Design and Analysis of Impact Attenuator for UiTM Formula SAE Car 2016

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ABSTRACT

Formula SAE is an international student design competition which is held in various parts of the world that challenges students to conceive, design, fabricate and test a single seater race car. Teams are evaluated based on business and technical presentations and dynamic performance of their cars. One of the important requirements of the Formula SAE car is its impact attenuator. It must be designed to absorb the specified impact energy. It is placed at the front of the car to protect the driver in the event of frontal collision. A new impact attenuator design which consists of aluminium circular tubes and L-sections arranged in a specified configuration is proposed. The attenuator is subjected to static and impact loads to determine its initial peak force (IPF), crush force efficiency (CFE) and specific energy absorption (SEA). Simulation result for static loading is validated by experiment. Minimum absorbed impact energy of 7350 J with peak acceleration of less than 40g and mean deceleration of 20g as specified in Formula SAE regulations is achieved. Impact simulation result also confirms that the impact attenuator is able to fulfil the requirements.

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Keywords: *Impact analysis, Formula SAE, stacked tubes, impact attenuator, energy absorption.*

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Introduction

Widespread occurrence of crash incidents globally resulted in the highest number of fatality [1]. Structural crashworthiness is important to ensure safety of occupants in crash events. Impact attenuator is a device that is installed on structures to absorb impact energy. It was first invented by an Austrian Mercedes engineer in 1937 [2]. The device decelerated the vehicle by absorbing the impact energy through its crumple zone during the crash. Various impact attenuators and tubular structures subjected to static and dynamic loadings were studied in the last few decades. They include frusta [3], buckling initiators [4], slotted and double wall tubes [5, 6], multi-cornered tubes [7], bi-structures design [8], corrugated tubes [9] and patterned tubes [10]. The aim of the study is to determine the crush performance of the structures. The knowledge gained can be use for designing structures such as drop cargo, sports equipment, machineries and sensitive goods packaging.

Formula SAE car requires an impact attenuator that must be designed and installed in accordance to Formula SAE rules and regulation [11]. The impact attenuator must be able absorb a minimum impact energy of 7350 J and provides a peak and mean decelerations of 40g and 20g respectively. Worcester Polytechnic Institute (WPI's) Formula SAE team [12] tested two types of attenuator design and the honeycomb-foam combination was proven to be the more superior design. G. Belingardi and J. Obradović [13] carried out crash analysis of composite and aluminium attenuators. They concluded the composite attenuator as the more suitable sacrificial structure for a racing car. Meanwhile, S. Boria and G. Forasassi [14] analysed a new impact attenuator design which was constructed from combination of honeycomb sandwich panels and aluminium sheets. The design fulfilled the average acceleration of 20g as required by rules. Figure 1 is the example of impact attenuator that is installed in Formula SAE car.

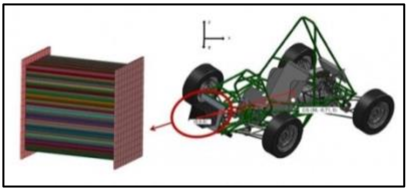


Figure 1: Example of impact attenuator [15]

A. Ghani et al. proposed a novel design of impact attenuator for UiTM single seater "Eco Challenge Car" [16]. It comprised of stacked toroidal tubes with a central tube. Recently, computer simulation has been widely used to design and analyse the structure as an effective way to reduce the time and cost of fabrication.

K.N. Anyfantis [17] predicted the response of expanded aluminium sheets joined by spot welding as impact attenuator structure using LY-DYNA software. It was shown that the structure has good energy absorption due to its uniform force-displacement curves. Conical frustum tube has been studied by A. Oshinibosi [18] to find the best combination and desired performance criteria by using surrogate method. The result showed that the conical frustum shell-type has high specific energy absorption and the impact force was minimized during collision.

Generally during impact, high initial force was achieved during the beginning of deformation. The force dropped as the structure progressively buckled, the car decelerated and kinetic energy was dissipated. Energy absorption can be maximized with the laterally arranged tubes constrained so that more plastic hinges can be produced [19]. Z. Fan et al. conducted dynamic lateral crush on empty and sandwich tubes [20] where the sandwich tubes exhibited good energy absorption. MA Yahaya et al. [21] carried out experimental study of aluminium honeycomb sandwich panels subjected to foam projectile impact and found out that it has good impact energy absorption.

There are endless possibilities of impact attenuator design yet to be explored. As customer demand for safety increases and environmental issues have become more important, designers and researchers need to find new ways to address these demands.

