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Numerical and experimental analysis of an impact attenuator in sandwich material for racing application

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Abstract— **An impact attenuator is a device used to protect the vehicle from damage during a collision, thus preventing the risk of injury to the driver. Nowadays, with the increasing number of accidents that occur during the car races due to an increase of speed on the track, the pilot's safety has become a major area of research. The designer must be able to design and realize structures able to dissipate the greater amount of kinetic energy with progressive and controlled crushing, in order to avoid high deceleration peaks dangerous to humans. Given the complexity of the dynamic phenomenon and the use of new materials, time and cost of development tend to grow; these can be reduced to a large extent through the use of software dedicated to the finite element modelling of explicit type. The simulated results, however, cannot be used directly without any validation with the experimental results. The purpose of this work is to compare, numerically and experimentally, the dynamic response of a frontal impact attenuator made of sandwich panels in aluminum, properly glued and riveted in order to obtain a truncated pyramidal structure. The results of the finite element analysis, obtained through the use of the solver LS-DYNA, proved to be in good agreement with the experimental data, confirming the quality of the numerical simulation. Moreover, the designed impact attenuator met the requirements imposed by regulation, having load peaks and average deceleration during impact less than 40 g and 20 g, respectively.**

Index Terms—**Impact Attenuator, Sandwich Structure, Energy Absorption, LS-DYNA, Design.**

I. INTRODUCTION

Among the aspects to take into account during car design, crash safety is one of the most important features. Crashworthiness is defined as the capability of a vehicle structure to provide adequate protection to its passengers from injuries in case of a collision. The safety requirements are outlined by the organizers of the specific competition [1]; to meet these regulations, designers must manufacture energy absorbing systems, called impact attenuators, that are fitted in front of the survival area occupied by the driver.

Impact attenuators are generally made up of aluminum, honeycomb, CFRP composites or a combination of these materials that provides maximum protection. The deformation of such components should absorb the higher level of kinetic energy while transmitting low load uniformly to the rest of the vehicle and maintaining an acceptable level of crushing. The use of lightweight materials may contribute also to improving the acceleration performance and fuel economy of the vehicle. Before attempting to implement such impact attenuators on an actual vehicle, a component level study has to carry out in order to verify the design concept.

Many researchers have investigated the behavior of empty or foam-filled aluminum tubes or honeycomb structures under axial quasi-static and dynamic loading conditions using both experimental and numerical techniques [2-8]. According to the published literature, honeycomb structures are relatively light in weight and also absorb more energy than other combinations (i.e. empty or foam filled). However, a large block of honeycomb occupying the whole volume of the impact attenuator is a heavy and expensive option and it may be advantageous to combine the benefits of thin-walled structures and honeycombs in a single system.

Therefore, the present paper deals with experimental and numerical investigations of the behavior of a sandwich structure, with external layers in aluminum and internal aluminum honeycomb core, with a truncated pyramidal shape subjected to axial dynamic loading. The explicit finite element code, implemented in LS-DYNA, was used to simulate the collapse mode of such structure and predict the energy-absorption capability. The numerical results were compared with the corresponding experimental test data. The overall aim is to investigate whether a sandwich structure can be used for designing an impact attenuator for a racing car competition, such as the Formula SAE one; after the comparison between numerical and experimental data it was observed how the proposed configuration is able to absorb impact energy with deceleration values coherent with the specific homologation references.

II. DESIGN OF THE IMPACT ATTENUATOR

The impact attenuator is such safety structure installed on car that could absorbs energy during frontal collision by progressive deformation; failure of this system might compromise the passenger health. Such structure should be lightweight, resistant and deformable enough to limit the force and deceleration transmitted to driver; moreover it is necessary to design it according specific technical regulation. As for the FSAE rules, the impact attenuator must be

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installed in front of the bulkhead and must be at least 200 mm long, 100 mm high and 200 mm wide. It should not penetrate the front bulkhead in the event of an impact and the total deformation must be limited to the attenuator. During the impact of the vehicle, with a mass of 300 kg, against a solid and non-yielding barrier at the velocity of 7 m/s, the attenuator must give an average deceleration less than 20 g, a peak lower than 40 g and the total energy absorbed must be at least 7350 J [1].

