INVESTIGATION OF THE MOST EFFICIENT SOLUTION FOR A SPECIFIC VEHICLE IMPACT ATTENUATOR IN CFRP MATERIAL

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Keywords: crashworthiness, CFRP, sandwich structure, impact design.

Abstract

Nowadays each manufacturers of racing cars must pass specific crash tests controlled by legislation, before seeing their vehicles in the market and in motor racing. In order to ensure the driver's safety in case of high-speed crashes, special impact structures must be designed to absorb the race car's kinetic energy and limit the decelerations acting on the human body. This target is achieved thanks to the synergy between the numerical and experimental analysis. Investigated in this work is the best solution, in terms of energy absorption, for a specific frontal impact attenuator in CFRP material with sandwich structure. In particular, given the geometry reported by the manufacturer, it has been necessary to find the right combination of material and lamination varying the stacking sequence and its disposition. Analyses have been conducted by the joining of numerical models, performed using LS-DYNA, with experimental tests, through an instrumented drop tower equipment.

1 Introduction

Racing cars demonstrate maximum vehicle driving performance resulting from high-tech developments in the area of lightweight materials and aerodynamic design. Since extreme racing speeds may lead to severe accidents with high amounts of energy involved, special measures are taken in order to ensure the drivers safety in case of high-speed crashes. Besides the driver's protective equipment (like helmet, harness or head and neck support device) and the circuit's safety features (like run-off areas and barriers), the race car itself is designed for crashworthiness and possesses special sacrificial impact structures that absorb the race car's kinetic energy and limit the decelerations acting on the human body [1-3].

This research work involves the examination of the front crash structure of a Formula Ford car used to protect the driver in the event of a frontal impact. At the moment every manufacturer has a different method of producing their crash structures. Recently it is preferable to make such structures in composite materials, thanks to their low density and high strength [4]. Excellent performance of composites, sometimes better than those of the metallic similar structures, can be obtained by choosing appropriately the mechanical (the stacking sequence, the number of layers, the type and quantity of fibers and matrix) and geometrical (the beam section shape, the wall thickness, the extremity joints) design parameters [5].

Each Formula Ford car, before racing on circuit, must have a FIA certification regards safety and must boast, among the safety features, a front carbon crash structure. According the

article 15.3 of the Technical Regulation for Formula Ford with 1600cc EcoBoost engine [6], that refers to the survival cell and frontal protection, an impact absorbing structure must be fitted in front of the space frame. This structure must be solidly attached to it and must pass specific static and dynamic FIA tests in the presence of an FIA/ASN technical delegate in an approved testing center. To test the attachments of the frontal impact-absorbing structure to the space frame structure, a static side load test must be performed on a vertical plane passing 400 mm in front of the front wheel axis. A constant transversal horizontal load of 30 kN must be applied to one side of the impact absorbing structure using a pad 100 mm long and 300 mm high. The center of area of the pad must pass through the plane mentioned above and the midpoint of the height of the structure at that section. After 30 seconds of load application, there must be no failure of the structure or of any attachment between the structure and the space frame. Moreover the space frame must be capable of withstanding a frontal impact test. For the purposes of this test, the total weight of the trolley and test structure must be 560 kg and the impact velocity at least 12 m/s. The resistance of the test structure must be such that during the impact the average deceleration of the trolley does not exceed 25 g. Furthermore, all structural damage must be contained within the frontal impact structure.
Design an impact attenuator able to pass imposed regulation can be achieved thanks to the synergy between the numerical and experimental analysis. Investigated in this research work is the best solution, in terms of energy absorption, for a Formula Ford frontal crashbox made of sandwich structure with skins in CFRP material and honeycomb core in aluminum. In particular, given the geometry reported by the manufacturer, it has been necessary to find the right combination of material and lamination varying the stacking sequence and its disposition. Firstly analyses have been conducted by the joining of numerical models, performed using LS-DYNA, with experimental tests, through an instrumented drop tower equipment. The first aim is, in fact, to reproduce as best as possible the crushing process with finite element code using the experimental results conducted on a preliminary impact attenuator with a lower impact energy. The devices available in the laboratory did not allow, in fact, to obtain impact energies comparable with those required by regulation. Only after having reproduced the dynamic behavior of the crashbox numerically, it is possible to change the stratigraphy of the skin to find the best configuration for homologation requirements. By performing these numerical tests is possible to have a baseline as well as a comparison between different solutions which will lead to improve driver safety in a front impact situation.