Design Configuration

Based on the stacked tubes design from previous study where a combination of axial and lateral crushing of tubular structures resulted in excellent energy absorption, a new design of impact attenuator is proposed as shown in Figure 2(a). This design consists of circular aluminium tubes arranged in lateral direction and closely-packed. There are five layers with each layer having four tubes. Each layer of tubes is arranged in an orthogonal direction to one another. An L-section is placed at each corner of the stacked tubes. Thin strips are used to further secure the stacked tubes in their positions. The combination of circular tubes and L-sections loaded in lateral and axial directions is expected to improve the crush performance. Figure 2(b) shows the dimension of the impact attenuator design.

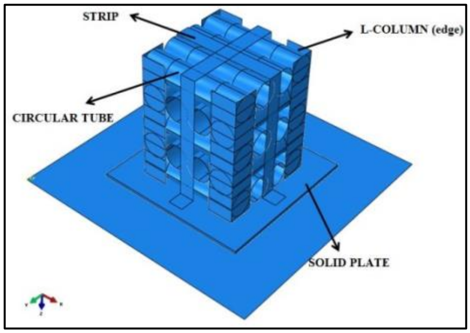


Figure 2 (a): Isometric view of impact attenuator for UiTM FSAE car

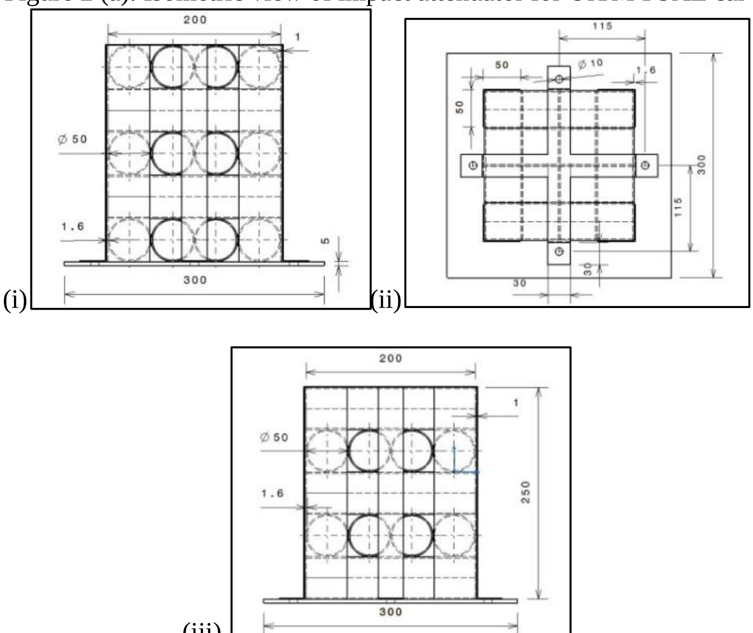


Figure 2 (b): Dimension of the impact attenuator; (i) Front view, (ii) Top view and (iii) Left view

Simulation and Experimental Test of Impact Attenuator

Simulation using ABAQUS Explicit was done to determine the crushing behaviour of the proposed impact attenuator. The material properties are given in Table 1 and were obtained from tensile test, carried out in accordance to ASTM E8 standards. Impact attenuator is subjected to axial static and impact loading.

In static loading, the impact attenuator was compressed by 70% of its original height so that effective deformation of the structure can be achieved. Coefficient of friction for all contacting surfaces is set at 0.25. For impact analysis, mass of 300 kg (weight of the Formula SAE car) and velocity of 8.3 m/s are assigned to the top plate. Types of element and number of nodes for the model are shown in Figure 3.

Static experiment is conducted using an INSTRON 3382 Universal Testing Machine. Simulation result is validated by experimental results.

Table 1: Properties of Aluminium Alloy AA6063-T5

Density	*2700 kg/m3*
Ultimate tensile strength (UTS)	*220 MPa*
Yield strength	*180 MPa*
Young's Modulus	*65 GPa*
Poisson's Ratio	*0.3*
Plastic strain at UTS	*0.1*

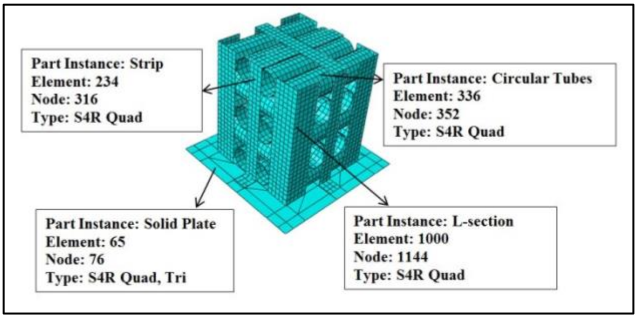
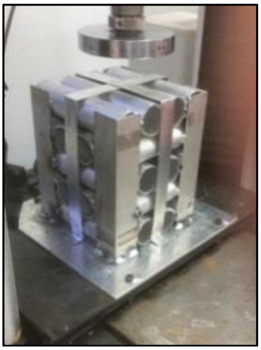


Figure 3: Finite element assembly of the impact attenuator.

