



Performance of Multi-Layered Honeycomb Sandwich Composites Under Low-Velocity Impact

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Abstract

This paper presents experimental, analytical, and numerical models to predict the response of multi-layered sandwich composite panels under low-velocity impact loading. The sandwich panels' skin consists of a twill carbon-reinforced epoxy resin, whereas the core comprises a 2D Nomex honeycomb. The panels are then subjected to transverse impact loading to investigate their impact behaviour. Analytical models were developed based on a spring-mass system to predict the dynamic behaviour of the striker-multi-core-sandwich plate domain and, finally, to determine the contact force history, which represents the main novelty of this research. The analytical models incorporate the effect of variable core and skin distribution to identify the most suitable combination for four designs under impact loading. These experimental results are then used to verify analytical and numerical models constructed in LS-Dyna. The finite element models of the honeycomb-reinforced sandwich panels are also investigated using MAT-054 and MAT-26 material cards in LS-Dyna to find the most economical computational approach. Finally, the energy-absorption characteristics calculated by analytical models are used to evaluate the performance of the multi-layered sandwich composite and to provide design recommendations. The specific energy absorption (SEA) under low-velocity impact (8.8 J) was compared across designs, revealing that the single-core panel achieved the highest SEA (78 J/kg), while multi-core and thicker builds provided higher stiffness and peak loads but reduced mass efficiency.

Keywords Multi-core · Sandwich composites · Low-velocity impact · Analytical model · LS-Dyna

Introduction

Sandwich composites are widely employed in aerospace structures such as fuselage panels, control surfaces, and spacecraft modules due to their excellent stiffness-to-weight ratio [1, 2]. Historically, these structures have been widely adopted in aircraft fuselages, control surfaces, and spacecraft modules, where the combination of high load-bearing capacity and low weight is critical for structural efficiency [1]. The aerospace industry has also relied on sandwich composites for nacelles, landing gear doors, and secondary structures because of their favourable balance of weight reduction and structural integrity [2].

Unlike metals, which dissipate energy primarily through plastic deformation, sandwich composites rely on

progressive damage mechanisms such as delamination, fibre fracture, core shear, and local indentation [3]. The crash-worthiness of sandwich composites, defined by their ability to absorb impact energy while preventing catastrophic failure, has attracted extensive research interest [4]. While these mechanisms enable high specific energy absorption, they also complicate the prediction of impact responses. For instance, Gibson and Ashby [5] noted that the mechanical response of cellular solids such as honeycombs is governed by complex nonlinear deformation and collapse modes, making modelling difficult.

Most of the published literature available focuses on experimental and numerical simulations to investigate the behaviour of single-layered structures under low-velocity impact [6–8]. Analytical studies have also focused on these traditional sandwich panels, which typically feature two facesheets and a single low-density core [9, 10]. Although such studies have advanced the understanding of impact behaviour, they do not capture the potential of multi-layered sandwich constructions.

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As noted in recent research, multi-layered configurations, incorporating multiple cores and additional facesheets, may alter stiffness, strength, and energy absorption capacity in ways that differ significantly from classical single-core panels [11–13]. It has also been observed that double-layered honeycomb sandwich panels show higher energy absorption than single-layered panels of the same material [14]. Despite this potential, the performance of multi-layered sandwich composites under low-velocity impact conditions has not been extensively explored.

A recent experimental study by Fan et al. [15] explored tandem Nomex honeycomb sandwich panels with multiple core types and impactor shapes at energies up to 125 J. They found that hybrid cores increased the initial contact force by around 34.84% and reduced the maximum impact displacement by 51% relative to single-layered panels, highlighting the superior crashworthiness performance of multi-core structures compared to traditional single-core ones. However, this behaviour depends on the ply thickness and overall material density.

Analytical approaches have historically been pursued as a means to predict impact behaviour more efficiently than finite element simulations. Plate theories, such as first-order shear deformation theory (FSDT), have formed the basis for simplified models of impact response. However, while effective for preliminary predictions, these models have often proven too simplistic for complex sandwich architectures. Numerical models, particularly finite element simulations with codes such as LS-Dyna, provide greater accuracy but remain computationally intensive, limiting their utility in early-stage design optimisation.

This study addresses a research gap by investigating the performance of multi-layered CFRP/Nomex honeycomb sandwich composites under low-velocity drop-weight impact. Four different panel configurations were designed, fabricated, and tested experimentally, varying the number of cores and facesheets. An analytical model based on FSDT was developed and extended to multi-layered architectures, and its predictions were benchmarked against both experimental results and finite element simulations. By combining experimental, analytical, and numerical approaches, this work seeks to provide a comprehensive understanding of the crashworthiness of multi-layered sandwich structures and to establish a reliable yet efficient modelling framework for aerospace applications.

Experimental Studies

Four different $100\text{ mm} \times 100\text{ mm}$ configurations were made using CFRP and Nomex honeycomb for investigation within the scope of this research. Figure 1 depicts all the

thicknesses and dimensions used to manufacture the samples for each configuration.

From the mass comparison in Table 1 can be noted that compared to the density of the CFRP skins, the density of the Nomex honeycomb core is negligible; therefore, the number of CFRP layers influences the mass of the overall sandwich panel.

Sandwich Panels Manufacturing

Sandwich composite panels are made using three standard processes. Co-curing, the first technique, involves bonding both skins to the core, either with or without the use of an adhesive film, and curing the entire sandwich panel in a single step [16]. Co-bonding, the second technique, entails two curing procedures [17]. One skin is attached to the core while the other is curing. The third technique, called secondary bonding, consists of three curing steps: the two skins are first cured separately, then they are adhered to the core and cured once more [16]. In this research, the first method (co-curing) was used to manufacture the samples due to its simplicity, lower cost, and improved bonding strength. Figure 2 shows the final samples after they were cut, cured and trimmed (The British Standard of BS EN ISO 14125 has been used to manufacture the test specimens).

Experimental Testing Process

For this research, a drop-tower impact test was used to simulate low-velocity/low-energy impact. Impact velocity was calibrated using the kinetic energy principle after determining the desired energy of the impactor, and impact force, displacement, and velocity were measured using sensors.

A hemispherical impactor with a diameter of 20 mm , a mass of 4.37 kg , and the velocity of 2 m/s is used to produce around 8.8 J of energy (see Fig. 3). Two specimens are tested for each configuration, and the maximum standard deviations recorded during the tests are presented in Table 2.

Figure 4 shows the mean load-time responses for the two specimens tested in each configuration. This highlights the relative stiffness and peak load capacity of the different sandwich designs under low-velocity impact. Configuration 4 reaches the highest mean peak load, followed by Configuration 2, 3, and lastly 1.

