making the mesh of the finite element model.

The model includes two different types of material: steel S275JR UNI EN 10025 (Fe430) and 6082T6 aluminum alloy (this type of the alloy is anticorrosion material good for energy absorption but quite hard for bending). Impact attenuator and links were made of that aluminum alloy and the car body frame was made of steel. Also, attenuator includes two beams which have different properties in comparison with the rest of attenuator. Parts of the attenuator with attachments are presented in Fig. 5.

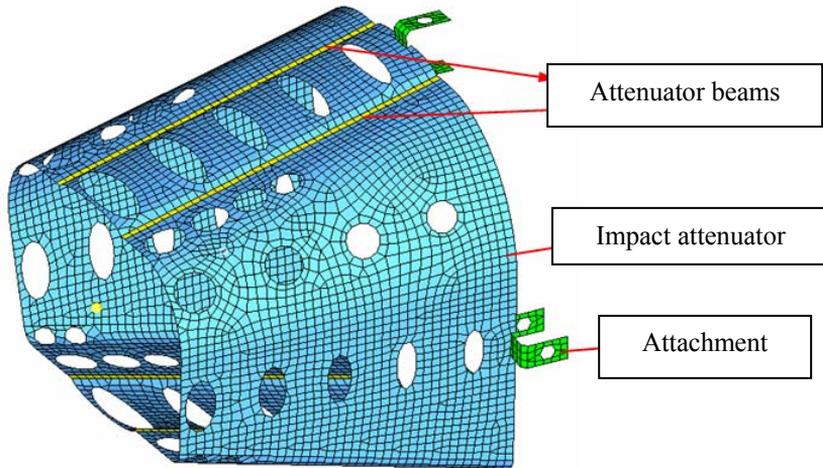


Fig. 5. Parts of the impact attenuator

For this simulation materials were modelled as isotropic with elastic – plastic characteristics. The strain hardening behaviour of materials is a major factor in structural response as metal working processes or plastic instability problems. A proper description of strain hardening at large plastic strains is generally imperative. For many plasticity problems, the hardening behaviour of the material is simply characterized by the strain – stress curve of the material. The incremental plasticity theory is generally used in computational methods. Plasticity models are written as rate – dependent or independent. As it is well well known, with a rate – dependent model the strain rate does affect the constitutive law. This is true for a large class of metals at low temperature relative to their melting temperature. Most isotropic elastic – plastic materials laws in solver Radioss use Von Misses yield criteria (Radioss Theory Manual 2005).

The description of the material characteristic will be made by means of the Johnson – Cook elastic - plastic law (Law 2 in RADIOSS). The stress vs. plastic strain law is as follows:

$$\sigma = (a + b\varepsilon_p^n) \left(1 + c \ln \frac{\dot{\varepsilon}}{\dot{\varepsilon}_0}\right) (1 - T^m) \quad (1)$$

where σ is flow true stress (elastic + plastic components), ε_p is plastic strain (true strain), a is yield stress, b is hardening modulus, c is strain rate coefficient, $\dot{\varepsilon}$ is strain rate, $\dot{\varepsilon}_0$ is reference strain rate, m is temperature exponent.

The Johnson-Cook model is a well-accepted and numerically robust constitutive material model, commonly utilized in modeling and simulation studies. The Johnson-Cook (JC) model assumes that the slope of the flow stress curve is affected by strain hardening, strain rate sensitivity and thermal softening behaviors. Each of these sensitivities is represented by a proper multiplying factor in the constitutive equation. According to this law the material behaves as linear elastic when the equivalent stress is lower than the yield stress. For higher value of stress, the material behavior is plastic. This law is valid for brick, shell, truss and beam elements. Adopted values of above parameter can be seen in Table 1:

Property	Aluminum alloy 6082T6	Steel S275JR UNI EN 10025 (Fe430)
Density	$2.7 * 10^{-3} \text{ g/mm}^3$	$7.8 * 10^{-3} \text{ g/mm}^3$
Poisson's ratio	0.33	0.3
Young's modulus	70000 MPa	206000 MPa
Yield strength	428.5 MPa	275 MPa
Hardening parameter	327 MPa	591.4 MPa
Hardening exponent	1	0.7108
Strain rate coefficient	0.00747	0.03
Ultimate strength		430 MPa

Table 1. Material properties

The properties of parts are: impact attenuator shell thickness is 1.5mm, impact attenuator beam thickness is 3mm, attachment thickness is 3mm and tube thickness is 2.5mm.

