

Article

Effect of Ply Orientation and Face-Sheet Thickness on PSD-Based Random Vibration Response of Honeycomb Sandwich Structures

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Abstract

Electric vehicle battery enclosure systems under vibrational loading may experience structural damage, reducing the lifespan of lithium-ion cells. Honeycomb sandwich structures are widely used in EV battery systems due to their high stiffness-to-weight ratio and superior vibration performance. The present study is undertaken to investigate the dynamic behaviour of honeycomb sandwich panels using ANSYS Workbench. A three-dimensional model of a sandwich structure consisting of an aluminium core and composite face sheets is developed in ANSYS. Modal analysis and power spectral density-based random vibration analysis are performed to examine the impact of face-sheet thickness (0.5–2.5 mm) and layup configuration on structural performance. The modal analysis reveals that the natural frequencies increase considerably with thickness up to 1.5 mm due to increased bending stiffness. The results of random vibration analysis demonstrate a substantial reduction in total deformation and equivalent stress with increasing thickness. Among the configurations studied, the [0C/0G]/Core/[0G/0C] layup demonstrates the most favourable response under the present PSD loading condition due to fibre alignment with the principal loading direction. The findings are further converted into practical design guidelines for electric vehicle battery enclosures, including an appropriate face-sheet thickness range of 1.0–1.5 mm and fibre orientations aligned with the principal loading direction, while considering the associated mass penalty. These findings provide a design-oriented framework for selecting thickness and layup configuration to achieve a practical balance between vibration resistance and weight.

Keywords: random vibration analysis; honeycomb sandwich structure; electric vehicle battery enclosure; composite laminates; structural optimisation; design guidelines; sustainable mobility



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1. Introduction

The rapid expansion of electric vehicle (EV) technology has created a growing need for battery systems that are not only efficient but also lightweight and structurally robust [1].

In modern EVs, the battery stores energy and contributes to the overall structural integrity of the vehicle [2]. As a result, its mechanical performance becomes as critical as its electrochemical behaviour [3]. During operation, battery packs are exposed to various dynamic loads originating from road irregularities, drivetrain excitations, and external disturbances [3]. Unlike idealised loading conditions, these excitations are irregular and continuously varying, and are more appropriately characterised as random in nature [1,2]. Prolonged exposure to such vibration environments can gradually deteriorate structural performance, leading to issues such as loosening of connections, damage to internal components, and overall reduction in battery efficiency [4,5]. For this reason, the design of battery enclosures must account for stochastic vibration loading. Ensuring sufficient structural integrity under these conditions is essential for enhancing durability, maintaining safety, and improving the long-term reliability of electric vehicle systems [5].

Battery enclosures have traditionally been made using metallic materials such as aluminium and steel, mainly because of their strength and durability [6,7]. These materials perform well structurally, but they also add significant weight to the vehicle [8]. This added mass can reduce energy efficiency and limit the driving range, both of which are key concerns in electric vehicle design [9]. Because of this, attention has gradually shifted toward lightweight alternatives that can offer similar or even better mechanical performance [10,11]. Among these, honeycomb sandwich structures have attracted considerable interest. Their main advantage lies in their ability to provide high stiffness while keeping the overall weight low. In addition, they offer good energy absorption and improved vibration-damping behaviour [11–13]. A typical honeycomb sandwich structure is made up of a lightweight cellular core, often aluminium, placed between two composite face sheets [14,15]. This arrangement helps distribute loads more evenly and improves structural stability [16]. Such characteristics make these structures more resistant to bending and dynamic loading, which is particularly important for applications such as EV battery enclosures [17–19].

Recent studies have examined the use of honeycomb sandwich structures in EV battery enclosures [20–23]. Many of these works show that such designs can reduce structural weight without compromising the strength and stiffness needed to protect the battery [23]. Experimental and simulation-based studies also point out that honeycomb-based systems perform well under vibration, shock, and impact loading, which is important for maintaining structural safety during operation [21]. Finite element analysis (FEA) has become a common approach in this area. It helps in understanding how these structures behave under different loading conditions, including their natural frequencies, deformation patterns, and stress distribution [24,25]. Another important aspect is the use of composite materials, such as carbon and glass fibres, in the face sheets. These materials allow engineers to adjust properties such as stiffness and strength by varying fibre orientation, thereby making the structure more efficient for specific loading conditions [26–28].

Despite these advancements, much of the existing research still focuses on basic vibration analysis of sandwich structures. Many finite element (FE) studies have examined the influence of geometric parameters on the free vibration of sandwich panels. Boudjemai et al. [29] performed modal analysis of honeycomb sandwich structures used in satellite structural design to determine the natural frequencies. The influence of skin thickness, core thickness and cell size on the natural frequency was examined. Sakar and Bolat [30] performed free vibration analysis of honeycomb structures to determine the modal parameters of the structure. The influence of face-sheet thickness, cell angle and core thickness on the modal characteristics was investigated. Jweeg [31] performed vibrational analysis to determine the effect of core height and face plate thickness on the natural frequency of the honeycomb structure. Nikhil et al. [32] conducted modal analysis

to determine the vibration response of the honeycomb structure. It was found that cell wall thickness, face-sheet thickness and core height had a remarkable influence on the natural frequencies. Pourriahi et al. [33] performed vibrational analysis to determine the influence of geometric parameters on the vibration behaviour of the honeycomb panel. It was observed that changing the face plate thickness rather than core height had a greater impact on the vibration characteristics of the panel. Kormanikova et al. [34] carried out a free vibrational analysis to determine the natural frequencies of the honeycomb panel. The influence of panel length, core thickness and fibre orientation on the natural frequency was examined. Upreti et al. [35] performed vibrational analysis of honeycomb sandwich structures and observed that face-sheet thickness had a considerable impact on the vibrational characteristics. Jweeg et al. [36] performed a numerical investigation on the effect of face thickness, core height and cell angle on the natural frequency. It was concluded that natural frequency increases with an increase in face-sheet thickness and core height, whereas it decreases with an increase in cell angle. Patro et al. [37] studied the effect of cell size, face-sheet thickness and core thickness on the vibration behaviour of a honeycomb sandwich structure. Kumar et al. [38] performed modal analysis of a honeycomb sandwich panel to determine the modal parameters of the structure. It was noticed that core material and face-sheet thickness had a considerable impact on the modal parameters. Haris and Lee [39] performed finite element analysis to investigate the vibration behaviour of the honeycomb battery back. The effects of battery placement into the honeycomb core were examined. Hamza et al. [40] conducted a numerical study to determine the effect of cell size, core height and thickness of cell wall on the natural frequency of the honeycomb panel. Brethee et al. [41] carried out an FE study to examine the vibration behaviour of a honeycomb sandwich structure. It was observed that the natural frequency of the panel increased with an increase in cell size and core height. However, wall thickness and rib angle had a marginal influence on the natural frequency.

Mohanty et al. [42] investigated the vibration behaviour of a honeycomb structure by performing an FE study. Different types of geometry, such as hexagonal, square, triangular and circular, were considered for the core. The effect of the geometry of the core, the material of the core, and the face-to-core thickness ratio on the natural frequency was examined. Overall, these studies offer useful insights into structural performance, but they do not fully represent the complex and unpredictable vibration conditions experienced during real driving. In actual automotive environments, vibration inputs are better described using power spectral density (PSD) functions. These functions capture how vibration energy is distributed over a range of frequencies [43,44]. Random vibration analysis based on PSD, therefore, provides a more realistic way to evaluate structural response, as it accounts for the statistical nature of loading [45,46]. This makes it possible to estimate deformation, stress, and fatigue-related effects more accurately.

Although several studies have investigated the vibration behaviour of honeycomb sandwich panels, most have focused on modal characteristics such as natural frequencies and mode shapes under idealised loading conditions. Studies examining the PSD-based random vibration response of hybrid composite honeycomb structures for EV battery enclosure applications remain limited. Furthermore, the combined influence of face-sheet thickness and hybrid carbon–glass layup configuration under stochastic road-induced excitation has not been systematically investigated. The present study addresses this gap by performing parametric variation of thickness and layup configuration under PSD-based random vibration analyses and translates the findings into quantitative design recommendations for EV battery enclosure structures.

A three-dimensional model is developed in ANSYS Workbench, consisting of an aluminium honeycomb core and composite face sheets made from carbon and glass fibre

laminates. The analysis begins with a modal analysis to determine the natural frequencies and corresponding mode shapes, which helps us understand the dynamic characteristics of the structure. Subsequently, random vibration analysis is carried out using the PSD approach to represent real-world loading conditions. The structural response is evaluated in terms of total deformation and equivalent stress under these stochastic excitations. A parametric study is also included by varying key design parameters such as face-sheet thickness (0.5–2.5 mm) and various layup configurations. Furthermore, this study aims to convert the numerical results into design-oriented guidelines for EV battery enclosures. This study aligns with key UN Sustainable Development Goals, particularly SDG 9 (Industry, Innovation and Infrastructure).

2. Methodology

2.1. Theory

The theoretical basis of the present finite element model follows first-order shear deformation theory (FSDT), which is suitable for moderately thick laminated and sandwich structures because transverse shear deformation is included. The displacement field may be written as

$$u(x, y, z, t) = u_0(x, y, t) + z\theta_y(x, y, t) \quad (1)$$

$$v(x, y, z, t) = v_0(x, y, t) - z\theta_x(x, y, t) \quad (2)$$

$$w(x, y, z, t) = w_0(x, y, t) \quad (3)$$

where u_0 , v_0 , and w_0 are the mid-plane displacements, and θ_x and θ_y denote the rotations of the normal about the x and y axes, respectively.