Figure 5 presents the mean load-displacement responses for the two specimens tested in each configuration. Configuration 4 exhibits the highest mean peak load and stiffness, followed closely by Configuration 2. Configurations 1 and 3 show lower load-carrying capacity, with Configuration 1 being the most compliant.

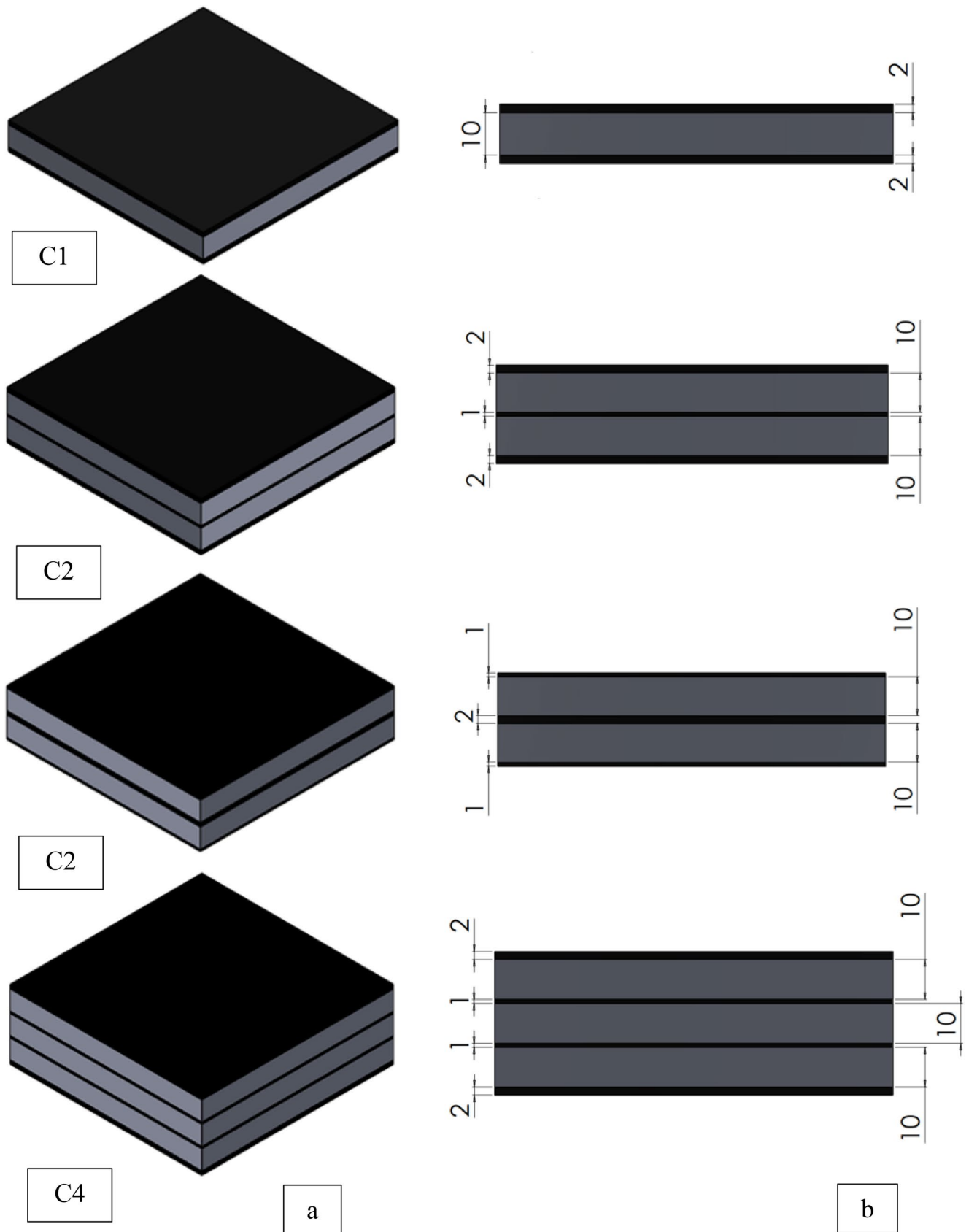


Fig. 1 CAD drawings of all configurations C1 – C4: (a) isometric view and (b) side view (Dimensions in mm)

Table 1 Mass comparison of configurations

Configuration	Number of plies	Number of cores	Mass (<i>kg</i>)
<i>C1</i>	12	1	0.060
<i>C2</i>	15	2	0.080
<i>C3</i>	12	2	0.065
<i>C4</i>	18	3	0.100

Following testing, the specimens' impact faces showed no apparent defects, as expected given that low-velocity impacts result in barely visible impact damage (BVID). The specimens were cut in half and examined under a microscope to better assess the extent of damage caused by the low-velocity impact. This section details the observations made from the cut samples.

The impacted specimens for various designs are shown in Fig. 6. Post-impact microscopy of all the specimens demonstrates characteristic barely visible impact damage (BVID) morphology, with a localised crushed zone directly

under the impact location where the honeycomb cell walls have kinked and densified due to partial skin-core debonding over a couple of cells and delamination within the upper facesheet.

The absorbed energy and mass of each configuration are presented in Table 3, and then used to calculate specific energy absorption (SEA).

Configuration 1 (single core) shows the best crash-worthiness performance with the highest SEA value of $78.4 \pm 0.4 \text{ J/kg}$, which absorbed the highest energy per unit mass, indicating efficient structural design for energy absorption. Configuration 3 (double core) performs well, with a slightly lower SEA of $72.4 \pm 3.3 \text{ J/kg}$ compared to configuration 1 due to the increase in weight. Although configurations 2 and 3 both have two cores, configuration 2 has more CFRP plies, resulting in a greater weight increase than configuration 1 but less than configuration 3. This causes the SEA of configuration 2 to decrease significantly