A rigid wall with a mass of 300 kg is added, it models the obstacle impacted during the test. The structure is moving with initial velocity of 7m/s. There is friction between the rigid wall and attenuator. The friction coefficient has the value of 0,2 between attenuator components and 0,4 between rigid wall and attenuator.

When analyzing the impact attenuator structure linked to the body frame, not all the body frame was modeled. Constraints were added in substitution of the rear part of car body, that was cut off, because it is not relevant for the present analysis.

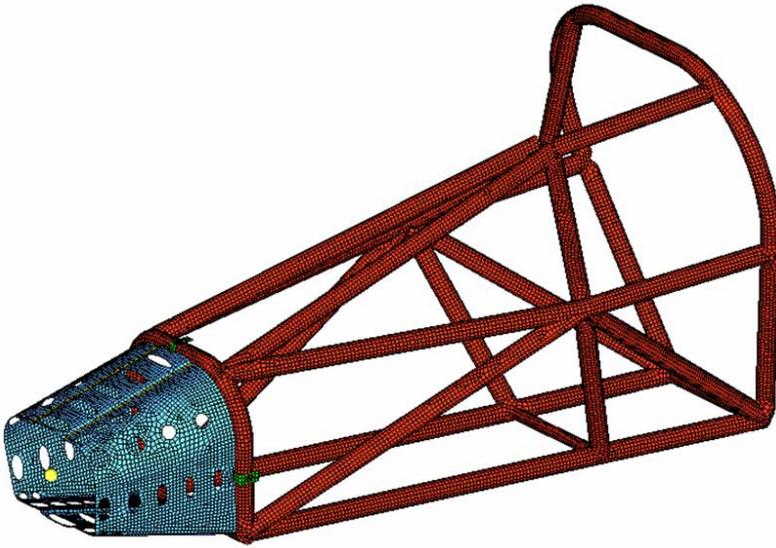


Fig. 6. Finite element model of the structure

In finite element model, there are several contacts between parts which do not allow for penetration. Figures 6 and 7 show the finite element model with mesh and constraints.

Complete model, which consists of impact attenuator (first solution), linkages and front part of the car body, has 181255 elements and 38705 nodes. We used 3 node and 4 node element types.

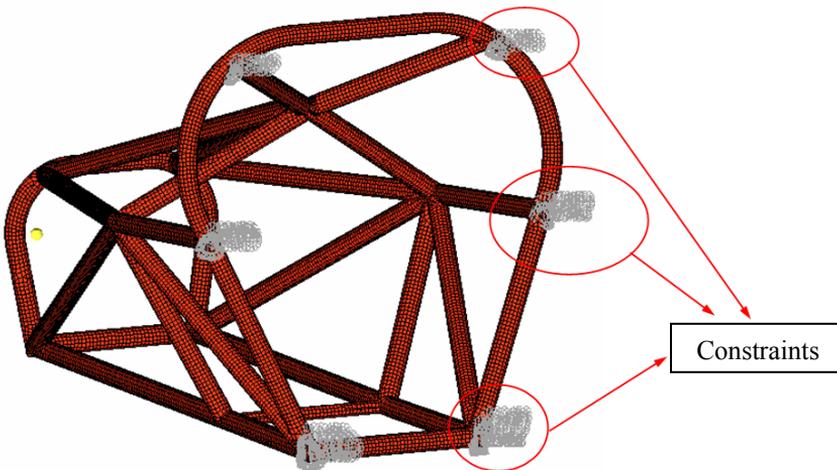


Fig.7. Finite element model with specific view on constraints

4. Results

Most of the relevant results obtained by the performed analysis will be illustrated and discussed in the following paragraphs. They were got by numerical simulation with solver Radioss v44.