Several studies have been carried out in developing impact attenuators. Munusamy *et al.* [2] designed an attenuator based on an array of empty and honeycomb-filled aluminum tubes. Belingardi *et al.* [9] designed and numerically simulated a crash box in aluminum with a truncated pyramidal structure with rounded edges. The design also included strategically placed holes in the skins. Zarei *et al.* [10] designed an optimum filled crash absorber design. Heimbs *et al.* [11] investigated the crash behavior of the nose cone of an F1 racing car impact structure. Obradovic *et al.* [12] showed that carbon fibre composites perform very well in crash if a careful analysis is conducted. Wang *et al.* [13] examined a composite impact attenuator with half-wave beam structure using also optimization techniques. Some FSAE team used rectangular prism made of aluminum honeycomb or steel truncated pyramid for their crash protection; some design included a hollow interior within the impact attenuator, effectively reducing the weight.

The impact attenuator analyzed in such study was made of aluminum sandwich material with an honeycomb core separating two thin skins. Honeycomb has low density and high energy absorption capability; it is ideal for impact structures where weight saving is important. The shape chosen for the crash-box is a truncated pyramid (Figure 1). The five sandwich panels are joined to each other with aluminum L-shaped profiles riveted and bonded on the skins with epoxy adhesive. Unfortunately for secrecy, it is not possible to indicate the real geometric values of the impact attenuator and the specific materials used during manufacturing.

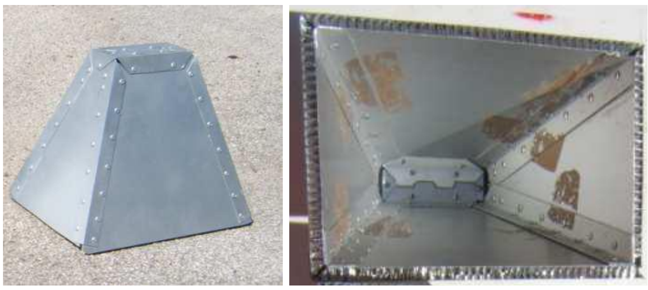


Figure 1: Impact attenuator: external and internal view.

III. NUMERICAL SIMULATIONS

The simulations have been performed modelling the impact attenuator with finite element analysis. The FE model has been implemented in LS-DYNA environment employing the material properties obtained from experimental tests, such as out-of-plane compression test, three point bending test, buckling under in-plane compression and in-plane shear test. Due to the symmetry, only half attenuator was modelled (Figure 2), in order to reduce also the computational time. This can be done only using appropriate boundary conditions on the separation nodes.

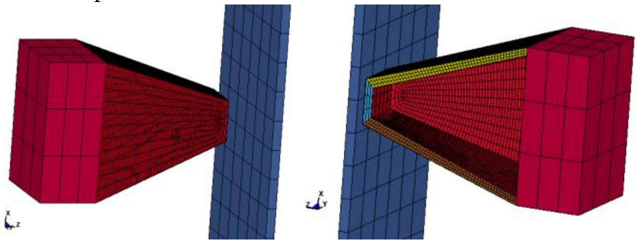


Figure 2: Numerical model of the impact attenuator.

Material type 24 (*MAT_PIECEWISE_LINEAR_PLASTICITY) of the LS-DYNA library is used for the aluminum skins. The final failure of the external skins during the impact loading is characterized by a failure criterion. Failure means a loss of stiffness and is detected when elongation of elements in any direction reaches the allowable strain. Among the different approaches available in modelling honeycomb structure, the core was simplified as an orthotropic continuum based on its effective material properties with solid elements (*MAT_HONEYCOMB). The car body and the barrier were idealized as rigid structures (*MAT_RIGID) assuming therefore that the total kinetic energy must be absorbed by the only impact attenuator.

The total system, consisting of the impact attenuator and the car body idealized as a 300 kg mass attached to it, impacts the barrier wall at the velocity of 7 m/s, as imposed by technical regulation. Such contact was modelled using a node to surface contact, while in order to prevent elements penetration of the impact attenuator during crushing a single surface contact was added. The elements representing the core are directly connected to the skin nodes at the interface using a tied break contact.

The explicit dynamic simulation using LS-DYNA can start only after fixing some control parameters, such as termination time, time-step and output requests.

IV. TESTING PROCEDURES

According to the FSAE rules and in order to evaluate the crushing capability of the designed and built impact attenuator, dynamic experimental test has been planned and performed. The test procedure is represented in Figure 3. A mass of 300 kg is dropped from a height of 2.5 m, thus getting an impact velocity of 7 m/s, as requested. A rapid unfastening system (a piston driven by pressurized air) operates the drop of the mass which is guided by two steel ropes.

The accelerometers, used to measure the mass deceleration, were placed on top of the mass. The impact attenuator was constrained by screws to a base of steel and concrete in order to reproduce the connection with the frame. In order to capture the physical phenomenon an high speed camera was used; pictures were recorded at a rate of 500 frames per second.