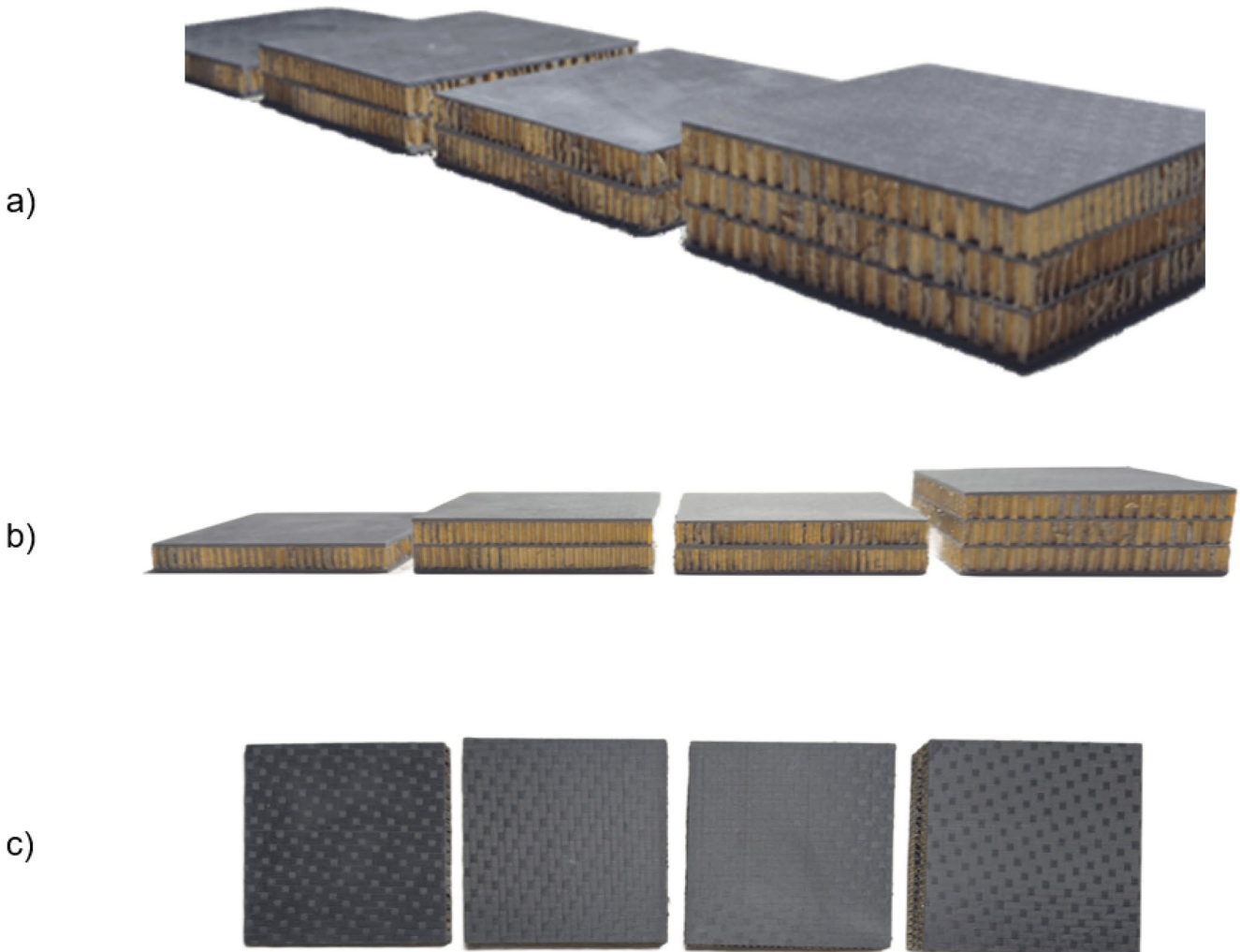
**Fig. 2** Final sandwich panel specimens in different views

Fig. 3 (a) Drop tower test facility and (b) Hemispherical impactor

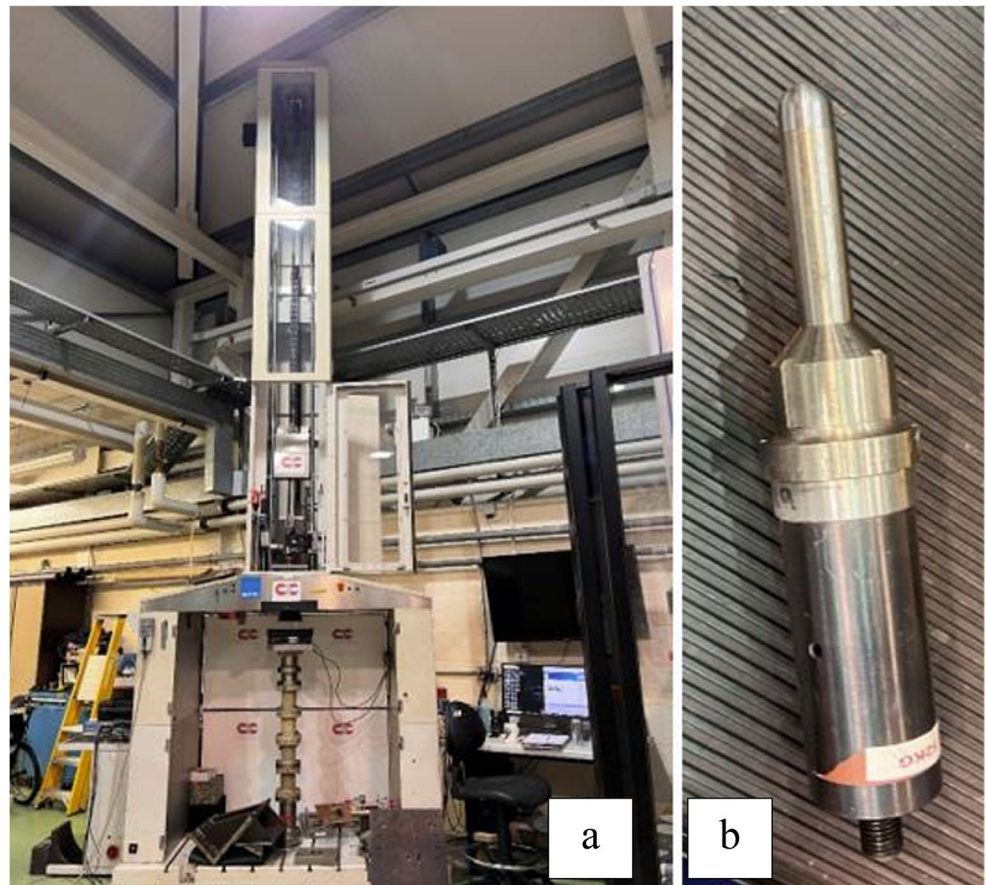


Table 2 Maximum standard deviation in load and displacement

Configuration	Standard Deviation	
	Load (<i>kN</i>)	Displacement (<i>mm</i>)
<i>C1</i>	± 0.25	± 0.3
<i>C2</i>	± 0.23	± 0.8
<i>C3</i>	± 0.49	± 1.0
<i>C4</i>	± 0.08	± 0.3

to a value of $45.4 \pm 1.8 \text{ J/kg}$ compared to configuration 3, decreasing its efficiency in low-velocity impact scenarios. Configuration 4 has the lowest SEA out of all configurations

at $37.7 \pm 0.8 \text{ J/kg}$, indicating that it is the least mass-efficient configuration and likely an overdesign for low-velocity impact scenarios.

