Design of Attenuator for FSAE Car for Improved Impact Performance

Bidkar R.R.[1], Patole P.B.[2]

1Student, Department of Mechanical Engineering, Bharati Vidyapeeth's College of Engineering, Kolhapur, Maharashtra

2Professor, Department of Mechanical Engineering, Bharati Vidyapeeth's College of Engineering, Kolhapur, Maharashtra

Abstract: *The objective of this paper is focused on detailed analysis of the crash behavior of the Impact attenuator structure that was designed to equip the formula SAE car is presented. The design of the energy absorbing structure has to allow a progressive force evolution, avoiding force peaks (i.e. deceleration peaks). It is very important to design impact attenuators in order to protect the driver from any serious wound, in case of any mishap. The impact attenuator serves dual purpose, it protects the race car structures as well as the driver. It absorbs the crash energy in a controlled manner, thus offering the required protection. So, the goal of crashworthiness is an optimized vehicle structure that can be absorbed the crash energy by a controlled vehicle deformations while maintaining adequate space so that the residual crash energy can be managed by the restraint systems to minimize crash loads transfer to the vehicle occupants. This paper is dealing with the study of different material for impact attenuator using honeycomb structured sandwich panel and its analysis by using Hypermesh and explicit solver LS-DYNA. The results of this paper show that the impact attenuator absorbs the total kinetic energy at the time of collision.*

***Keyword**: Crashworthiness, Formula SAE, Honeycomb structures, Impact attenuator, LS-DYNA.*

I .Introduction

Automobile industry has progressed through different phases. As a part of this progression since 1950's, Motor sports and Auto racing are the most famous sports in the world. Despite of being a dangerous sport, a lot of people get attracted towards it. Many drivers have lost their lives in the fatal crashes occurring during these sports. Racing cars may roll over the track causing the car to be shattered, which is one of the clichéd images at any car racing accident.

Formula 1 motorsport is a platform for maximum race car driving performance resulting from high-tech developments in the area of lightweight materials and aerodynamic design. In order to ensure the driver's safety in case of high-speed crashes, special impact structures were designed to absorb the race car's kinetic energy and limit the decelerations acting on the human body [1]. The impact attenuator serves dual purpose - it protects the race car structures as well as the driver. It absorbs the crash energy in a controlled manner, thus offering the required protection.Crash tests and numerical simulations carried out for designing the safety elements in a car.

A sandwich construction provides excellent structural efficiency, with high ratio of strength to weight. Other advantages offered by sandwich construction are elimination of welding, superior insulating qualities and design versatility. Even if the concept of sandwich construction is not very new, it has primarily been adopted for non-strength part of structures in the last decade [2]. Sandwich structures were widely used in many important engineering to resist the blast loading however, very few investigations have been carried out to study the dynamic mechanical response and energy absorption of sandwich plate with asymmetric face sheet [3]. Honeycomb structures are natural or man-made structures that have the geometry of a honeycomb to allow the minimization of the amount of used material to reach minimal weight and minimal material cost. There are different types of honeycomb core structures like square, hexagonal, tetrahedral, pyramidal [4].

The mechanical behavior of aluminum hexagonal honeycombs subjected to out-of-plane dynamic indentation and compression loads has been investigated numerically using LS-DYNA [1, 2, 5]. The aim of present research paper is to simulate the race car to find the stresses and deformation of front buckle head and the design of honeycomb structured panels which can be used in FSAE vehicle as impact attenuator.

1. Experimental Procedure

1.1 Study of different types of metallic honeycomb structures

Honeycombs can be categorized as asymmetrical, symmetrical, reinforced and curved, according to geometrical parameters and cell connectivity. Initial methods which were developed by researchers cannot be used for symmetrical, curved and reinforcement type honeycombs. Flexible, having a symmetrical configuration were studied and analyzed to obtain the crush strength equation. In asymmetrical honeycomb configuration, symmetry along the cell connectivity does not exist. It is shown below in Fig.1. Whereas In symmetrical honeycomb configuration, symmetry exists along the cell connectivity, as shown in Fig.2.