The in-plane strain vector is expressed as

$$\{\varepsilon\} = \{\varepsilon^0\} + z\{\kappa\} \quad (4)$$

$$\{\varepsilon\} = \begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \gamma_{xy} \end{Bmatrix} \quad (5)$$

$$\{\varepsilon^0\} = \begin{Bmatrix} \frac{\partial u_0}{\partial x} \\ \frac{\partial v_0}{\partial y} \\ \frac{\partial u_0}{\partial y} + \frac{\partial v_0}{\partial x} \end{Bmatrix}, \quad \{\kappa\} = \begin{Bmatrix} \frac{\partial \theta_y}{\partial x} \\ -\frac{\partial \theta_x}{\partial y} \\ \frac{\partial \theta_y}{\partial y} - \frac{\partial \theta_x}{\partial x} \end{Bmatrix} \quad (6)$$

where $\{\varepsilon^0\}$ represents the mid-plane strain vector and $\{\kappa\}$ represents the curvature vector.

The transverse shear strain vector is expressed as

$$\{\gamma\} = \begin{Bmatrix} \gamma_{yz} \\ \gamma_{xz} \end{Bmatrix} = \begin{Bmatrix} \frac{\partial w_0}{\partial y} - \theta_x \\ \frac{\partial w_0}{\partial x} + \theta_y \end{Bmatrix} \quad (7)$$

For the k th orthotropic lamina, the transformed stress–strain relationship in the global laminate coordinate system is given by

$$\begin{Bmatrix} \sigma_x \\ \sigma_y \\ \tau_{xy} \end{Bmatrix}^{(k)} = \begin{bmatrix} \bar{Q}_{11} & \bar{Q}_{12} & \bar{Q}_{16} \\ \bar{Q}_{12} & \bar{Q}_{22} & \bar{Q}_{26} \\ \bar{Q}_{16} & \bar{Q}_{26} & \bar{Q}_{66} \end{bmatrix}^{(k)} \begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \gamma_{xy} \end{Bmatrix}^{(k)} \quad (8)$$

$$\{\sigma\}^{(k)} = [\bar{Q}]^{(k)} \{\varepsilon\}^{(k)} \quad (9)$$

where $[\bar{Q}]^{(k)}$ is the transformed reduced stiffness matrix of the (k^{th}) ply.

The laminate extensional, coupling, and bending stiffness matrices are obtained by integrating the transformed ply stiffnesses through the laminate thickness:

$$A_{ij} = \sum_{k=1}^n \bar{Q}_{ij}^{(k)} (z_k - z_{k-1}) \quad (10)$$

$$B_{ij} = \frac{1}{2} \sum_{k=1}^n \bar{Q}_{ij}^{(k)} (z_k^2 - z_{k-1}^2) \quad (11)$$

$$D_{ij} = \frac{1}{3} \sum_{k=1}^n \bar{Q}_{ij}^{(k)} (z_k^3 - z_{k-1}^3) \quad (12)$$

where z_k, z_{k-1} are the top and bottom coordinates of the ply.

$[A]$, $[B]$, and $[D]$ represent the extensional, bending–extension coupling, and bending stiffness matrices, respectively. This formulation shows that the bending stiffness is governed by both ply orientation and ply location through the transformed stiffness matrix and the distance from the laminate mid-plane.

The free vibration behaviour of the sandwich structure is governed by the equation of motion:

$$[M]\{\ddot{u}\} + [K]\{u\} = 0 \quad (13)$$

where $[M]$ and $[K]$ denote the mass and stiffness matrices, respectively, and $\{\ddot{u}\}$ and $\{u\}$ represent acceleration and displacement vectors.

For modal analysis, the free vibration response is obtained by solving the eigenvalue problem:

$$[K]\{\phi_i\} = \omega_i^2 [M]\{\phi_i\} \quad (14)$$

where ω_i represents the natural frequency of the i th mode (rad/s) and $\{\phi_i\}$ denotes the corresponding mode shapes.

The natural frequency (Hz) is given by

$$f_n = \frac{1}{2\pi} \sqrt{\frac{k}{m}} \quad (15)$$

where k -stiffness and m -mass.

In a honeycomb sandwich structure (Figure 1), the face sheets are located away from the neutral axis and therefore carry most of the tensile and compressive bending stresses, while the honeycomb core primarily provides transverse shear stiffness and maintains separation between the face sheets. The bending stiffness of a symmetric sandwich panel may be approximately expressed as

$$D = E_f \frac{bt_f^3}{6} + E_f \frac{bt_f d^2}{2} + E_c \frac{bh_c^3}{12} \quad (16)$$

where E_f and E_c are the elastic moduli of the face sheet and core, respectively, b is the panel width, t is the thickness of each face sheet, c is the core thickness, and d is the distance between the centrelines of the upper and lower face sheets. This expression indicates that the bending stiffness of a sandwich panel is strongly influenced by elastic moduli, face-sheet thickness, and separation distance between the face sheets.

As per pure bending theory,

$$\frac{M}{I} = \frac{\sigma}{y} = \frac{E}{R} \quad (17)$$

$$\text{Bending Moment, } M = EI\kappa \quad (18)$$

$$\text{Bending stress, } \sigma = E y \kappa \quad (19)$$

where y is the distance of the extreme fibre from neutral axis, E is the modulus of elasticity, I is the moment of inertia, κ is the curvature and R is the radius of curvature.

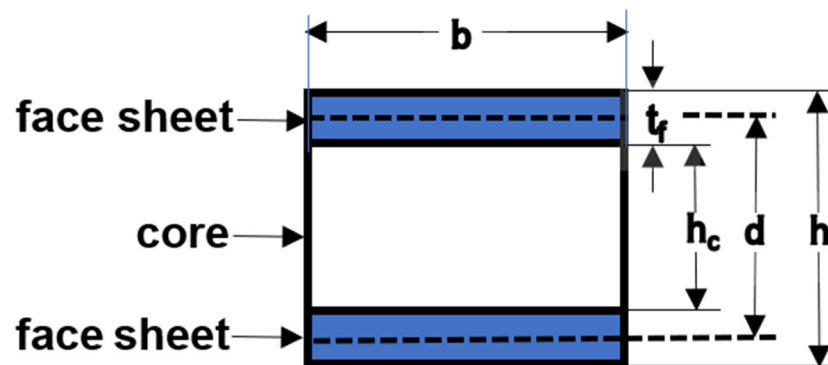


Figure 1. Idealised cross-section of the honeycomb sandwich panel.

The natural frequency of a structure is related to its stiffness-to-mass ratio. Therefore, increasing the effective bending stiffness of the face sheets, either by increasing thickness or by aligning fibres with the principal bending direction, increases the natural frequency and reduces the random vibration response.

In the present study, the composite face sheets are modelled using layered orthotropic laminate properties, while the honeycomb core is represented as an equivalent orthotropic material. This modelling approach follows established laminated composite and sandwich structural theories rather than introducing a new analytical plate formulation.

In PSD-based random vibration analysis, the excitation is described statistically using a power spectral density function, which represents the distribution of vibration energy over frequency. For a linear system subjected to stationary random excitation, the response is evaluated in the frequency domain using the modal superposition approach. The mean-square response is obtained from the area under the response PSD curve, and the RMS response is given by

$$x_{RMS} = \sqrt{\int_0^{\infty} S_x(f) df} \quad (20)$$

where $S_x(f)$ is the response PSD. For an approximately zero-mean Gaussian random response, the statistical peak response can be estimated using the three-sigma criterion:

$$x_{3\sigma} = 3x_{RMS} \quad (21)$$

Under this assumption, approximately 99.7% of the response amplitudes lie within $\pm 3\sigma$. Therefore, the three-sigma deformation and stress values reported in this study are used as conservative statistical estimates for comparing the vibration severity of different face-sheet thicknesses and layup configurations. These values should not be interpreted as deterministic maximum responses or fatigue-life predictions. The PSD–RMS relationship and three-sigma statistical estimate used in this study follow standard random vibration theory [47]. These equations provide the theoretical basis for interpreting the ANSYS modal and random vibration results.

Definition of Base Acceleration PSD Input for ISO 8608 Class G Road Profile

ISO 8608 [48] provides spatial road displacement PSD. Therefore, in the present work, it was used to define the relative roughness scaling between road classes, while

the acceleration PSD was prescribed as a base excitation input for ANSYS. According to ISO 8608, the roughness scaling ratio between Class G and Class A is expressed as

$$\frac{G_d(\eta_0)_{Class\ G}}{G_d(\eta_0)_{Class\ A}} = \frac{65,536 \times 10^{-6}}{16 \times 10^{-6}} = 4096 \quad (22)$$

where $G_d(\eta_0)$ is the road roughness coefficient at the reference spatial frequency η_0 .

A baseline acceleration PSD reference value of $2.5 \times 10^{-4} \text{ g}^2/\text{Hz}$ at 10 Hz was adopted for the Class A profile. The corresponding Class G acceleration PSD level at 10 Hz was then obtained by applying the ISO 8608 roughness scaling ratio:

$$G_a(10)_{Class\ G} = G_a(10)_{Class\ A} \times 4096 \quad (23)$$

$$G_a(10)_{Class\ G} = 2.5 \times 10^{-4} \times 4096 = 1.024 \text{ g}^2/\text{Hz}$$

Thus $1.024 \text{ g}^2/\text{Hz}$ was used as the reference acceleration PSD value for the ISO 8608 Class G road profile. The acceleration PSD values at the remaining frequencies were calculated using an inverse-square frequency variation:

$$G_a(f) = G_a(10) \left(\frac{10}{f} \right)^2 \quad (24)$$

where $G_a(f)$ is the acceleration PSD at frequency (f).

These values were applied as the base acceleration PSD input ANSYS Workbench 2023 R2 (ANSYS, Inc., Canonsburg, PA, USA) for the PSD-based random vibration analysis.

2.2. Modelling

The honeycomb sandwich structure is modelled as a three-dimensional configuration consisting of an aluminium honeycomb core and composite face sheets. The honeycomb cell structure is shown in Figure 2. The different dimensional parameters of the honeycomb core are shown in Table 1. Aluminium 5052 is selected as the core material. The face sheets are modelled as composite laminates using the ANSYS ACP (Composite PrepPost) module. Each face sheet was modelled as a four-ply carbon/glass laminate for all thickness cases. The total face-sheet thickness was varied from 0.5 mm to 2.5 mm by adjusting the thickness of each ply while keeping the number of plies constant. Accordingly, the ply thicknesses used for the 0.5, 1.0, 1.5, 2.0, and 2.5 mm face sheets were 0.125, 0.250, 0.375, 0.500, and 0.625 mm, respectively.