Analytical Modelling of the Multi-Layered Honeycomb Sandwich Panels

Shear effects across the thickness must be taken into account due to the anisotropy of composites and their high ratio of Young's modulus to shear modulus. Thus, the first-order

Fig. 4 Mean load vs. time comparison for all configurations

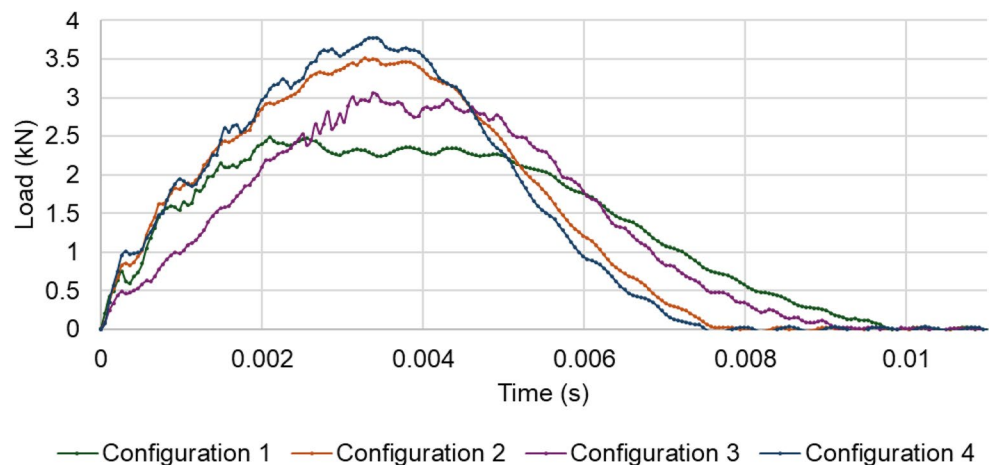


Fig. 5 Mean load vs. displacement comparison for all configurations

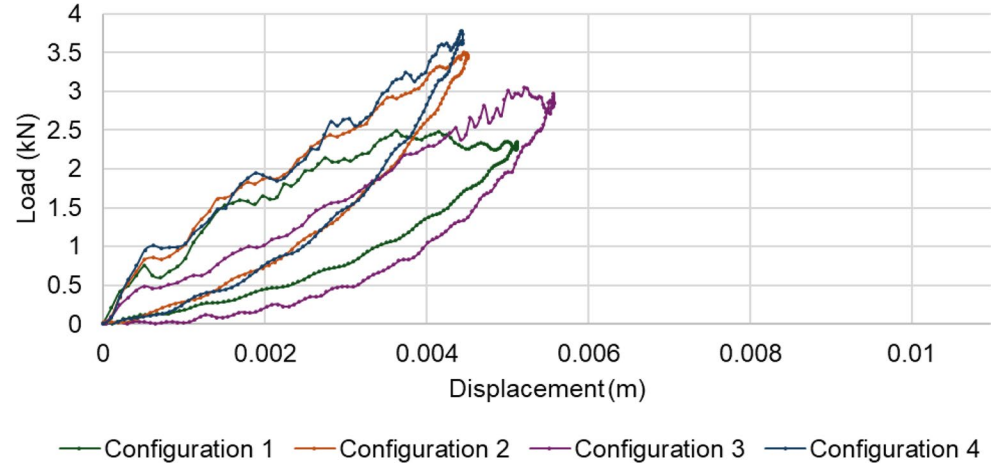
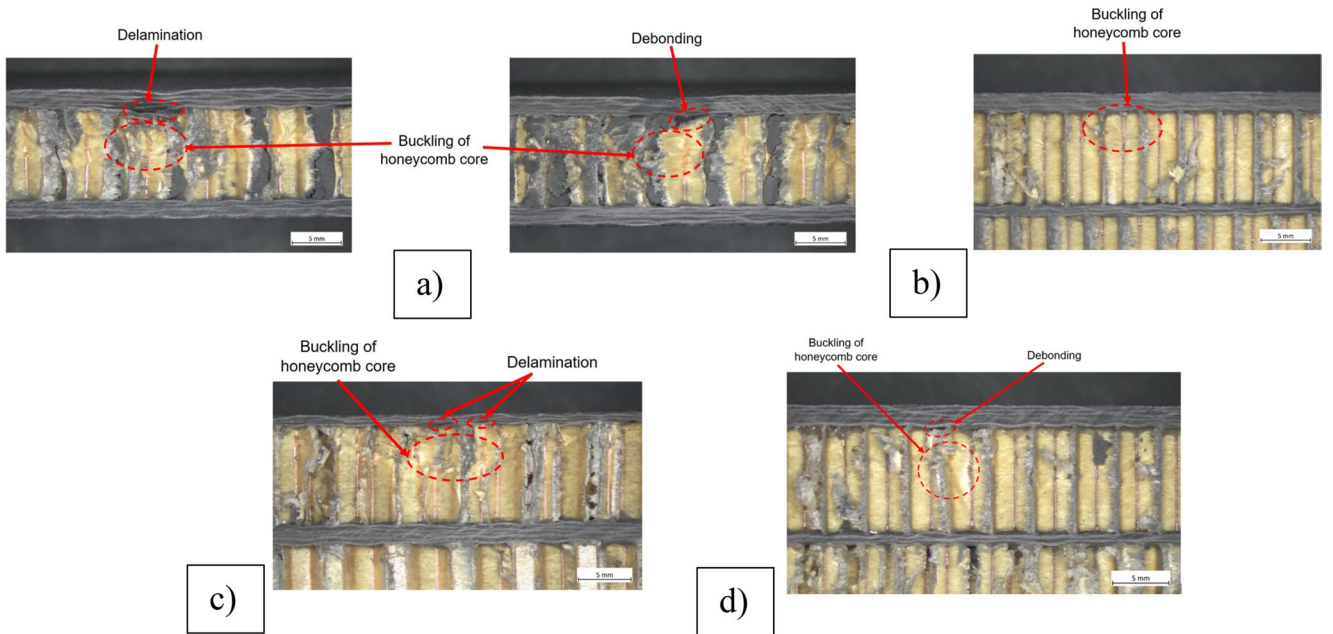


Fig. 6 Damage assessment for: (a) Configuration 1, (b) Configuration 2, (c) Configuration 3, and (d) Configuration 4

Table 3 Mean energy absorbed, sample mass and SEA of each configuration

Configuration	Peak Load (kN)	mean $E_{absorbed}$ (J)	m_{sample} (kg)	mean SEA (J/kg)
C1	2.49 ± 0.04	4.70 ± 0.03	0.060	78.4 ± 0.4
C2	3.51 ± 0.17	3.63 ± 0.15	0.080	45.4 ± 1.8
C3	2.97 ± 0.35	4.71 ± 0.20	0.065	72.4 ± 3.3
C4	3.78 ± 0.05	3.78 ± 0.08	0.100	37.7 ± 0.8

shear deformation theory (FSDT) was used in this study to model plate behaviour. The governing displacement field was written as shown in the Eq. (1) to (3).