2 Geometrical and material characterization of the impact attenuator

The analyzed crashbox (Figure 1) consists of a truncated pyramidal structure with an almost rectangular section. The pyramidal structure makes it possible to obtain a major stability during progressive crushing, while the rectangular section has rounded edges to avoid stress concentrations [7]. The design of sacrificial structure has been completed with a trigger which consists in a smoothing (progressive reduction) of the wall thickness in order to reduce the resisting section locally. This trigger is intended both to reduce the value of the force peak and to initialize structure collapse in a stable way. For the preliminary impact attenuator was used three different wall thickness zones, as shown in Figure 1.
Unfortunately, for reasons of secrecy it is not possible to report the geometric dimensions and the sequence of lamination used.
The impact attenuator is manufactured with a sandwich structure, made of CFRP skins with two different prepreg and a honeycomb core in aluminum. Table 1 reports the main mechanical properties of the used materials.

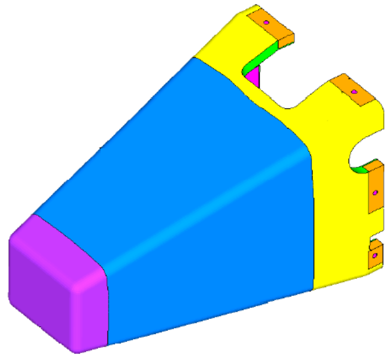


Figure 1. Impact attenuator with the preliminary stratigraphy

Property	GG200-HTA5131 Prepreg	GG630-UTS5631 Prepreg	1/8-5052-0.001 Honeycomb
Density [kg/m^3]	1400	1500	72
Young modulus in fibre direction [MPa]	45000	60000	10
Young modulus in transverse direction [MPa]	45000	60000	10
Poisson's ratio	0.04	0.06	0.05
Shear modulus in direction xy [MPa]	2500	3500	10
Shear modulus in direction xz [MPa]	3000	3000	483
Shear modulus in direction yz [MPa]	3000	3000	214
Longitudinal tensile strength [MPa]	500	800	1
Transverse tensile strength [MPa]	500	800	1
Longitudinal compressive strength [MPa]	300	500	1
Transverse compressive strength [MPa]	300	500	1
In plane shear strength [MPa]	60	50	1

Table 1. Mechanical properties of the used materials

3 Experimental test

The dynamic experimental tests were performed at the Picchio S.p.A. plant in Ancarano (TE, Italy) using a drop weight test machine with an impact mass of 413kg and an initial velocity of about 8.6m/s. During the tests every crashbox was supported at the bottom edge with four metallic angular (Figure 2). The acceleration of the mass and the velocity at impact were measured using an accelerometer with 180g full-scale and a photocell, respectively.

After the tests, the diagram representing the variation of the deceleration with the time are analyzed and filtered with a CFC60 filter (Figure 3). The filter was used to eliminate the high frequency content introduced by vibration and noise and to give a relatively smooth trace that can be replicated. Integrating the deceleration signal one and two times it has been possible to obtain a pattern of velocity and displacement in time. Moreover some important parameters for crash resistance characterization have been computed and reported in Table 2. In particular the peak deceleration is the maximum filtered deceleration, the average deceleration is obtained from the beginning of the impact to the instant when the velocity vanishes, the absorbed energy (Figure 5-b) corresponds to the area under the load-shortening diagram (Figure 5-a) and residual height is the height of the crushed impact attenuator.

