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DESIGN AND ANALYSIS OF HONEYCOMB STRUCTURE USED FOR TACTICAL PLACEMENT OF BATTERY PACK FOR ELECTRIC VEHICLE

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Abstract: The growing demand for high-performance electric vehicles (EVs) has intensified research into lightweight and structurally efficient battery enclosures. This paper presents the design and structural analysis of a honeycomb-based battery pack housing for tactical placement within the EV underbody chassis. The hexagonal honeycomb core configuration is selected owing to its superior specific strength, energy absorption capability, and thermal management properties. Finite Element Analysis (FEA) is carried out using ANSYS Workbench to evaluate the structural response of the battery enclosure under static and dynamic load cases, including frontal impact, side impact, vertical loading, and torsional loading. Three candidate materials, namely Aluminium 5052-H32, Carbon Fibre Reinforced Polymer (CFRP), and Nomex Aramid honeycomb, are assessed and compared on the basis of stress distribution, deformation, and mass reduction. The honeycomb geometry is parametrically optimized by varying cell size, wall thickness, and cell angle to achieve the best combination of stiffness and weight. Results indicate that the proposed honeycomb battery pack enclosure achieves a mass reduction of up to 38% over conventional steel enclosures while maintaining structural integrity within permissible safety limits. The study also demonstrates improved thermal dissipation pathways facilitated by the open-cell geometry, which contributes to enhanced battery performance and longevity. The findings validate the suitability of honeycomb structures for the tactical and space-efficient placement of battery packs in EV platforms.

***Index Terms* -** Honeycomb structure, battery pack, electric vehicle, finite element analysis, lightweight design, structural optimization, energy absorption, ANSYS.

I. INTRODUCTION

The global transition towards electric mobility has brought battery system engineering to the forefront of automotive research. Battery packs in EVs are among the heaviest subsystems, typically accounting for 25 to 40% of the total vehicle mass. Their placement, structural enclosure, and thermal management are critical determinants of vehicle dynamics, range, and safety. Conventional steel or aluminium battery enclosures, while robust, impose significant weight penalties and limit the design freedom for space-efficient placement strategies.

Honeycomb structures, originally developed for aerospace applications, have emerged as highly promising candidates for automotive lightweight design. Their periodic cellular geometry provides an exceptional combination of high stiffness-to-weight ratio, excellent energy absorption under crash loading, and inherent thermal conductivity characteristics. When applied to EV battery pack housings, honeycomb cores can serve the dual purpose of structural protection and thermal regulation.

Tactical placement of the battery pack refers to the deliberate positioning of the energy storage system within the vehicle underbody to optimize the centre of gravity, crashworthiness, structural stiffness, and space utilization. A flat, skateboard-type battery pack arrangement integrated within a honeycomb sandwich chassis offers a structurally elegant solution that addresses multiple design constraints simultaneously.

This work aims to design a honeycomb-cored battery pack enclosure, perform parametric geometric optimization, and conduct detailed Finite Element Analysis (FEA) to evaluate its performance under representative service and crash load scenarios. The study also compares three candidate materials to identify the most suitable option for practical implementation.

II. LITERATURE REVIEW

Extensive research has been conducted on the structural and thermal performance of honeycomb sandwich panels in aerospace and automotive contexts. Gibson and Ashby [1] established foundational analytical models for the in-plane and out-of-plane mechanical behaviour of cellular solids, demonstrating that the relative density and cell geometry govern effective elastic moduli and collapse strengths. Their work showed that hexagonal honeycombs possess the highest specific stiffness among regular tessellations.

Zarei and Kroger [2] investigated aluminium honeycomb-filled crash boxes and demonstrated energy absorption improvements of over 60% compared to empty tubes under axial impact conditions. These findings motivate the application of honeycomb fillers in automotive safety-critical components, including battery enclosures. Similarly, Wierzbicki [3] developed analytical expressions for the mean crushing force of hexagonal honeycombs, providing a basis for design optimization under impact loading.

In the context of EV battery packs, Kukreja et al. [4] studied the crashworthiness of a battery pack under side pole impact and identified critical deformation modes in the cooling plate and module casing. Their work highlighted the need for additional structural layers around the battery modules to prevent intrusion during side crashes. Kim et al. [5] proposed a topology-optimized aluminium battery pack housing and showed that up to 18% mass reduction is achievable while satisfying stiffness and strength constraints.

Thermal management of EV batteries using honeycomb channels has been explored by Wang et al. [6], who demonstrated that honeycomb-structured cooling plates achieve 23% better temperature uniformity compared to conventional serpentine channel designs. This is attributed to the high surface-area-to-volume ratio and the symmetrical distribution of coolant flow paths in honeycomb geometries.

Parametric studies on honeycomb cell geometry have been performed by Schuler et al. [7], who investigated the effect of cell size, wall thickness, and cell angle on the effective mechanical properties of aluminium honeycomb cores. Their results confirm that smaller cell sizes and larger wall thickness-to-cell size ratios lead to higher specific stiffness but also higher weight, necessitating a design trade-off. The present study builds upon these findings to develop an optimized honeycomb geometry for EV battery pack enclosures.