$$u(x, y, z, t) = u^0(x, y, t) + z\psi_x(x, y, t) \quad (1)$$

$$v(x, y, z, t) = v^0(x, y, t) + z\psi_y(x, y, t) \quad (2)$$

$$w(x, y, t) = w^0(x, y, t) \quad (3)$$

where u^0 , v^0 , w^0 are the displacements of the composite plate in x , y and z directions at the mid-plane, and ψ_x and ψ_y are the shear rotations in x and y directions. The plate equations developed by Whitney and Pagano [18] are used, and further reduced for the case of specially orthotropic

laminates, i.e. ($B_{ij} = 0$, $A_{16} = A_{26} = D_{16} = D_{26} = 0$), and are shown in Eq. (4) to (6).

$$D_{11} \frac{\partial^2 \psi_x}{\partial x^2} + D_{66} \frac{\partial^2 \psi_x}{\partial y^2} + (D_{12} + D_{66}) \frac{\partial^2 \psi_y}{\partial x \partial y} - k_{sh} A_{55} \left(\psi_x + \frac{\partial w}{\partial x} \right) = I \frac{\partial^2 \psi_x}{\partial t^2} \quad (4)$$

$$(D_{12} + D_{66}) \frac{\partial^2 \psi_x}{\partial x \partial y} + D_{66} \frac{\partial^2 \psi_y}{\partial x^2} + D_{22} \frac{\partial^2 \psi_y}{\partial y^2} - k_{sh} A_{44} \left(\psi_y + \frac{\partial w}{\partial y} \right) = I \frac{\partial^2 \psi_y}{\partial t^2} \quad (5)$$

$$k_{sh} A_{55} \left(\frac{\partial \psi_x}{\partial x} + \frac{\partial^2 w}{\partial x^2} \right) + k_{sh} A_{44} \left(\frac{\partial \psi_y}{\partial y} + \frac{\partial^2 w}{\partial y^2} \right) + q(x, y, t) = \rho h \frac{\partial^2 w}{\partial t^2} \quad (6)$$

where w is plate displacement in z -direction at mid-plane, ψ_x and ψ_y are the shear rotations in x and y directions, k_{sh} is the shear correction factor, where $k_{sh} = \frac{\pi^2}{12}$, h is the plate thickness, t is time, and ρ is density.

The system of differential equations along with the boundary conditions outlined by Khalili et al. [19] define the solution of the vertical displacement w and shear rotations, ψ_x and ψ_y , for a simply supported rectangular plate of dimensions $a \times b$ under the load case q . The solution to the dynamic problem is based on the double expansion of loads, displacements, and rotations functions as Fourier series for the supported boundary condition [19]. By substituting the solutions of the governing equations and the load into the first-order shear deformation theory plate, it becomes Eq. (7)

$$[L_{ij}] \begin{bmatrix} A_{mn} \\ B_{mn} \\ W_{mn} \\ X_{mn} \\ Y_{mn} \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \\ \rho h \ddot{W}_{mn}(t) - Q_{mn}(t) \\ 0 \\ 0 \end{bmatrix} \quad (7)$$

where the symmetric coefficient matrix $[L_{ij}]$ are explained and detailed by Gong et al. [16]. There is no curvature in the plate, and where A_{44} and A_{55} are extensional shear matrix coefficients. k_{sh} is shear correction factor taken as $\pi^2/12$.

Equation (8) is then reduced to a single linear second-order differential equation using a method developed by Birman et al. [20] to give Eq. (8).

$$\ddot{W}_{mn}(t) + \omega_{mn}^2 W_{mn}(t) = \frac{Q_{mn}(t)}{\rho h} \quad (8)$$

where $\omega_{mn}^2 = -\frac{1}{K_{mn} \rho h}$ and $K_{mn} = \frac{\tilde{L}_{33}}{\det L}$. $\tilde{L}_{ij}$ are cofactors of the element L_{ij} in $\det L$. After calculating natural frequency, the spring-mass approach employed by authors [21] was used to model the dynamic behaviour, where an analytical function for the contact force history is expressed in Eq. (9).

$$F_{(t)} = K_2^* [A_1 (C_1 - 1) \sin \omega_1 t + A_2 (C_2 - 1) \sin \omega_2 t] \quad (9)$$

Equivalent Stiffness Matrix

The $[A]$ and $[D]$ stiffness matrices used as inputs in the impact model need to be adjusted for both woven fabric and sandwich cores. A woven layer is considered orthotropic when there is an equal number of fibres in the $+\Theta$ and $-\Theta$ directions in the $x_1 - x_2$ plane. For such a layer, the elements of the stiffness matrix are calculated from Eq. (10).

$$\overline{Q}_{ij}^{woven} = \frac{1}{2} \left[(\overline{Q}_{ij})_{+\Theta} + (\overline{Q}_{ij})_{-\Theta} \right] \quad i, j = 1, 2, 6 \quad (10)$$

Where $(\overline{Q}_{ij})_{+\Theta}$ and $(\overline{Q}_{ij})_{-\Theta}$ are the elements of the stiffness matrices of the plies oriented in the $+\Theta$ and $-\Theta$ directions respectively. Equation (10) when expanded, results in Eqs. (11) to (17).

$$\overline{Q}_{11}^{woven} = c^4 Q_{11} + s^4 Q_{22} + 2c^2 s^2 (Q_{12} + 2Q_{66}) \quad (11)$$

$$\overline{Q}_{22}^{woven} = s^4 Q_{11} + c^4 Q_{22} + 2c^2 s^2 (Q_{12} + 2Q_{66}) \quad (12)$$

$$\overline{Q}_{12}^{woven} = c^2 s^2 (Q_{11} + Q_{22} - 4Q_{66}) + (c^4 + s^4) Q_{12} \quad (13)$$

$$\overline{Q}_{66}^{woven} = c^2 s^2 (Q_{11} + Q_{22} - 2Q_{12}) + (c^2 - s^2)^2 Q_{66} \quad (14)$$

$$\overline{Q}_{16}^{woven} = 0 \quad (15)$$

$$\overline{Q}_{26}^{woven} = 0 \quad (16)$$

Once the $[\overline{Q}]^{woven}$ matrix has been calculated, it can be used to calculate $[A]$ and $[D]$ matrices as they normally would be. For sandwich composites, the $[A]$ and $[D]$ matrices are adjusted to include the effect of the core. For a single core sandwich composite the formulation used is as shown in (17) and (18) [22].

$$[A] = 2[A]^t \quad (17)$$

$$[D] = \frac{1}{2} d^2 [A]^t + 2[D]^t \quad (18)$$

where $[A]^t$ and $[D]^t$ are the stiffness matrices for a single skin of thickness t , and d is the distance between the mid-plane of the upper and lower skin. For multiple cores, the formulation in (19) and (20) was developed.

where N is the number of skins in the composite, $[A]^{tn}$ and $[D]^{tn}$ are the stiffness matrix of skin n with thickness t_n , and $d_{n \text{ to } (n+1)}$ is the distance between the midplane of skin n and skin $n + 1$. It is also important to note that $n = 1$ is used to represent the skin where the impactor strikes, whilst $n = N$ is the bottom skin of the sandwich composite, as shown in Fig. 7.