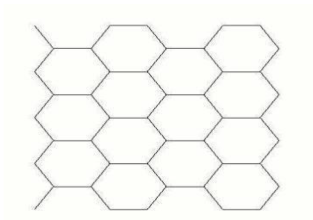


Figure 1: Regular hexagonal honeycomb

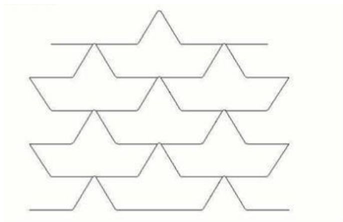
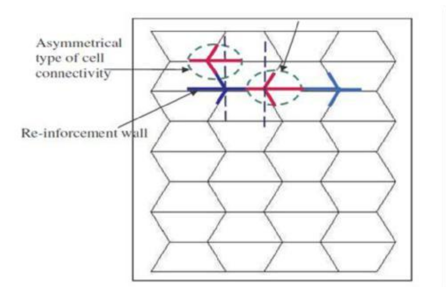


Figure 2: Form grid

In reinforced honeycomb configuration, honeycombs are reinforced with walls, in order to increase the crush properties, as shown in Fig. 3. In case of energy absorption of metallic honeycomb Three different axis are used to evaluate the strength of the honeycomb T-direction - which represents thickness or cell depth, L-ribbon direction, W-transverse direction, as shown in Fig.4.



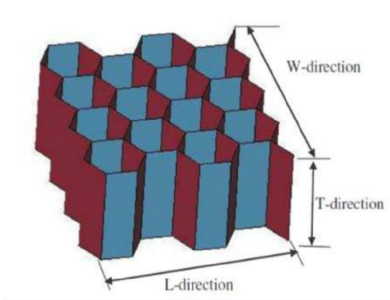


Figure 3: Half hexagonal (reinforced configuration) Figure 4: Cell structure of a honeycomb In-plane properties and out-of plane properties defines the strength characteristics of the honeycomb. In-plane properties obtained when the load is applied in L and W axis. In case of out-of plane properties, they are obtained when the load is applied in out-of-plane direction, which is the effective buckling or crushing axis. The crush strength obtained by the in-plane axes is less compared to the out-of-plane axis. E.g. Energy

absorption, which is higher if the honeycomb is compressed in T-direction, which is out-of-plane direction. The crush strength of a honeycomb can be evaluated by crushing it between a rigid base and a punch at quasi-static velocity. The schematic diagram of the same is shown in Fig.5.

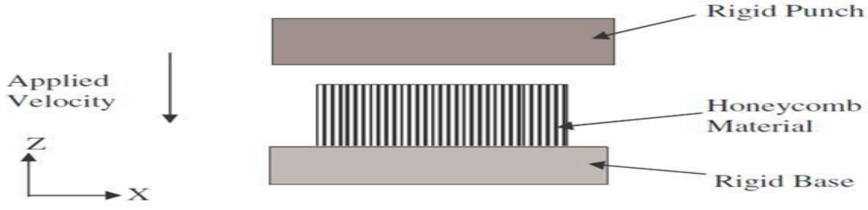


Figure 5: schematic diagram of honeycomb crushed in out-of-plane direction

The Load-deflection curve, when the honeycomb is crushed in an out-of-plane direction can be represented through Fig.6.Peak load occurs when the bonds between the inter-connected cells break. A high strength-to-weight ratio is observed in case of honeycombs which absorb higher energy with a lower material weight. Buckling of honeycomb is uniform, which can be seen from the mean crush load curve, which represents the average force being absorbed by the honeycomb during the crush. These features improve the cushioning effect during impact crush. Honeycomb crushing progresses until it reaches the compaction phase, known as densification region. Beyond this, there is a drastic increase in the load due to locking up of cells.

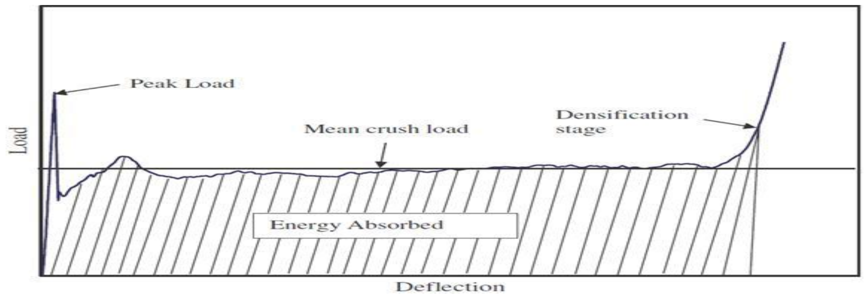


Figure 6: schematic diagram of load deflection curve of honeycomb crushed in out-of-plane direction

1.2 Study of impact attenuator requirements as per FSAE rules

According to the FSAE rules, the impact attenuator must be:

A) Installed in the forward of front bulkhead.

B) Minimum 7.8 in or 200 mm long. Its length should be oriented along the fore/aft axis of the frame. At least 3.9 in or 100 mm high, along with 7.8 in or 200 mm width for a minimum distance of 7.8 in or 200 mm ahead of the front bulkhead such that it does not penetrate the front bulkhead in case of an impact or crash.