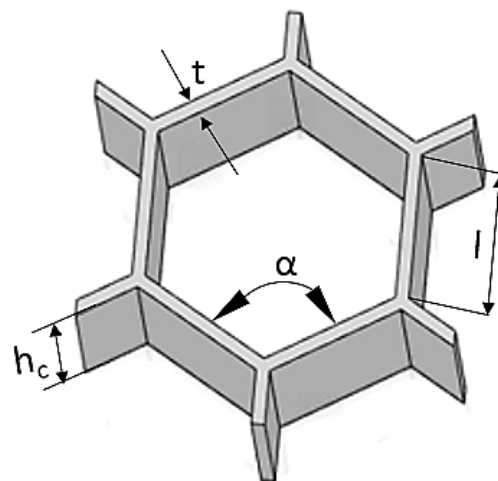


Figure 2. Honeycomb cell structure.

Table 1. Honeycomb Dimensions.

Parameters	Dimensions
Edge Length (l)	15 mm
Angle between edges (α)	120°
Height of the core (hc)	32 mm
Wall thickness (t)	0.5 mm

The following layup configurations are considered in the present study:

1. [90C/90G]/Core/[90G/90C], 2. [60C/−30G]/Core/[−30G/60C], 3. [45C/−45G]/Core/[−45G/45C], 4. [90C/0G]/Core/[0G/90C], 5. [0C/90G]/Core/[90G/0C], 6. [0C/0G]/Core/[0G/0C].

The core, i.e., Aluminium 5052, is modelled as an equivalent orthotropic material [35] while the face plate is made of carbon/glass fibre-reinforced composite laminates (Table 2).

Table 2. Material properties [35].

	Aluminium 5052	Carbon Fibre UD	Glass Fibre UD
Density (kg/m ³)	83	1490	2000
E ₁ (MPa)	1.48	1.21×10^5	45,000
E ₂ (MPa)	0.49	8600	10,000
E ₃ (MPa)	2129	8600	10,000
ν_{12}	0.58	0.27	0.3
ν_{23}	7.5×10^{-5}	0.4	0.4
ν_{13}	2.4×10^{-4}	0.27	0.27
G ₁₂ (MPa)	0.72	4700	5000
G ₂₃ (MPa)	253	3100	3846.2
G ₁₃ (MPa)	524	4700	5000

The fabric, stacking sequence, fibre orientations, ply thickness and number of plies are defined using ANSYS Composite PrepPost (ACP). In ACP, the stackup is defined with appropriate fabric and the corresponding angle. A rosette is defined to indicate the zero-degree fibre direction. An oriented selection set is specified for defining the layup area, the direction of stackup and the fibre direction (Figure 3). The directions of carbon and glass fibres for the [0C/90G]/Core/[90G/0C] layup configuration are shown in Figure 4. The composite model was then transferred to the Modal analysis module of ANSYS Workbench for performing free vibration analysis.

The effect of anisotropic material behaviour on the dynamic response is determined. The mesh discretization is carried out to accurately represent both the global structural behaviour and local deformation characteristics of the honeycomb sandwich configuration. The composite face sheets are modelled using shell 181 elements defined within the ACP framework, allowing accurate representation of laminate stacking sequences and directional material properties. The honeycomb core is modelled using shell 181 elements to capture the hexagonal cellular geometry and associated structural interactions. It is a 4-node element with 6 degrees of freedom at each node: translations in the x, y and z directions and rotations about the x, y and z-axes. Figure 5 depicts the meshed model of the honeycomb sandwich structure.

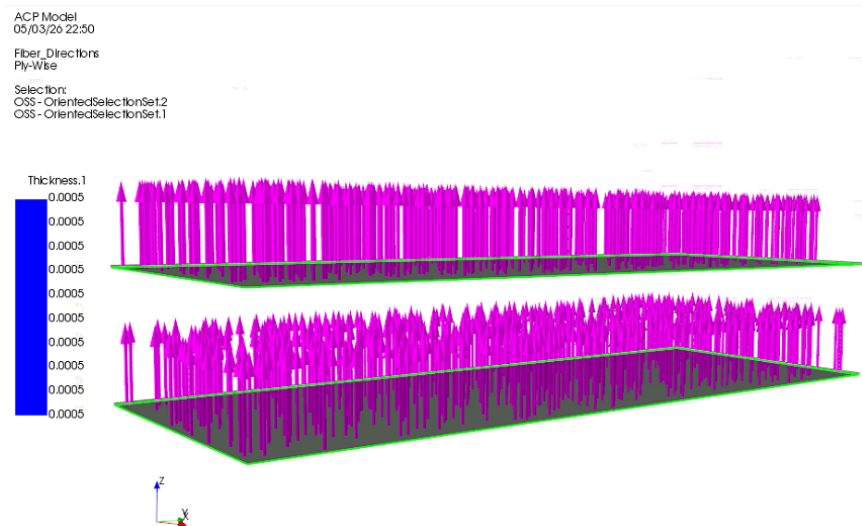


Figure 3. Laminate stackup direction. The arrows indicate the direction of stackup.

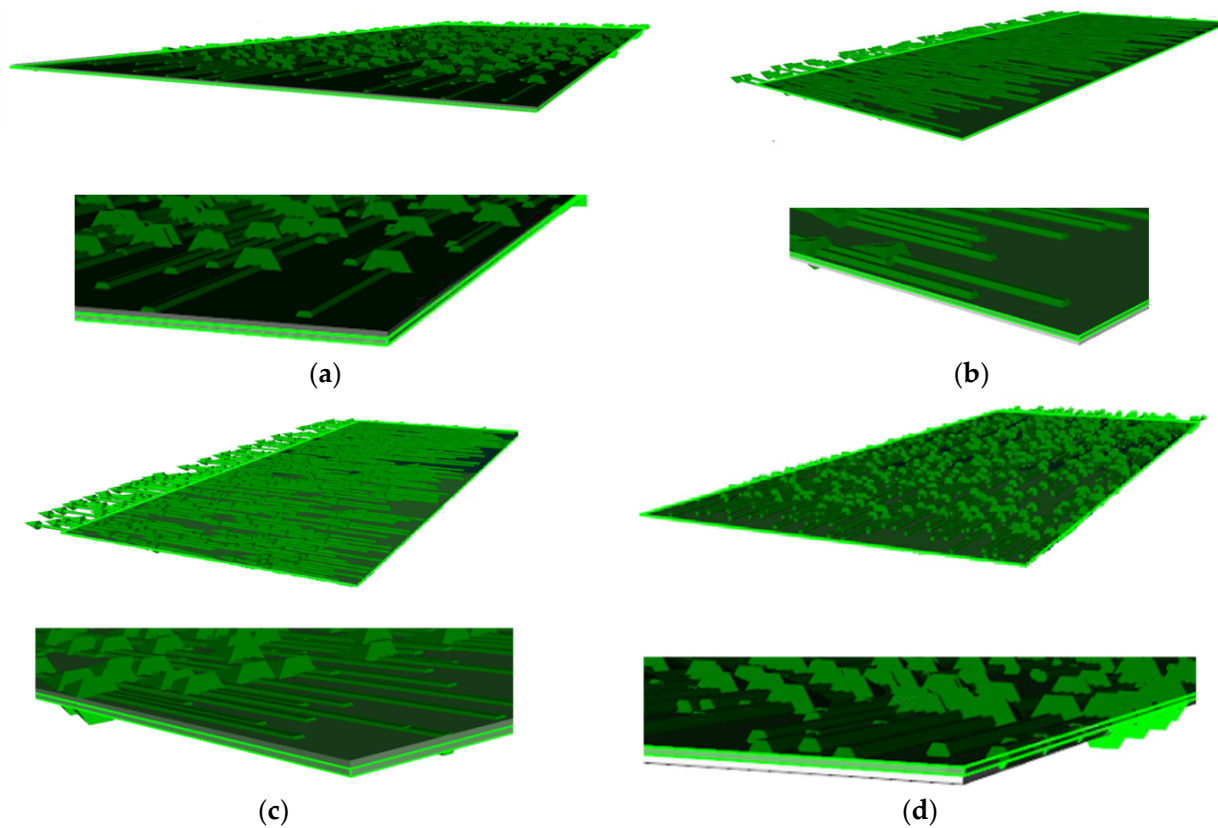


Figure 4. Fibre directions in face plates for [0C/90G]/Core/[90G/0C] layup configuration in ACP: (a) 0° carbon fibre in lower face plate, (b) 90° glass fibre in lower face plate, (c) 90° glass fibre in upper face plate and (d) 0° carbon fibre in upper face plate.

The one end of the honeycomb panel is fixed at the face sheets, while the other end is fixed at the core. This is similar to the boundary conditions adopted by Upreti et al. [35]. These support locations correspond to their attachment to the chassis bed of the EV. The adopted boundary condition represents an idealised constrained mounting condition used for comparative modal and random vibration analysis. Since the same boundary condition was applied to all thickness and layup configurations, the results provide a consistent basis for comparing the relative influence of face-sheet thickness and fibre orientation. Figure 6 shows the boundary conditions applied on the panel.