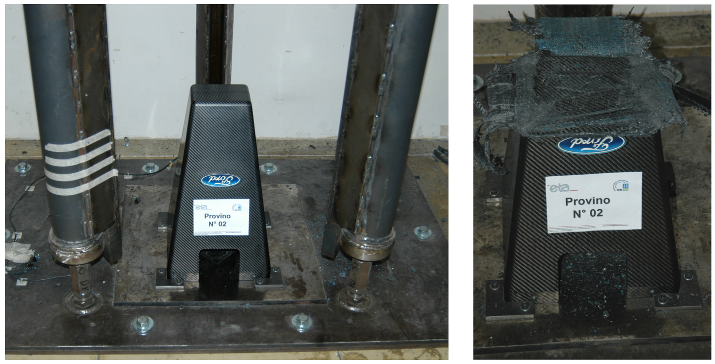


Figure 2. Crash-box before and after impact

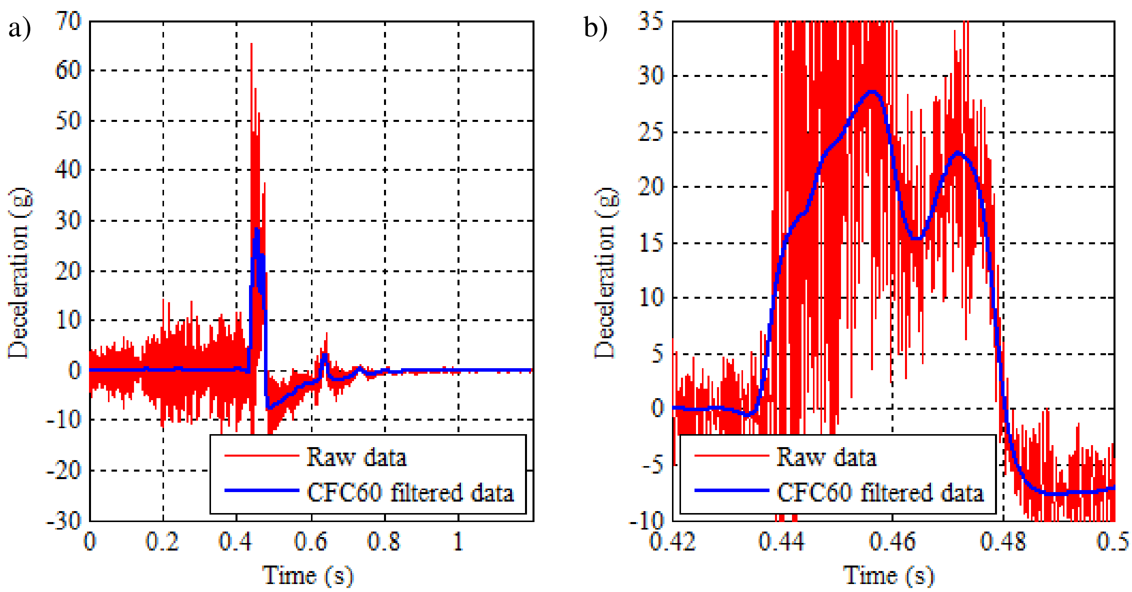


Figure 3. Deceleration vs time: a) total acquisition, b) impact phase

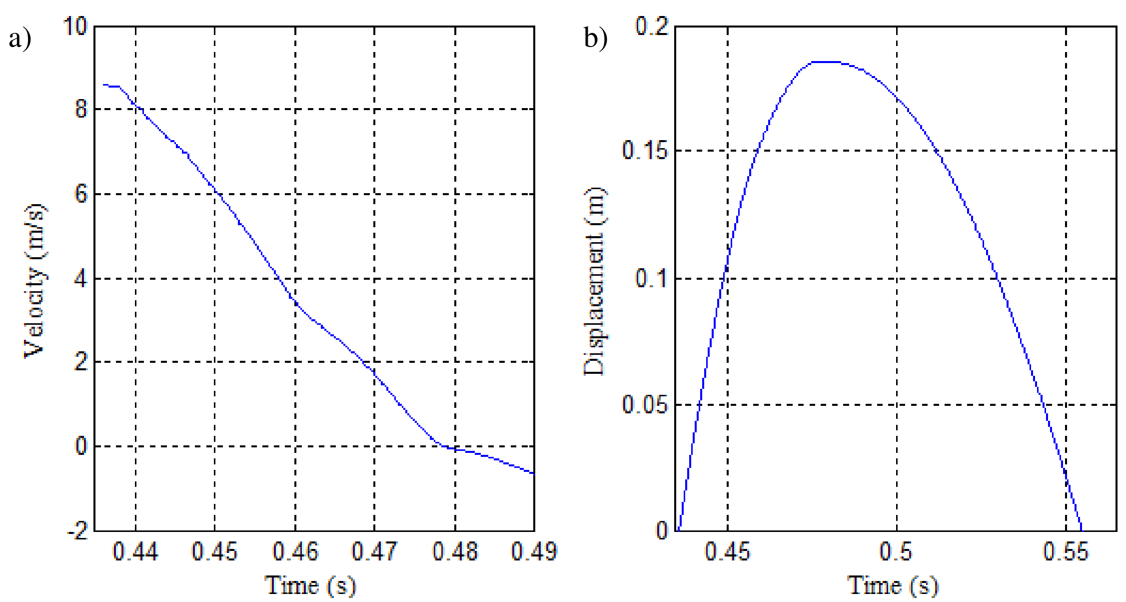


Figure 4. a) Velocity vs time, b) Displacement vs time

Peak deceleration [g]	Average deceleration [g]	Absorbed energy [J]	Residual height [m]	Impact time [s]
28.5	20.65	15290	0.315	0.0425

Table 2. Experimental crash-test results

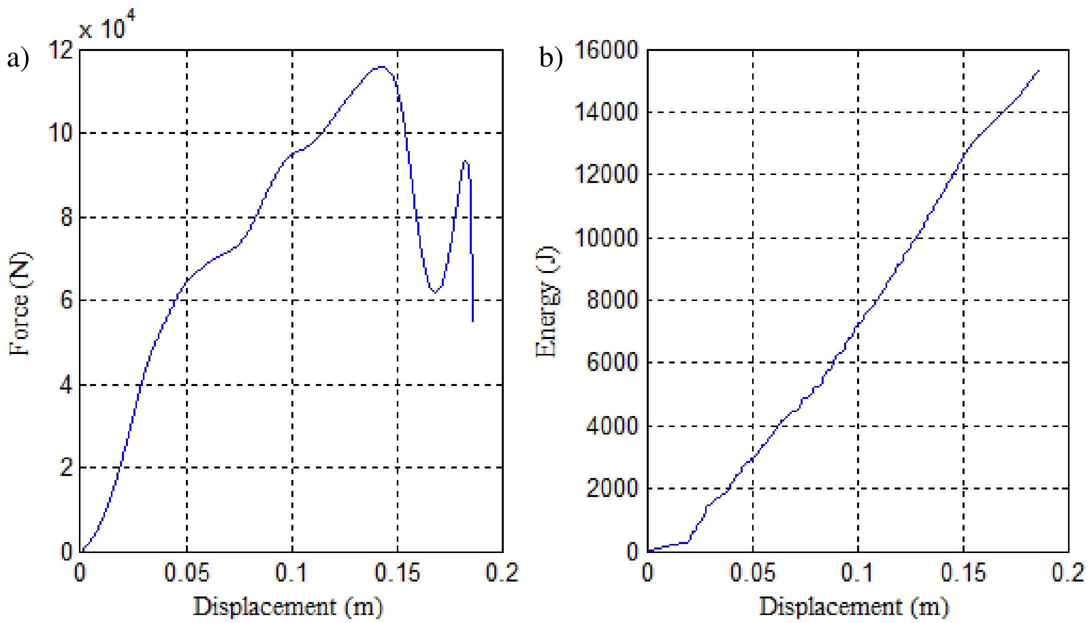


Figure 5. a) Force vs displacement, b) Energy vs displacement

4 FEA and LS-DYNA

In order to find the best configuration of crashbox able to absorb the total energy required by technical regulation, maintaining the average deceleration under the maximum permissible value, finite element analysis has been conducted using the non-linear dynamic code LS-DYNA [8]. The impact attenuator is modeled by four-node shell elements with Belytschko-Tsay formulation. A multi-layered shell is used with one integration point per layer. To define composite numerical model the card *PART_COMPOSITE is used. According to this card, the laminate has a thickness defined by the sum of each individual layer. Laminate theory is also activated with LAMSHT parameter in *CONTROL_SHELL card, to correct for the assumption of a uniform constant shear strain through the thickness of the shell. A master-surface to slave-node contact is defined between the impact surface and the nodes of the tubes. Then a self-contact algorithm, based on the penalty formulation, is defined on the impact attenuator surface to provide the friction effect between its parts during deformation and to prevent the elements penetration. Particular attention was given to the material definition of composites. The codes, in fact, contain different materials models that implement composite fabric with various failure criteria. In particular, for the LS-DYNA library the material type 55 was used for modeling impact attenuator, thanks to its ability to give a numerical behaviour near to the experimental ones [9-10]. To avoid ductile behaviour with folding, it is important to change the element strength at some point of the collapse evolution. This is obtained thanks to a time-step failure parameter (TFAIL), which defines a limit to the element effective strain. This allows to reproduce the brittle behaviour of the material and, at the same time, to control the time-step value and therefore to reduce the analysis CPU time.

