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Effect of Impact Attenuator Geometry on Head Injury Criterion (HIC) in NxGV Challenge Vehicles

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ABSTRACT – *The Next Generation Vehicle (NxGV) Challenge, organized by PERODUA, SAE Malaysia, and the Malaysia Automotive, Robotics and IoT Institute (MARii) is a student competition featuring race cars similar to those in Formula SAE. These vehicles are powered by electric motors with a maximum power limit of 70 horsepower. A mandatory rule of the competition requires each vehicle to be equipped with an impact attenuator mounted at the front of the chassis. The driver's safety during frontal collisions is critically influenced by the design of the impact attenuator. This study investigates the effect of varying impact attenuator geometries on the Head Injury Criterion (HIC), a widely accepted metric for evaluating the severity of head injuries in crash scenarios. Using finite element analysis, several attenuator configurations differing in shape were subjected to standardized impact conditions. The resulting head acceleration data were used to calculate HIC values, allowing for a comparative assessment of design performance. Results indicate that a hollow design of impact attenuator for this study allows for greater material deformation during impact, enhancing energy absorption and contributing to lower Head Injury Criterion (HIC) values. The findings underscore the importance of geometric optimization in attenuator design to enhance driver safety and meet NxGV Challenge regulatory requirements. This research provides practical insights for Formula Student teams aiming to design efficient, lightweight, and regulation-compliant impact attenuators.*

KEYWORDS: Head Injury Criterion (HIC), dynamic analysis, impact attenuator, NxGV

1. INTRODUCTION

In high-performance motorsport competitions such as the Next Generation Vehicle (NxGV) Challenge, vehicle safety remains a critical aspect of the design. The NxGV Challenge, hosted by PERODUA, SAE Malaysia, and the Malaysia Automotive, Robotics and IoT Institute (MARii), is a student engineering competition inspired by Formula SAE, where teams build and race single-seat electric vehicles. These vehicles are powered by electric motors limited to 70 horsepower and are subject to strict safety and design regulations. One of the mandatory safety components outlined in the rules is the impact attenuator, which must be installed at the front of the chassis to absorb energy during a frontal collision and reduce the forces transmitted to the driver.

The primary function of an impact attenuator is to deform in a controlled manner upon impact, thereby dissipating kinetic energy and minimizing the deceleration forces experienced by the occupant (Belingardi & Obradovic, 2010). To ensure adequate performance, the Formula SAE regulations require that the impact attenuator be at least 200 mm in length, with a minimum height of 100 mm and width of 200 mm. However, within these constraints, the geometry and internal structure of the attenuator can significantly influence its effectiveness. A poorly designed attenuator may fail to absorb sufficient energy, while an overly stiff design might transmit excessive force to the driver, increasing the risk of injury.

To assess occupant safety, metrics such as the Head Injury Criterion (HIC) are widely used in automotive crash analysis (Mihradi et al., 2017). HIC quantifies the likelihood of head injury based on the magnitude and duration of acceleration experienced by the head during an impact. Lower HIC values indicate a lower probability of serious injury, and according to the Injury Assessment Reference Value (IARV) proposed by the Alliance of Automobile Manufacturers in 1998, a HIC value below 700 is considered to represent an acceptable safety threshold.

Several previous studies have explored the effect of attenuator geometry and material selection on crash performance in Formula SAE (FSAE) vehicles. Arora and Venkatachalam (2022) conducted a parametric simulation study using aluminum foam and evaluated twelve different attenuator designs with varying geometries and relative densities. Their results emphasized the importance of geometry in achieving effective energy absorption while complying with FSAE rules. Similarly, Wheatley and Zaeimi (2020) investigated three different attenuator shapes made from polypropylene honeycomb and found that a four-tier pyramid configuration provided superior energy absorption and deceleration characteristics, which were further validated through destructive testing.

Coppola et al. (2020) adopted a combined numerical and experimental approach to optimize a lightweight honeycomb sandwich attenuator, demonstrating that carefully engineered internal cavities can reduce weight while promoting progressive deformation. Additionally, Pang and Tristian (2014) validated a nonlinear finite element model of an FSAE impact attenuator using experimental data, reinforcing the value of simulation tools like LS-DYNA and ABAQUS for accurate prediction of crash behaviour. These studies collectively highlight that both material properties and internal geometry significantly influence crashworthiness and driver safety, an insight that directly informs the current investigation into head injury outcomes using the Head Injury Criterion (HIC).