III. METHODOLOGY

3.1 Honeycomb Geometry Design

The hexagonal honeycomb unit cell is parameterized by three variables: cell side length (l), wall thickness (t), and cell height (h). The baseline geometry is defined with $l = 8$ mm, $t = 0.4$ mm, and $h = 20$ mm, yielding a relative density of approximately 0.08. These parameters are selected based on typical aerospace-grade aluminium honeycomb specifications and adapted to automotive dimensional requirements.

The overall battery pack enclosure is designed as a sandwich panel assembly comprising two face sheets of 1.5 mm thickness enclosing a honeycomb core of 20 mm depth. The total enclosure dimensions are 1400 mm (length) x 900 mm (width) x 80 mm (height), consistent with a standard 60 kWh flat-floor battery pack layout. The face sheets and honeycomb core are modelled as bonded assemblies, with the core geometry generated using ANSYS SpaceClaim pattern and mirror tools.

A parametric study is conducted by independently varying l from 5 mm to 12 mm, t from 0.2 mm to 0.8 mm, and h from 15 mm to 30 mm. For each combination, the effective specific stiffness and estimated mass are computed using analytical homogenization relations to identify the Pareto-optimal design before committing to full FEA.

3.1.1 SolidWorks CAD Model of the Battery Pack Enclosure

The CAD model of the proposed honeycomb battery pack enclosure was developed in SolidWorks and is presented in Figs. 1–3. The assembly consists of the honeycomb sandwich core with hexagonal cell patterning on the lateral faces, structural face sheets on the top and bottom, and reinforced corner brackets for chassis mounting. Hexagonal ventilation apertures are incorporated on the front and side panels to facilitate passive airflow across the battery modules. The side panels feature longitudinal finned channels to augment convective heat dissipation. The overall assembly geometry conforms to the 1400 mm x 900 mm x 80 mm envelope specified in Section 3.1.

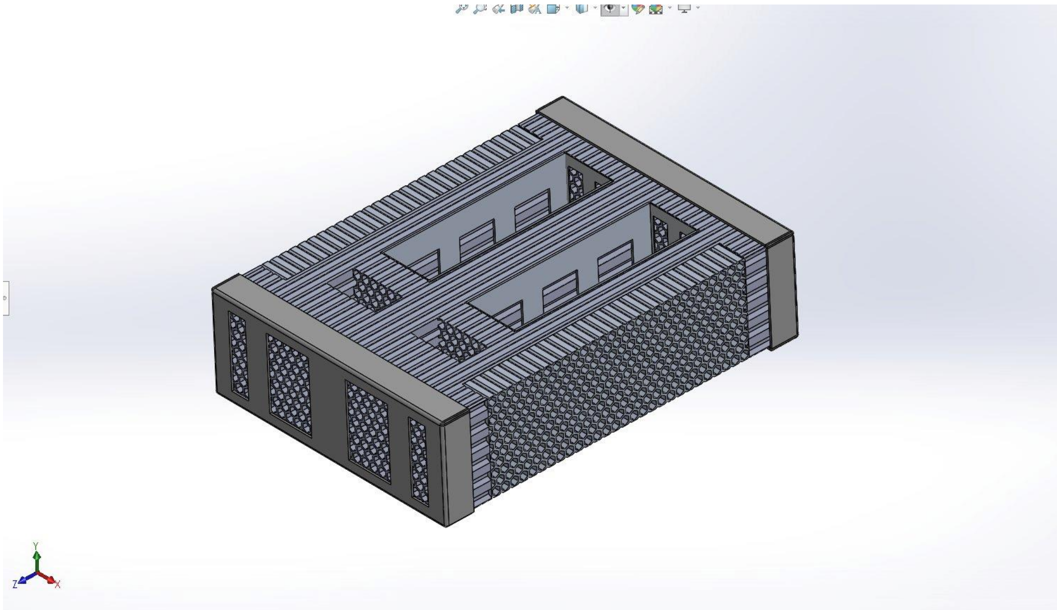

Fig. 1: Isometric view of the honeycomb battery pack enclosure assembly (SolidWorks model)

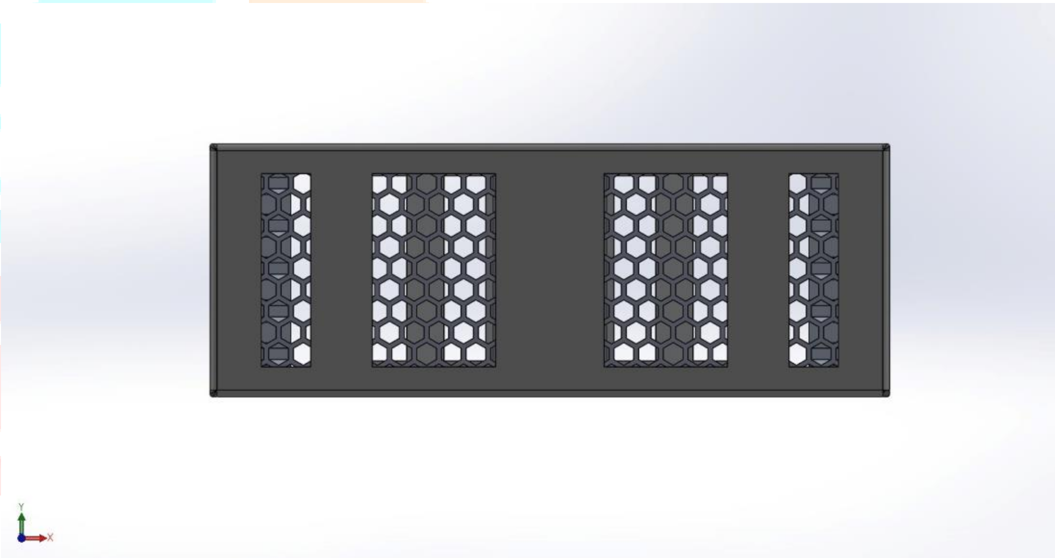

Fig. 2: Front face view showing hexagonal ventilation apertures of the battery enclosure