The total system (Figure 6-a) is constituted by internal and external shells in composite material, honeycomb in aluminum and rigid mass as 3D elements. The FE model (Figure 6-b) has 13178 nodes and 35039 elements. For the symmetry of the impact attenuator only half geometry has been modeled, inserting the correct symmetry constraints in the cutting plane. As regards the initial conditions, two different impact energies have been set: initially the rigid moving mass have a finite mass of 206.5 kg and an initial velocity of about 8.6 m/s; after, in order to reproduce the homologation requirements, a mass of 280 kg and an initial velocity of 12 m/s have been used. Moreover the crash-box is constrained in all degree of freedom at its base for 10 mm, as dictated by the experimental evidence.

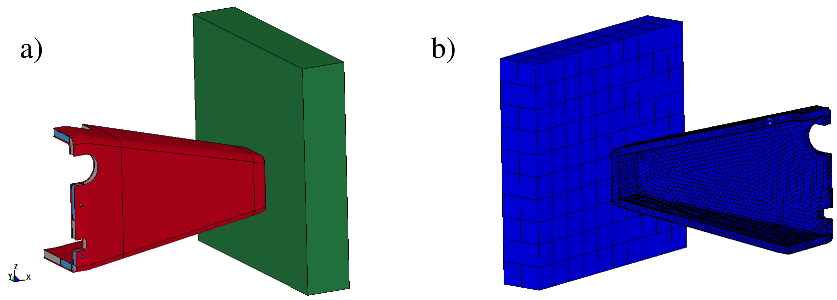


Figure 6. Total system: a) CAD model, b) FE model

5 Comparison between numerical and experimental analysis

The numerical and experimental deceleration versus shortening curves for the crashbox under the first impact conditions (total energy lower than technical regulation) are compared in Figure 7, while some crash parameters obtained from experimental test are reported with numerical results in Table 3 showing also the committed error. As the experimental data, also the numerical ones are filtered with the same filter CFC60.

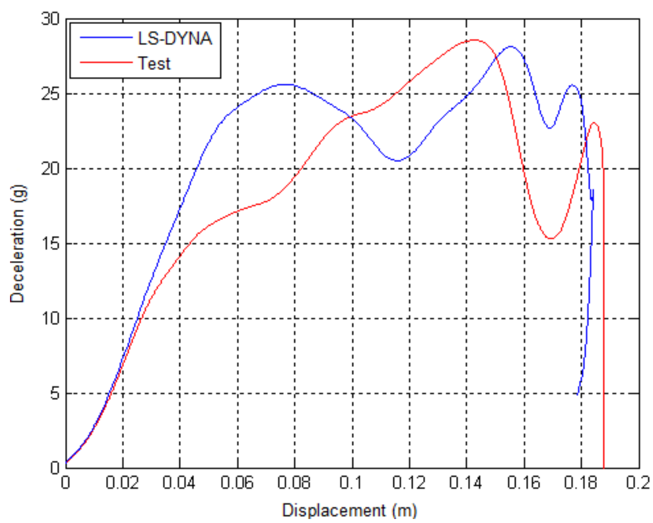


Figure 7. Deceleration vs displacement diagram for numerical and experimental test

	Test	LS-DYNA	Error %
Total crush [m]	0.185	0.182	1.6
Average deceleration [g]	20.65	21.58	4.5
Maximum deceleration [g]	28.51	28.12	1.4

Table 3. Comparison between experimental and numerical results

The difference in behavior in the early stages of impact is due to the fact that the crashbox tested had been subjected to static side push; the propagation of internal structural cracks resulting from the static test could be the main reason of the loss of stiffness. Despite this structural difference the numerical model is able to reproduce the crash behavior with errors less than 5%, also reproducing the detachment of the skin outside from the lower face of the sandwich (Figure 8).

