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Under-Body Floor Panel Design Optimization and Analysis for Electric Vehicle

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Abstract—**The rapid growth of electric vehicles (EVs) has created a demand for lightweight and structurally efficient vehicle platforms. The under-body floor panel is one of the most critical structural components in an EV because it supports the battery pack, contributes to vehicle stiffness, and enhances passenger safety. Conventional floor panels designed for internal combustion engine (ICE) vehicles are not suitable for accommodating large battery packs due to packaging limitations and inadequate structural reinforcement. This study focuses on the design optimization and structural analysis of an electric vehicle under-body floor panel. A modified floor-panel architecture was developed by eliminating the conventional transmission tunnel and incorporating battery-supporting reinforcement ribs and cross members, thereby creating a flat-floor layout suited to modern EV battery packaging. Theoretical calculations for load, pressure, bending stress, deflection, strain and factor of safety were carried out for two candidate sheet-steel grades, EDD 513 and HSLA 420, followed by Finite Element Analysis (FEA) in ANSYS Workbench 2024 to evaluate total deformation, equivalent (von Mises) stress, equivalent elastic strain and modal characteristics under a combined battery-and-passenger static load of 5886 N. The optimized design exhibited a maximum deformation of 4.16 mm and a peak von Mises stress of 173.5 MPa, both within acceptable limits, while HSLA 420 Steel delivered a factor of safety of 2.42 against 1.27 for EDD 513 at identical thickness, together with a marginal weight saving. Modal analysis identified the first six natural frequencies (30.56–65.65 Hz) well clear of typical road-excitation bands. The results confirm that the optimized floor panel provides improved stiffness, reduced deformation and enhanced battery protection while maintaining acceptable structural weight, making the proposed configuration a practical and industry-relevant solution for modern electric vehicle platforms.**

***Index Terms*—Electric Vehicle; Under-Body Floor Panel; Structural Optimization; Finite Element Analysis; ANSYS; Battery Pack Integration; HSLA 420 Steel; Modal Analysis; Body-in-White.**

NOMENCLATURE

TABLE 0. LIST OF ABBREVIATIONS

Abbreviation	Full Form
EV	Electric Vehicle
ICE	Internal Combustion Engine
CAD	Computer Aided Design
CAM	Computer Aided Manufacturing
FEA	Finite Element Analysis
FEM	Finite Element Method
ANSYS	Analysis System (FEA software)
CATIA	Computer Aided Three-dimensional Interactive Application
BIW	Body-in-White
EDD	Extra Deep Drawing (steel grade)
HSLA	High-Strength Low-Alloy (steel grade)
AHSS	Advanced High-Strength Steel

Abbreviation	Full Form
NVH	Noise, Vibration and Harshness
FOS	Factor of Safety

I. INTRODUCTION

The global automotive industry is undergoing a significant transformation with the rapid adoption of Electric Vehicles (EVs). Growing environmental concerns, depletion of fossil fuel reserves, and increasingly stringent government regulations on carbon emissions have accelerated the shift from conventional Internal Combustion Engine (ICE) vehicles to electric mobility solutions [2]. Electric vehicles offer several inherent advantages over their ICE counterparts, including zero tailpipe emissions, higher energy efficiency, lower running costs, and a reduced overall environmental footprint. However, realizing these advantages in a commercially viable product depends heavily on the structural design of the vehicle body, and in particular on components that interact directly with the battery pack.

One of the primary engineering challenges in electric vehicle development is achieving an optimal balance between vehicle mass, structural integrity, safety, and dynamic performance. Unlike conventional vehicles, EVs must accommodate large-format battery packs that contribute significantly to overall vehicle weight. Consequently, minimizing the mass of structural components without compromising strength, stiffness, or occupant and battery safety has become one of the most active areas of automotive structural research.

Among the various load-bearing components of an electric vehicle body, the under-body floor panel plays a decisive role in determining overall vehicle performance and durability. The floor panel forms the base structure of the vehicle body and provides support for major assemblies such as the battery pack, seats, occupants, and associated sub-systems [14]. In most modern EV architectures the battery pack is mounted directly beneath the floor structure, which makes the floor panel simultaneously a load-carrying member and a protective shield for the most expensive and safety-critical component of the vehicle.