First of all we will see the results obtained in the simulation of the impact attenuator alone (the first design solution) in comparison with the results obtained in the simulation of the improved solution of the impact attenuator linked to the frame structure. Results have been analyzed by means of the software code Hyper View. The duration of the simulation is 50ms. In Figs. 8 and 9 is shown the crushed shape of the two different models in two subsequent positions during the impact: in Fig. 8 we are about at 14ms of the crash event while in Fig. 9 we are at the end of the crash. On the left side are shown the results obtained when considering the impact attenuator structure alone while on the right side we see the results obtained when taking into account also part of the frame structure linked to the frontal one. Both structures have almost the same behavior and absorption of the energy in the first half of the simulation, which is presented in Fig. 8. But, at the end of the simulation, we can see in Fig. 9 that the impact attenuator linked to the frame structure is more deformed, in particular in the lateral walls. This means that the structure response becomes softer and the crushing load decreases. The difference in displacement and deformation is not such a big with respect to the independent attenuator at the end of the simulation.

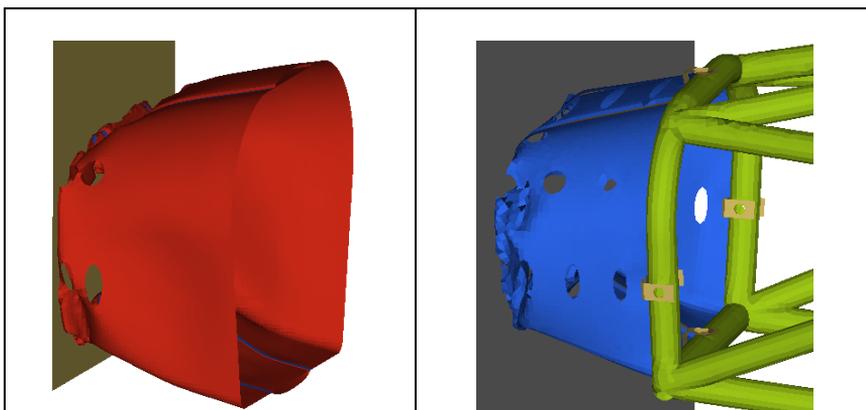


Fig. 8. The position of two structures during the impact at 14ms of crash event

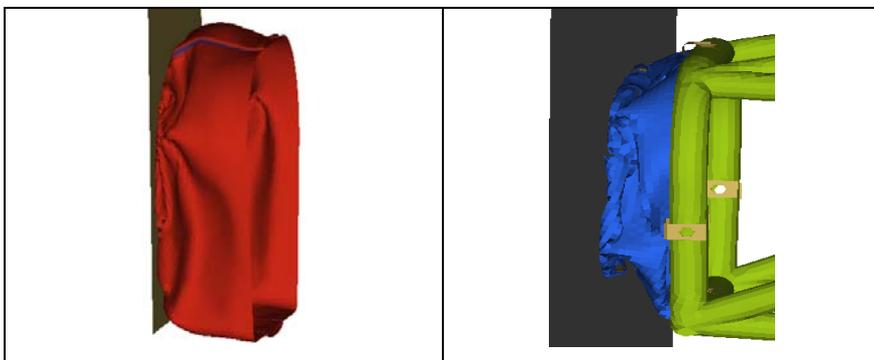


Fig. 9. The final position of two structures during the impact

The stress field is also of interest in this research. Figure 10 shows the von Mises stress field of the structures from the frontal view. The picture shows the stresses at the shell outer surfaces. The scale on the left side of the figures is presenting almost the same values of the stresses. But, the critical areas are not similar and we can see that the stress concentration is smaller when we consider the structure with links to car body tubes. Only a small part of the stress field is transmitted during the crash from attenuator to the bulkhead and the frame. It means that this design solution of the impact device is absorbing the largest part of the received energy. Links are nearly uniformly spaced on the bulkhead with which the attenuator is connected.