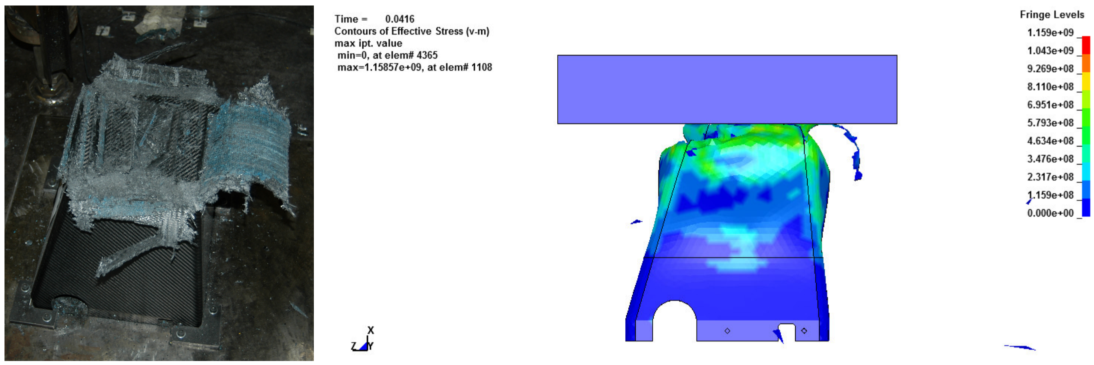


Figure 8. Comparison between experimental and numerical crush

6 Investigation on the best solution

After having obtained a good correlation between numerical and experimental results in the first case of low impact energy reached with the drop tower, the attention was focused on the impact condition imposed by technical regulation. Therefore, the same impact attenuator was numerically tested with an initial velocity of 12 m/s and a total energy of 20160 J, obtaining a total crush equal to 0.252 m, an average and maximum deceleration of 30.58 g and 42.14, respectively. Since it is not able to overcome the homologation test, it was decided to modify the stacking sequence and disposition of the CFRP skins. Figure 9 reports the different versions of the impact attenuator taken into account: versions 2 and 6 analyze the crush behavior using four or six subdivision zones with different thickness, version 3 considers the crashbox made of the same lamination without thickness change, version 4 reduce the gap between the thinner laminations in the top of the crashbox, version 5 takes instead into account an oblique lamination into the second zone.

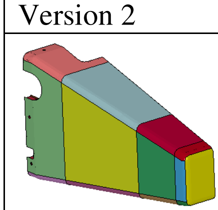
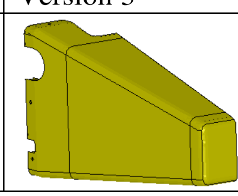
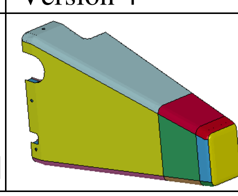
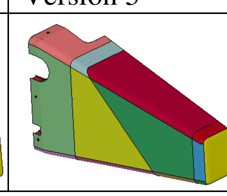
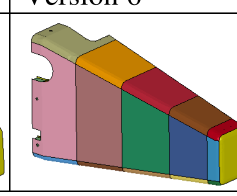
Version 2	Version 3	Version 4	Version 5	Version 6
				

Figure 9. Different versions of impact attenuators

	Version 2	Version 3	Version 4	Version 5	Version 6
Total crush [m]	0.335	0.270	0.295	0.329	0.357
Average deceleration [g]	27.13	24.96	25.41	28.45	24.46
Maximum deceleration [g]	61.72	32.25	37.79	50.43	39.87

Table 4. Comparison among different stratigraphy version

The numerical results can be summarized in term of total crush, average and maximum deceleration, as reported in Table 4. The third and the sixth version seem to have a better behavior in term of energy absorption; the growing progression in the lamination thickness is not so needful, because the geometry in itself leads to an increase of the section with the advance of the crushing. Modifying the orientation in the disposition of the laminate is not advantageous and it is also complex from the point of view of practical realization.

7 Conclusions

The present research work describes the numerical and experimental investigation on the energy absorbing capability of a specific frontal impact attenuator, for a Formula Ford racing car, made of a CFRP sandwich structure. Initially, to calibrate the numerical model, experimental crash tests were conducted using a drop tower suitably instrumented with an impact energy lower than that required by the technical regulations. Only after having obtained a good correlation between numerical and experimental data, the same impact attenuator was numerically tested under real conditions showing a crush behavior unable to pass the homologation. Then it was found the best compromise regarding the lamination thickness and its disposition, without changing the initial geometry, in order to obtain an impact attenuator able to verify the real test conditions.

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