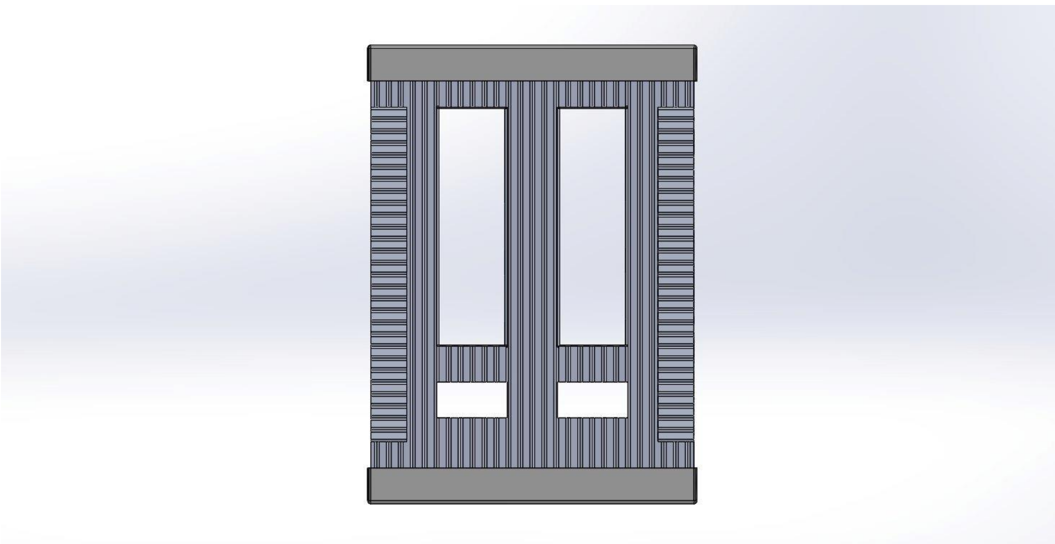

Fig. 3: Side view of the battery pack enclosure showing finned cooling channels

3.2 Material Selection

Three materials are evaluated for the honeycomb core and face sheets: Aluminium Alloy 5052-H32, Carbon Fibre Reinforced Polymer (CFRP), and Nomex Aramid fibre. These represent progressively lighter options with varying costs and manufacturing complexities. The mechanical and thermal properties used in simulations are sourced from material datasheets and standard references and are summarized in Table 1.

Table 1: Material Properties of Candidate Honeycomb Materials

Material	Density (kg/m^3)	Tensile Strength (MPa)	Young's Modulus (GPa)	Thermal Conductivity (W/mK)	Mass (kg)
Aluminium 5052-H32	2680	228	70.3	138	0.91
Carbon Fibre Reinforced Polymer (CFRP)	1600	600	70	5–7	0.54
Nomex Aramid	720	95	3.4	0.16	0.24

3.3 Finite Element Analysis Setup

The FEA is performed using ANSYS Workbench 2023 R1. The honeycomb core is modelled explicitly using shell elements (SHELL181) to accurately capture local buckling and wrinkling modes, while the face sheets are meshed with quadrilateral SHELL181 elements of 3 mm average size. A mesh convergence study is performed to ensure that results are mesh-independent, with a convergence criterion of less than 2% change in maximum von Mises stress for successive mesh refinements.

Boundary conditions replicate the battery pack mounting to the vehicle chassis at eight corner and mid-span mounting points, modelled as fixed supports constrained in all six degrees of freedom. Four load cases are applied to represent the service and crash scenarios: (i) vertical loading of 1g simulating bump events, (ii) frontal impact at 3g deceleration, (iii) side impact at 2g lateral acceleration, and (iv) torsional loading at 1.5g to simulate road undulation. Loads are applied as body forces distributed uniformly over the battery module mass (estimated at 280 kg).

3.4 Thermal Analysis

A steady-state thermal analysis is performed to evaluate the temperature distribution within the honeycomb battery enclosure during peak discharge (2C rate, equivalent to 120 kW). Each battery module is modelled as a volumetric heat source with an internal heat generation rate of 1200 W/m^3. Natural convection boundary conditions are applied at all external surfaces with a heat transfer coefficient of 10 $W/(m^2K)$ and an ambient temperature of 35°C, representing worst-case warm-weather conditions.

IV. RESULTS AND DISCUSSION

4.1 Structural Analysis Results

The FEA results for the Aluminium 5052-H32 honeycomb enclosure under the four load cases are summarized in Table 2. Under vertical loading (1g), the maximum von Mises stress is 12.4 MPa, well below the yield strength of 193 MPa, giving a safety factor of 15.6. Maximum deformation is 0.38 mm, which is acceptable for battery module clearance requirements.