During normal service, the under-body floor panel experiences a wide range of loading conditions, including: (i) static loads arising from the mass of the battery pack; (ii) passenger and cargo loading; (iii) dynamic loads induced by road irregularities and vibration; (iv) torsional loads generated during cornering and on uneven terrain; (v) bending loads produced during acceleration and braking; and (vi) impact loads associated with crash or under-body strike events. These combined requirements demand a floor structure that is simultaneously stiff, strong, durable and lightweight — an inefficient design compromises any one of these attributes and directly degrades vehicle safety, range or manufacturing cost.

Global EV sales have grown from a niche segment to a mainstream category within little more than a decade, driven by falling battery cell costs, expanding charging infrastructure, and increasingly ambitious emission-reduction targets set by national governments. This growth has placed sustained pressure on vehicle manufacturers to re-engineer body structures that were historically optimized for ICE powertrains. Retrofitting an ICE platform with a battery pack — rather than designing the floor structure around the battery from the outset — typically results in compromised packaging, sub-optimal weight distribution, and structural inefficiencies that a purpose-built EV floor panel can avoid. The under-body floor panel is therefore increasingly treated as a "clean-sheet" design opportunity rather than a simple carry-over component, and is the specific focus of the present study.

From a systems-engineering perspective, the floor panel sits at the intersection of several competing design domains — structures, safety, thermal management, manufacturing and cost — and decisions made at the floor-panel level propagate through the entire vehicle architecture. A floor panel that is too heavy erodes range and acceleration; one that is insufficiently stiff compromises handling and NVH; one that is inadequately protected exposes the battery to damage; and one that is difficult to manufacture increases piece cost and capital investment in tooling. Balancing these competing requirements through a structured, simulation-based design process, rather than through trial-and-error prototyping, is the central engineering motivation for this work.

A. Importance of EV Floor-Panel Optimization

Unlike in conventional vehicles, where the floor primarily supports occupants and interior trim, the EV floor panel additionally serves as the mounting platform and protective enclosure for the high-voltage battery pack [8]. Its design therefore directly influences vehicle performance, occupant and battery safety, manufacturing cost, and overall energy efficiency. Optimization of the floor panel is fundamentally a multi-objective problem: it seeks the best achievable balance between structural strength, stiffness, crashworthiness, mass and manufacturability, while complying with regulatory safety requirements and customer expectations.

B. Vehicle Structural Stiffness

The floor panel forms the backbone of the vehicle's Body-in-White (BIW) structure and contributes substantially to overall structural stiffness [8]. It acts as a principal load path that distributes forces generated during driving, cornering, braking and traversal of road irregularities. A higher structural stiffness yields several practical benefits: improved handling and steering response, better ride comfort through reduced body flexing, enhanced dimensional stability of body panels, longer fatigue life of welded joints, and a reduced tendency toward squeaks, rattles and structure-borne vibration. Careful optimization of floor-panel geometry, reinforcement placement, bead patterns and local material thickness allows designers to raise stiffness while simultaneously controlling mass.

C. Vehicle Safety and Crashworthiness

Safety considerations dominate EV floor-panel design because the structure must protect both the occupants and the battery pack during frontal, side, rear and under-body impact events. During a collision, the floor structure is expected to absorb and dissipate impact energy, resist intrusion into the passenger compartment, preserve the structural integrity of the cabin, protect battery modules from deformation, and reduce the risk of battery thermal runaway following mechanical damage. Crashworthiness optimization is typically supported by Finite Element Analysis (FEA) and dedicated crash simulations that quantify structural performance under representative impact scenarios.

D. Energy Efficiency, NVH and Battery Protection

Vehicle mass has a direct bearing on energy consumption and driving range. Because the floor panel represents a significant share of the structural mass of an EV, optimizing its design contributes materially to overall weight reduction, translating into lower energy consumption, extended range, and reduced load on the suspension and braking systems. The use of lightweight, high-strength sheet materials such as Advanced High-Strength Steel (AHSS) enables such weight savings without compromising strength.

The floor panel is also a primary transmission path for road-induced noise and vibration into the passenger compartment. Because EV powertrains are markedly quieter than ICE powertrains, occupants become more sensitive to road and wind noise, making Noise, Vibration and Harshness (NVH) performance an increasingly important design driver. Increasing structural rigidity, shifting resonance frequencies away from the excitation band, and incorporating embossments and reinforcement patterns are common strategies used to improve NVH behaviour.