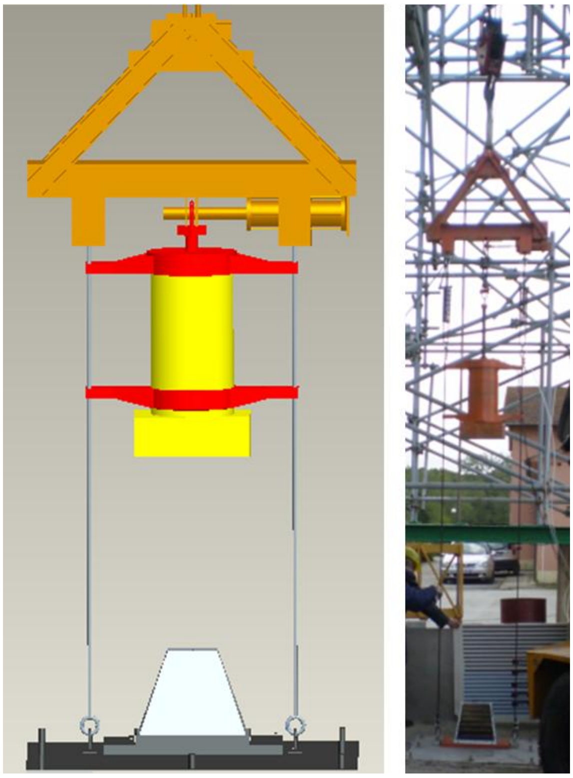
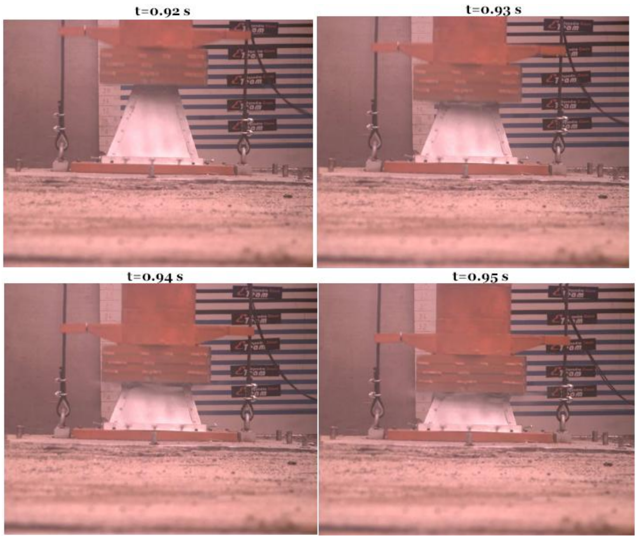


Figure 3: Testing method: schematic and real.

V. RESULTS AND DISCUSSION

Figure 4 shows the impact attenuator crushing during the real test every 0.01 seconds. From the pictures it is evident how the impact attenuator is able to absorb kinetic energy through the formation of plastic folds that start from the top of the structure and tend to close on themselves in a progressive manner during crushing.



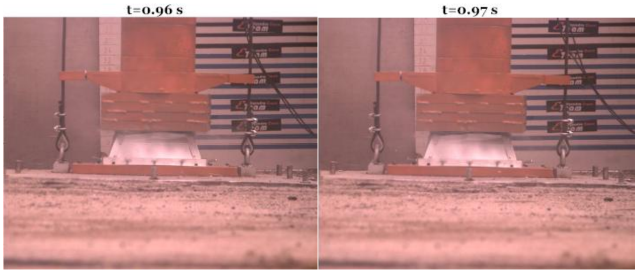


Figure 4: Frames of the impact every 1 ms (impact starts at time 0.92 s).

Figure 5 shows the structure after the test. It is evident how the part of the structure not responsible of the energy absorption remain almost perfectly intact.

Figure 5: Impact attenuator after test: external and internal view.

Figure 6 shows the acceleration raw data versus time acquired by the accelerometer during the experimental test; the values have been recorded with a sampling frequency of 5 kH. It is evident from the signal that the impact begins at about 0.922 s and ends at about 0.972 s. Since the deceleration peak of the raw data is greater than 40 g, it has been filtered with the Channel Filter Class (CFC) 60 (100 Hz), as required by the FSAE Rules. Also the filtered data are shown in Figure 6: the deceleration peak of the filtered data is about 31.1 g and the average deceleration, computed on the time interval 0.922-0.972 s, is about 14.2 g.