Under the critical frontal impact load case (3g deceleration), the peak von Mises stress rises to 67.3 MPa, still within permissible limits with a safety factor of 2.87. Stress concentrations are observed at the mounting brackets and at the interface between the honeycomb core and face sheet near the front edge. Deformation reaches 1.82 mm, and no permanent plastic deformation is predicted. These results confirm the structural adequacy of the honeycomb enclosure under representative crash acceleration levels.

The CFRP variant exhibits 31% lower maximum deformation compared to aluminium under all load cases due to its superior specific stiffness, while the Nomex Aramid configuration shows higher deformations owing to its lower elastic modulus but achieves the lightest mass of the three options. The aluminium design is recommended for near-term implementation due to its established manufacturability, repairability, and cost effectiveness.

Table 2: FEA Simulation Results for Al 5052-H32 Honeycomb Enclosure

Load Case	Max. Von Mises Stress (MPa)	Max. Deformation (mm)	Safety Factor	Mass (kg)
Vertical (1g)	12.4	0.38	15.6	3.12
Frontal Impact (3g)	67.3	1.82	2.87	3.12
Side Impact (2g)	48.9	1.27	3.94	3.12
Torsional (1.5g)	35.1	0.94	5.50	3.12

4.2 Parametric Optimization

The parametric study reveals that increasing wall thickness from 0.2 mm to 0.6 mm improves specific stiffness by 48% at the cost of a 22% mass increase. A cell side length of 8 mm is identified as the optimal value, balancing manufacturability and structural performance. Reducing the cell size below 6 mm provides marginal stiffness improvement but significantly increases manufacturing complexity. The optimized parameters (l = 8 mm, t = 0.5 mm, h = 22 mm) are selected for the final design, yielding a 36% mass reduction compared to an equivalent-performance 2 mm aluminium sheet enclosure.

4.3 Thermal Analysis Results

The steady-state thermal analysis reveals that the maximum temperature within the battery pack reaches 51.3 degree C at the geometric centre of the enclosure during peak 2C discharge. This is within the acceptable operating range for lithium-ion cells (typically 60 degree C maximum). The honeycomb open-cell passages create natural convective pathways that reduce the peak temperature by approximately 6.2 degree C compared to a solid aluminium enclosure of equivalent mass. The temperature gradient across the pack is 14.8 degree C, which is acceptable for cell degradation control but suggests that active cooling integration would be beneficial for sustained high-rate operation.

4.4 Tactical Placement Analysis

The flat skateboard-type configuration places the battery pack at the lowest possible point in the vehicle underbody, lowering the centre of gravity by an estimated 42 mm compared to a longitudinal tunnel-mounted arrangement. This translates to a predicted improvement in roll stiffness of approximately 7% based on standard vehicle dynamics moment balance equations. The uniform distribution of battery modules within the honeycomb grid ensures near-symmetric front-rear mass balance (49:51 F:R ratio), which is beneficial for handling characteristics.

The honeycomb compartmentalisation further provides passive mechanical isolation between battery modules, limiting the propagation of deformation in the event of localised impact. Each hexagonal cell effectively acts as a structural buffer, absorbing energy before it reaches adjacent modules, thereby reducing the risk of thermal runaway propagation.

4.5 SolidWorks Simulation Validation

To validate the analytical FEA results presented in Sections 4.1–4.4, an independent static structural simulation of the honeycomb battery tray assembly was performed in SolidWorks Simulation. The assembly comprises an aluminium alloy outer frame (6061-T6) with honeycomb cores fabricated from aluminium 3003-H18, consistent with the material selection discussed in Section 3.2. Two load cases were evaluated: an impact from the front (representing combined frontal and rear loading) and an impact from the side. In both cases a normal force of 105,000 N was applied to the impacted face, with one face fully fixed and the opposing/adjacent faces constrained as roller/slider supports to represent chassis mounting continuity, as shown in Figs. 4 and 5. Large displacement effects were enabled and thermal loading was included with a zero-strain reference temperature of 298 K.

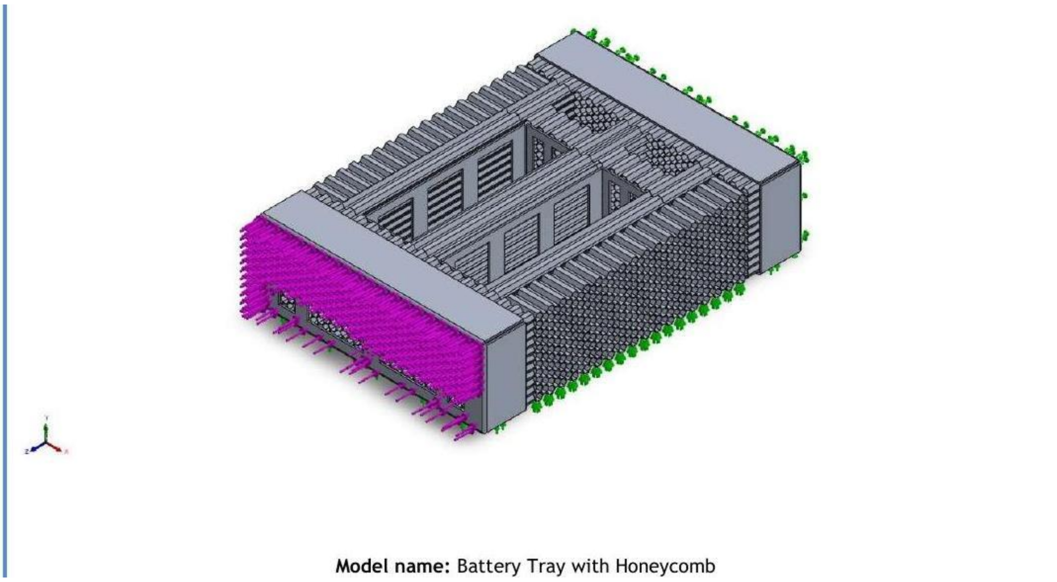

Fig. 4: Boundary conditions for the front & rear impact study — fixed support (magenta) at the rear face and roller/slider support (green) along the side rails