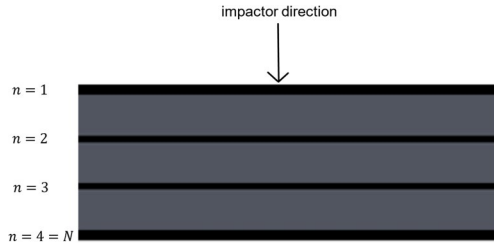


Fig. 7 Notation used for the mathematical model

$$[A] = \sum_{n=1}^N [A]^{t_n} \quad (19)$$

$$[D] = \frac{1}{2N} \sum_{n=1}^N d_{n \text{ to } (n+1)}^N [A]^{t_n} + \sum_{n=1}^N [D]^{t_n} \quad (20)$$

Validation of Analytical Model

The analytical models are compared with the experimental results to capture the initial slope, peak load, and the time to peak. It over-predicts the peak for Configuration 1, as shown in Fig. 8a, about 3.5 kN in the model versus about 2.6 kN in the lab, due to the experimental graphs having a plateaued peak, whilst the analytical model is simplified and is unable to capture this behaviour. However, the analytical model reproduces the initial slope of the load-time

curve and the time to peak (time at which impactor velocity = 0 mm/ms) well at about 4.0 ms in the model versus 4.5 ms in the lab, confirming that the model accurately predicts the stiffness of the sandwich panel.

In Configuration 2, the correlation between the model and experiment was strong, as shown in Fig. 8b), the peak load in the analytical model was around 3.6 kN versus about 3.5 kN in the lab. The initial slope of the load-time curve and the time to peak (time at which impactor velocity = 0 mm/ms) also matched well at about 3.9 ms in the model versus 3.7 ms in the lab.

In Configuration 3, the correlation between the model and experiment was slightly weaker but still within acceptable error, as shown in Fig. 8c. The peak load in the analytical model was around 3.1 kN versus about 3.3 kN in the lab. The initial slope of the load-time curve and the time to peak also matched well at about 4.1 ms in the model versus 4.4 ms in the lab.

In Configuration 4, the correlation between the model and experiment is the strongest as shown in Fig. 8d, with the peak load in the analytical model and experimental data matching exactly at 3.8 kN. The initial slope of the load-time curve and the time to peak (time at which the impactor velocity = 0 mm/ms) also matched well at about 3.3 ms in the model versus 3.6 ms in the lab. As the stiffness of the specimen increases, the accuracy of the predictive analytical

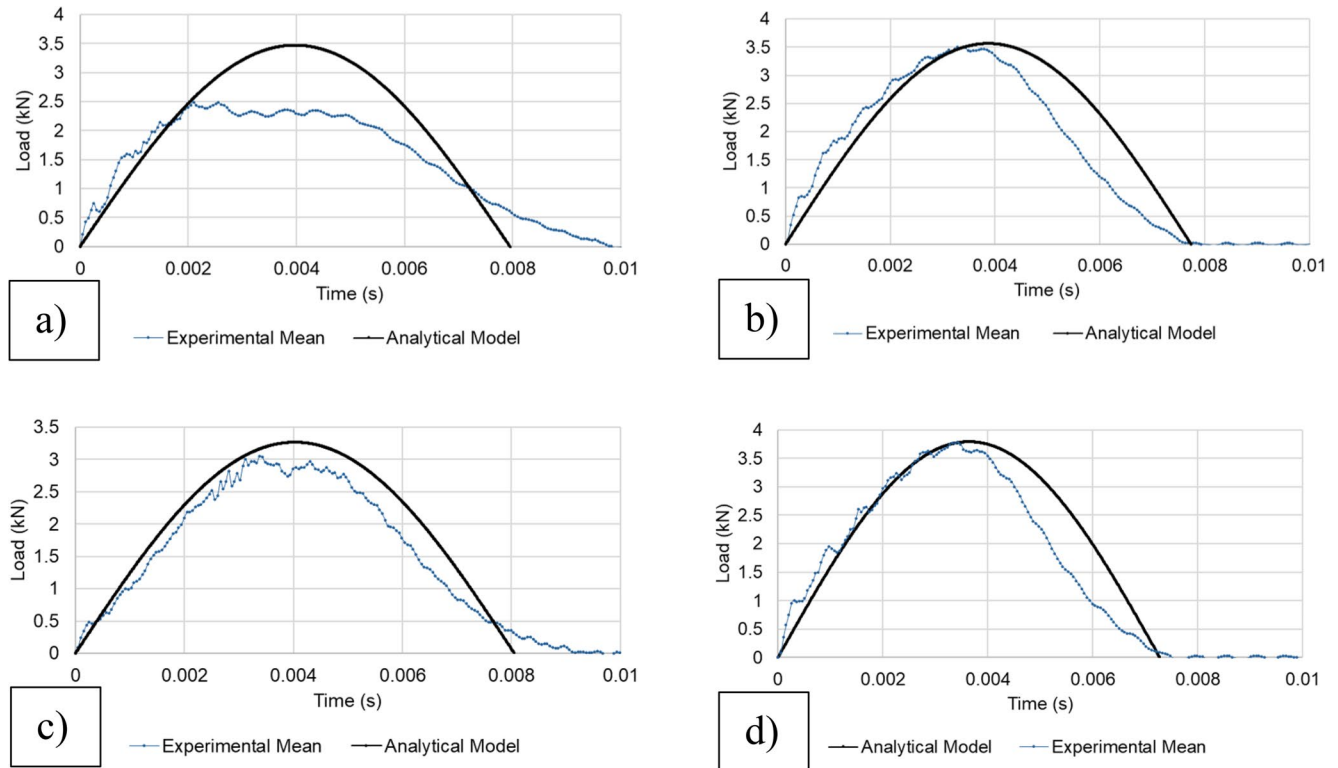


Fig. 8 Predicted load-time curve superimposed over experimental data for: (a) Configuration 1, (b) Configuration 2, (c) Configuration 3, and (d) Configuration 4

model also increases, resulting in more accurate estimates of stiffness, time to peak, and peak load for more complex multi-layered designs.

Finite Element Modelling

For the objectives of this research, numerical simulations were performed using LS-Dyna. LS-PrePost V4.11 was used to set up the model and analyse the results, while LS-Run R4 was used to run the simulation. The hemispherical impactor was modelled as a sphere using solid elements and a rigid material model (MAT-020), with its weight adjusted based on the sphere's density. The CFRP skins were modelled using shell elements and an enhanced composite damage material model (MAT-54), whilst the core was modelled using solid elements and a honeycomb material model (MAT-026).