Building on these foundations, this study aims to investigate the effect of impact attenuator geometry on driver head injury, specifically by analyzing HIC values obtained from crash simulations. Using finite element analysis (FEA) with LS-DYNA, three attenuator designs were evaluated: a hollow structure, a design with internal cross-shaped reinforcement, and one with a grid-like structure. The simulation results focus on deformation behaviour, head acceleration, and HIC values to determine which configuration provides the best balance between structural integrity and occupant safety.

2. METHODOLOGY

2.1 Material Model

This study investigates the effect of different attenuator geometries on Head Injury Criterion (HIC) values through finite element analysis. All designs and simulations adhere to the impact attenuator requirements specified in the Formula SAE (FSAE) regulations, which mandate a minimum length of 200 mm, a minimum height of 100 mm, and a minimum width of 200 mm. The baseline impact attenuator currently implemented on the vehicle measures 300 mm in length and 210 mm in both height and width. The extended length provides additional space for controlled deformation, creating a more effective crumple zone to absorb collision energy before it reaches the chassis structure. The impact attenuator is constructed from mild steel and is modelled in LS-DYNA using the Johnson-Cook material model, which accurately captures the material's strain rate-dependent behaviour under dynamic loading conditions. The material property values used in the simulation are summarized in Table 1.

TABLE 1: Mild steel Johnson Cook material model properties

Density	**Shear Modulus**	**Young's Modulus**	**Poison Ratio**	
7.850e-g kg/mm^3	80 GPa	205 GPa	0.3	
Parameter A	**Parameter B**	**Parameter N**	**Parameter C**	**Melting Temperature**
0.214	0.234	0.643	0.076	373 ^{o}C

2.2 Geometry Setup

Three different impact attenuator designs were developed and analyzed in this study. The first design, referred to as Design A, consists of a hollow rectangular box structure with 3 mm thick outer walls and an empty internal cavity. Design B maintains the same external dimensions as Design A but incorporates a single cross-shaped internal divider, also 3 mm thick, which divides the cavity into four equal square sections. Design C builds upon Design B by introducing a grid-like internal structure with additional dividers, creating a total of nine equally sized square compartments within the box. The geometry of each impact attenuator design is illustrated in Figure 1.

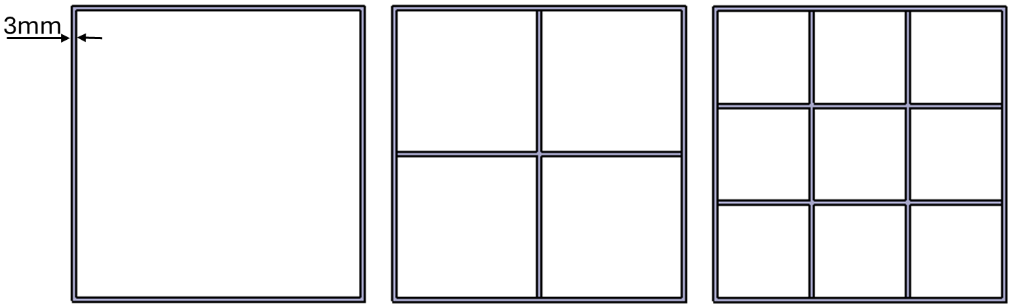


FIGURE 1: Front view of the inner geometry for the impact attenuator – Design A (Left), Design B (Center), and Design C (Right)

2.2 Analysis Setup

For the crash analysis, a simplified vehicle chassis geometry was imported into the simulation environment. To isolate the performance of the impact attenuator, the chassis was modelled as a rigid body, ensuring that the collision energy is primarily absorbed by the attenuator itself rather than dissipated through chassis deformation. The mass of the chassis was set to 300 kg to simulate the overall weight of the vehicle in accordance with FSAE requirements.

A 50th percentile male crash test dummy model was positioned within the chassis and restrained using a five-point seatbelt system. The dummy was used to measure the head acceleration profile during the impact, from which the Head Injury Criterion (HIC) values were calculated.

A rigid wall was placed in front of the impact attenuator to serve as the collision target. Rather than accelerating the entire vehicle toward the wall, the rigid wall was instead accelerated toward the stationary vehicle and dummy. This approach eliminated the need for a stabilization phase of the dummy within the chassis, thereby significantly reducing the total simulation time. The speed of collision was set at 40 km/h, which is higher than the 25.2 km/h impact speed typically required by FSAE regulations. This increased velocity was chosen to impose more severe loading conditions, providing a more rigorous evaluation of each attenuator design's energy absorption performance and occupant protection capability. The simulation setup is illustrated in Figure 2.