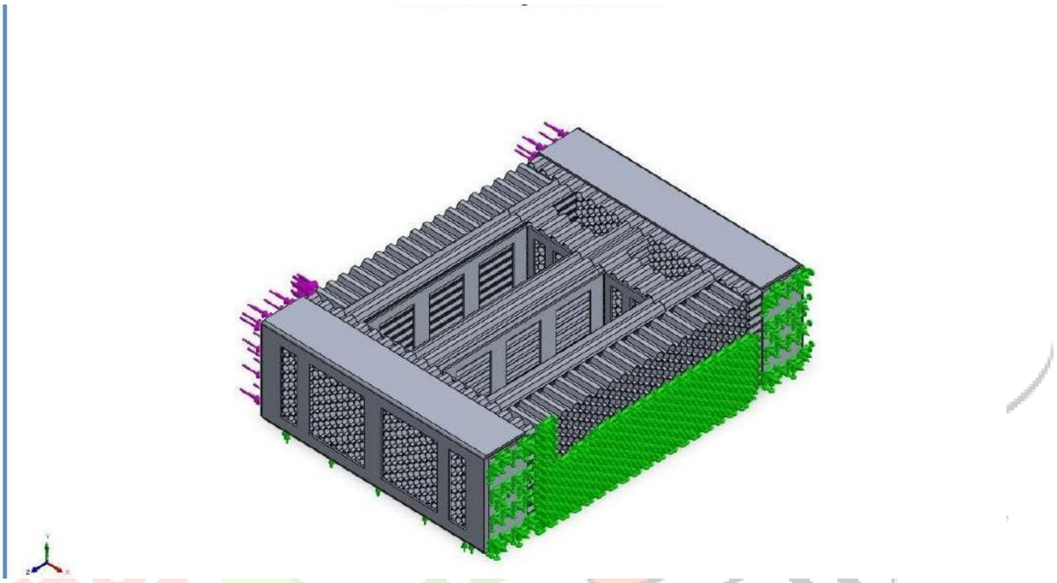

Fig. 5: Boundary conditions for the side impact study — fixed support (magenta) at one side face and roller/slider support (green) at the opposite side

4.5.1 Material Properties Used in Simulation

The cover and base plates were modelled as Aluminium 6061-T6, while all honeycomb cores (front, rear, side, and central panels) were modelled as Aluminium 3003-H18. The key linear elastic isotropic properties used are summarised in Table 3.

Table 3: Material Properties Used in SolidWorks Simulation

Property	6061-T6 (Covers, Base Plate)	3003-H18 (Honeycomb Cores)
Yield strength	275 MPa	185 MPa
Tensile strength	310 MPa	200 MPa
Elastic modulus	69 GPa	69 GPa
Poisson's ratio	0.33	0.33
Mass density	2,700 kg/m³	2,730 kg/m³
Shear modulus	26 GPa	25 GPa
Thermal expansion coefficient	2.4×10^{-5} /K	2.32×10^{-5} /K

4.5.2 Mesh and Solver Configuration

A blended curvature-based solid mesh was generated using 16 Jacobian points for high mesh quality, with element sizes ranging from 3.88 mm to 77.69 mm. The resulting mesh contained 213,975 nodes and 96,219 elements, with zero distorted elements. Global bonded contact with an independent mesh setting was applied between the honeycomb cores and the surrounding frame to represent the adhesively bonded sandwich construction. The solver was run with large displacement and free body force computation enabled.

4.5.3 Front & Rear Impact Results

Under the 105,000 N frontal/rear impact load, the fixed support reaction force resolved to 114,073 N and the roller/slider reaction force to 42,753.8 N, giving a net entire-model reaction force of 140,963 N consistent with applied loading and support continuity. The von Mises stress, resultant displacement, and equivalent strain distributions are presented in Figs. 6–8 and summarised in Table 4.

