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RESEARCH ARTICLE

Evaluation of Automotive Bio-Composites Crash Box Performance

X.Y. Ang[1], C.S. Hassan[1]*, S.Y. Soh[1], E.U. Olugu[2], N.F. Abdullah[1], L.J. Yu[1], N. Abdul Aziz[3]

[1]Faculty of Engineering, Technology & Built Environment, UCSI University, Kuala Lumpur, Malaysia
[2]School of Mechanical, Industrial & Aeronautical Engineering, University of the Witwatersrand, Johannesburg, South Africa
[3]Department of Mechanical and Manufacturing Engineering, University Putra Malaysia, Selangor, Malaysia

ABSTRACT - In the automotive industry, sustainable materials, such as bio-composites, are progressively being adopted due to their lightweight feature, which reduces vehicle weight, fuel consumption and pollutants emissions. Bio-composites are renewable and biodegradable, making them more environmental-friendly. However, limited investigations into the use of bio-composites in crash box applications have indicated that they lack the impact strength to fully absorb collision energy. This study aims to compare the crashworthiness performance of crash boxes made from OPEFB fiber/epoxy and kenaf fiber/epoxy composites, with conventional steel and carbon fiber/epoxy using LS-DYNA quasi-static simulations. Six different crash box designs are proposed: square, hexagonal, decagonal, hexagonal 3-cell, hexagonal 6-cell, and decagonal 10-cell structure, to evaluate the effect of these designs on crash box performance. The results show that bio-composite crash boxes are inferior to traditional materials in terms of energy absorption and specific energy absorption, but they yield better performance in crush force efficiency. In terms of design, decagonal 10-cell structure produces the highest specific energy absorption and energy absorption for bio-composites. Hence, optimization is performed on the OPEFB fibre/epoxy decagonal 10-cell crash box, aiming to increase energy absorption capability by varying the thickness, perimeter, and length of the crash box. The design is optimized by increasing thickness and maintaining length and perimeter. Compared to the original design, the optimized design improves energy absorption by 59% and specific energy absorption by 19%. The optimized design is then subjected to both quasi-static and impact loading tests, revealing that the optimized OPEFB fibre/epoxy crash box design exhibits 44% lower energy absorption than steel under quasi-static load, but it demonstrates a 56% increase in crush force efficiency and a 6 % increase in specific energy absorption. Under impact load, it shows a 91% increase in specific energy absorption compared to the traditional square steel crash box.

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1.0 INTRODUCTION

A crash box is typically a sacrificial device installed between the car bumper component and longitudinal rails. Most vehicles utilise crash boxes to reduce the force exerted on the front rail during low-speed crashes and buffer the impact force [1]. Through structural deformation, the crash box transforms the kinetic energy of the collision into a strain energy [2]. When a car is involved in a head-on crash, the energy should be transferred in such a way that the bumper transmits it to the right and left crash boxes, forcing them to compress and allowing the collision energy to be transmitted to the front rail [3]. The crash performance of a crash box can be evaluated using its crashworthiness. Crashworthiness refers to a component's capacity to tolerate loads below a specific threshold while minimising damage in circumstances of extreme dynamic loads. The features of a crashworthy system are the impact's kinetic energy must be dissipated in a regulated way, a survival space needs to be maintained for the safeguarded parts, and the stresses and accelerations that these parts are subjected to must be minimized [4]. Crashworthiness, in terms of automotive engineering, refers to a vehicle's ability to prevent passengers from suffering serious injuries or even fatalities in a potentially fatal collision [5]. Crashworthiness has been evaluated using a variety of metrics. Specific energy absorption (SEA), crush force efficiency (CFE), collision force and crumple distance are the indicators that can be used to describe a structure's energy absorption capacity in a systematic manner [3].

The ability of automotive parts to absorb energy during the collision event can be measured using a metric called Energy Absorption (EA). It is the total amount of energy that a crash box absorbs by deforming plastically during an impact. When crush force is plotted against crush stroke, EA is the area under the curve. It is recommended that the EA of the crash box be high when it is deforming to absorb as much collision energy. Specific energy absorption (SEA) is the energy absorbed per unit mass (m) of the crash box due to plastic deformation during impact. The optimal requirement is to maximize the amount of impact energy absorbed while minimizing the amount of mass required; therefore, the optimal SEA should be high. Components with exceptional performance-to-weight ratios are greatly valued in today's modern automobile sector. Peak crush force (F_{peak}) is the first peak impact force experienced by the crash box that is necessary to initiate plastic deformation. An ideal crash box should have low peak collision force. It must be below the threshold limit (equivalent acceleration levels of 20 g) to prevent severe injury. The optimal F_{peak} of the crash box should

***CORRESPONDING AUTHOR | C.S. Hassan** | ✉ suhana@ucsiuniversity.edu.my

have accelerations that are within the range that humans can safely tolerate and a gradual increase in time to achieve F_{peak} [6]. EA divided by the corresponding deformation displacement (φ) yields the mean crush force (F_{mean}). Crush force efficiency (CFE) is the ratio of F_{mean} to F_{peak}. After F_{peak}, the crush force curve normally decreases and is accompanied by fluctuations. Crush force should achieve consistency so that energy may be absorbed as effectively as possible. It is not a straightforward measurement of crash performance, instead, it is a measurement of crush force consistency that optimizes the EA. Higher CFE is favorable. If CFE approaches 100 %, the crash box's EA will be the maximum [6].