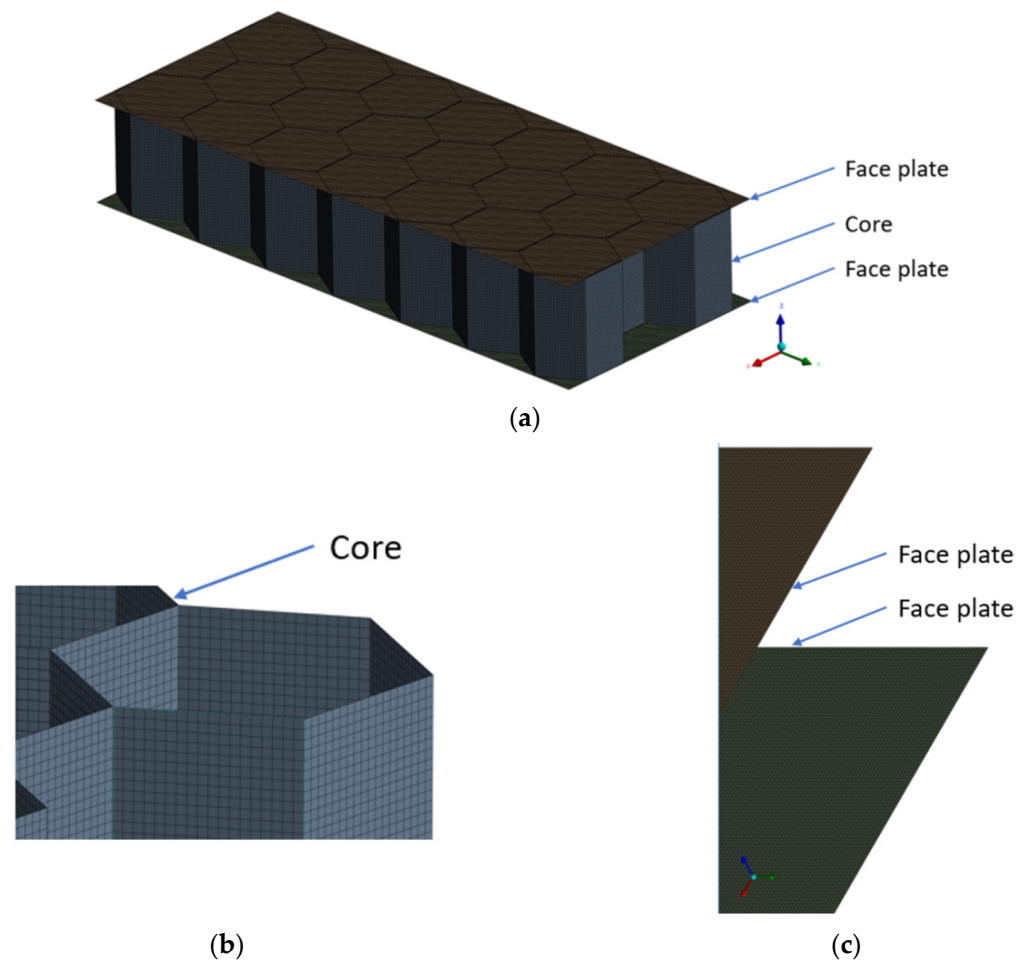


Figure 5. Meshed model of (a) honeycomb panel, (b) core and (c) face plates.

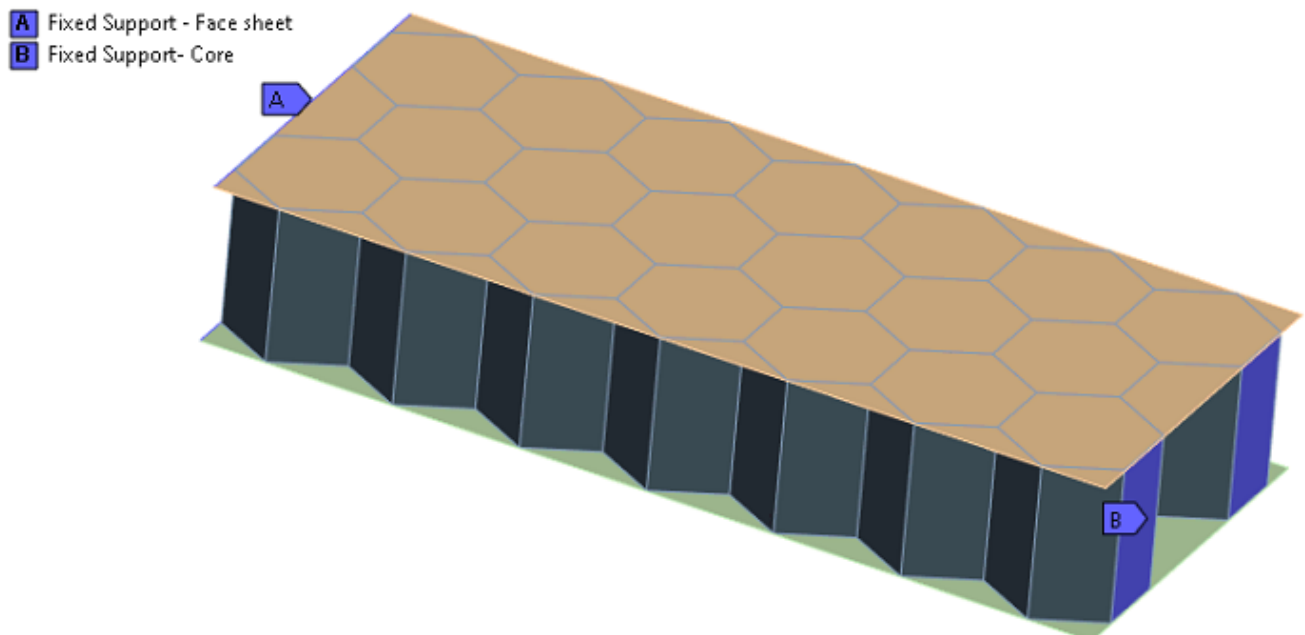


Figure 6. Boundary conditions applied on the honeycomb structure.

A mesh convergence study is carried out to determine the appropriate number and sizes of the elements necessary in modal analysis to ensure that further refinement in the mesh would yield negligible changes in the results. It was performed us-

ing various element sizes, ranging from 2 mm to 0.75 mm. As it is known that stiffness changes considerably with thickness and layup configurations, a mesh convergence study for the $[45C/-45G]/Core/[-45G/45C]$ configuration, along with two extreme cases $[90C/90G]/Core/[90G/90C]$ and $[0C/0G]/Core/[0G/0C]$ at thicknesses of 0.5 mm and 2.5 mm, is considered. The results are shown in Figure 7. It can be observed that the curves pertaining to the different configurations at a thickness of 2.5 mm overlap with each other due to the negligible influence of layup configuration on natural frequency at this thickness. The optimum number of elements (192,310 elements) corresponding to an element size of 0.875 mm was chosen for further analyses. This selection provides a good balance between the accuracy of the solution and computational cost for better accuracy of FEA.

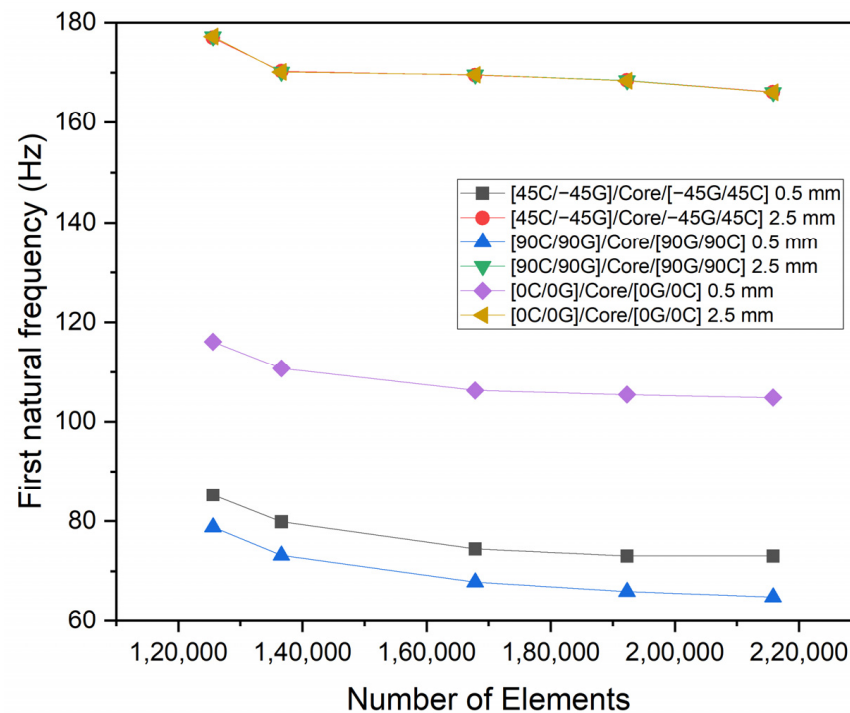


Figure 7. Mesh convergence plot.

3. Results and Discussion

3.1. Modal Analysis

Modal analysis is carried out to determine the natural frequencies and mode shapes of the structure. The results obtained from the analysis are also compared with values reported in the literature. This comparison helps verify the accuracy of the finite element model and builds confidence in the reliability of the simulation approach.

3.1.1. Comparison with Literature

The natural frequencies obtained in the present study were compared with the results reported by Upreti et al. [35] as presented in Table 3. The quantitative comparison shows that the first natural frequency follows the same general increasing trend with increasing face-sheet thickness. The natural frequencies computed from the present study's $[90C/0G]/Core/[0G/90C]$ configuration are compared with the results reported by Upreti et al. [35] as presented in Table 3. Across all the thickness values considered, the results follow a trend consistent with the reported study, supporting the benchmarking of the modelling approach. It can be observed that the fundamental natural frequency (global bending mode) agrees reasonably well with that of Upreti et al. [35], particularly at lower thickness values, with an error of 1.86% at 0.5 mm and 5.07% at 1.0 mm. However, higher

modes show larger deviations. For example, at 0.5 mm thickness, the natural frequency of the fifth mode reported by Upreti et al. was 179.64 Hz, whereas the corresponding value obtained in the present study was 162.74 Hz, giving a percentage difference of 9.41%. The maximum deviation of 28.48% was observed for the second mode at 0.5 mm thickness. The deviations might be due to the difference in fibre orientation, mesh refinement and behaviour of local modes. In the present study, the [90C/0G]/Core/[0G/90C] configuration was considered, whereas in the reported study [35], fibre orientation was not mentioned. The difference in the fibre orientation could result in a difference in the bending stiffness, which in turn might cause a difference in the frequency. Furthermore, the present study used an element size of 0.875 mm (coarser mesh) whereas the reported study adopted an element size of 2 mm (refined mesh). It is known that a model with a coarser mesh used in the reported study [35] would not be able to capture the behaviour of the local mode accurately, which could result in a lower natural frequency compared to that in the present study.