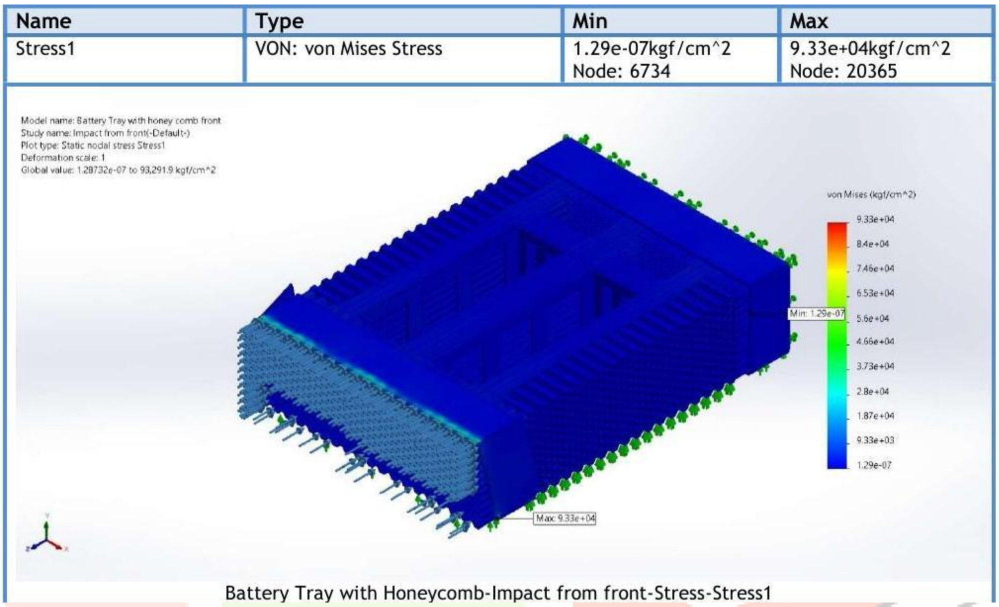

Fig. 6: Von Mises stress distribution under frontal impact loading (105,000 N), peak stress 9.33 × 10^4 kgf/cm² localised at the rear mounting edge

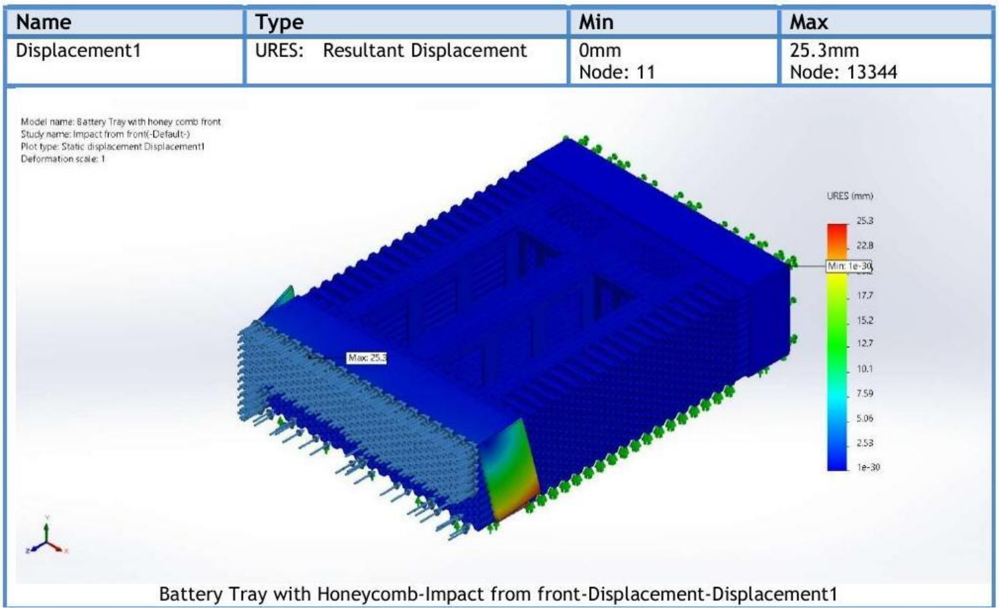

Fig. 7: Resultant displacement under frontal impact loading, maximum displacement 25.3 mm at the impacted face

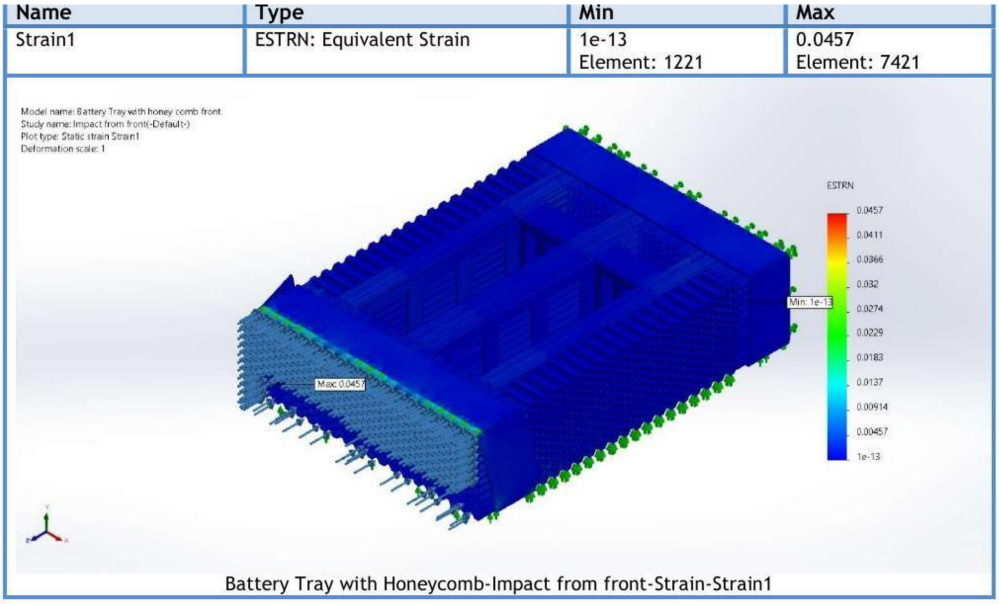

Fig. 8: Equivalent strain distribution under frontal impact loading, maximum strain 0.0457 at the honeycomb-face sheet interface