Li et al. [3] stated that other than SEA and CFE, the crashworthiness of the crash box also depends on crumple distance. Even with good energy absorption and buffering capabilities, the crumple distance of the crash box may not meet the safety criteria. Providing sufficient residual room for the passengers will be challenging if the crumple distance is long. Therefore, crumple distance is another important measurement to estimate the vehicle's penetration risk under maximum deformation. The crashworthiness of a crash box can be affected by several factors, including the design of the crash box and its material. In kinetic energy absorption, thin-walled structures are frequently utilized because of their low cost, high capacity to take in a great deal of energy, and low weight. Many factors can affect the effectiveness of a thin-walled column's ability to absorb energy, including material characteristics, cross-section shape and wall thickness. One of the most essential of these criteria is the cross-section design [7]. Alexander [8] first presented a theoretical model to predict the mean crush force for a circular tube's axially symmetric (concertina) folding pattern. This model assumes that the creases in a concertina collapse mode are straight, however, this is not the case. The wrinkles of concertina collapse are with a curve profile. Therefore, this idealized theory's best guess for mean axial force is derived by Jones [9]. After some time, this theory was updated to include the folding radius, and Abramowicz [10] incorporated the effective crushing distance circular, resulting in the tube collapsing axially across a distance. A circular tube may collapse in concertina (or axisymmetric) mode, diamond mode, or a combination of the two. The radius (R) to thickness (t) ratio determines how these circular tubes deform. Some research indicates that the tube deforms axisymmetrically if R/t is between 40 and 45. Otherwise, the tube buckles in a diamond mode as the ratio increases [9]. Wierzbicki and Abramowicz [10] investigated square column axial crushing and presented the Super Folding Element (SFE) hypothesis. Utilizing kinematics plasticity, they constructed a theory of the crush behavior of thin walls and discovered the equation for the mean crush load while addressing the energy balance for a square tube. Mahmood and Paluszny [11] discovered that non-compact sections exist for small thickness-to-width ratios (t/b), which are massive irregular folds that cause bending instability due to fold abnormalities. The impact of geometry is less significant as the t/b ratio increases, and the material strength attribute drives the collapse mode, resulting in post-buckling stability. Hussain et al. [12] investigated the crash performance of glass fiber-reinforced plastic (GFRP) composite crash boxes with four distinct cross sections using LS-Dyna software. The crash boxes were exposed to drop-weight impact testing, with a velocity of 16 km/h applied to the impactor in a manner analogous to the RCAR test. The greatest SEA was achieved using a decagonal crash box, with 6427.1 J/kg, followed by the hexagonal, cylindrical, and square crash box. Reddy et al. [6] investigated the crashworthiness of SS304 stainless steel tubes with various cross-sectional geometries such as triangle, square, rectangle, pentagon, hexagon, octagon, and circle. The finite element software used was ABAQUS, with the impactor given a starting velocity of 0.1 mm/s axially. It was reported that pentagonal or hexagonal cross-sections offer the best balance of stability regarding crush force, CFE, SE, and EA regarding crashworthiness. Due to its greater SEA capacity, interest in multi-cell columns has increased [13]. Using a Simplified Super Folding Element (SSFE) theory, Chen and Wierzbicki [14] investigated the performance of single-cell, double-cell, and triple-cell columns when subjected to quasi-static impact loading. Aluminum extrusion was used for the column walls. They found that double and triple cells outperform single cells by roughly 15% in SEA. Tang et al. [7] introduced a cylindrical multi-cell column (CMC) as a novel form of a multi-cell column. This work explored the crash performance of a square multi-cell column (SMC), a CMC, and a square column made of aluminium extrusion AA 6060 T4. The SEA of the CMC was 144.90% more than that of the square column and 49.94% greater than that of the SMC. The CFE of CMC was raised by 92.33% compared to the square column and 51.66% compared to SMC. Tang et al. [7] proceeded to carry out a parametric analysis to determine the effect of geometrical factors on the crashworthiness of CMC and discovered that the column with the thicker wall has greater energy absorption, specific energy absorption, and peak force.

While studies on the crash box performance of synthetic fiber composites have seen significant growth, the exploration of bio-composite potential remains limited, despite several studies [15,16,17] highlighting the potential of natural fibers for energy absorption applications and conceptual design analysis [18]. Hence, there is a vital need for further research to advance the utilization of bio-composites in crash boxes and to evaluate their crash performance compared to conventional materials. Ataollahi et al [19] investigated the crashworthiness properties of silk fiber/epoxy composite crash boxes through a quasi-static compression test with a 30 kN load and 20 mm/min speed. The length of the square cross-section of the silk fiber/epoxy crash box was varied. It was reported that the tube with a shorter length resulted in the highest SEA. However, the energy absorption capability of silk fiber/epoxy laminated with 12 layers was not as good as that of conventional materials. Yan and Chouw [20] studied the energy absorption capability of bidirectional woven flax fiber/epoxy composites with 7.4 threads/cm in the warp and 7.4 threads/cm in the weft, circular tubes with a varied inner diameter, number of plies and length-to-diameter ratio. The investigation was conducted with a quasi-static compression experiment with 100 kN load and 10 mm/min speed. It was found that the optimal design in the study yielded SEA and CFE that is higher than that of conventional metal crash tubes. Notably, the limited studies on bio-composite crash boxes suggest that, when appropriately designed, bio-composites can achieve higher SEA than conventional materials. The current study investigates the parameters that can contribute to the enhanced crash performance of bio-composite, utilizing

OPEFB fiber/epoxy and kenaf fiber/epoxy composites, crash boxes by first experimenting with different designs. Then, optimization is performed by varying the crash box's thickness, perimeter, and length. The performance of the optimized bio-composite crash box is benchmarked against conventional steel and carbon fiber-reinforced epoxy crash boxes. This study utilizes finite element analysis with LS-DYNA to investigate the behavior of bio-composite crash boxes under explicit dynamic axial load. This study aims to examine parameters that can contribute to the enhanced crash performance of bio-composites crash box, to access the energy absorption, collision force and crumple distance of the proposed improved crash box using finite element analysis and to evaluate the optimized bio-composites crash performance against the conventional crash box.

2.0 METHODS AND MATERIALS

2.1 Crash Box Design and Optimization Strategy

Six designs of the crash box are proposed and modeled in LS-Prepost, as shown in Figure 1. The reference model is the square cross-section thin-walled structure as shown in Figure 1(a). According to Hussain *et al.* [12], hexagonal and decagonal cross-sections are the two geometries with the highest SEA compared to other geometries. Therefore, the second design employs a hexagonal cross-section, while the third design is a thin-walled structure with a decagonal cross-section, as shown in Figure 1(b) and 1(c). The fourth, fifth and sixth designs are developed by modifying the previous designs to incorporate multicells in the tube to study how the adoption of multicell affects the crashworthiness performance. The design modifications and optimization are based on the TRIZ method [18], employing the principles of 'Local Geometry' and 'Mechanics Substitution,' as depicted in Table 1. In this study, hexagonal 3-cell, hexagonal 6-cell, and decagonal 10-cell thin-walled crash boxes are developed. These designs are depicted in Figure 1(d),1(e) and 1(f).

Table 1. Crash box design strategy based on TRIZ recommendations

TRIZ solution principle	TRIZ recommended solution	Strategy of design
#28. Mechanics substitution	Change from fixed to dynamic attributes, from unstructured to structured features	Improve the hexagonal design by adding supporting structures to become multi-cell hexagonal structure.
#3. Local Geometry	Make each component of an item operate under circumstances optimal for its functioning.	Developing the optimal cross-section. Optimize the thickness, perimeter, and length of the thin-walled structure.