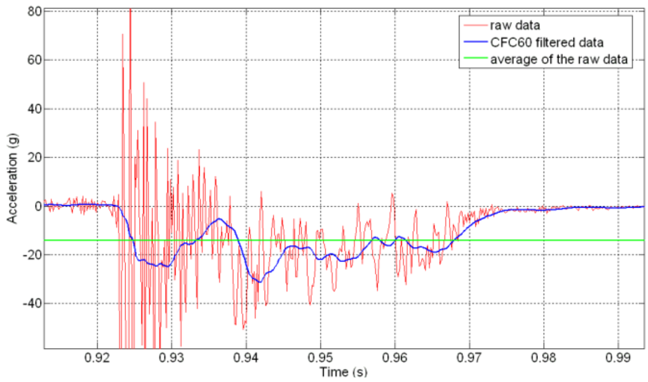


Figure 6: Acceleration vs. time: raw data, filtered data and average value.

Also from the numerical modelling it is possible to reproduce the crushing phenomenon. Figure 7 shows the numerical final deformation of the impact attenuator. It is evident the plastic folding at the top of the structure and a more intact configuration on the bottom of the same, as in real test.

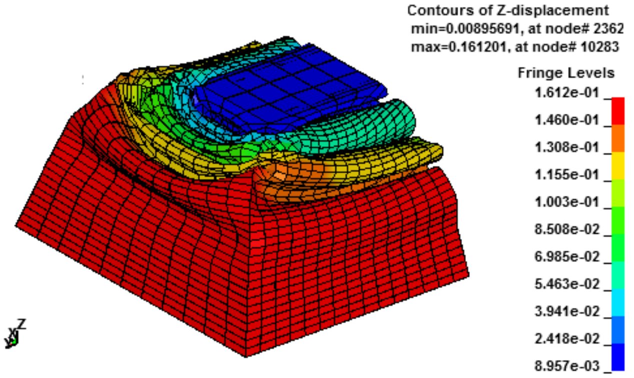


Figure 7: Final deformation of the impact attenuator from the numerical point of view

The acceleration signal versus time can be recorded during simulation thanks to output controls. Figure 8 reproduces the variation of numerical accelerations in time and it can be observed how such trend is very close to the experimental case, except for small differences. It is due to the simplifications adopted in the numerical model respect to real structure.

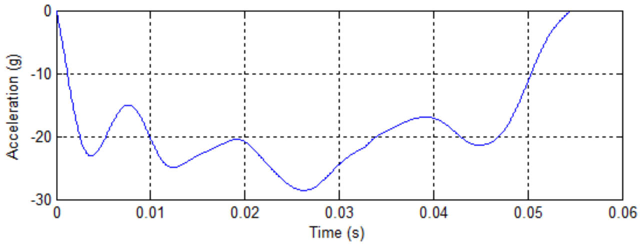


Figure 8: Acceleration signal obtained from numerical model

The following table summarizes the main results of the experimental and numerical tests for such impact attenuator with a truncated pyramidal structure.

	Experimental	Numerical
time at the beginning of impact	0.922 s	0 s
time at the end of the impact	0.972 s	0.048 s
average deceleration	14.2 g	15.9 g
peak deceleration	31.1 g	28.6 g
final length of the attenuator	0.109 m	0.114 m

Table 1: Results of the experimental and numerical tests.

It can be concluded that the designed impact attenuator is able to satisfy the specific homologation requirements of the FSAE, therefore can be used in front of racing car bulkhead during competition.

VI. CONCLUSIONS

A new impact attenuator design for a racing car based on an aluminum sandwich structure has been proposed in this paper. Experimental test and numerical simulations of the axial dynamic crushing of such truncated structure have been carried out. The following observations have been made from this study:

- the results showed that the designed impact attenuator met the criteria for energy absorption, as stated by FSAE Rules, during the physical crash testing;
- the final deformation is characterized by plastic buckling with a series of folds that proceed from the top to the bottom of the impact attenuator with a progressive crushing;
- the numerical model is able to reproduce sufficiently well the crushing phenomenon despite the simplification adopted;
- a more precise numerical representation could better copy final deformation with a consequently growth of simulation times.

Therefore, the numerical modelling procedure outlines in this paper can be used with confidence for designing the proposed lightweight impact attenuator using such thin-walled sandwich structure.

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- Crash behaviour of automotive structures using the code LS-DYNA;
- Experimental characterization of energy absorbers with different materials (metal and composite) and geometries in uniaxial loading conditions (static and dynamic).