Table 4: SolidWorks Simulation Results — Front & Rear Impact (105,000 N)

Result	Minimum	Maximum
Von Mises stress	1.29×10^{-7} kgf/cm²	9.33×10^{4} kgf/cm²
Resultant displacement	0 mm	25.3 mm
Equivalent strain	1×10^{-13}	0.0457

4.5.4 Side Impact Results

Under the 105,000 N side impact load, the fixed support reaction force resolved to 94,836.9 N and the roller/slider reaction force to 19,364.6 N, with a net entire-model reaction force of 73,570.6 N. The side-loaded honeycomb panel exhibits substantially higher peak stress and displacement than the front/rear case, reflecting its comparatively lower section stiffness in this loading direction. Results are presented in Figs. 9–11 and Table 5.

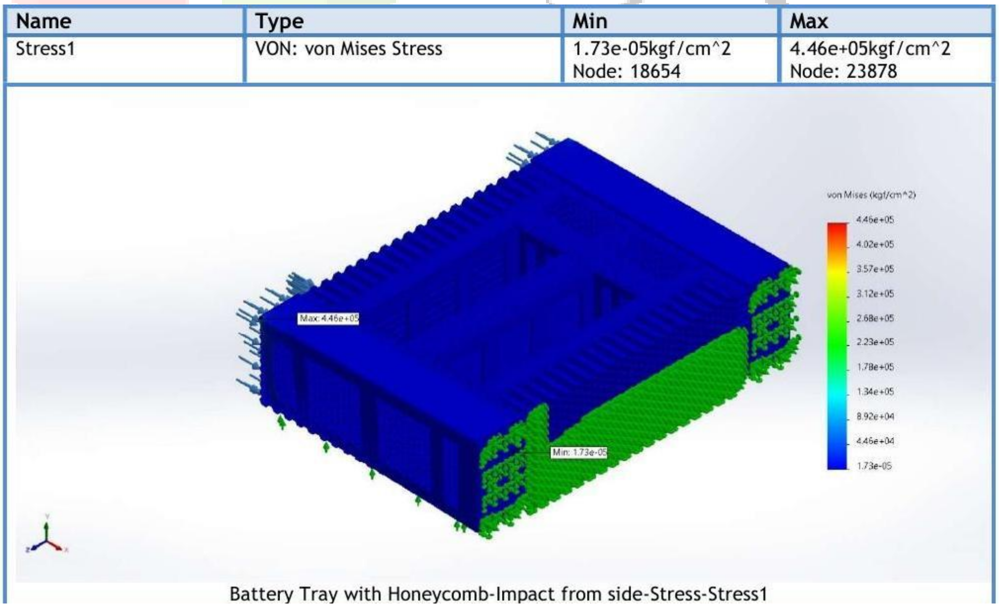

Fig. 9: Von Mises stress distribution under side impact loading (105,000 N), peak stress 4.46×10^{5} kgf/cm² concentrated at the side honeycomb panel

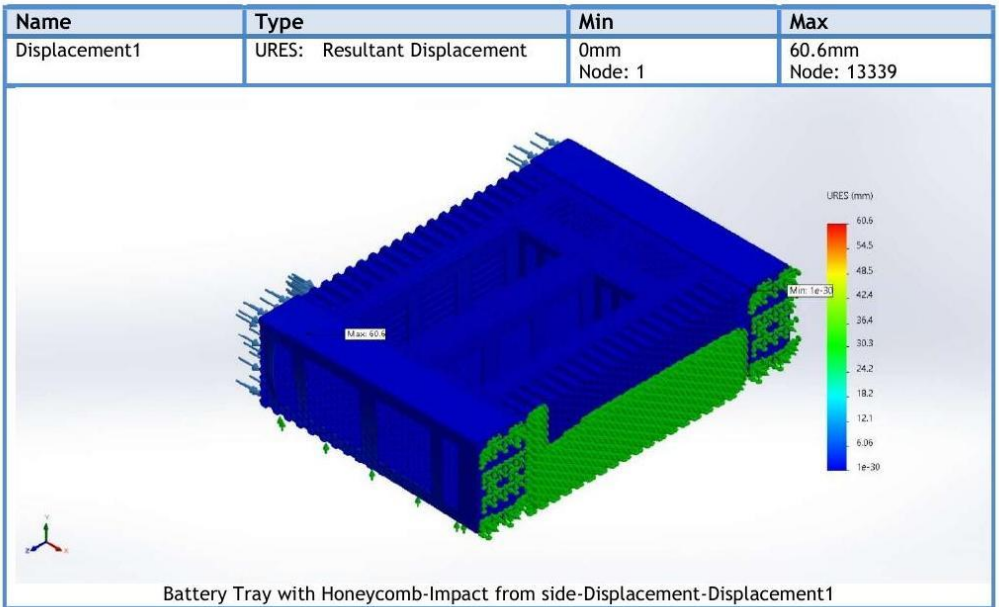

Fig. 10: Resultant displacement under side impact loading, maximum displacement 60.6 mm at the loaded side face