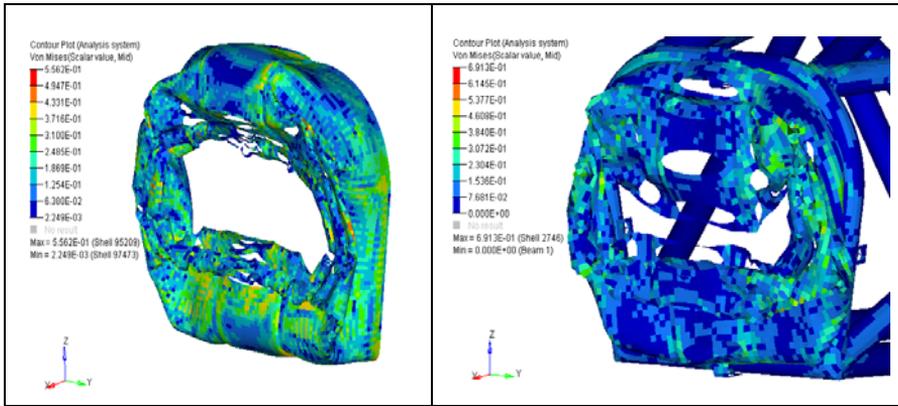


Fig. 10. Von Mises stress field of the independent impact attenuator (first design solution) and assembly structure (with improved attenuator design)

On the diagram of kinetic energy in Fig. 11 we can see responses of three different models: the impact attenuator without body frame (independent structure), the assembly which consists of attenuator (first design solution) connected with body frame by attachments (complete structure) and assembly which consists of impact attenuator (improved design solution) connected with car body (improved complete structure). The same notation of these three different designs will be used in other diagrams. It is worth of note that all curves are starting from the initial energy value of 7350 J. The behavior of the complete and independent structures during the crash impact is almost the same until the middle of the crash evolution, after which the kinetic energy is absorbed faster by the complete structure than by the alone structure. Also, the trend of the improved structure curve is almost the same as with other curves, but the kinetic energy is absorbed slowly. The zero value of the kinetic energy is reached after 28 ms of the simulation in the case of complete structure, after 37 ms of the simulation for alone structure and 44 ms when we have complete structure with improved attenuator design. This means that the improved structure has a more progressive collapse. In Fig. 12 we can see the diagram of the history of the velocity in x direction for two cases: independent attenuator structure and improved complete structure. The before mentioned results are confirmed.

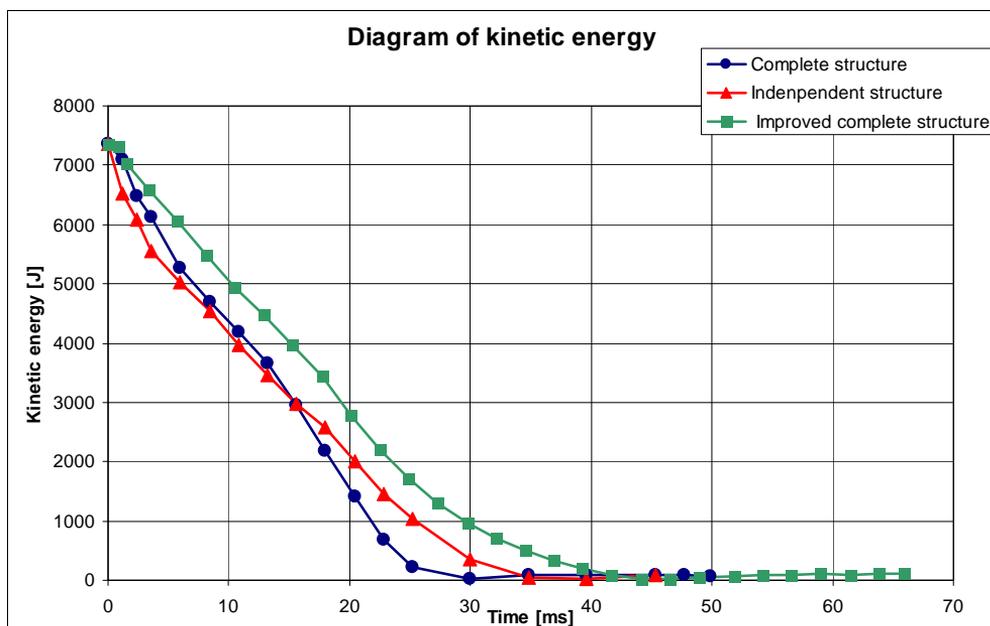


Fig. 11. Comparison of kinetic energy of three models: complete structure, complete structure with improved attenuator design and independent attenuator