Table 3. Comparison of natural frequencies with those of Upreti et al. [35].

Thickness (mm)	Mode	Upreti et al. [35]	Present Study	% Difference
0.5	1	73.164	74.523	1.86%
0.5	2	124.630	160.130	28.48%
0.5	3	168.630	160.250	4.97%
0.5	4	173.990	160.920	7.51%
0.5	5	179.640	162.740	9.41%
1	1	123.400	117.140	5.07%
1	2	167.040	161.130	3.54%
1	3	179.640	161.330	10.19%
1	4	183.240	164.170	10.41%
1	5	184.960	164.380	11.13%
1.5	1	179.640	161.420	10.14%
1.5	2	180.480	161.840	10.33%
1.5	3	183.230	164.630	10.15%
1.5	4	184.960	164.960	10.81%
1.5	5	188.670	165.000	12.55%
2	1	179.640	161.570	10.06%
2	2	183.240	165.290	9.80%
2	3	184.960	165.320	10.62%
2	4	188.670	165.700	12.17%
2	5	188.910	166.060	12.10%
2.5	1	179.640	170.280	5.21%
2.5	2	183.240	170.710	6.84%
2.5	3	184.960	170.760	7.68%
2.5	4	188.670	171.030	9.35%
2.5	5	188.910	171.150	9.40%

3.1.2. Mode Shapes

The deformation patterns associated with the first five natural frequencies are shown in Figure 8. The first mode corresponds to global bending of the panel, with a larger amount of bending deformation happening at the face sheet. The deformation is uniform across the structure, indicating that the response is mainly dominated by bending. The higher-order modes (second mode to the fifth mode) correspond to localised deformation in the core/cell-wall regions, while the face sheets undergo less deformation. This indicates that the vibration is more localised rather than distributed across the entire structure. The mode shapes clearly illustrate a progression from global bending in the lower modes to more localised behaviour at higher modes.

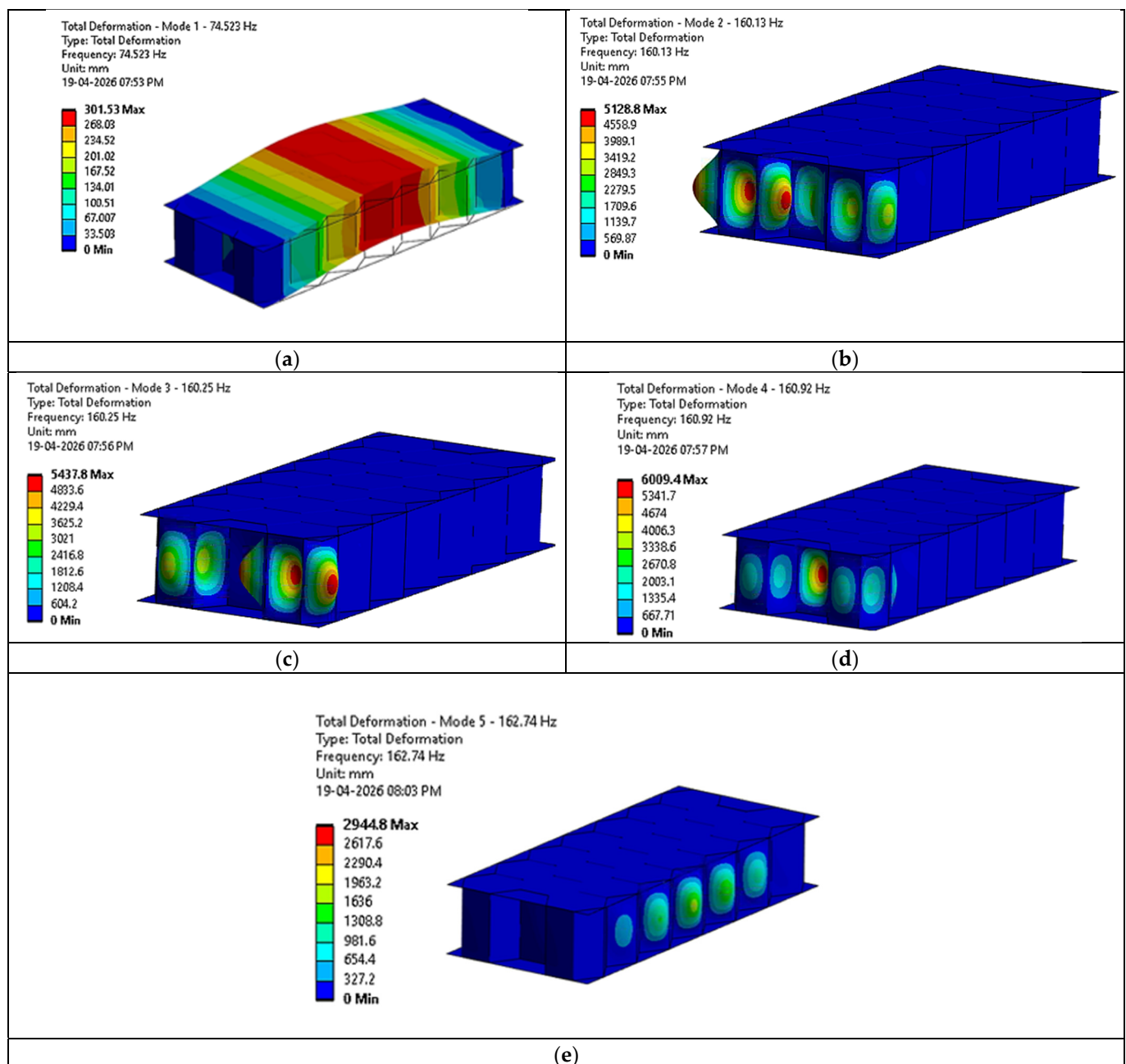


Figure 8. (a) First global bending mode; (b–e) higher-order localised core/cell-wall deformation modes.

3.1.3. Effect of Face Plate Thickness on Natural Frequency

The natural frequencies for different face-sheet thicknesses and fibre orientations are presented in Table 4. Figure 9 shows the variation in fundamental frequency with the thickness of the face sheet. The increase in thickness from 0.5 mm to 1.5 mm resulted in a rapid rise in natural frequency. It was observed earlier that the first mode of vibration is dominated by bending. Also, it is known that bending stiffness is directly proportional to the cube of thickness. This is the reason for the sudden increase in stiffness (or frequency), which in turn causes the rapid rise in natural frequency. For thickness above 2.0 mm, the natural frequencies were found to converge at 170 Hz across all layups. However, it can be observed from Table 4 that the natural frequencies corresponding to the higher modes are marginally affected by increasing the thickness of the face plates.

Table 4. First five natural frequencies for different face plate thicknesses for various layup configurations.

Configuration	Thickness (mm)	Mode 1 (Hz)	Mode 2 (Hz)	Mode 3 (Hz)	Mode 4 (Hz)	Mode 5 (Hz)
[90C/90G]/Core/[90G/90C]	0.5	67.774	151.97	160.13	160.25	160.92
	1	95.077	161.13	161.33	164.17	164.38
	1.5	139.94	161.42	161.84	164.63	164.96
	2	161.57	165.29	165.32	165.7	166.06
	2.5	170.14	170.66	170.75	171.03	171.14
[60C/−30G]/Core/[−30G/60C]	0.5	72.507	160.13	160.25	160.92	162.74
	1	111.61	161.13	161.33	164.17	164.38
	1.5	161.42	161.83	164.63	164.96	165
	2	161.57	165.29	165.32	165.7	166.06
	2.5	170.28	170.71	170.76	171.03	171.15
[45C/−45G]/Core/[−45G/45C]	0.5	73.006	160.13	160.25	160.92	162.74
	1	112.64	161.13	161.33	164.17	164.38
	1.5	161.42	161.83	164.63	164.96	165
	2	161.57	165.29	165.32	165.7	166.06
	2.5	170.14	170.66	170.75	171.03	171.14
[90C/0G]/Core/[0G/90C]	0.5	74.523	160.13	160.25	160.92	162.74
	1	117.14	161.13	161.33	164.17	164.38
	1.5	161.42	161.84	164.63	164.96	165
	2	161.57	165.29	165.32	165.7	166.06
	2.5	170.28	170.71	170.76	171.03	171.15
[0C/90G]/Core/[90G/0C]	0.5	83.213	160.13	160.25	160.92	162.74
	1	139.75	161.16	161.37	164.23	164.38
	1.5	161.42	161.83	164.63	164.96	165
	2	161.57	165.29	165.32	165.7	166.06
	2.5	170.14	170.66	170.75	171.03	171.14
[0C/0G]/Core/[0G/0C]	0.5	106.27	160.13	160.25	160.92	162.74
	1	161.16	161.37	164.23	164.38	164.48
	1.5	161.42	161.83	164.63	164.96	165
	2	161.57	165.29	165.32	165.7	166.06
	2.5	170.14	170.66	170.75	171.03	171.14