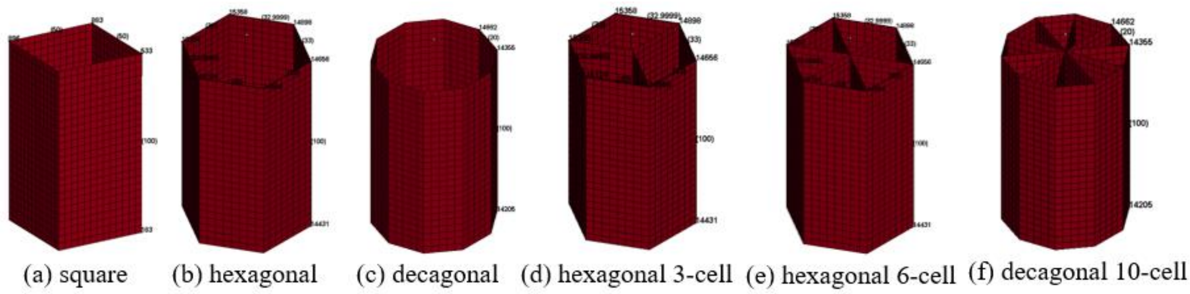


Figure 1. Proposed designs of crash box

2.2 Finite Element Analysis

2.2.1 Model Analysis

The crash boxes are modeled by four-node shell elements, as their reliability in forecasting the failure process in terms of collapse manner and the number of formed folds in thin-walled structures has been verified [21]. All models have a length of 100 mm, thickness of 1.5 mm and perimeter of 200 mm as baseline design parameters. Figure 2 shows the analysis model simulating the quasi-static test. The impactor and the bottom base are modeled in LS-Prepost using eight-node solid elements, as they are considered rigid bodies. To replicate a typical sedan car, which weighs 1100 kg and the mass is evenly distributed throughout the two crash boxes on the frontal structure for energy absorption, a 550 kg mass element is added to the impactor's center node [22].

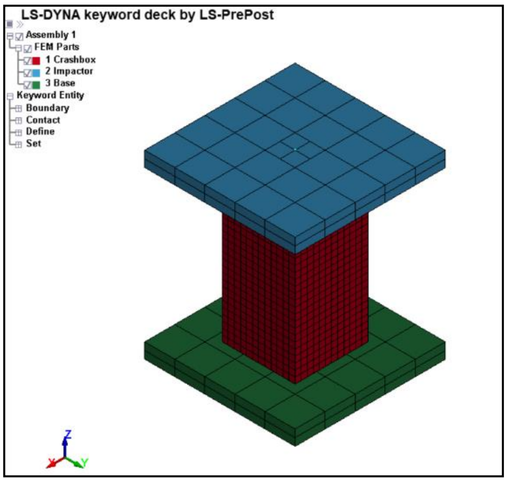


Figure 2. Quasi-static element model

The shell element dimensions of every design of the crash boxes are adjusted to 4 mm x 4 mm, which was priorly converged to match the validated experimental work of Karantza and Manolakos[23]. The bottom ends of the crash box are set to be fixed, where the DOFs of bottom nodes are constrained against any displacement and rotation. Impactors are loaded at a continuous rate of 4 km/h or 1.1111 m/s until they reach a maximum vertical motion of 60 mm using BOUNDARY_PRESCRIBED_MOTION_RIGID card, in compliance with Federal Motor Vehicle Safety Standard (FMVSS) 581 low-velocity impact standard [24].

Boundary conditions for the interface contacts are implemented to prevent penetration between the interacting parts. To prevent the nodes of the crash box shell elements from penetrating the surface of the impactor, the AUTOMATIC_NODES_TO_SURFACE contact algorithm is first applied between the crash box and the impactor. The rigid impactor is set as the master, while the softer crash box is the slave. Coulomb friction conditions are also considered using static and dynamic friction coefficients 0.2. Besides, AUTOMATIC_SINGLE_SURFACE contact type is utilized to model the interaction of crash box shell elements during fold formulation in the crashing process and to prevent self-penetration of the crash box surface, static and dynamic friction coefficients of 0.2 are used. The center of Mass Constraint Option (CMO) is assigned as EQ.+1 to apply constraints to global directions, where translations in x and y, as well as rotations in x, y and z axes, are all confined. This permits the impactor to move vertically along the z-axis only. For simulation control, CONTROL_ACCURACY is enabled. Termination time is set to 40ms by utilizing the CONTROL_TERMINATION card. The methodology employed in this study was replicated based on Karantza and Manolakos[23] which has been validated against experimental work.

A comprehensive evaluation is conducted on each of the six crash box designs, encompassing all four materials: steel, carbon/epoxy, OPEFB/epoxy, and kenaf/epoxy. The evaluation process involves subjecting the designs to quasi-static compression simulation, enabling the assessment of their crashworthiness through key parameters such as peak crush force, mean crush force, CFE, EA and SEA. Following this initial assessment, the crash box design demonstrating superior crashworthiness for bio-composites is carefully selected. Subsequently, an optimization phase is implemented to enhance the performance of the chosen design. This optimization process systematically alters crucial design parameters, including thickness, perimeter, and length, all conducted under quasi-static conditions. Thickness optimization is explored utilizing varied thicknesses from 1 mm to 2.25 mm, the perimeter of the crash box is changed from 175 mm to 250 mm, and the length of the crash box is varied from 80 mm to 120 mm.

The final step involves subjecting the optimized bio-composite crash box to impact test simulation, providing valuable insights into its dynamic crashworthiness performance as the quasi-static test cannot replicate the decreasing initial velocity from the starting velocity to rest. For a better representation of the real collision, the conventional steel crash box and bio-composites optimized crash box design are subjected to a low-velocity impact test, in which the impactor is subjected to an initial velocity of 4 km/h, according to the low velocity impact standard by FMVSS 581 [24]. In addition, a point mass of 550 kg is applied to the impactor because the average car weighs 1100 kg and has 2 crash boxes.

2.2.2 Material

MAT_054 simulates two bio-composites and one synthetic composite: OPEFB fiber and kenaf fiber-reinforced epoxy composites and carbon fiber-reinforced epoxy. The fiber orientation in composites made with the MAT 054 card is assigned parallel to the axial load. The composite material properties used were based on literature by Hassan, et al. [25], Bruno and Baskaran [26], Manap, et al. [27] and Feraboli and Wade [28]. Since 1018 steel has a high stiffness due to a high Young's modulus value, it is chosen as the conventional metal that would be utilized as the benchmark for measuring crashworthiness performance. For impactor material, MAT_020 is utilized due to the impactor's higher stiffness and is assumed to be undeformable rigid bodies. The materials of the impactor and base are steel, and the material properties are referenced from a study by Karantza and Manolakos [23]. The material card data of the bio-composites and synthetic composites are shown in Table 2.