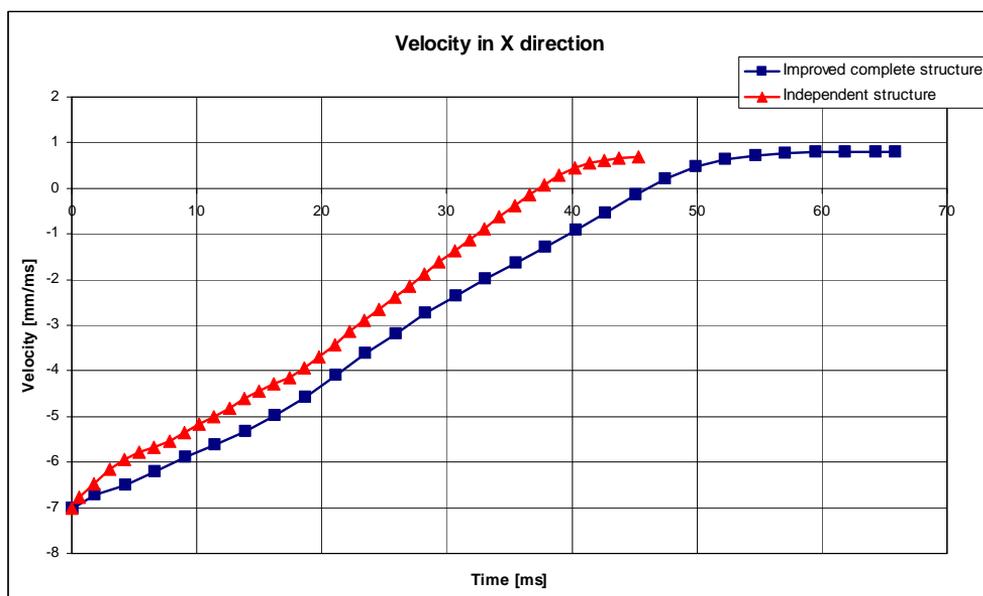


Fig. 12. Numerical results of the velocity of the improved complete structure and independent attenuator

Figure 13 shows the diagram of the evolution of deceleration during the time for the independent attenuator and for the assembly that consists of body frame and first designed impact attenuator. The first peak of the structure alone is due to the initialization of the structure collapse and has the value of $0,72 \text{ mm/ms}^2$, which is $72g$. This very high value was almost

halved in the case of complete structure, when it became 46 g. The force-displacement diagram is presented in Fig. 14. The force is directly dependent on acceleration, and that's why the behaviour is almost the same. The first maximum peak of the complete structure is about 97 kN and for independent structure 210 kN.

In both diagrams in Figs. 13 and 14 it is well visible that, a part from the initial peak, the independent impact attenuator response is quite good, being close to the optimal absorber solution. The response of the assembly (complete structure) is giving good results with respect to initial pick of the force and acceleration. Further, it is well visible that the stiffer behaviour of the complete structure in the final part of the crash event leads to higher values of the force and acceleration.

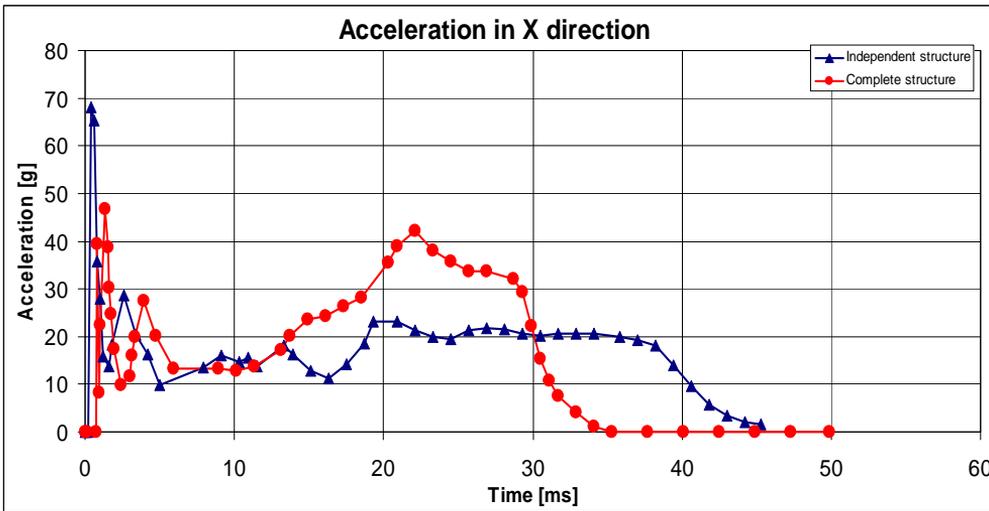


Fig. 13. Comparison of decelerations for two models: independent attenuator and complete structure