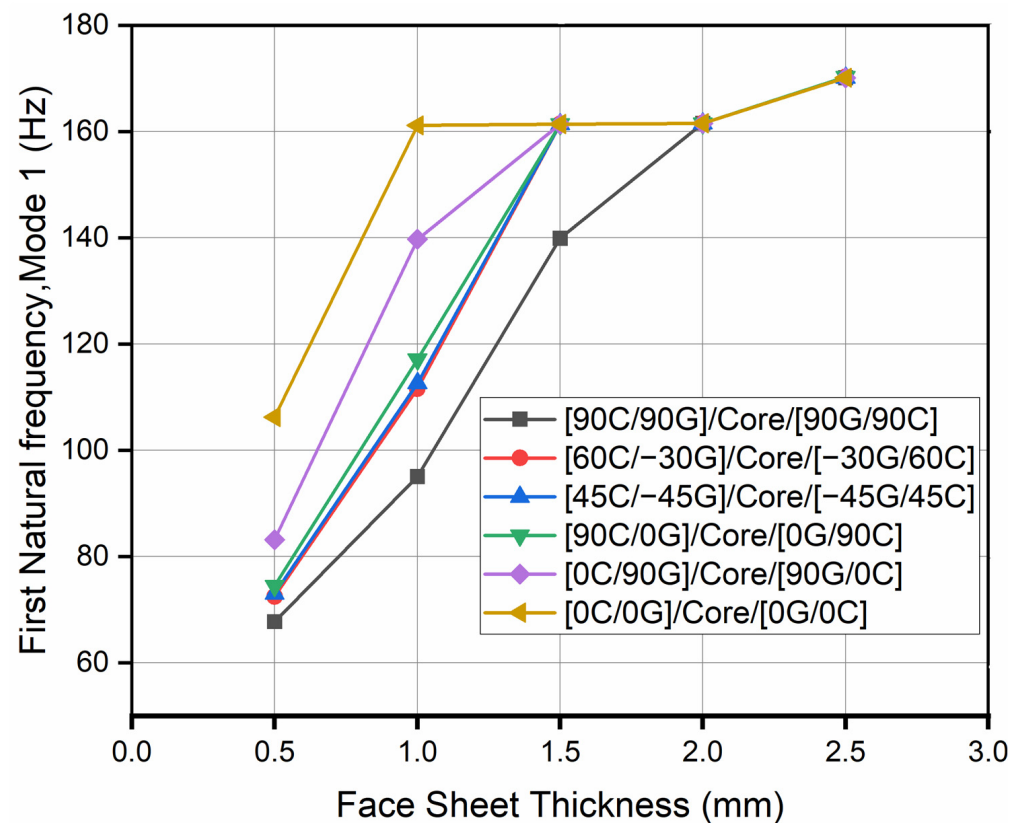


Figure 9. Variation in natural frequencies with thickness and layup configurations.

3.1.4. Effect of Layup Configuration on Natural Frequency

The difference in natural frequency between the layup configurations is significant for face plate thicknesses between 0.5 mm and 1.5 mm as depicted in Figure 9. For instance, at a thickness of 0.5 mm, changing the layup configuration from [90C/90G]/Core/[90G/90C] to [0C/0G]/Core/[0G/0C] increased the first natural frequency by 22.8%. The increase in the fundamental frequency was marginal for thicknesses of 2 mm and 2.5 mm, regardless of the layup configuration. This means that as the structure gets stiffer, its modal response becomes less sensitive to changes in the layup configuration. It can also be noted from Table 5 that the natural frequencies of the higher modes are not affected by a change in the layup configuration.

Table 5. PSD data for speed of 60 kmph.

Frequency (Hz)	Acceleration PSD (g^2/Hz)
10	1.024
20	0.256
40	6.4×10^{-2}
60	2.844×10^{-2}
80	1.6×10^{-2}

Here, $g = 9.81 \text{ m/s}^2$.

(a) Random Vibration Analysis

For random vibration analysis, a base excitation PSD is applied in a direction perpendicular to the structure (global Y-direction), representing vertical road-induced vibrations. ISO 8608 Class G road condition corresponding to off-road terrain (speed of 60 kmph) was considered to achieve a conservative design of the honeycomb panel. Since ISO 8608 defines

road roughness in terms of spatial displacement PSD, the present study used ISO 8608 only to define the relative road-class scaling. The acceleration PSD applied in ANSYS was prescribed as a base excitation profile, with the Class G reference value at 10 Hz obtained by scaling a baseline Class A acceleration PSD level using the ISO 8608 Class G/Class A roughness ratio. The remaining PSD values were then calculated using the inverse-square frequency relation, as described in Section Definition of Base Acceleration PSD Input for ISO 8608 Class G Road Profile. The acceleration PSD input used in the random vibration analysis is listed in Table 5.

The random vibration response is computed using the mode superposition method, where contributions from individual modes are combined to obtain the overall structural response. In this approach, the structural response is represented using a finite number of retained modes; therefore, the modal truncation order must be selected carefully. In the present study, the retained modal range was checked with respect to the applied PSD excitation bandwidth (Table 6). Since the base acceleration PSD was specified over 10–80 Hz, the highest retained natural frequency was kept above 80 Hz for all analysed configurations.

Table 6. Modal truncation verification.

Thickness (mm)	Layup	Modes Retained	Highest Retained Frequency (Hz)	PSD Range (Hz)	Coverage Ratio
0.5	[90C/90G]/Core/[90G/90C]	6	160.92	10–80	2.0115
1.0			164.38		2.05475
1.5			164.96		2.062
2.0			166.06		2.07575
2.5			171.14		2.13925

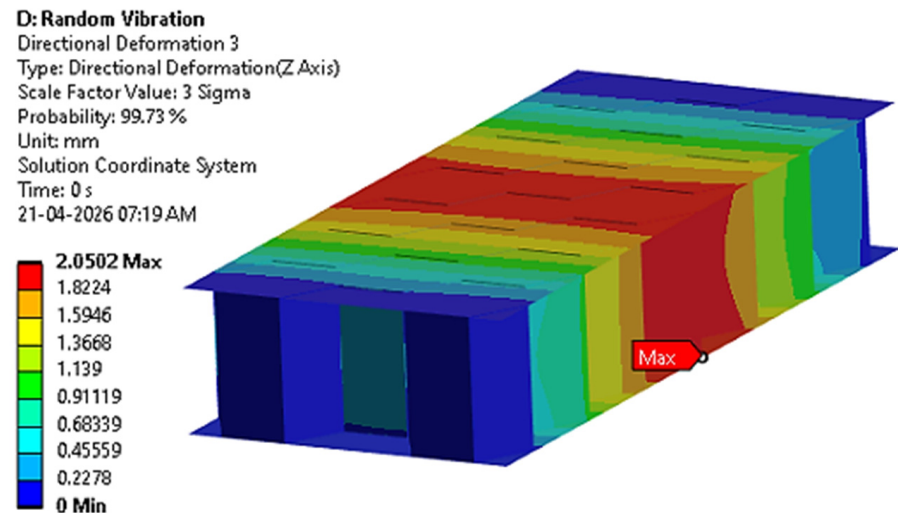
The [90C/90G]/Core/[90G/90C] layup is suitable because it is expected to have the lowest bending stiffness and lower natural frequencies (Table 5), and therefore it is more critical to check whether the retained modal range covers the PSD bandwidth of 10–80 Hz. For the representative 0.5 mm configuration, six modes were retained, and the highest retained natural frequency was 160.92 Hz, giving a frequency coverage ratio of 2.04 with respect to the maximum PSD frequency of 80 Hz.

Modal damping was adopted in the random vibration analysis with a damping ratio of 2%. The response is evaluated in terms of total deformation and equivalent (von Mises) stress under three-sigma conditions. A parametric study is conducted by varying face-sheet thickness and fibre orientation, and the resulting changes in dynamic response are analysed to identify performance trends and optimal configurations.

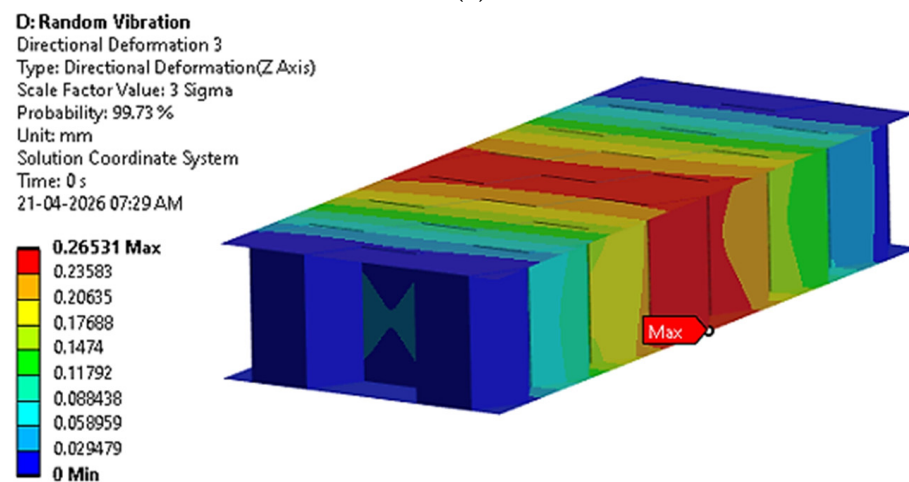
The random vibration behaviour of the composite honeycomb sandwich structure is studied using PSD-based excitation, which represents the type of loading encountered under real operating conditions. The influence of the thickness and layup configurations on the dynamic response of the structure is studied by observing the changes in the total deformation and equivalent stress under three-sigma loading conditions.

i. Total deformation

The total deformation contour plot corresponding to three-sigma loading conditions for the [45C/−45G]/Core/[−45G/45C] configuration is depicted in Figure 10. The total deformation experienced by the panel was maximum at the centre and minimum at the ends. This is because the structure is supported at the ends. The deformation contours indicated a consistent distribution pattern across all configurations.



(a)



(b)

Figure 10. Total three-sigma deformation: (a) 0.5 mm thickness, (b) 1.0 mm thickness.

Effect of Thickness on the Total Deformation

The variation in total deformation with face-sheet thickness and layup configuration is further illustrated in Figure 11. It can be observed from the graph that there is a significant reduction in the total deformation as the thickness is varied from 0.5 mm to 2.5 mm. This implies that the deformation is strongly controlled by the bending stiffness of the face sheets. In honeycomb panels, bending stresses are taken up by the face sheets, whereas the core takes up the shear stresses. Hence, there is a significant decrease in deformation as the thickness of the face sheet increases. This trend is similar to the findings of Kumar et al. [48], who reported that increased face-sheet thickness led to greater stiffness and reduced deformation in sandwich structures.