Table 2. Material card data for the bio-composites and synthetic composite

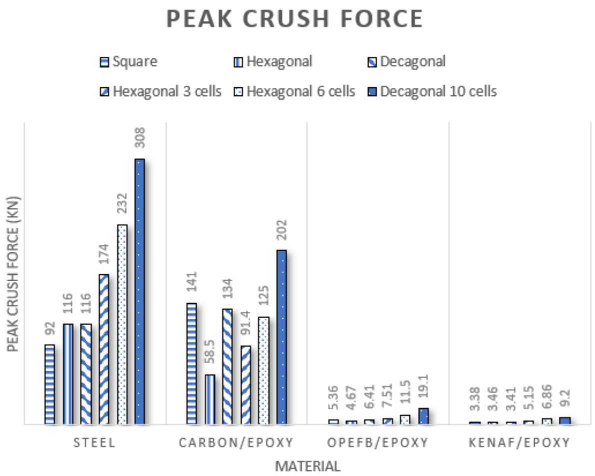
MAT_054 Material Card Data for OPEFB/Epoxy bio-composite.								
Variable	MID	RO	EA	EB	EC	PRBA	PRCA	PRCB
Value	1	1.12e-6	1.09968	0.83823	-	0.17	-	-
Variable	GAB	GBC	GCA	XC	XT	YC	YT	SC
Value	0.35755	0.35755	0.35755	0.21304	0.06638	0.06318	0.02956	-
MAT_054 Material Card Data for Kenaf/Epoxy Bio-composite.								
Variable	MID	RO	EA	EB	EC	PRBA	PRCA	PRCB
Value	1	1.29e-6	4.12	2.05	-	0.1945	-	-
Variable	GAB	GBC	GCA	XC	XT	YC	YT	SC
Value	1.472	0.855	1.472	0.01203	0.04218	0.01184	0.00445	-
MAT_054 Material Card Data for Carbon Fibre/Epoxy Composite.								
Variable	MID	RO	EA	EB	EC	PRBA	PRCA	PRCB
Value	1	1.57e-6	124.7951	8.4116	8.4116	0.0204	-	-
Variable	GAB	GBC	GCA	XC	XT	YC	YT	SC
Value	4.2058	4.2058	4.2058	1.4479	2.19943	0.19857	0.048882	0.154443

3.0 RESULTS AND DISCUSSION

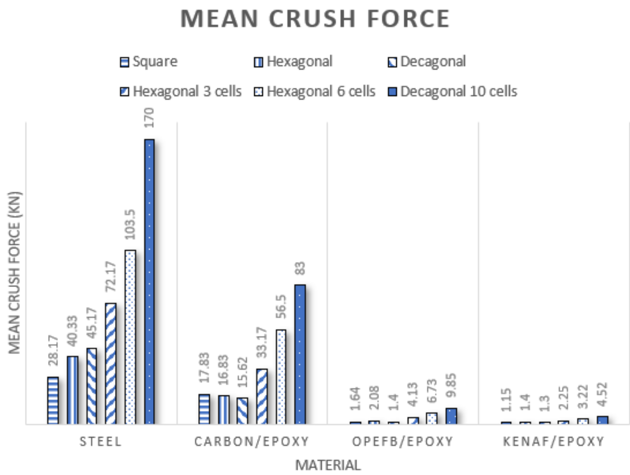
3.1 Design Evaluation under Quasi-Static Load

Quasi-static simulation is conducted to evaluate the crashworthiness of steel, carbon/epoxy, OPEFB/epoxy, and kenaf/epoxy with all 6 designs proposed. The low-velocity impact is such that the top impactor moves downwards at a constant speed of 4 km/h. Peak crush force, mean crush force and CFE obtained are shown in Figure 3. As shown in Figure 3(a), the peak crush force of OPEFB/epoxy and kenaf/epoxy crash boxes are significantly lower than crash boxes made of steel and carbon/epoxy. This might be due to the low stiffness and strength of bio-composites. This low peak force is desirable for an ideal crash box as the passengers will experience no sudden large impact force. In terms of design, the hexagonal cross-section will produce the lowest peak force for carbon/epoxy and OPEFB fiber/epoxy. Similar findings were observed in the study by Reddy, et al. [6], where a hexagonal cross-section tube yielded the lowest peak force compared to other geometries. For steel and kenaf/epoxy crash boxes the lowest peak force is achieved by square cross-section. For all 4 types of materials, the largest peak force is obtained from a decagonal 10-cell design. This result can be supported by a study from Chen and Wierzbicki [14], where increasing the number of cells in a multicell column also raises the peak force. This might be due to the larger surface area available to absorb kinetic energy. Besides, mean crush force is also an important parameter. A higher mean crush force is desired, resulting in higher energy absorption. Figure 3(b) shows the mean crush force of the 6 designs with different materials. The mean crush force of the bio-composites is also significantly lower than the conventional steel and carbon/epoxy crash boxes. Bio-composites do not perform as well as steel and carbon/epoxy in energy absorption. For crash box design, the decagonal 10-cell design performs the best, with the significantly highest mean force for each type of material. This is comparable to a study by Chen and Wierzbicki [14], where increasing the number of cells in a multicell column effectively raises the mean force. The decagonal cross-section has the lowest mean force for carbon/epoxy and OPEFB/epoxy, whereas the square cross-section has the lowest mean force for steel and kenaf/epoxy. Similarly, in the study by Reddy, et al. [6], the mean force is the least when using a mild steel tube with a square cross-section, compared to other geometries. CFE is also evaluated, which is the ratio of mean force to peak force. A higher CFE means impact energy is absorbed effectively. Therefore, a higher CFE is favorable. The CFE of the designs and materials are shown in Figure 3(c). The CFE of bio-composites (OPEFB fiber/epoxy and kenaf/epoxy) are comparable to conventional steel and carbon/epoxy. The OPEFB fiber/epoxy hexagonal 6-cell design performs best with the highest CFE. This shows that bio-composite crash boxes can have a very high CFE when carefully designed, which means a higher load uniformity. For both steel and kenaf/epoxy materials, decagonal 10-cells achieve the highest CFE. For carbon fiber/epoxy and OPEFB fiber/epoxy, the hexagonal 6-cell design has the highest CFE.