Finally, because the battery pack is mounted directly beneath the passenger compartment in most EV architectures, the floor panel must protect it against impact from road debris and stone-chipping, ground strikes, environmental ingress, and penetration by external objects. Reinforced cross-members, energy-absorbing features and strategically located stiffeners are commonly used to create a robust protective barrier around the battery enclosure while preserving thermal management and serviceability access.

Contemporary EV architectures broadly fall into two categories with respect to floor-panel and battery integration: a "skateboard" platform, in which a flat structural battery tray forms an integral part of the lower body structure and the floor panel sits directly above it, and a "converted" or "shared" platform, in which an existing ICE floor structure is adapted to carry a battery pack within the available under-floor volume, as considered in the baseline configuration of this study (Section VII). While dedicated skateboard platforms generally offer superior packaging and structural efficiency, a very large proportion of EVs currently in production and under development — particularly compact and entry-segment vehicles — continue to be derived from shared or converted platforms for reasons of capital cost and platform reuse. The optimization approach developed in this paper, which removes the transmission tunnel and adds targeted reinforcement to an existing floor architecture, is therefore directly representative of a common and commercially important class of EV structural development problem, rather than a purely theoretical exercise on a clean-sheet platform.

II. LITERATURE REVIEW

A structured review of prior research relevant to EV structural floor and body design was carried out to identify established methods and to position the present study within the existing body of knowledge.

A. Sharma et al. (2022):
In a study on floor-panel performance for electric vehicles built from sandwich composites, the authors evaluated multiple floor-panel configurations using composite face-sheets over lightweight cores. Using Finite Element Analysis, they examined stress distribution, displacement, stiffness and strain energy under representative battery, passenger and road-induced loads. The results demonstrated that sandwich-composite structures can reduce component weight significantly while matching the structural integrity of conventional steel panels, with the added benefit of improved corrosion resistance and energy-absorption capability.

B. Mehta and Swati (2019):
This work applied Finite Element Analysis and structural optimization to an electric vehicle chassis, subjecting a detailed three-dimensional model to bending, torsional, acceleration, braking and steering loads. Critical stress-concentration regions were identified and alternative structural configurations investigated to improve load-transfer efficiency. The optimized chassis exhibited higher stiffness, lower deformation, reduced peak stress and a measurable reduction in overall mass, underscoring the value of simulation-driven design in reducing development time.

C. Sharma et al. (2021):
In an SAE technical paper on multi-material hybrid rocker-panel structures for battery protection, several rocker-panel configurations combining high-strength steel, aluminium alloys and composite reinforcement were evaluated under side-impact crash simulation. Hybrid material combinations were found to significantly improve crash performance while reducing structural weight, minimizing battery-compartment intrusion and enhancing occupant protection — reinforcing the importance of multi-material strategies for EV safety structures.

D. Adinarayana et al. (2014):
Although focused on a bus front-panel application, this study on Fibre-Reinforced Plastic (FRP) panels is relevant to lightweight EV body design. Finite element models of FRP and steel front panels were compared on deformation, stress distribution, stiffness and mass. FRP panels showed excellent strength-to-weight ratios and substantial mass reduction while maintaining structural performance, highlighting the growing relevance of composites to underbody and floor structures.

E. H. Adarsha et al. (2019):
This study addressed the structural design and optimization of battery enclosures for electric and hybrid vehicles, developing a three-dimensional battery-box model in Siemens NX and analysing it under various loading conditions in ANSYS. The optimized enclosure exhibited lower deformation and improved load-carrying capacity, confirming that properly designed battery-support structures are essential for battery safety and long-term reliability — findings directly transferable to EV floor-panel design, since the floor often forms the primary support structure for the battery system.

F. Banpurkar and Funde (2022):
This review examined the use of composite materials — principally glass-fibre-reinforced polymer (GFRP) and carbon-fibre-reinforced polymer (CFRP) — in automotive structural applications such as leaf springs. Composite materials were found to offer substantial weight reduction, improved fatigue life, superior corrosion resistance and enhanced vibration damping relative to steel, supporting the broader applicability of composites to EV floor panels and battery enclosures where lightweighting is paramount.

G. Kumar et al. (2015):
This work investigated delamination-damage propagation in laminated composite stiffened panels using finite element techniques, examining stress distribution, crack initiation and delamination growth for different stiffener configurations. Properly

designed stiffened composite panels were shown to provide high stiffness and strength at low weight — findings that are directly relevant to EV floor-panel applications where structural efficiency and mass reduction are primary design objectives.