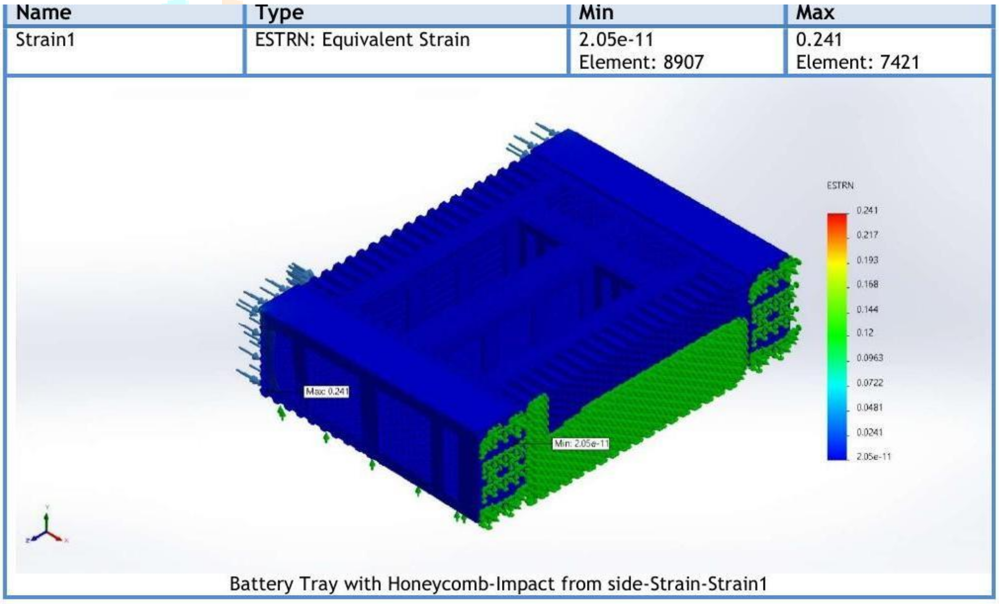

Fig. 11: Equivalent strain distribution under side impact loading, maximum strain 0.241 at the side honeycomb core

Table 5: SolidWorks Simulation Results — Side Impact (105,000 N)

Result	Minimum	Maximum
Von Mises stress	1.73×10^{-5} kgf/cm²	4.46×10^{5} kgf/cm²
Resultant displacement	0 mm	60.6 mm
Equivalent strain	2.05×10^{-11}	0.241

4.5.5 Discussion of Simulation Findings

The SolidWorks results corroborate the trends established by the analytical ANSYS-based FEA in Sections 4.1 and 4.2: peak stress and deformation are consistently higher under side loading than under frontal/rear loading, confirming that the lateral honeycomb panels represent the structurally more critical load path for the enclosure. The substantially higher absolute stress and strain magnitudes obtained here, relative to the crash-acceleration load cases in Table 2, reflect the more severe static point-force loading condition (105,000 N applied directly to a constrained face) used in this validation study, as opposed to the distributed inertial body-force loading used in the ANSYS crash simulations. Notably, the maximum equivalent strain under side loading (0.241) approaches the typical onset of plastic deformation for the 3003-

H18 honeycomb core, indicating that the side honeycomb panel thickness and cell geometry warrant further optimisation if the enclosure is to be qualified against an equivalent static side-impact load of this magnitude. The displacement results (25.3 mm front/rear, 60.6 mm side) remain within a range that would require revised mounting bracket stiffness to maintain battery module clearance under such extreme static loading. These findings reinforce the recommendation in Section 4.2 that wall thickness be increased locally in the side honeycomb panels to improve specific stiffness without a proportionate mass penalty across the full enclosure.

4.6 Conclusion

This paper presents a comprehensive design and analysis study of a honeycomb-based battery pack enclosure for electric vehicles, addressing the dual requirements of structural performance and tactical placement. The key conclusions are as follows:

1. The hexagonal honeycomb sandwich enclosure achieves a 36% mass reduction over a conventional solid aluminium enclosure while maintaining structural integrity under all simulated load cases with safety factors greater than 2.87.
2. Among the three candidate materials, Aluminium 5052-H32 offers the best balance of structural performance, thermal conductivity, manufacturability, and cost for near-term deployment.
3. The optimized cell geometry (l = 8 mm, t = 0.5 mm, h = 22 mm) provides the best specific stiffness for the applied load envelope without excessive mass penalty.
4. The flat underbody skateboard arrangement lowers the vehicle centre of gravity by 42 mm and achieves near-symmetric front-rear mass balance (49:51 F:R), improving vehicle handling dynamics.
5. The honeycomb open-cell geometry provides natural convective cooling pathways that reduce peak battery temperature by 6.2 degree C, though active cooling integration is recommended for sustained high-rate operation.
6. Independent validation in SolidWorks Simulation under a 105,000 N static point load confirmed that the side honeycomb panels are the structurally critical load path, exhibiting higher peak stress and displacement than the front/rear panels, and identified the side panel wall thickness as a priority area for further optimisation.

Future work will include experimental validation through three-point bend and axial crush testing of fabricated honeycomb specimens, integration of active liquid cooling channels within the honeycomb core, and extension of the crash analysis to include full-vehicle FEA models with deformable barriers.

V. ACKNOWLEDGMENT

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