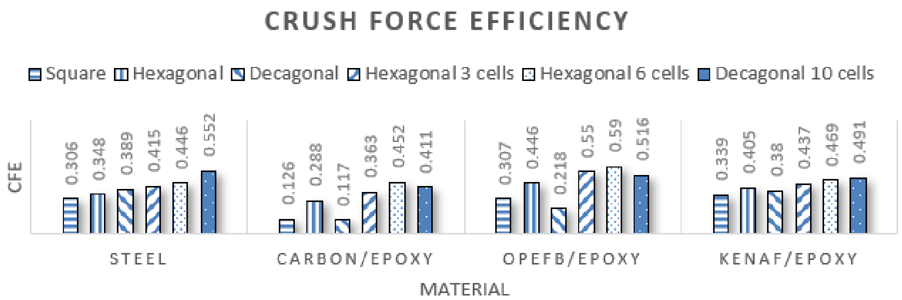
EA obtained from the simulation is shown in Figure 4(a). It is recommended that the EA of the crash box to be high when it is deforming to absorb as much collision energy and reduce damage to vehicles and passengers. Bio-composites do not perform as well as conventional materials in terms of EA due to their lower stiffness and strength than conventional materials. This could suggest that the bio-composite crash box has a poorer energy absorption capacity since it deforms more easily under an impact load [29]. The manufacturing method and the bio-composite material's natural variation may contribute to the poor stiffness and strength [30]. Comparing hexagonal and decagonal designs, decagonal yields a higher EA than hexagonal designs for steel. Research from Reddy, et al. [6] also produced similar findings, where stainless steel crash boxes with decagonal design produce a higher EA than hexagonal ones. For composites such as carbon/epoxy, OPEFB/epoxy and kenaf/epoxy, hexagonal design performs better in EA than decagonal design. This might suggest that hexagonal design performs better in initiating the folding mechanisms in composites and produces higher EA for composites. In addition, increasing the number of cells effectively raises EA, especially for OPEFB fiber/epoxy composites, where hexagonal 3-cell and 6-cell designs boost EA by 98% and 226%, respectively, compared to hexagonal designs. This is in accordance with the study by Chen and Wierzbicki [14], where increasing the number of cells in a multicell column can increase the EA significantly. The best crash box is the decagonal 10-cell structure that absorbs the greatest energy for each material. With the most cells, the greater surface area for deformation allows force distribution and energy absorption. However, it should be noted that increasing the number of cells has a potential drawback of an increase in mass, which might affect the crash box's SEA. Therefore, it is important to study the SEA in this case. Figure 4(b) depicts the SEA of the crash boxes with different materials. Carbon/epoxy outperforms other materials due to synthetic composite's low weight, high strength and stiffness qualities, which result in a high strength-to-weight ratio and improved energy absorption per unit mass [29]. For bio-composites, SEA is significantly lower than that of conventional material. Bio-composite crash boxes have lower masses than steel, but their EA is too low compared to steel and carbon/epoxy, resulting in poor SEA. To maximize energy absorption, the bio-composite crash box should be optimized.



(a) Peak crush force of different materials using different designs



(b) Mean crush force of different materials using different designs



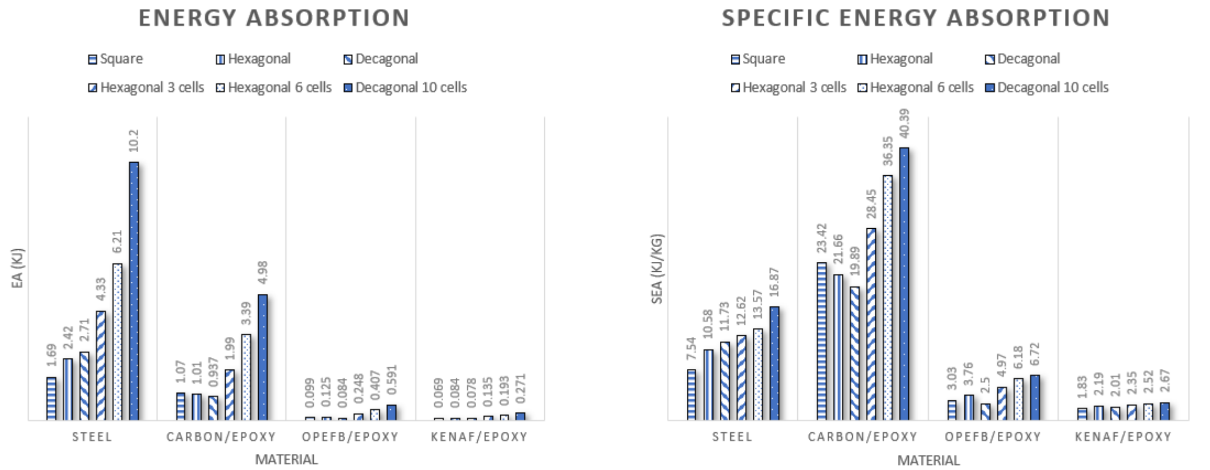
(c) CFE of different materials using different designs

Figure 3. Crashworthiness parameters obtained from simulation results

3.2 Optimization under Quasi-Static Load

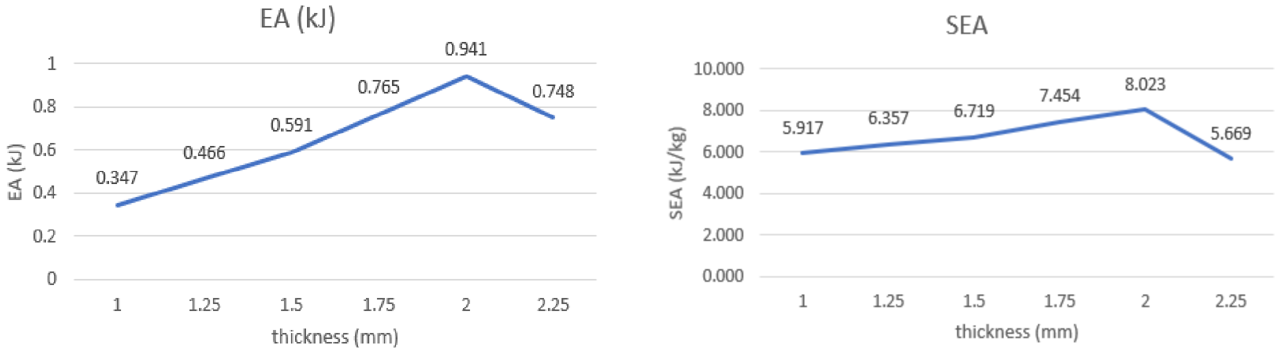
For all proposed designs for bio-composite crash boxes, their EA and SEA are not comparable to the conventional steel material. Hence, optimization is required to maximize the energy capability of the bio-composite materials. The design that can yield the highest EA and SEA for bio-composite is the decagonal 10-cell design, therefore, it is selected to be optimized by varying the thickness, perimeter and length. OPEFB fiber/epoxy is used between the two bio-composite materials since it outperforms kenaf/epoxy in various aspects such as EA, SEA and CFE.