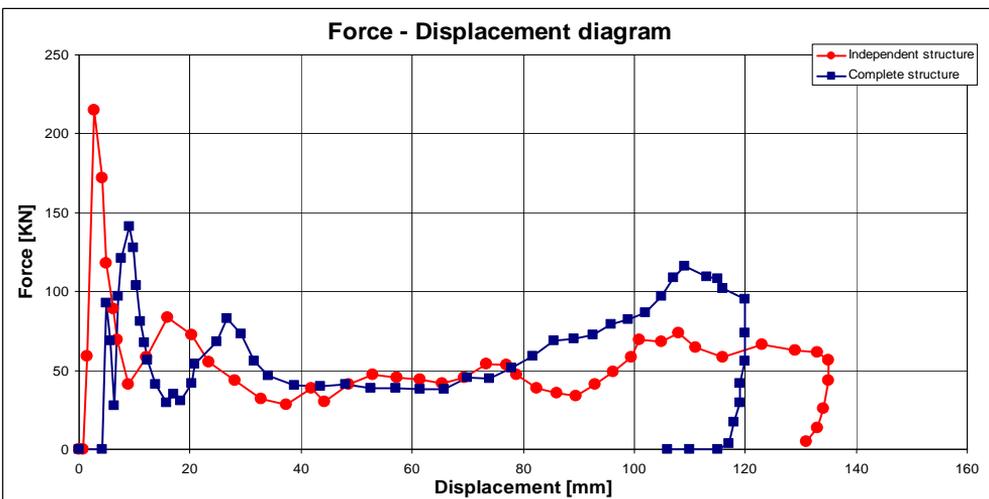


Fig. 14. Force – displacement diagram for independent attenuator and complete structure

In both cases a decrease of the initial peak values can be obtained by a proper modification of the impact attenuator structure that includes crush initialisation triggers. We further present comparison of results between independent attenuator (first solution) and assembly that consists of the improved attenuator design with additional triggers (additional holes, different position of holes, removed thin plates).

Figure 15 shows the diagram of the evolution of deceleration during the time for the independent attenuator and for the assembly consisted of body frame and improved designed impact attenuator. The initial peak of the improved complete structure acceleration is 52g (20 g less with respect to the independent attenuator solution). Between 6 and 25 ms of the simulation process, the behavior of these two structures is almost the same. It is also well visible on figure 9, because the deformation in the first part of the process is almost the same. The force displacement diagram is presented in Fig. 16. The first maximum peak for the improved complete structure is about 152 kN.

The improved structure is giving very good results, almost close to optimal. After initial peaks, we obtain a nearly flat diagram of the impact force, that means a nearly constant value of the deceleration. Also, it is recommended in SAE rules that “average deceleration of the vehicle must not exceed 20 g”. It is well visible in Fig. 15 that the average acceleration of the assembly that consists of car body and improved complete structure is about 14 g.