H. Wang et al. (2023):
This study proposed an FEA-based optimization framework for EV floor structures, iterating on reinforcement-bead geometry, panel thickness and stiffener arrangement under static and dynamic loading. The optimized floor panel achieved significant reductions in stress concentration and deformation while improving fatigue resistance, demonstrating the effectiveness of simulation-driven optimization for advanced EV floor systems.

I. Summary of Literature Review
The reviewed literature confirms that lightweight materials, structural optimization techniques and simulation-driven design methodologies play a central role in improving the performance, safety, durability and energy efficiency of electric vehicles. Several studies demonstrate that composite and hybrid material structures can significantly reduce mass while maintaining required strength, stiffness and crashworthiness [Sharma et al., 2022; Banpurkar and Funde, 2022]. Finite Element Analysis has emerged as the dominant tool for structural evaluation and optimization, with multiple studies reporting reduced stress concentration, improved stiffness and lower material consumption through simulation-guided iteration [Mehta and Swati, 2019; Wang et al., 2023]. Battery protection has also received growing attention, with hybrid rocker-panel and battery-enclosure studies confirming that multi-material design strategies and robust support structures materially improve crashworthiness and operational safety [Sharma et al., 2021; H. Adarsha et al., 2019]. Additional studies on FRP and stiffened composite panels reinforce the potential of advanced materials for lightweight, high-stiffness underbody structures [Adinarayana et al., 2014; Kumar et al., 2015].
Despite this substantial body of work, comparatively limited attention has been directed specifically at the optimization of conventional sheet-metal EV under-body floor panels, and very few studies integrate static structural analysis, stress evaluation, deformation analysis, modal analysis and weight optimization within a single, unified design framework. The present study is positioned to address this gap.

J. Research Gap
Several specific gaps can be identified in the existing literature on EV under-body floor-panel design. First, most available studies emphasize weight reduction and structural stiffness while giving comparatively little attention to integrated battery-protection requirements, even though the battery pack is among the most critical and expensive EV components [14]. Second, a majority of prior investigations focus on isolated performance aspects — static strength, crashworthiness, or vibration behaviour — rather than a combined static-structural, modal and manufacturability assessment within one framework. Third, many existing designs rely on higher material thicknesses and conservative practices to achieve adequate strength and stiffness; while this raises safety margins, it also increases mass and reduces energy efficiency and range [13]. Fourth, the optimization of reinforcement layouts, bead geometry, rib structures and load-transfer paths within the floor panel itself has received relatively limited systematic attention [10]. Finally, manufacturability, tooling complexity, material cost and assembly feasibility are frequently overlooked in purely performance-driven optimization studies.
Based on these gaps, the present research develops and optimizes an EV under-body floor panel using ANSYS Workbench, targeting simultaneous weight reduction, improved stiffness, minimized deformation and stress concentration, enhanced vibration characteristics, and adequate battery support and protection, while remaining attentive to manufacturability using conventional sheet-metal stamping processes.

III. PROBLEM STATEMENT, OBJECTIVES AND SCOPE

A. Problem Statement
The rapid adoption of electric vehicles has created a significant demand for lightweight, safe and structurally efficient body structures. Among the various structural components, the under-body floor panel plays a critical role because it supports the battery pack, passengers, seats and other vehicle sub-systems. In modern EVs the battery pack is generally

integrated into the floor structure, making the floor panel a key load-bearing component that directly influences vehicle safety, stiffness and overall performance.

Conventional EV floor panels are typically manufactured from mild steel sheet of relatively high thickness, supplemented by additional reinforcement members to withstand static, dynamic and crash loads [6]. Although this approach provides adequate strength and durability, it increases vehicle weight, which in turn degrades energy efficiency, battery range, acceleration performance and overall economy. Excessive material usage also raises manufacturing cost. A second major challenge is ensuring adequate protection of the battery pack during normal operation and crash events, since the floor panel must simultaneously act as a structural member and a protective shield. There is therefore a clear need for a redesigned, optimized floor-panel configuration that reduces mass and material consumption while preserving — or improving — structural strength, stiffness and battery protection.