Thickness optimization is explored using varied thicknesses from 1 mm to 2.25 mm. As shown in Figure 5(a), when the thickness of the crash box is increased, EA increases almost linearly until the thickness reaches 2.0 mm. However, beyond 2.0 mm, EA decreases. This trend was also observed in the research by Porwal, et al. [22], where a conical tube made of aluminium was subjected to low-velocity impact. Thicker crash boxes can absorb energy more, but the EA will reach a certain saturation point beyond a certain thickness. SEA also exhibits a similar trend as EA as shown in Figure 5(b). This shows that up to 2 mm thickness, although the increase in thickness results in an increase in mass, the improved energy absorption capacity still compensates for the additional weight, causing SEA to increase.



(a) EA of different materials using different designs (b) SEA of different materials using different designs

Figure 4. EA and SEA obtained from simulation results

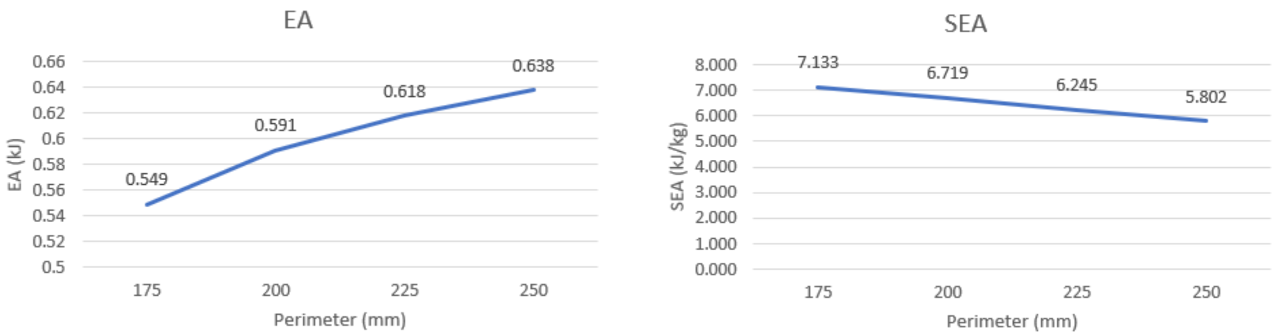


(a) EA of decagonal 10-cell crash box with varied thickness

(b) SEA of decagonal 10-cell crash box with varied thickness

Figure 5. Simulation results with varied thicknesses

The perimeter of the crash box is changed from 175 mm to 250 mm. EA and SEA obtained are shown in Figure 6. As the perimeter increases, EA rises as well. However, SEA reduces quite significantly if the perimeter is increased. Therefore, the increase in EA due to the increase in perimeter is insufficient to compensate for the increase in mass, causing SEA to reduce.

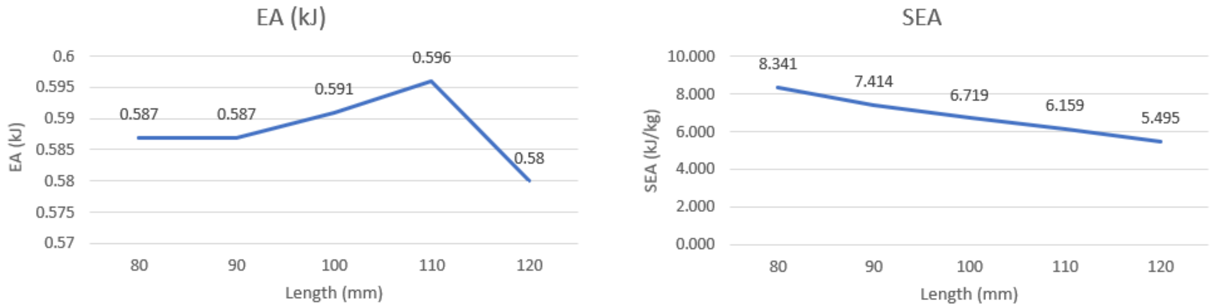


(a) EA of decagonal 10-cell crash box with varied perimeter

(b) SEA of decagonal 10-cell crash box with varied perimeter

Figure 6. Simulation results with varied perimeter

From Figure 7(a), EA rises only slightly when the length increases from 80 mm to 110 mm. Beyond 110 mm, EA decreases. SEA of the crash boxes with varied lengths is as shown in Figure 7(b), revealing a decrease in SEA when length is increased. It can be concluded that the change in the length of the crash box does not obviously affect the EA but reduces the SEA values significantly. Research by Ataollahi, et al. [19] also produced a comparable outcome, where silk/epoxy crash box with the shortest length produced the highest SEA. Since the crushing displacement is maintained at 60 mm, and the amount and surface area undergoing deformation are still the same, hence results in equivalent EA. Without an increase in EA, the greater mass caused by the lengthening of the crash box will cause a decline in EA.



(a) EA of decagonal 10-cell crash box with varied length

(b) SEA of decagonal 10-cell crash box with varied length

Figure 7. Simulation results with varied lengths

The optimal thickness for the decagonal 10-cell design is 2 mm, which produces the maximum SEA and EA. The length has an insignificant effect on EA but increasing it will cause SEA to reduce dramatically. Hence, it is maintained at the original value, which is 100 mm. An increase in perimeter increases EA but lowers SEA due to mass increase, so the original perimeter of 200 mm remains. Hence, a decagonal 10-cell design with 2 mm thickness, 100 mm length, and 200 mm perimeter is optimal. The optimized design is then evaluated and compared against the design before optimization and the steel conventional design (square cross-section) as shown in Table 3. The force-displacement graphs of the conventional square steel design, OPEFB fiber/epoxy decagonal 10-cell, and OPEFB fiber/epoxy optimized design are shown in Figure 8.

Table 3. Comparison of crashworthiness performance of steel and OPEFB/epoxy crash box.