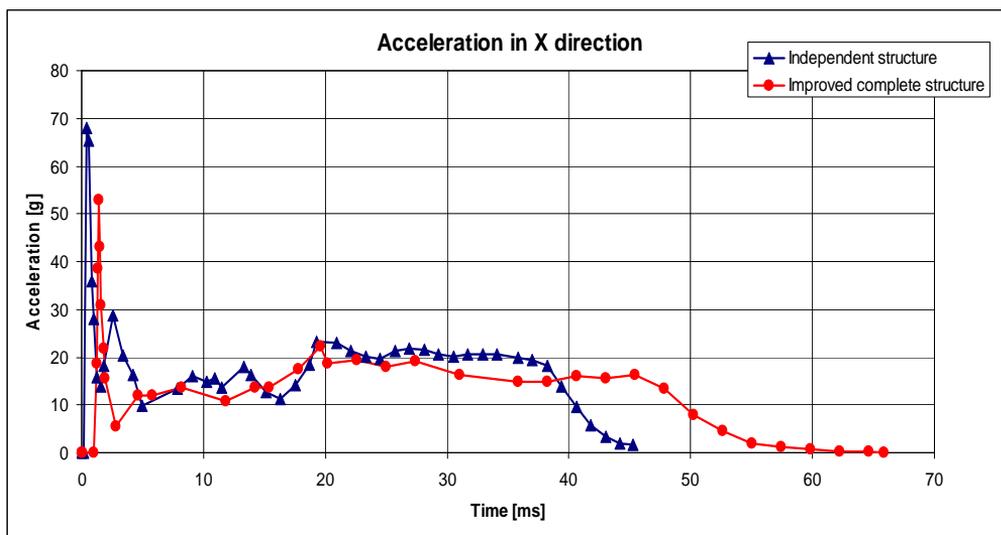


Fig. 15. Comparison of decelerations for two models: independent attenuator and improved complete structure

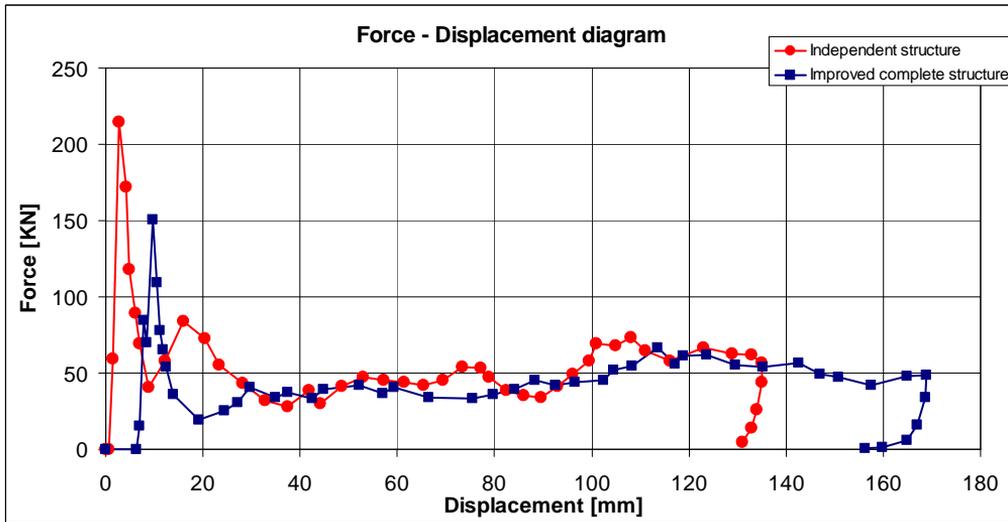


Fig. 16. Force – displacement diagram for independent attenuator and improved complete structure

5. Conclusions

In the paper is presented a detailed analysis of the crash behavior of the impact attenuator structure that was designed to equip the formula SAE car of the Politecnico di Torino racing team. The simulations performed by the finite element methods reveal some very important facts that also support the reliability of the numerical results

First of all, the design of the energy absorbing structure has to allow a progressive force evolution, avoiding the presence of force peaks (i.e. deceleration peaks). The ideal solution is a flat diagram of the resisting force vs. time.

As a second important point, the design and position of the links between the energy absorbing structure and the car frame structure are found to be essential to determine the crashworthiness of the vehicle in case of front impact.

From the analysis of the diagrams, we can conclude that the suggested solution for the energy absorbing structure and for the links is quite good from the point of view of kinetic energy absorption. Analyzing the deceleration diagrams, the influence of the frame structure on the results can be observed. Presence of a quite high first peak of the load is notable, which is due to the initialization of the structure collapse. In any case, the value of the maximum deceleration (that is in correspondence to the first peak) in the case of improved complete structure is about 0.52 m/ms^2 (52g). Also, the average deceleration of the improved complete structure is 14g. These values fit well with the requirements of the SAE 2008 rules, as they require that the adopted solution must provide an average deceleration lower than 0.2 m/ms^2 (20g) along the Y – axis. It means that crush initialization triggers improved significantly the structural response.

All these results confirm that the designed structure is very good from the point of view of frontal impact, with good absorption of the impact energy. Also, the stresses on the absorbing structure are lower than in the case of independent attenuator.

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