C) In case the attenuator is honeycomb or foam filled, a 0.060 in or 1.5 mm solid steel or 0.157 in or 4.0 mm solid aluminum metal plate must be assembled with the impact attenuator. The size of metal plate and front bulkhead must be same and both must be bolted or welded together, attached directly and firmly to the front bulkhead.

D) The attenuator must not be a part of non-structural bodywork. This attachment must be built to provide a suitable load path for both transverse as well as vertical loads in case of off-centre and off-axis impacts.

G) In case it is not an integral attachment with the frame, i.e. welded, a minimum of four 8 mm grade 8.8 (5/16 in grade 5) bolts must fasten the impact attenuator and front bulkhead. Such that, when mounted on the front of car (with a total mass of 300 kg) and run into a non-yielding, solid impact barrier (with impact velocity of 7 m/s), would give an average deceleration of the car not exceeding 20g (where g = 9.8 m/s2).

1.3 Preparation of test model

The preparation of test model and simulation processes to be carried out on the test model are separated as per the use of the software for the test, which is shown in following Fig.7. The simulation test will follow the same to get the result.

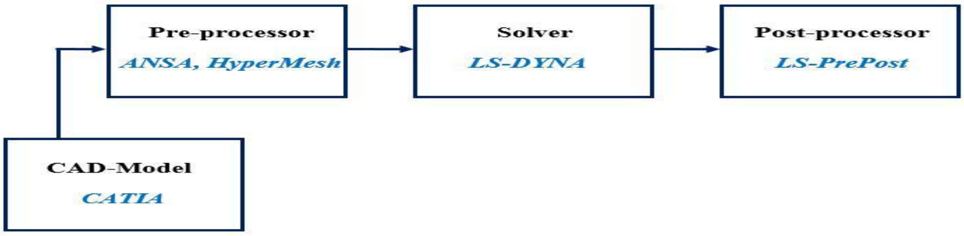


Figure 7: Flow diagram of the simulation test process.

Initial CAD was generated from CATIA and merged into the Hypermesh as .iges file as shown in Fig.8.

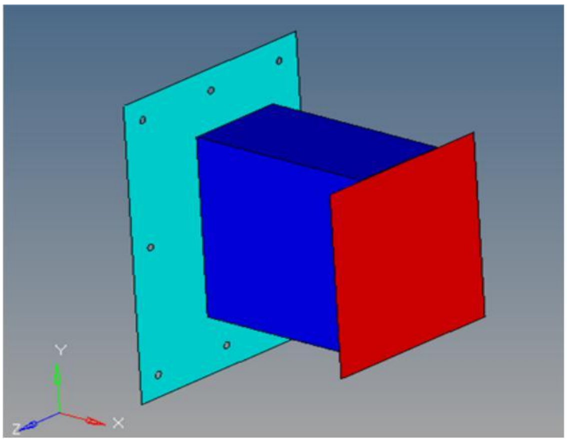


Figure 8: Initial CAD input into Hypermesh

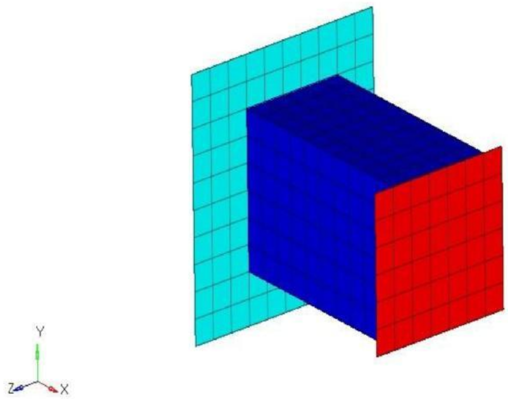


Figure 9: Mesh model in Hypermesh

As shown in figure, it is initial design fulfilling the FSAE rules (regarding dimensions). After taking this CAD into the Hypermesh, it was first meshed with shell elements at mid surface of the CAD as per the given criteria. After meshing the model it will appear as shown in Fig.9.

The mesh layout and its quality is dependent upon the topology of the surface. Small features on its surface geometry controls the size as well as quality of the elements. Small radii or holes, below the defined limits, also needs to be removed. Otherwise, it may degrade the quality of the mesh. Beads and ribs, below a certain defined criteria, can also be removed. Overall, the mesh should be throughout uniform. Any transition occurring from a coarse mesh to a fine mesh, should always be done smoothly.