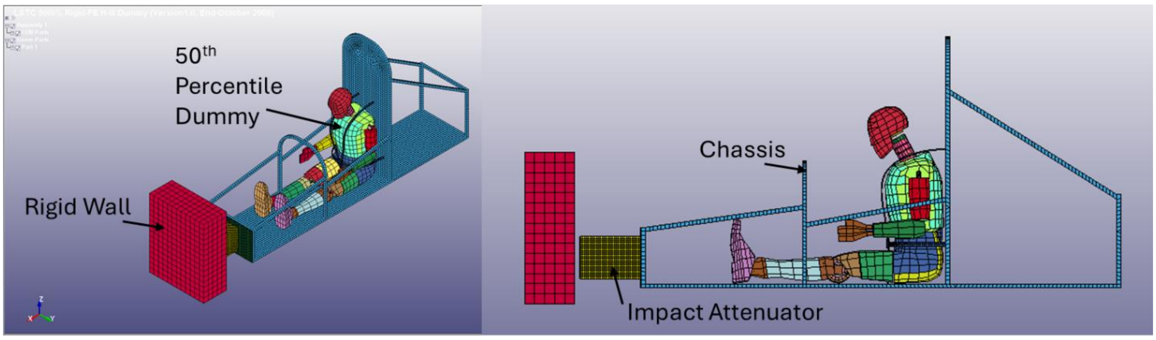


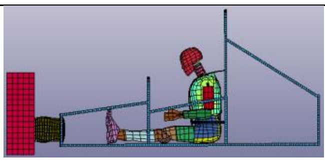
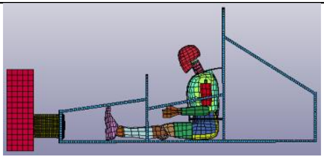
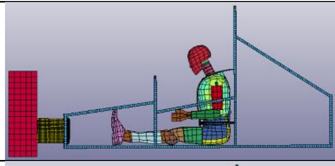
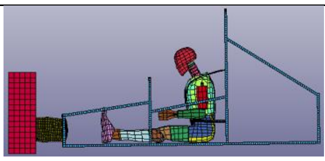
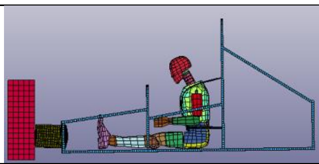
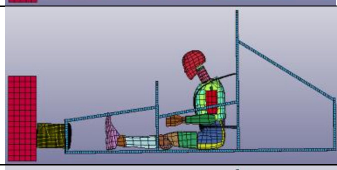
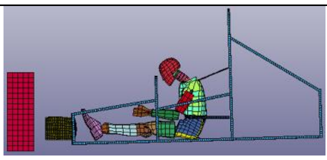
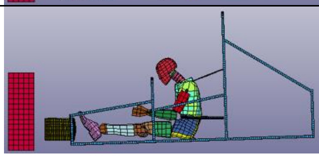
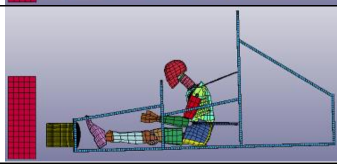
FIGURE 2: Dynamic analysis setup. Isometric view (left) and side view (right)

3.0 RESULTS AND DISCUSSION

To evaluate the energy absorption behaviour of each impact attenuator geometry, the deformation progress was analyzed at selected time intervals of 10 ms, 15 ms, and 30 ms from the onset of impact. These key time frames were chosen to observe the dynamic crushing behaviour and to assess the relative performance of each design during the impact event. The animated deformation results for each design at the selected time points are summarized in Table 2.

At 10 ms, Design A exhibited the most prominent deformation among the three geometries. The hollow structure allowed it to compress rapidly upon contact with the rigid wall, indicating early energy absorption and a higher rate of material collapse. In contrast, Designs B and C, which featured internal reinforcements, showed less deformation at this stage. The added structural elements appeared to provide increased resistance, delaying the onset of significant crushing.