It can be seen from the graph that deformation values are very high in the range of 0.29 mm to 2.4 mm for a thickness of 0.5 mm. As the thickness is increased, there is an increase in the bending stiffness, causing the deformation to drop. A sharp decline in deformation is observed when the thickness is increased from 1.0 mm to 1.5 mm, confirming that this region of the graph acts as a stiffness transition zone, where the stiffness gain is very high. Between 1.5 mm and 2.0 mm, and further to 2.5 mm, there is a gradual reduction in deformation. This indicates that it is not advantageous to increase the thickness beyond 1.5 mm, as the stiffness gain is marginal.

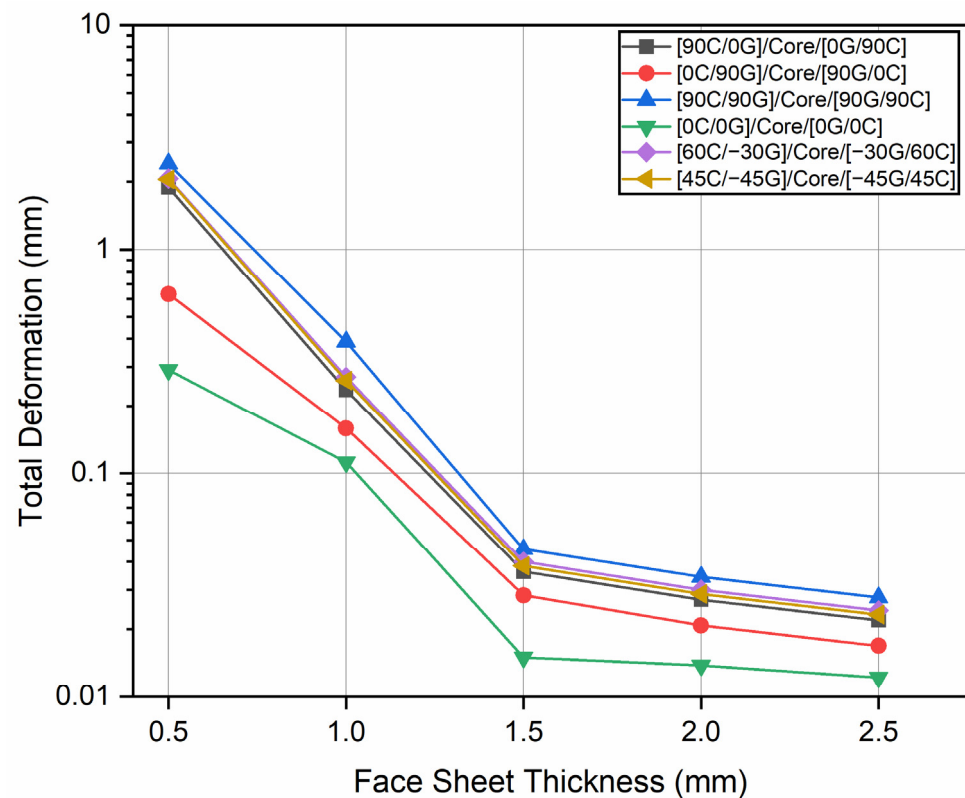


Figure 11. Variation in total deformation with face-sheet thickness and layup configuration.

Effect of Layup Configuration on the Deformation

The layup configuration has a significant impact on the deformation of the panel throughout the thickness range. It can be observed from Figure 11 that the [0C/0G]/Core/[0G/0C] configuration undergoes minimum deformation for all the thickness values. For instance, it undergoes a deformation of 0.291 mm for a face plate thickness of 0.5 mm. This is because the fibre directions are aligned with the principal loading directions, exhibiting a higher effective modulus, resulting in a high bending stiffness. In contrast, the [90C/90G]/Core/[90G/90C] configuration experiences maximum deformation irrespective of the thickness. For instance, it undergoes a deformation of 2.404 mm for a thickness of 0.5 mm. This is because the fibres are oriented transverse to the principal loading directions, leading to a low bending stiffness. The [45C/-45G]/Core/[-45G/45C] configuration shows intermediate behaviour. It offers better stiffness compared to the [90C/90G]/Core/[90G/90C] configuration, but lesser stiffness compared to the [0C/0G]/Core/[0G/0C] configuration. Moreover, the result also shows the importance of hybridization. Since the stiffness offered by carbon fibres is greater compared to that provided by glass fibres, it is better to place them at the outermost layers because these layers experience the highest bending stress.

In general, the findings demonstrate that the deformation is governed by the combined effect of thickness, stack-up configuration and hybridization. The significant reduction in deformation is possible by altering the thickness, whereas the relative ranking among the laminates is decided by the fibre orientation. For EV applications, the [0C/0G]/Core/[0G/0C] configuration is a suitable candidate since it exhibits the lowest deformation and stress response.

ii. Equivalent Stress

The equivalent (von Mises) stress plots corresponding to three-sigma loading conditions are presented in Figure 12. It can be seen from Figure 12 that for a thickness of 0.5 mm,

the maximum value of equivalent stress is located near the edges of the face sheets. As the thickness is increased, the bending stiffness increases, resulting in a lower stress magnitude corresponding to 1.0 mm thickness. The location of the maximum value of equivalent stress remains the same.

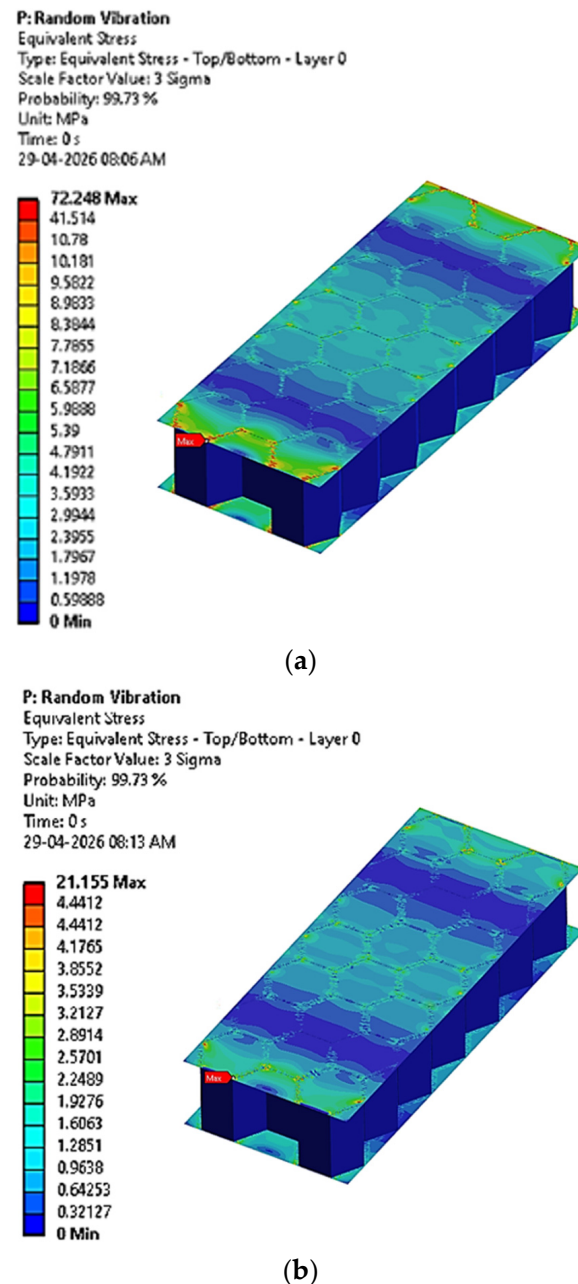


Figure 12. Equivalent stress (three-sigma) for (a) 0.5 mm thickness and (b) 1.0 mm thickness.

Effect of Thickness on the Equivalent Stress

The variation in equivalent stress with face-sheet thickness and layup configuration is depicted in Figure 13. It can be seen from Figure 13 that equivalent stress values are very high, ranging from 31.58 MPa to 82.337 MPa for a thickness of 0.5 mm. As the thickness is increased from 0.5 mm to 2.5 mm, there is a considerable decrease in the equivalent stress. This implies that thicker face sheets provide a higher bending stiffness, resulting in lesser curvature and, in turn, cause a significant stress reduction. The sharp decline in stress occurs when the thickness changes from 1.0 mm to 1.5 mm. It can also be noted that there is a gradual reduction in stress magnitude with changes in thickness from 1.5 mm

to 2.0 mm and from 2.0 mm to 2.5 mm, due to the lesser stiffness gain. Kumar et al. [48] reported that increasing face-sheet thickness improves bending rigidity and stiffness of composite sandwich structures, which supports the present reduction in equivalent stress with thickness.

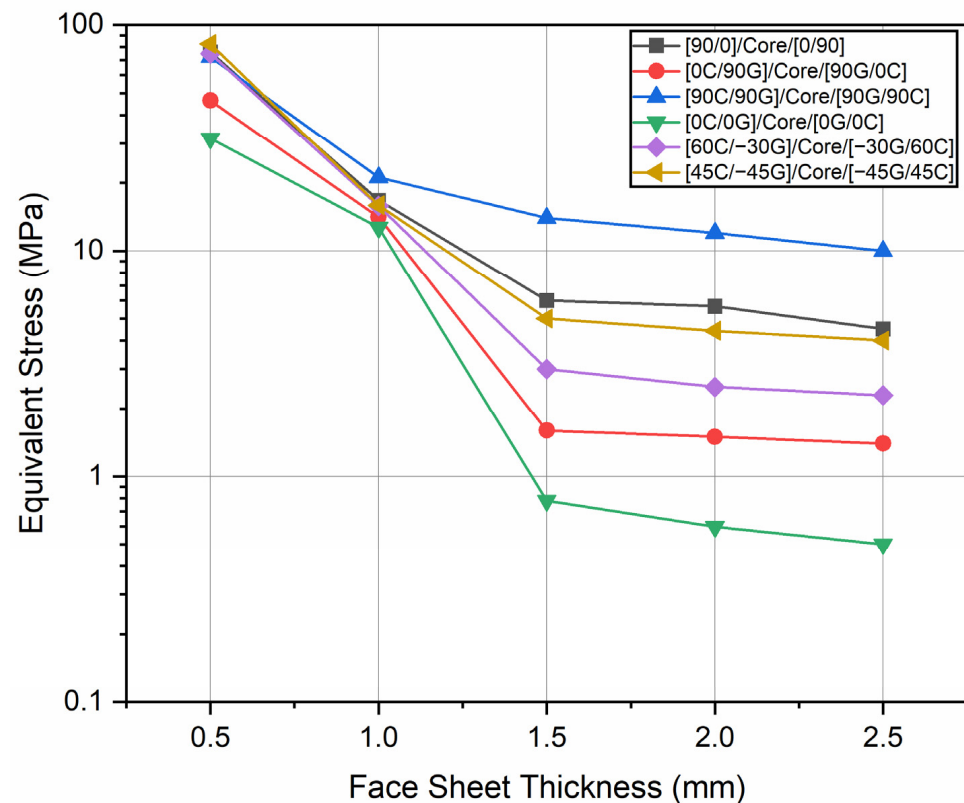


Figure 13. Variation in equivalent stress with face-sheet thickness and layup configuration.