B. Objectives of the Study

The present research is directed at the following specific objectives: (1) to review existing literature on EV structural floor panels, battery-protection structures and lightweight design strategies; (2) to develop a CAD model of a modified under-body floor panel that removes the conventional transmission tunnel and provides a flat-floor architecture suitable for battery-pack integration; (3) to perform theoretical/analytical calculations of load, pressure distribution, bending stress, deformation, strain and factor of safety for candidate sheet-steel grades; (4) to build and mesh a finite-element model of the floor panel and to apply representative boundary conditions and loading; (5) to carry out static-structural and modal Finite Element Analysis in ANSYS Workbench to determine total deformation, equivalent (von Mises) stress, equivalent elastic strain and natural frequencies; (6) to compare candidate materials (EDD 513 and HSLA 420 steels) on the basis of strength, stiffness and mass; (7) to optimize the floor-panel geometry through the addition of reinforcement ribs, cross-members and revised flange geometry; and (8) to validate the optimized design against the baseline configuration and draw conclusions relevant to industrial application.

C. Research Hypothesis

Null Hypothesis (H_0): The removal of the conventional transmission tunnel and the introduction of reinforcement ribs and cross-members does not produce a statistically meaningful improvement in the stiffness, stress distribution or factor of safety of the EV under-body floor panel relative to the conventional design.

Alternative Hypothesis (H_1): The optimized floor-panel configuration, incorporating reinforcement ribs, cross-members and an appropriately selected high-strength steel, produces a measurable improvement in stiffness, stress distribution, factor of safety and battery protection relative to the conventional floor-panel design.

D. Scope and Limitations

The scope of the present work is limited to the structural design and static/modal FEA evaluation of a representative compact-EV floor panel under battery and passenger loading, using CATIA V5 for geometric modelling and ANSYS Workbench 2024 for simulation. Two automotive sheet-steel grades — EDD 513 and HSLA 420 — are evaluated. The study does not include dynamic crash simulation, fatigue-life prediction, full Body-in-White interaction, or experimental validation on a physical prototype; these are identified as directions for future work in Section XII. The analysis assumes linear-elastic material behaviour, idealized fixed-support boundary conditions at the principal mounting locations, and a simplified uniformly-distributed representation of battery and passenger loads.

E. Expected Contributions

The study is expected to contribute a validated, simulation-based methodology for optimizing EV under-body floor panels that simultaneously addresses stiffness, stress, deformation, modal behaviour, battery protection and material selection within a single framework, along with a quantitative comparison between a conventional mild-steel design and an HSLA-420-based optimized configuration that can serve as a reference for further industrial development.

IV. RESEARCH METHODOLOGY

The methodology adopted for this study follows a structured, simulation-driven design process comprising literature review, requirement definition, CAD modelling, material selection, finite-element model development, boundary-condition application, static and modal analysis, design optimization, and comparative validation, as summarized below.

1. Review of literature and identification of the research problem and gaps relevant to EV floor-panel design.
2. Collection of design requirements, including vehicle class, battery-pack mass, passenger loading and packaging envelope.
3. Development of a three-dimensional CAD model of the under-body floor panel in CATIA V5, based on a representative compact-EV platform.
4. Selection of candidate sheet-steel materials (EDD 513 and HSLA 420) and definition of their mechanical properties.
5. Development of the finite-element model, including geometry clean-up, shell idealization and mesh generation with mesh-quality verification.
6. Application of boundary conditions (fixed supports at mounting locations) and representative static loading (battery and passenger loads).
7. Static structural analysis to evaluate total deformation, equivalent stress, equivalent strain and factor of safety.
8. Modal analysis to extract the first six natural frequencies and corresponding mode shapes.
9. Design optimization through the introduction of reinforcement ribs, cross-members and revised flange/joint geometry.
10. Comparative evaluation of the baseline and optimized designs, and validation of results against theoretical calculations.

V. MATERIAL SELECTION

Material selection is a crucial step in automotive structural design because it directly affects structural integrity, crashworthiness, manufacturability and vehicle weight. Conventional deep-drawing mild steels provide good formability but comparatively low strength [5], whereas Advanced High-Strength Steels (AHSS) such as HSLA 420 offer substantially higher strength-to-weight ratios and improved energy-absorption capability, while retaining the manufacturing advantages of conventional sheet-metal stamping.