The geometry of the core and skins was given a 2.5 mm hex mesh based on mesh sensitivity studies. Accurate findings were captured by maintaining a ten-element mesh along the thickness of the geometries.

The bottom edges of the plate were fixed in the z-direction, but were allowed to move in the x and y directions to imitate how the experimental setup held the specimen. Figure 9 shows the contacts that were defined in the LS-Dyna setup. The contacts defined are as follows:

- A: impactor – honeycomb has an automatic-surface-to-surface contact.
- B: impactor – top skin has an automatic-single-surface contact.
- C: impactor – bottom skin has an automatic-single-surface contact.
- D: top skin – honeycomb has an automatic-single-surface-tiebreak contact.
- E: bottom skin – honeycomb has a tiebreak-node-to-surface contact.

Finally, an initial velocity of 2.01 m/s was applied to the nodes of the impactor.

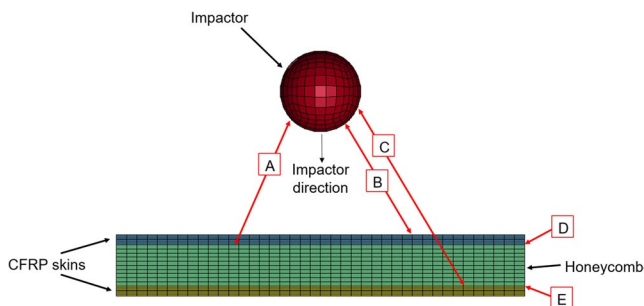


Fig. 9 LS-Dyna setup

Validation of Finite Element Studies

Finite element analysis in LS-Dyna was used to provide an additional verification of the experimental and analytical results. The force–time curves obtained from the simulations generally reproduced the initial stiffness and the time to peak with reasonable accuracy. However, differences were observed in the unloading phase due to the modelling of failure through element deletion.

In Configuration 1, the numerical prediction captured the initial slope of the load–time curve and the rise to peak with reasonable accuracy, as shown in Fig. 10a. The peak load was around 2.5 kN in the simulation compared with about 2.6 kN in the lab, demonstrating close agreement. The time to peak was predicted at approximately 4.0 ms in the model compared with 4.5 ms in the laboratory results. However, once the peak was reached, the simulated curve dropped more sharply than the experimental data, reflecting the fact that element deletion removed failed regions abruptly, whereas the physical specimen retained some residual load-carrying capacity after localised core crushing and delamination.

Configuration 2 showed a stronger correlation, as shown in Fig. 10b. The peak load in the FEA was around 3.4 kN, close to the 3.5 kN recorded in the lab. The time to peak was 3.8 ms compared with 3.7 ms experimentally, indicating excellent agreement in stiffness and timing. As in Configuration 1, the simulation's unloading behaviour diverged from the experiment, with a steeper decline due to the simplified failure representation.

For Configuration 3, the numerical model predicted a peak load of about 3.2 kN, compared with 3.3 kN in the lab, as shown in Fig. 10c. The time to peak was 4.0 ms in the simulation versus 4.4 ms in the lab, the model again demonstrated its ability to reproduce structural stiffness with reasonable accuracy. The unloading path, however, showed the same tendency to underestimate residual capacity, resulting in earlier load decay than in the experimental curve.

Configuration 4 gave a close match between simulation and experiment. As shown in Fig. 10d the peak load in the FEA was 3.6 kN compared with 3.8 kN in the lab. The time to peak was 3.6 ms in both the FEA and lab, again demonstrating that the global stiffness was well captured. The unloading response fell earlier in the numerical model than in the experiment, but the overall shape of the curve and the main impact characteristics were consistent.

High-velocity Impact of Multi-core Laminates

The above configurations have also been analysed for the high-velocity impact energy of 143.7 J to determine the

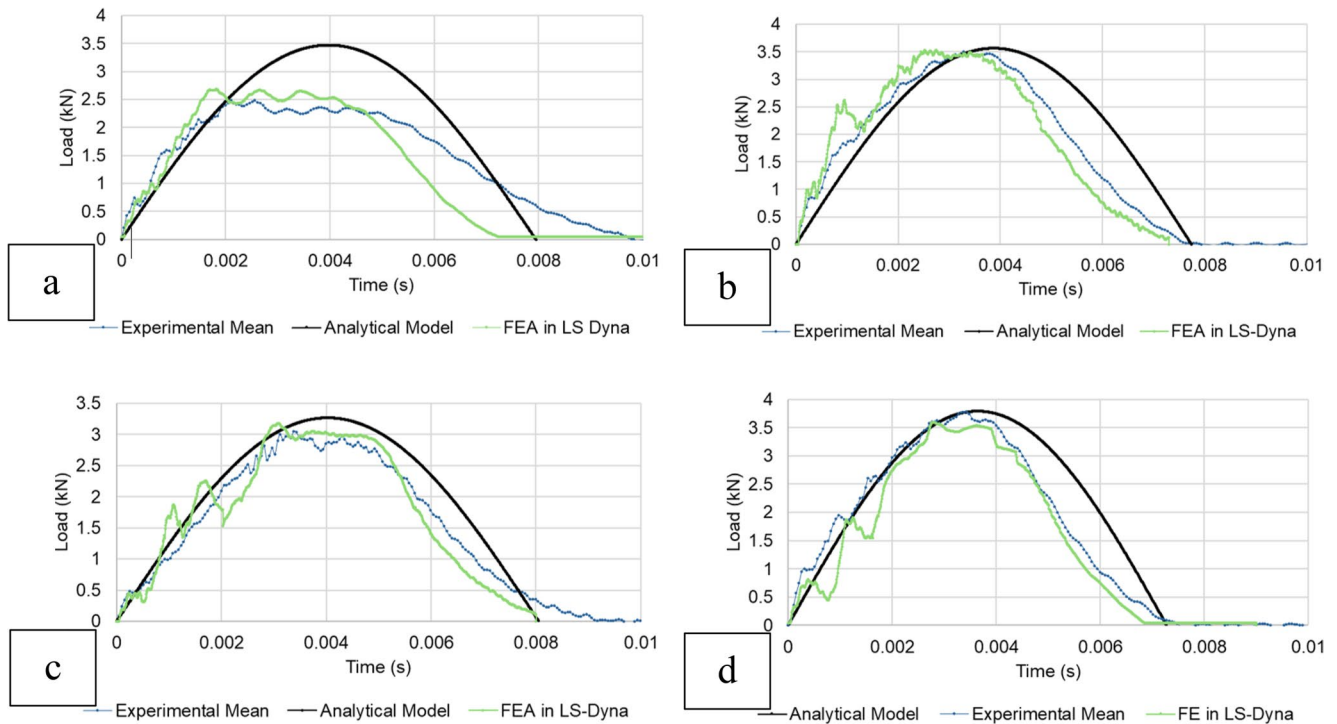


Fig. 10 Force-time comparison of experimental data, analytical model and FEA for: (a) Configuration 1, (b) Configuration 2, (c) Configuration 3, and (d) Configuration 4

energy-absorption capability of these designs relative to the low-velocity impact studies conducted in this research. The numerical results indicate that multi-core laminates are mainly suitable for high-velocity applications, including multi-core and shell components, in terms of the energy absorption capability of composite structures. Increasing cores and skins through the thickness of structures introduces various damage and failure scenarios. In particular cases, the arrangement of cores within a sandwich panel affects the number of pick loads in the force history during the impact event. The specific energy absorption has been tripled from 47 J/kg (Configuration 1) to 138 J/kg (Configuration 4) as shown in Fig. 11.