TABLE 2: Animation Plot during collision for all three geometries

Time (ms)	Design A	Design B	Design C
10			
15			
30			

By 15 ms, the deformation process had progressed further for all designs. In Design A, the crash box continued to compress significantly and remained in full contact with the rigid wall, demonstrating sustained energy absorption. However, for Designs B and C, slight separation from the rigid wall was observed. This indicates that the initial structural stiffness provided by the internal dividers had already absorbed a portion of the impact energy, reducing the continued need for contact and deformation. The separation also suggests that the reinforced designs had begun to rebound or complete their major deformation phase earlier than Design A.

By 30 ms, all three impact attenuator designs had fully separated from the rigid wall, indicating the conclusion of the primary deformation phase. At this stage, the main difference among the geometries lies in their final deformed shapes. The final deformed shapes of each impact attenuator design after the collision event are presented in Figure 3. Among the three, Design A exhibited the most significant deformation, while Design B showed an intermediate level of crushing, and Design C experienced the least amount of structural collapse.

In addition to the deformation of the crash boxes themselves, the displacement of the anti-intrusion plate, an important safety feature designed to prevent intrusion into the driver compartment, was also evaluated. The maximum penetration depth of the anti-intrusion plate into the chassis was recorded for each design. For Design A, the deepest penetration point measured was 46 mm. For Design B, the intrusion increased slightly to 58 mm, and for Design C, the maximum penetration reached 65 mm. The comparison of head acceleration responses for each impact attenuator geometry is presented in Figure 4. The graph illustrates the variation of head acceleration over time, highlighting the dynamic effect of attenuator design on occupant safety during the crash event.

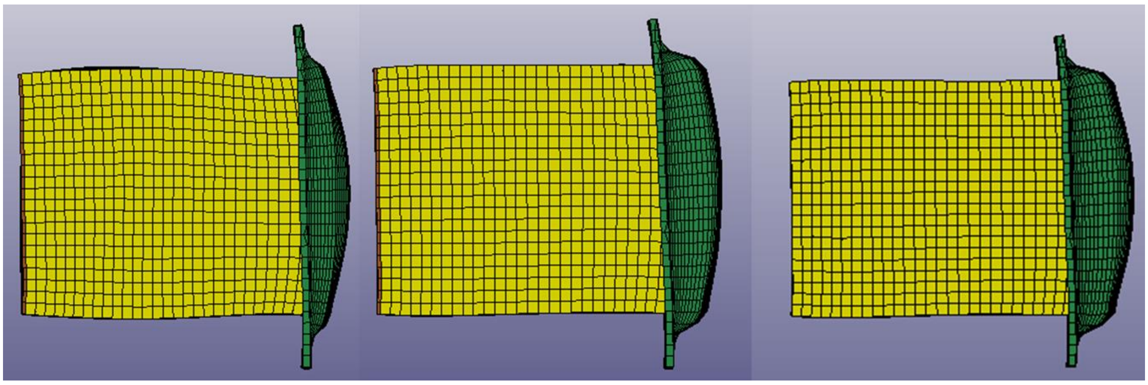


FIGURE 3: Final deformed shape of the impact attenuator (yellow) and anti-intrusion plate (green) for Design A (left), Design B (middle) and Design C (right)