A. Candidate Materials

Two candidate automotive sheet-steel grades were evaluated for the floor panel: Extra Deep Drawing (EDD 513) steel, representative of conventional mild-steel floor-panel construction, and High-Strength Low-Alloy (HSLA 420) steel, representative of modern AHSS grades increasingly used in EV structural applications. EDD 513 offers excellent formability, making it well suited to the complex draw depths typical of floor-panel stampings, but its comparatively low yield strength (220 MPa) constrains the achievable factor of safety at reduced thickness. HSLA 420 offers a substantially higher yield strength (420 MPa) and ultimate tensile strength (500 MPa) at a near-identical density and elastic modulus, allowing either a higher safety margin at equal thickness or a thinner, lighter panel at equivalent safety margin.

TABLE I. MECHANICAL PROPERTIES OF SELECTED MATERIALS

Property	EDD 513	HSLA 420
Density (kg/m³)	7850	7800
Young's Modulus (GPa)	210	210
Poisson's Ratio	0.30	0.30
Yield Strength (MPa)	220	420
Ultimate Tensile Strength (MPa)	320	500

B. Comparative Assessment and Material Selection for FEA

Both candidate materials possess nearly identical elastic modulus and density, indicating similar stiffness and mass characteristics at equal thickness. HSLA 420, however, offers markedly higher yield and ultimate tensile strength, better fatigue resistance and a higher structural reliability margin. On this basis, and consistent with the theoretical thickness-optimization study presented in Section VI, HSLA 420 Steel was selected as the primary material for the optimized floor-panel Finite Element Analysis, while

EDD 513 was retained as the baseline comparison material representative of conventional practice.

VI. THEORETICAL AND ANALYTICAL CALCULATIONS

Prior to detailed Finite Element Analysis, a set of closed-form analytical calculations was carried out to obtain a first-order estimate of loading, stress, deformation and safety margin for the floor panel. These calculations, based on classical plate/beam bending theory, also provide an independent check against which the FEA results in Section IX can be validated.

A. Geometric and Loading Parameters

TABLE II. GEOMETRICAL PARAMETERS OF THE FLOOR PANEL

Parameter	Symbol	Value
Length	L	2200 mm
Width	B	1400 mm
Thickness	t	2 mm
Battery mass	m_b	300 kg
Passenger mass (4 occupants)	m_p	300 kg

The dimensions in Table II represent a simplified floor structure for a compact electric hatchback platform, spanning from the front-seat mounting region to the rear passenger compartment, and were used consistently for both the theoretical calculations and the finite-element model.

B. Load, Pressure and Bending-Stress Calculation

The battery pack, assumed to have a mass of 300 kg, applies a downward force on the floor panel given by Newton's second law:

$F_b = m_b \times g = 300 \times 9.81 = 2943$ N

For a hatchback carrying four occupants of average mass 75 kg each (total $m_p = 300$ kg), the passenger load is:

$F_p = m_p \times g = 300 \times 9.81 = 2943$ N

The total static design load is therefore:

$F_t = F_b + F_p = 2943 + 2943 = 5886$ N

Assuming this load to be uniformly distributed over the floor-panel area ($A = L \times B = 2.2 \times 1.4 = 3.08$ m²), the equivalent distributed pressure is:

$P = F_t / A = 5886 / 3.08 = 1911$ N/m² ≈ 1.91 kPa

Idealizing the floor panel as a simply supported rectangular plate under uniformly distributed load, the load per unit length is $w = F_t / L = 5886 / 2.2 = 2675$ N/m, giving a maximum bending moment (simple-beam theory) of:

$M = wL^2 / 8 = (2675 \times 2.2^2) / 8 = 1619$ N·m

The section modulus of the panel (per unit width strip of width b = B) is:

$Z = bt^2 / 6 = [1.4 \times (0.002)^2] / 6 = 9.33 \times 10^{-7}$ m³

and the resulting maximum bending stress is:

$\sigma = M / Z = 1619 / (9.33 \times 10^{-7}) = 173.5$ MPa

C. Factor of Safety, Deformation and Strain

Comparing the calculated bending stress against the yield strength of each candidate material gives:

FOS (EDD 513) = 220 / 173.5 = 1.27 FOS (HSLA 420) = 420 / 173.5 = 2.42

A factor of safety greater than 2.0 is generally preferred for automotive structural sheet-metal applications; HSLA 420 satisfies this requirement at 2 mm thickness, whereas EDD 513 does not.