Conclusion

The mathematical model extends first-order shear deformation theory to multi-core sandwich panels by constructing laminate stiffness matrices for stacked skins and cores, coupled to a simplified impactor system. Across four panel configurations, it consistently reproduced the early-time response. Predictions of initial stiffness and peak load timing agreed closely with experiments, while peak load was slightly overestimated for the lightest panel and accurately

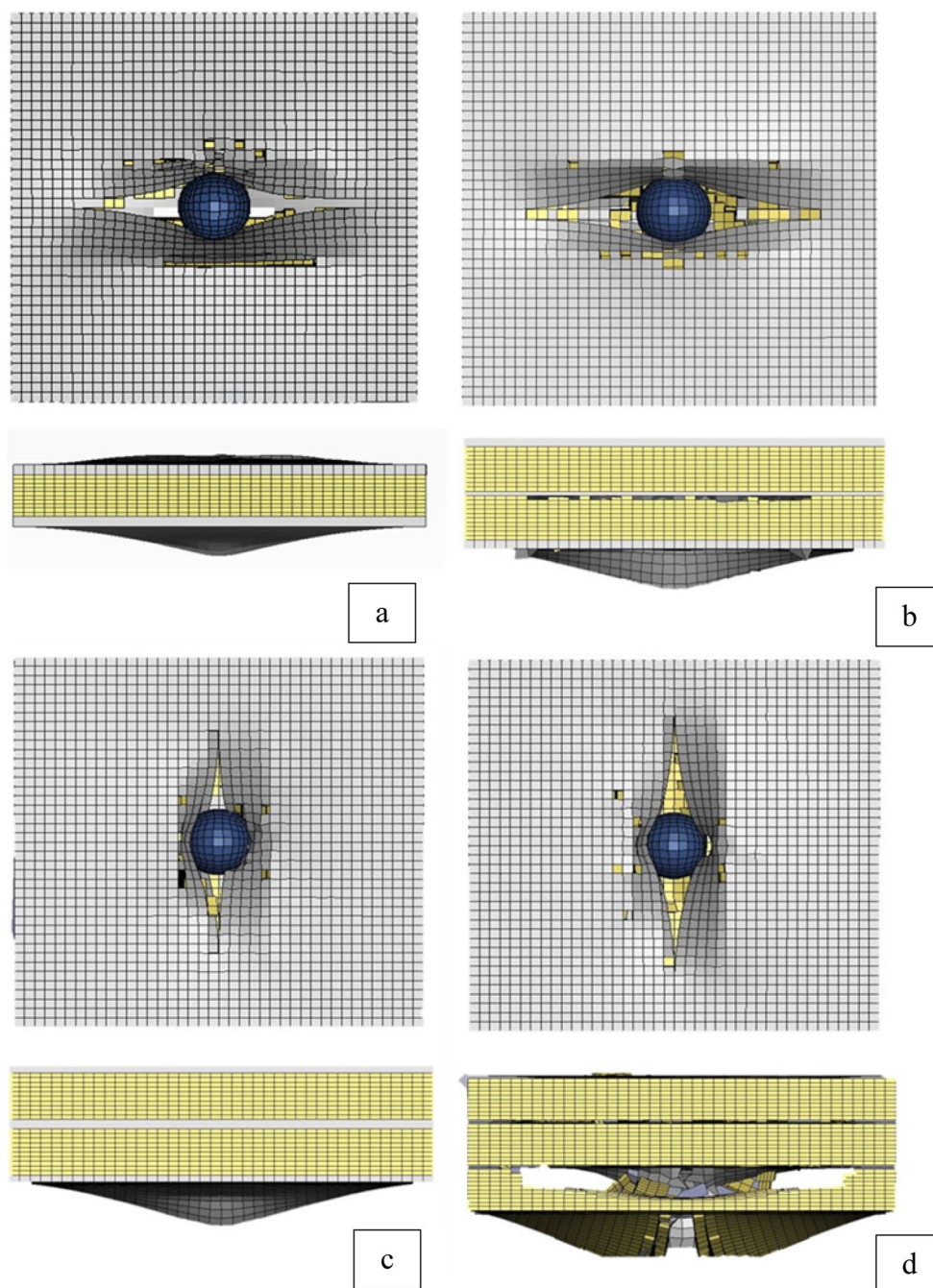
captured for the stiffer ones. Finite-element simulations in LS-Dyna supported these trends.

Experiments showed a clear trade-off between stiffness and specific energy absorption (SEA). Additional skins and cores increased stiffness and peak load, whereas the single-core panel achieved the highest SEA. Taken together, the tests, analytical model, and finite-element analyses suggest that thicker or multi-core panels are preferable when stiffness or peak load is critical, while lighter single-core panels are advantageous when mass-specific energy absorption is the priority. The model provides a fast and reliable way to screen such design choices before fabrication.

This work was limited to low-energy, low-velocity impact on panels of the tested geometries and layouts, with validation focused on early response and peak metrics. Within that scope, the model met its intended purpose and can be used with confidence to guide preliminary design. Future studies should extend the formulation to gently curved panels and configurations with cut-outs or inserts to assess their influence on peak load and SEA under high-energy impact in practical applications.

The finite element simulations reproduced the key global features of the impact response across all configurations, namely the initial stiffness, peak load, and time to peak, with an error of less than 10%. The primary limitation of the

Fig. 11 Plane and side views of the high-velocity impact response of four configurations under an impact energy of $143.7J$



numerical model was its treatment of post-peak behaviour, where element deletion resulted in a faster loss of load-carrying capacity than observed experimentally. Nevertheless, the close agreement in global response confirmed that the analytical predictions were reliable and that both experimental and numerical results support the validity of the proposed modelling approach.

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Data Availability The datasets generated during and/or analysed during the current study are available from the corresponding author upon request.

Declarations

Competing interests No competing interest is declared.

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