Impact Attenuator Fabrication

Figure 4(a) shows the image of fabricated impact attenuator on the universal testing machine. Circular tubes and L-sections were cut, arranged and welded together. Aluminium strips were bent and riveted on the base plate. Figure 4(b) shows the image of impact attenuator being assembled at the front of UiTM Formula SAE car. As stated by the regulation, a minimum of six M8 bolts are required to mount the attenuator on to the car.



(a)

(b)

Figure 4: (a) Fabricated impact attenuator, (b) Impact attenuator mounted on the Formula SAE car

Results and Discussions

Quasi-Static Analysis and Validation

Figure 5 shows the simulation and experimental force-displacement curves of the impact attenuator subjected to quasi-static loading. Both results showed good agreement where high initial peak forces occurred at the beginning of the crush. The IPFs from simulation and experiment differed by 11.9%. This was followed by slightly lower mean forces. As the structure is further compressed, there are increases of forces due to tubes and L-section locking with each other. Experimental result shows a more gradual increase of force and slightly lower maximum force compared to the simulation. In experiment, this was due to movement of the tubes when load is applied whereas in the simulation, all tubes are fully locked into positions. Circular tubes are flattened and the L-sections deformed in progressive buckling mode. Figure 6 shows the simulation and experimental final deformed shapes of the impact attenuator subjected to axial static loading. Experimental deformed shape showed a slightly less uniform crush pattern compared to the simulation. Again, this was due to the tubes moving sideways when load is applied. Welded sections may also affect the stiffness of the structure. This resulted in slightly inconsistent crush pattern.

Peak and average accelerations were calculated from the quasi-static force displacement curves. Taking mass of the Formula SAE car as 300 kg, a peak acceleration of 22.2g and average acceleration of 17.6g were obtained. The values fulfilled the Formula SAE rules.

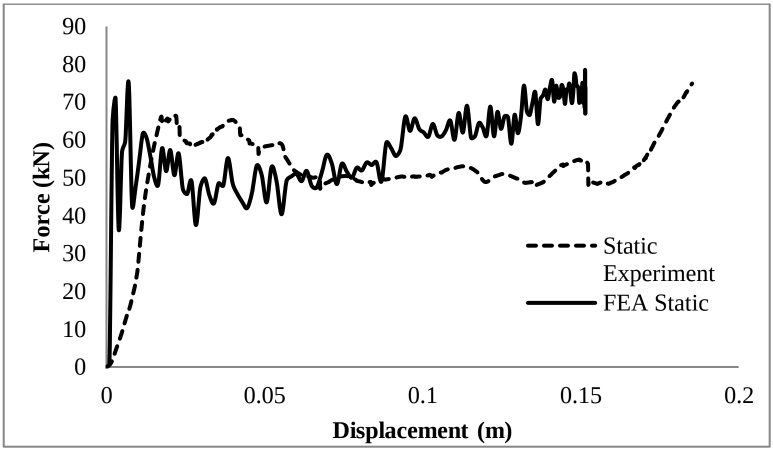


Figure 5: Simulation and experimental force-displacement curves of impact attenuator subjected to axial static loading

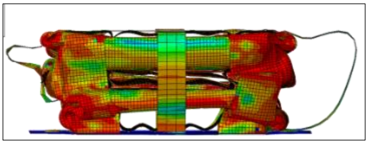
(a)  (b) 

Figure 6: Final deformation of the impact attenuator under axial static loading, (a) simulation and (b) experiment

Impact Analysis of Impact Attenuator

Even though result from the quasi-static experiment is sufficient to fulfil the Formula SAE requirements, an impact analysis was carried out to determine the effect of inertia. However, only impact simulation is pursued. Figure 7 shows the simulation force-displacement curve of impact attenuator under axial impact loading with impact mass of 300 kg and impact speed of 8.3 m/s.