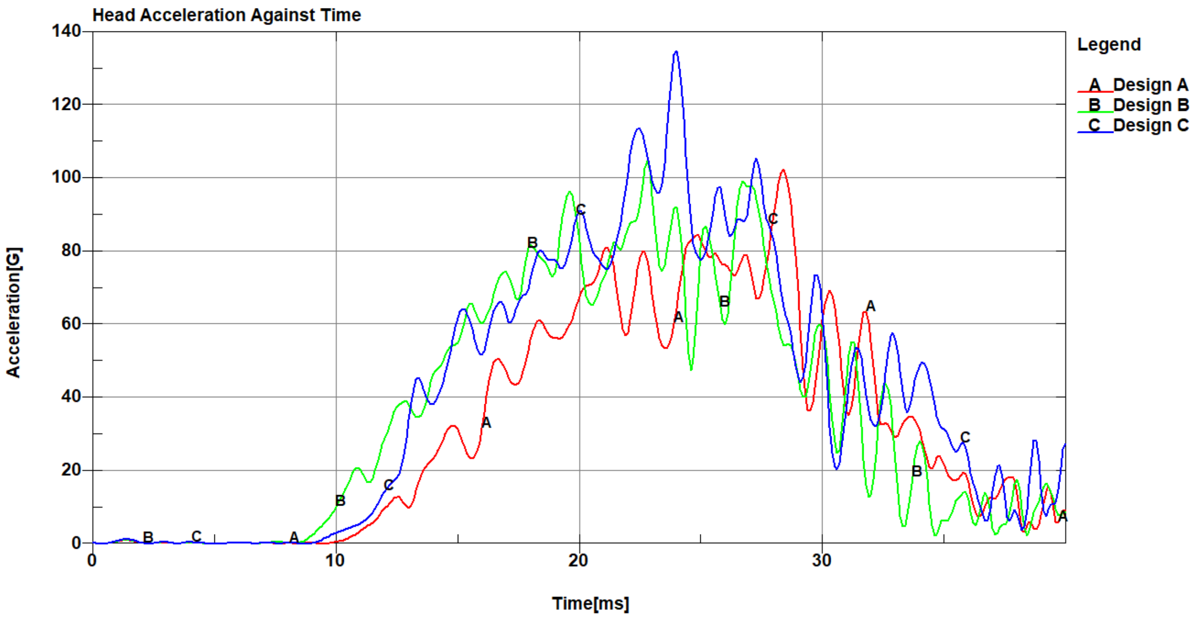


FIGURE 4: Comparison of head acceleration for each design

Design A recorded a peak head acceleration of 102.3 G, which occurred at 28.3 ms – the latest among the three. This delayed peak indicates a more gradual energy transfer, aligning with the extensive deformation observed in Design A. The hollow structure allowed for a longer duration of energy absorption, contributing to a smoother deceleration profile. In contrast, Design B reached a slightly higher peak of 104.11 G at 22.8 ms, while Design C achieved the highest maximum acceleration of 134.68 G at 23.9 ms. Both B and C showed earlier and steeper rises in acceleration compared to Design A, consistent with their stiffer structures and quicker deformation completion.

These trends demonstrate that while reinforced designs may offer structural rigidity, they can transmit higher and more abrupt acceleration loads to the occupant, potentially increasing the risk of head injury. Design A, with its delayed and lower peak acceleration, appears to be more effective in moderating impact forces experienced by the driver. The Head Injury Criterion (HIC) values for each impact attenuator design are compared in Figure 5. The HIC metric, which is derived from the magnitude and duration of head acceleration, serves as a key indicator of the potential severity of head injuries during a crash event.