Effect of Layup Configuration on the Equivalent Stress

The layup configuration has a strong influence on equivalent stress because fibre orientation controls directional stiffness and load transfer in the face sheets. It can be observed from Figure 13 that the [0C/0G]/Core/[0G/0C] configuration experiences minimum stress for all the thickness values. For instance, it undergoes a stress of 31.58 MPa for a face plate thickness of 0.5 mm. This is because the fibre directions are aligned with the principal loading directions, resulting in a higher effective modulus, which gives rise to a high bending stiffness. In contrast, the [90C/90G]/Core/[90C/90G] configuration experiences higher stress. For instance, it undergoes a stress of 72.25 MPa for a thickness of 0.5 mm. This is because fibre directions are perpendicular to the principal loading direction.

In general, the results reveal that the equivalent stress is governed by the combined effect of thickness, stack-up configuration and hybridization. The significant reduction in equivalent stress is possible by altering the thickness, whereas the relative ranking among the laminates is decided by the fibre orientation. The overall results of this study are further converted into practical design guidelines for EV battery enclosures, as presented in the following section.

4. Design Guidelines for EV Battery Enclosures

The present study provides important insights into the design of honeycomb sandwich structures subjected to random vibration environments typical of electric vehicle (EV) applications. Based on the numerical findings, the following design-oriented guidelines

are proposed to assist engineers in selecting appropriate configurations for improved structural performance.

4.1. Selection of Appropriate Face-Sheet Thickness

The thickness of the face sheet plays a major role in controlling the dynamic behaviour of the honeycomb sandwich structure. A considerable rise in natural frequency and bending stiffness is noted, corresponding to the increase in thickness from 0.5 mm to 1.5 mm. The thickness range of 1.0 mm to 1.5 mm is considered to be more favourable as it provides higher stiffness gain with the corresponding reduction in deformation and stress level. However, stiffness gain for a thickness beyond 1.5 mm is minimal.

4.2. Selection of Layup Configuration

The layup configuration of the composite face sheets has a major influence on the bending stiffness and vibration behaviour of the structure. The bending stiffness provided by the [0C/0G]/Core/[0G/0C] configuration is higher compared to that provided by the [90C/90G]/Core/[90G/90C] configuration. Accordingly, the deformation and stress would be lesser in the [0C/0G]/Core/[0G/0C] configuration compared to those in the [90C/90G]/Core/[90G/90C] configuration. Compared with the previous configurations, the [45C/−45G]/Core/[−45G/45C] configuration provides moderate bending stiffness and vibration behaviour. Therefore, the selection of a proper layup configuration aligned with the principal loading direction in conjunction with the appropriate hybridization scheme could enable the honeycomb structure to withstand the vibrational loading.

4.3. Thickness–Layup Interaction

The interaction between the thickness and layup configuration has a major influence on the vibration behaviour of the panel. It is possible to achieve a higher natural frequency along with lesser deformation and equivalent stress for honeycomb structures with thin face sheets (thickness < 1.5 mm) by selecting an appropriate layup configuration. However, for thicker face sheets, the layup configuration has a marginal influence on the vibration response. Hence, better vibration resistance is possible by the proper combination of thickness and layup configuration for thin panels, whereas it is decided by the thickness in the case of thicker panels.

4.4. Design Trade-Off

A compromise between stiffness and weight is necessary for the optimum design of honeycomb sandwich structures for EV battery enclosures. Increasing the face-sheet thickness improves bending stiffness and reduces total deformation and stress, but it also increases the structural mass of the panel. Similarly, alignment of fibres with the principal loading direction improves directional stiffness, but its influence becomes less significant at higher thickness values. Therefore, the best design cannot be selected only on the basis of minimum deformation or stress; the associated mass penalty must also be considered.

To support the recommended thickness range quantitatively, a mass-based vibration-performance comparison was carried out for the [0C/0G]/Core/[0G/0C] configuration. The panel mass, mass penalty relative to the 0.5 mm configuration, first natural frequency, and frequency per unit mass were calculated for different face-sheet thicknesses, as shown in Table 7. This comparison helps determine whether the improvement in vibration performance is sufficient to justify the additional mass introduced by increasing the face-sheet thickness.

The results show that the total panel mass increases continuously with face-sheet thickness. When the thickness is increased from 0.5 mm to 1.0 mm, the total panel mass increases from 0.061 kg to 0.085 kg, corresponding to a mass penalty of 39.65%. However,

the first natural frequency increases significantly from 106.27 Hz to 161.16 Hz. The frequency per unit panel mass also increases from 1751.93 Hz/kg to 1902.49 Hz/kg, indicating improved vibration performance per unit mass. When the thickness is further increased from 1.0 mm to 1.5 mm, the mass increases to 0.109 kg, but the first natural frequency changes only slightly from 161.16 Hz to 161.42 Hz. Beyond 1.5 mm, the mass penalty becomes more significant, reaching 118.95% at 2.0 mm and 158.60% at 2.5 mm, while the improvement in frequency remains comparatively small.

Table 7. Weight penalty and mass-normalised vibration–performance comparison for different face-sheet thicknesses.

Sl.No.	t_f (mm)	m_f (kg)	m_c (kg)	m_{tot} (kg)	Mass Penalty Relative to 0.5 mm (%)	f (Hz)	f/m_{tot} (Hz/kg)
1	0.5	0.024	0.037	0.061	-	106.27	1751.93
2	1	0.048	0.037	0.085	39.65	161.16	1902.49
3	1.5	0.072	0.037	0.109	79.3	161.42	1484.17
4	2	0.096	0.037	0.133	118.95	161.57	1216.52
5	2.5	0.120	0.037	0.157	158.6	170.14	1084.63

t_f is the face-sheet thickness, m_f is the combined mass of the top and bottom face sheets, m_c is the core mass, m_{tot} is the total panel mass, and f is the first natural frequency of the [0C/0G]/Core/[0G/0C] configuration.

Therefore, the 1.0–1.5 mm face-sheet thickness range provides a practical design window for EV battery enclosure panels. This range offers improved vibration resistance compared with the 0.5 mm configuration while avoiding the larger mass penalty associated with 2.0 mm and 2.5 mm face sheets. Among the layup configurations considered, [0C/0G]/Core/[0G/0C] showed the best vibration response under the present PSD loading condition because the fibres are aligned with the principal load-carrying direction. However, the final selection of thickness and layup should also consider allowable ply stresses, fatigue life, bonding/interface strength, cost, manufacturing feasibility, and joining constraints with the battery enclosure frame.

5. Limitations

The present study is limited to a linear PSD-based vibration assessment using the standard modal and random vibration modules of ANSYS Workbench. The honeycomb core was modelled using adopted equivalent orthotropic properties, and no new homogenisation strategy was developed. Nonlinear behaviour, progressive damage, adhesive/interface degradation, ply-level failure, detailed interlaminar stress analysis, fatigue-life prediction, and formal multi-objective optimisation were not included. The equivalent stress values were used only as comparative global stress-response indicators; a complete failure assessment would require maximum principal stress, Tsai–Wu failure analysis for the composite face sheets, and suitable core failure criteria. Modal participation factors and cumulative effective mass ratios were not explicitly reported for all parametric cases, although the retained modal frequency range was checked against the applied 10–80 Hz PSD excitation bandwidth. A detailed modal participation, effective mass, and modal truncation sensitivity study using progressively increasing numbers of modes should be included in future work. In addition, multidirectional vibration inputs, variable speed conditions, different road classes, alternative core topologies, graded layups, variable-thickness cores, boundary-condition sensitivity, and improved honeycomb homogenisation strategies should be considered to further improve the reliability of honeycomb sandwich battery enclosure designs.

6. Conclusions

The present study investigated the dynamic behaviour of honeycomb panels for electric vehicle (EV) battery enclosure applications using modal and PSD-based random vibration analyses. Modal analysis showed that the natural frequencies follow an increasing trend with an increase in the thickness, particularly in the lower thickness range (0.5–1.5 mm), due to increased bending stiffness. The modal results were consistent with the general frequency trend reported in the literature, particularly with respect to the increase in natural frequency with face-sheet thickness. However, differences observed in higher modes suggest that the comparison serves as a literature-based benchmark rather than a strict one-to-one validation. The layup configuration had a significant impact on the natural frequency at lower thickness values, with the [0C/0G]/Core/[0G/0C] configuration exhibiting the most favourable response among the layups considered under the present PSD loading condition. The random vibration analysis revealed that both total deformation and equivalent stress follow a decreasing trend when the thickness is increased from 1.0 mm to 1.5 mm. The [0C/0G]/Core/[0G/0C] configuration offers the best performance in reducing deformation and stress. An optimal balance between vibration resistance and weight is possible by properly selecting the face-sheet thickness and layup configuration when designing battery enclosures. In addition to the numerical findings, the study provides practical design guidelines for EV battery enclosures by identifying an appropriate face-sheet thickness range of 1.0–1.5 mm and optimal layup configurations under random vibration loading, while considering the associated mass penalty. These findings provide a design-oriented framework for selecting thickness and layup configuration to achieve a practical balance between vibration resistance and weight.

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