Material		Steel	OPEFB/epoxy	
Design		Square	Decagonal 10-cell	Optimized Design
Crashworthiness Parameters	F_{peak} (kN)	91.5	19.1	32.6
	F_{mean} (kN)	28.167	9.850	15.683
	EA (kJ)	1.690	0.591	0.941
	SEA (kJ/kg)	7.542	6.719	8.023
	CFE	0.308	0.516	0.481
	Mass (kg)	0.224	0.088	0.117

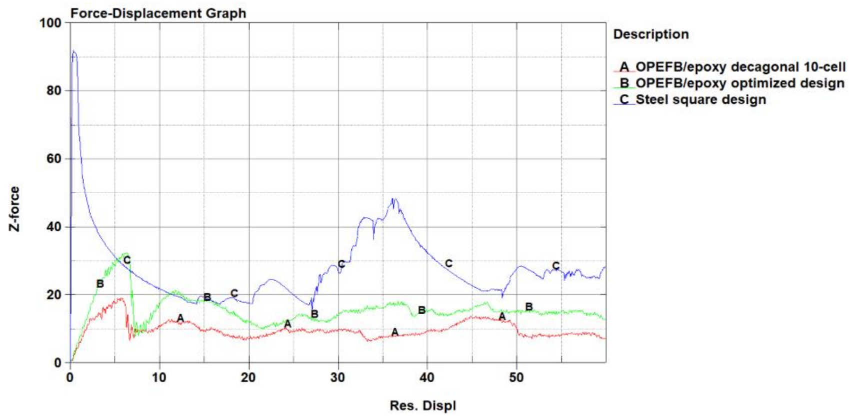


Figure 8. Force-displacement graph of steel and OPEFB/epoxy crash box

Compared to the original design, EA increases by 59%, SEA increases by 19%, but CFE decreases by around 6%. The decrease in CFE is still acceptable as the CFE is higher than the conventional steel crash box. Despite the improvements, the EA of the optimized OPEFB/epoxy is still 44% lower than the conventional steel design. The improved design cannot compensate for the bio-composite material's low stiffness and strength. However, the SEA of the optimized design is 6% higher than the steel crash box, which means it is more efficient in energy absorption in relation to its mass compared to steel. Besides, optimized OPEFB/epoxy crash boxes have 64% lower peak force than steel, which can be observed clearly in Figure 8. It prevents abrupt passenger impact and deceleration better than steel. In terms of CFE, OPEFB/epoxy optimized design outperforms steel by 56%, indicating that energy is absorbed more efficiently for OPEFB/epoxy. Lastly, OPEFB/epoxy is 48% lighter than steel.

3.3 Impact Test

The quasi-static test cannot replicate the decreasing of initial velocity from starting velocity to rest. For a better representation of the real collision, the conventional steel crash box and OPEFB/epoxy optimized design are subjected to low velocity impact test, in which the impactor is subjected to an initial velocity of 4 km/h, according to the low velocity impact standard by FMVSS 581 [20]. In addition, a point mass of 550 kg is applied to the impactor because the average car weighs 1100 kg and has 2 crash boxes.

The results of the impact tests are shown in Table 4. Both the EA for steel and OPEFB crash boxes are 0.340 kJ, which represents all the kinetic energy present in the impactor at the beginning, which is all transferred and absorbed by both crash boxes.

Table 4. Results of impact tests for steel and OPEFB/epoxy crash box

Material		Steel	OPEFB/epoxy
Design		Square	Optimized Design
Crashworthiness Parameters	Displacement (mm)	8.59	20.3
	F_{peak} (kN)	92	32.9
	F_{mean} (kN)	39.581	16.749
	EA (kJ)	0.340	0.340
	SEA (kJ/kg)	1.517	2.899
	CFE	0.43	0.509
	Mass (g)	0.224	0.117

For OPEFB/epoxy with an optimized design, the crushing displacement is higher than steel. Its peak force is 64% lower than steel, which means it can prevent sudden major impact and deceleration for passengers better than steel. Besides, the CFE of the optimized design is 18% higher than the steel crash box, indicating better energy absorption efficiency for OPEFB/epoxy. The performance of OPEFB/epoxy with optimized design is excellent in terms of SEA, where its SEA is 91% greater than that of the standard steel crash box. As both steel and OPEFB/epoxy absorb the same amount of energy during the low velocity impact, the lightweight property of the OPEFB/epoxy crash box contributes to its high SEA compared to steel. Comparing the mass of steel and OPEFB/epoxy crash box, the mass of steel with a square design is almost 2 times the mass of the OPEFB/epoxy with an optimized design. Hence, with the same amount of energy absorbed, the OPEFB/epoxy crash box with a lighter mass outperforms the steel crash box and can contribute to the reduction in fuel consumption in vehicles.

4.0 CONCLUSIONS

LS-DYNA quasi-static simulation is performed to compare bio-composites like OPEFB/epoxy and kenaf/epoxy against steel and carbon/epoxy crash boxes. To assess crash box performance, square, hexagonal, decagonal, hexagonal 3-cell, hexagonal 6-cell, and decagonal 10-cell crash boxes are proposed. Results show that bio-composite crash boxes perform better for CFE but worse for EA and SEA. Decagonal 10-cell bio-composites have the greatest SEA and EA. Hence, the OPEFB/epoxy decagonal 10-cell crash box's thickness, perimeter, and length are optimized to enhance EA. Increasing thickness raises the EA and SEA significantly, while altering the length has little effect on EA but causes SEA to reduce dramatically. An increase in perimeter causes EA to increase, but is compensated by a drop in SEA. Hence, the design is optimized by increasing thickness and maintaining length and perimeter. The optimized design improves EA by 59% and SEA by 19% over the original design. The last step is to compare the optimized design to a standard square steel crash box while being subjected to quasi-static and impact loads. For the quasi-static test, optimized OPEFB/epoxy design is 44% lower in EA than steel, but 56% higher in CFE and 6 % higher in SEA. When subjected to impact loading, both steel and OPEFB/epoxy crash boxes can absorb all the energy in the crash. Particularly, optimized OPEFB/epoxy design outperforms steel crash boxes by 91% in terms of SEA. Hence, the OPEFB/epoxy, with its lower mass but equivalent energy absorption, outperforms the steel crash box under low-velocity impact conditions and can help bring down vehicle fuel usage.

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6.0 REFERENCES

[1] A. M. Moghaddam, A. Kheradpisheh, and M. Asgari, "A basic design for automotive crash boxes using an efficient corrugated conical tube," *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering,* vol. 235, no. 7, pp. 1835-1848, 2021.

[2] E. Acar, M. Guler, B. Gerceker, M. Cerit, and B. Bayram, "Multi-objective crashworthiness optimization of tapered thin-walled tubes with axisymmetric indentations," *Thin-Walled Structures,* vol. 49, no. 1, pp. 94-105, 2011.

[3] Z. Li, Q. Yu, X. Zhao, M. Yu, P. Shi, and C. Yan, "Crashworthiness and lightweight optimization to applied multiple materials and foam-filled front end structure of auto-body," *Advances in Mechanical Engineering,* vol. 9, p. 168781401770280, 2017.