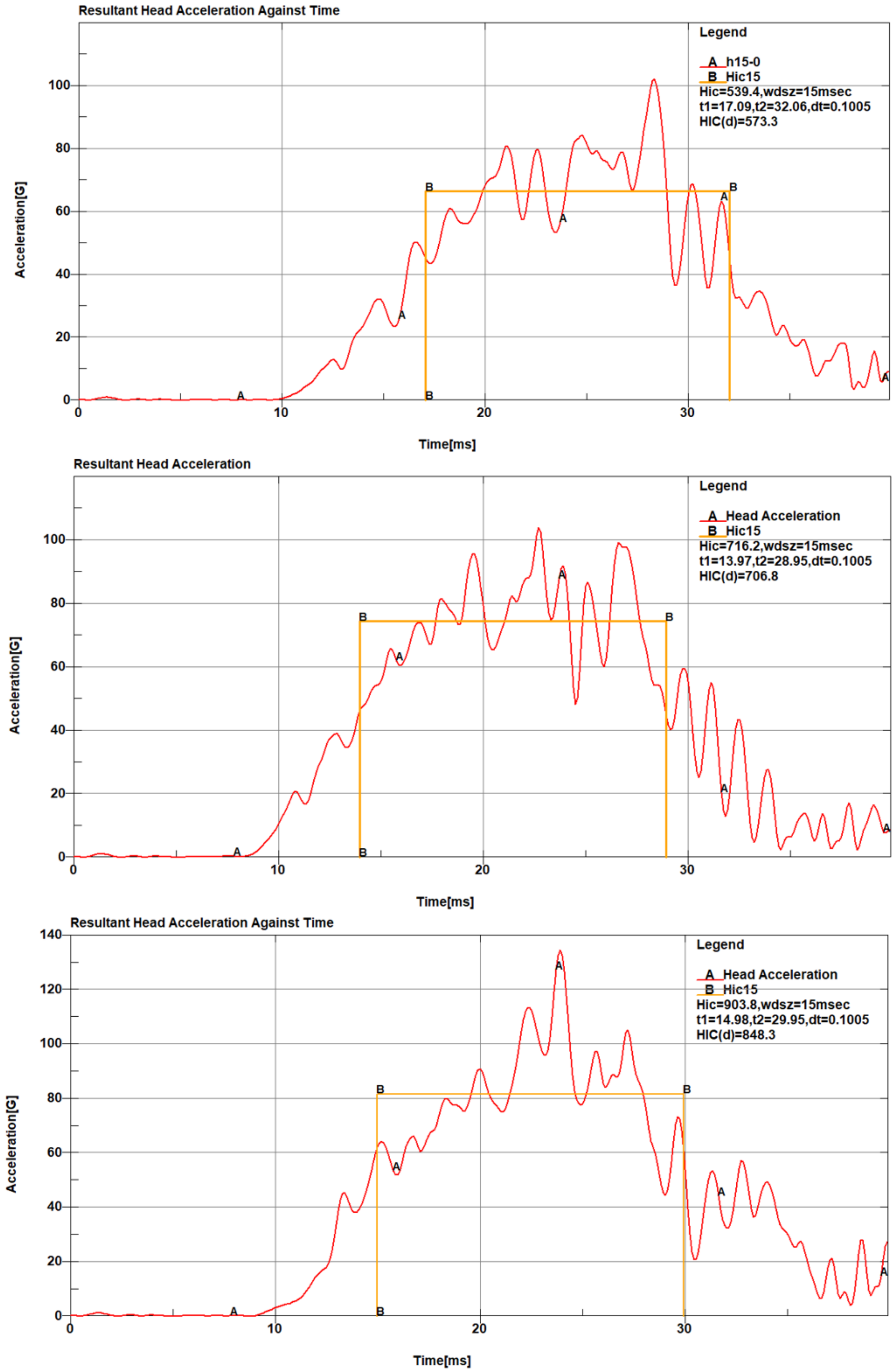


FIGURE 5: Comparison of Head Injury Criteria Value for each impact attenuator design – Design A (top), Design B (middle) and Design C (Bottom)

Design A yielded the lowest HIC value of 539.9, with the integration period extending from 17.09 ms (t_1) to 32.06 ms (t_2). This result is well below the injury threshold and reflects the design's ability to reduce the severity of head acceleration through prolonged energy absorption. The smoother head acceleration profile observed earlier is consistent with this relatively low HIC value.

Design B produced a higher HIC value of 716.2, with the integration window ranging from 13.97 ms to 28.95 ms. Although this value slightly exceeds the safety limit proposed by the 1998 Alliance of Automobile Manufacturers, it still shows an improvement compared to more rigid designs.

Design C, featuring the most internal reinforcement, recorded the highest HIC value of 903.8, calculated between 14.98 ms and 29.95 ms. This elevated value reflects the high peak acceleration and short energy dissipation period observed in the earlier results.

According to the Injury Assessment Reference Value (IARV) proposed in 1998, a HIC value below 700 is considered to represent a low risk of serious head injury. Only Design A meets this safety criterion, suggesting that increased structural stiffness in the attenuator may reduce deformation but potentially increases the risk of injury by transmitting greater forces to the driver.

4.0 CONCLUSION

This study investigated the influence of impact attenuator geometry on driver head injury in a student-built electric race car designed for the Next Generation Vehicle (NxGV) Challenge. Three different attenuator designs ranging from a simple hollow box to reinforced internal structures were evaluated through finite element simulations using LS-DYNA. The analysis focused on deformation behaviour, head acceleration, and Head Injury Criterion (HIC) values.

Among the three designs, Design A, the hollow structure, demonstrated the most extensive deformation, lower peak head acceleration, and the lowest HIC value of 539.9, well below the injury risk threshold of 700. Design B offered intermediate performance, with moderate stiffness and a HIC value of 716.2, slightly exceeding the safe limit. Design C, although structurally the most rigid, resulted in the highest head acceleration and a HIC value of 903.8, indicating increased injury risk.

These results highlight a critical trade-off between structural stiffness and occupant safety. While reinforcement can limit deformation, it may also lead to greater force transmission to the occupant. The findings suggest that a more deformable attenuator, like Design A, provides better protection by effectively absorbing impact energy and reducing the likelihood of serious head injury.

Future work may explore hybrid materials or multi-stage attenuator configurations to further optimize safety without compromising structural performance. To further strengthen the findings of this study, experimental validation is recommended, using physical crash testing of the proposed attenuator designs under controlled conditions. Such testing would help confirm the accuracy of the simulation results and provide real-world data to refine material models and improve design reliability for future studies.

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