After completion of mesh model all parts were assigned aluminum material i.e. A5052-H111 for Impact attenuator parts and A6082-T6 for plates. Elastic-plastic materials parts are modelled using this material card, which has a defined stress versus strain curve. Aluminum and steel parts like alloy steel AISI 4340 under

MQL mode with Nano fluid Co-relation Coefficient can be used [6]. This material card. Later, different control cards and load set up cards are given in Hypermesh for the impact attenuator such as, material card, rigid card, constraints card, contact card, load and boundary condition card and initial velocity generation card. When all control cards are applied then we export that Hypermesh file in .k format, which will be input file for the LS-DYNA solver to perform explicit simulation by using numerical method calculations. Shown in Fig.10 as an user interface for LS-DYNA

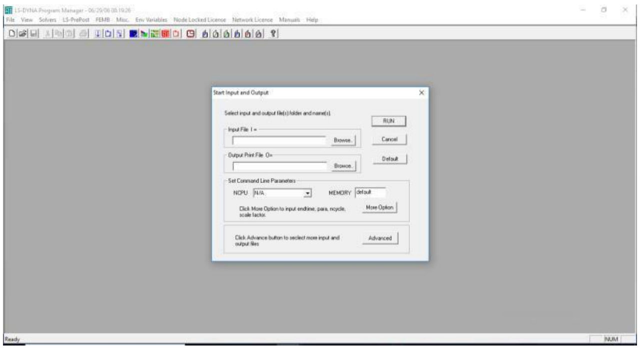


Figure 10: LS-DYNA interface

II. Result and Discussion

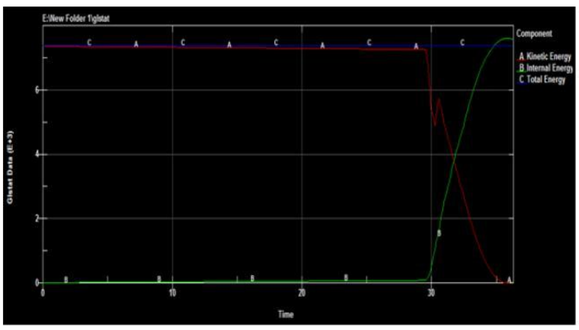
In this paper with the help of simulation software, different types of honeycomb structures were studied and desirable structure is selected for the impact attenuator bucklehead. The results which we got from the simulation test need to compare with the hand calculations done as per the criteria governed by FSAE for the front bucklehead.

In post-processing, first checked the Energy plots which consists of internal, kinetic, sliding and hourglass energy. Fig.11 shows the energy plots for first iteration. As shown in Fig.11, we can see that kinetic energy is reducing as this energy was absorbed by the structures and is converted to internal energy. We know that,

Kinetic Energy = $_2{}^1$ * $\quad {}^{*}{}_{2}$

Where, =mass of SAE car, = velocity of the vehicle so, Kinetic Energy = $_2{}^1$ *302.018 *7^2=7339.441 kJ

This kinetic energy is depicted in Fig.11. From figure, we can confirm that kinetic energy at zero millisecond is equal to the theoretical value as calculated.



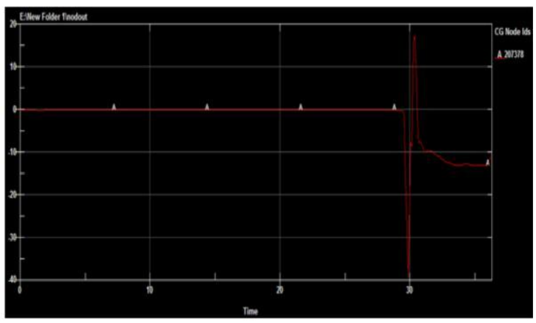


Figure 11: Energy plot

Figure12: Acceleration in g's vs. Time

The animation of the impact attenuator at 0, 15 and 30 ms. from Fig.12, it is clear that deceleration is very low as structure is having very less stiff. So need to make the structure still stiffer.

III. Conclusion

The following conclusions can be drawn from the simulation result obtained and compared with the mathematical calculation.

1. By using aluminum honeycomb sandwich panel we can improve the impact resistance of the front attenuator. This is analytically validated from the Fig.11 and Fig.12. And it is within limit of the FSAE regulations for the vehicles.

2. As in common practice we use steel sheet as the impact attenuator for FSAE vehicle which increase its weight, aluminum honeycomb sandwich panel can be a better option for reducing overall weight of the vehicle.

3. Honeycomb structure have good capability of absorbing load and transferring the same load in lateral direction as well, it will be advantageous in case of crash where any mishap can be avoided.

4. We can implement the same technique for the other panel of the other racing vehicle and for passenger vehicle as well, on the basis of the result outcome in the actual physical tests which will be future development of this research work.

5. Major disadvantage of the honeycomb structured panels is the cost of the vehicle will be increased, which is not a desirable condition in case of small vehicles.

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Vijay Ram Ghorpade Digitally signed by Vijay Ram Ghorpade Date: 2023.04.19 10:34:42 +05'30'