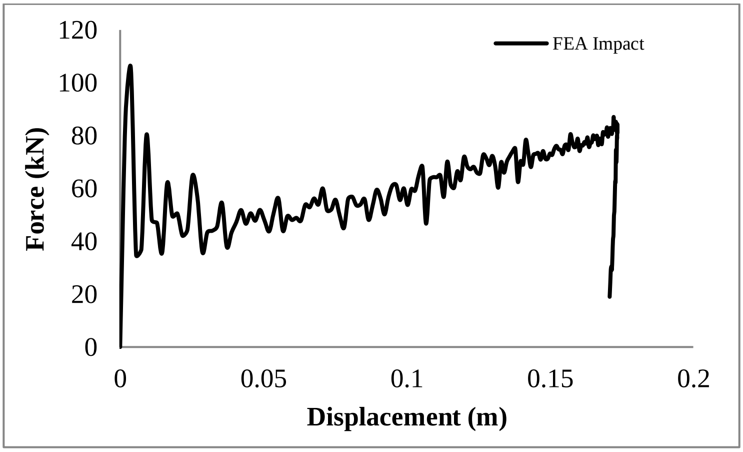


Figure 7: Simulation force-displacement curve of impact attenuator subjected to axial impact loading

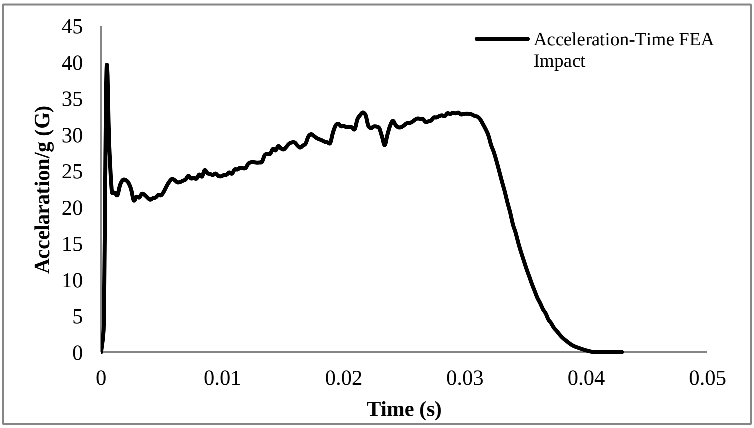


Figure 8: Acceleration-time curve of impact attenuator under axial impact loading

The curve is characterised by a high peak force in the initial stage and followed by fluctuating mean force which increased with displacement. Figure 8 shows the acceleration-time curve of the impact attenuator subjected to axial impact loading. A peak acceleration of 38.3g and an average acceleration of 22.1g were obtained. Although the values were higher than the values obtained from the experimental quasi static test, they are still

within the Formula SAE rules. Figure 9 shows the kinetic and internal energy graphs of the impact attenuator subjected to axial impact loading.
It can be seen that the attenuator is able to absorb the required impact energy where absorbed impact energy of 8841 J was recorded.

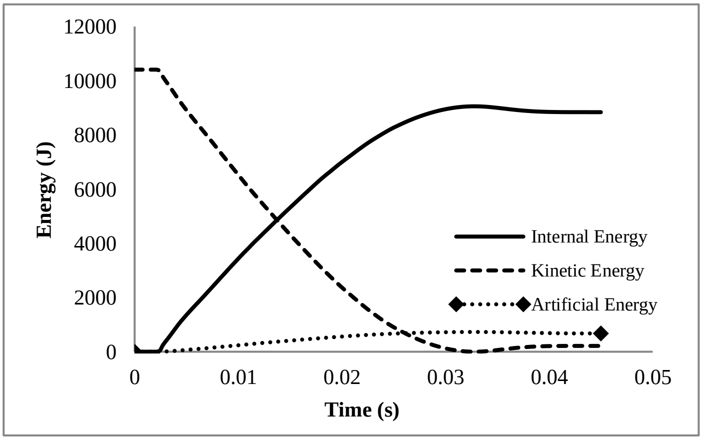


Figure 9: Energy-time curve of the impact attenuator under axial impact loading

Conclusions

A new impact attenuator design for the Formula SAE car is proposed. It is constructed from aluminium circular tubes and L-sections, arranged in a specified configuration. The attenuator was subjected to simulation and experimental axial static loading and simulation impact loading. Quasi-static simulation and experimental results have shown peak and average accelerations of 22.2g and 17.6g respectively. Absorbed energy was more than 7350 J and therefore the design complied with the Formula SAE rules. Impact simulation result also has shown that the design complied with the Formula SAE rules. Further improvement of the impact attenuator in terms of energy absorption and weight reduction shall be carried out for future work.

Acknowledgement

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