[4] S. A. Meguid, M. Attia, and A. Monfort, "On the crush behaviour of ultralight foam-filled structures," *Materials & Design,* vol. 25, no. 3, pp. 183-189, 2004.

[5] A. Sadighi, A. Eyvazian, M. Asgari, and A. M. Hamouda, "A novel axially half corrugated thin-walled tube for energy absorption under Axial loading," *Thin-Walled Structures,* vol. 145, p. 106418, 2019.

[6] T. Reddy, V. Narayanamurthy, and Y. Rao, "Study on crush tube geometric cross sections and topology for axial crashworthiness," *Defence Science Journal,* vol. 70, no. 3, p. 249, 2020.

[7] Z. Tang, S. Liu, and Z. Zhang, "Analysis of energy absorption characteristics of cylindrical multi-cell columns," *Thin-Walled Structures,* vol. 62, pp. 75-84, 2013.

[8] J. M. Alexander, "An approximate analysis of the collapse of thin cylindrical shells under axial loading," *The Quarterly Journal of Mechanics and Applied Mathematics,* vol. 13, no. 1, pp. 10-15, 1960.

[9] N. Jones, *Structural impact*. Cambridge University Press, 2011.

[10] T. Wierzbicki and W. Abramowicz, "On the crushing mechanics of thin-walled structures," *Journal of Applied Mechanics*, vol. 50, no. 4a, pp. 727-734 1983.

[11] H. Mahmood and A. Paluszny, "Design of thin walled columns for crash energy management—their strength and mode of collapse," *SAE Transactions,* pp. 4039-4050, 1981.

[12] N. N. Hussain, S. P. Regalla, Y. V. D. Rao, and A. M. Mohammed, "An experimental and numerical analysis on influence of triggering for composite automotive crash boxes under compressive impact loads," *International Journal of Crashworthiness,* vol. 7, no. 4, pp. 1-15, 2021.

[13] M. Kamran, P. Xue, N. Ahmed, M. Zahran, and A. Hanif, "Axial crushing of uni-sectional bi-tubular inner tubes with multiple outer cross-sections," *Latin American Journal of Solids and Structures,* vol. 14, pp. 2198-2220, 2017.

[14] W. Chen and T. Wierzbicki, "Relative merits of single-cell, multi-cell and foam-filled thin-walled structures in energy absorption," *Thin-Walled Structures,* vol. 39, no. 4, pp. 287-306, 2001.

[15] C.S. Hassan, P. Qiang, S.M. Sapuan, N.A. Aziz, M.Y.M. Zuhri, and R.A. Ilyas, "Unidirectional oil palm empty fruit bunch (OPEFB) fibre reinforced epoxy composite car bumper beam – effects of different fibre orientations on its crash performance," In: Sapuan SM and Ilyas RA, editors. *Biocomposite and synthetic composites for automotive applications*, Woodhead Publishing, 2021, pp. 233-253.

[16] Faris M. AL-Oqla, "Effects of intrinsic mechanical characteristics of lignocellulosic fibres on the energy absorption and impact rupture stress of low density polyethylene biocomposites," *International Journal of Sustainable Engineering*, vol. 14, no. 6, pp. 2009-2017, 2021.

[17] K.P. Wong, C.S. Hassan, P. Qiang, S.M. Sapuan, N.A. Aziz, and M.Y.M. Zuhri, "Crash behaviour of unidirectional oil palm empty fruit bunch fibre-reinforced polypropylene composite car bumper fascias using finite element analysis," *Functional Composites and Structures*, vol. 3, no. 4, p. 044001, 2021.

[18] N. Yusof, S. Sapuan, M. Sultan, and M. Jawaid, "Conceptual design of oil palm fibre reinforced polymer hybrid composite automotive crash box using integrated approach," *Journal of Central South University,* vol. 27, no. 1, pp. 64-75, 2020.

[19] S. Ataollahi, S. T. Taher, R. A. Eshkoor, A. K. Ariffin, and C. H. Azhari, "Energy absorption and failure response of silk/epoxy composite square tubes: Experimental," *Composites Part B: Engineering,* vol. 43, no. 2, pp. 542-548, 2012.

[20] L. Yan and N. Chouw, "Crashworthiness characteristics of flax fibre reinforced epoxy tubes for energy absorption application," *Materials & Design,* vol. 51, pp. 629-640, 2013.

[21] F. Pled, W. Yan, and C. E. Wen, "Crushing modes of aluminium tubes under axial compression," *arXiv preprint arXiv:1408.5390,* 2014.

[22] A. Porwal, A. Tripathi, and P. Krishnasamy, *Frontal Crash Worthiness Performance of Bi-Tubular Corrugated Conical: Structures under Axial Loads at Low Velocity*. 2020.

[23] K. D. Karantza and D. E. Manolakos, "Crashworthiness analysis of square aluminum tubes subjected to oblique impact: Experimental and numerical study on the initial contact effect," *Metals,* vol. 12, no. 11, p. 1862, 2022.

[24] N. H. T. S. Administration, "Federal motor vehicle safety standards and regulations," *US Government Printing Office, Washington, DC,* 1999.

[25] C. S. Hassan, C. W. Yeo, B. B. Sahari, S. M. Sapuan, and N. A. Aziz, "Mechanical properties of unidirectional oil palm empty fruit bunch (OPEFB) fiber reinforced epoxy composite," in *IOP Conference Series: Materials Science and Engineering,* vol. 206, no. 1: IOP Publishing, p. 012045, 2017.

[26] A. D. Bruno and M. Baskaran, "Analysing the mechanical properties of natural fiber reinforced polymer composites using FEA," *International Journal of Engineering Sciences & Research Technology,* vol. 3, no. 1, pp. 269-282, 2014.

[27] N. Manap, A. Jumahat, and A. Ludin, "Tensile and compressive properties of glass reinforcement in kenaf reinforced epoxy composite," *Advances in Environmental Biology,* vol. 8, pp. 2673-2682, 2014.

[28] P. Feraboli and B. Wade, "Simulating laminated composite materials using LS-DYNA material model MAT54: Single-element investigation," *Federal Aviation Administration Technical Report,* 02/01 2015.

[29] C. W. Isaac and C. Ezekwem, "A review of the crashworthiness performance of energy absorbing composite structure within the context of materials, manufacturing and maintenance for sustainability," *Composite Structures,* vol. 257, p. 113081, 2021.

[30] X. Li, Y. Wang, X. Xu, X. Cao, and R. Li, "Energy absorption characteristics of crash box of new honeycomb core structure with foam-filled," *International Journal of Automotive Technology,* vol. 22, no. 1, pp. 221-230, 2021.