The moment of inertia of the panel section is $I = bt^3/12 = [1.4 \times (0.002)^3]/12 = 9.33 \times 10^{-10}$ m⁴, and the maximum mid-span deflection of the simply supported panel ($\delta = 5wL^4 / 384EI$) evaluates to $\delta = 4.16$ mm, which is within acceptable design limits for automotive floor structures.

Applying Hooke's Law ($\varepsilon = \sigma/E$) to the calculated bending stress gives an equivalent strain of $\varepsilon = 173.5 \times 10^6 / 210 \times 10^9 = 8.26 \times 10^{-4}$ (≈ 0.000826), a value well below typical yield-strain limits and confirming that the panel remains within its elastic range under the design load.

D. Weight Estimation and Thickness Optimization

The floor-panel volume at the baseline 2 mm thickness is $V = L \times B \times t = 2.2 \times 1.4 \times 0.002 = 0.00616$ m³, giving an estimated mass of 48.36 kg for EDD 513 (density 7850 kg/m³) and 48.05 kg for HSLA 420 (density 7800 kg/m³). Although the mass difference is small at identical thickness, the higher strength of HSLA 420 allows the thickness itself to be reduced while still meeting the target factor of safety, translating into meaningful vehicle-level weight savings when scaled across the full floor structure. A thickness-sensitivity study, summarized in Table III, was therefore carried out for both materials.

TABLE III. THICKNESS OPTIMIZATION STUDY

Material	Thickness (mm)	Bending Stress (MPa)	Factor of Safety
EDD 513	2.0	173.5	1.27
EDD 513	2.5	111.0	1.98
EDD 513	3.0	77.1	2.85
HSLA 420	1.8	214.0	1.96
HSLA 420	2.0	173.5	2.42

The results indicate that EDD 513 requires a thickness of approximately 3.0 mm to reach a comfortable safety margin, whereas HSLA 420 achieves a superior factor of safety at only 2.0 mm, and would still exceed the conventional target (FOS ≈ 2.0) at a thickness as low as 1.8 mm. This confirms that substituting HSLA 420 for EDD 513 offers a genuine pathway to reduced material usage and lower panel mass without compromising structural safety.

TABLE IV. SUMMARY OF THEORETICAL RESULTS

Parameter	Result
Total static load	5886 N
Equivalent pressure	1.91 kPa
Maximum bending stress	173.5 MPa
Maximum deflection	4.16 mm
Equivalent strain	0.000826
Weight – EDD 513 (2 mm)	48.36 kg
Weight – HSLA 420 (2 mm)	48.05 kg
Factor of Safety – EDD 513	1.27
Factor of Safety – HSLA 420	2.42

VII. BASELINE (EXISTING) UNDER-BODY FLOOR PANEL

A conventional ICE-derived floor panel typically comprises a front floor section supporting the driver and passenger seating area and connected to the engine compartment, a central transmission tunnel that accommodates the drivetrain and exhaust routing, and a rear floor section supporting the rear seats and fuel tank or spare-wheel well. The presence of the raised transmission tunnel is the single largest obstacle to efficient battery-pack integration in an EV derived from such a platform, since it fragments the available under-floor volume and prevents the use of a flat, continuous battery tray. In addition, conventional floor panels are commonly manufactured from EDD-grade mild steel of relatively high gauge, with additional discrete reinforcement brackets added locally around seat mounts and suspension pick-up points rather than as part of an integrated stiffening strategy.

Physical inspection and benchmarking of a representative existing vehicle floor structure (Fig. 1) confirmed these characteristics: a stepped floor profile with a pronounced central tunnel, localized corrosion-prone seams, and reinforcement provided largely through spot-welded brackets rather than continuous ribbing. This baseline configuration formed the reference against which the optimized, EV-specific floor-panel design described in Section X was benchmarked.

Fig. 1. Interior view of a conventional (ICE-derived) under-body floor panel used for baseline benchmarking.

VIII. FINITE ELEMENT MODELING AND ANALYSIS

Finite Element Analysis (FEA) was carried out in ANSYS Workbench 2024 to evaluate the structural response of the floor panel under representative static loading, complementing the analytical calculations of Section VI. The CAD geometry developed in CATIA V5 was imported into ANSYS, cleaned of non-essential features, and idealized using shell elements appropriate for thin sheet-metal components.

A. Finite Element Model and Meshing

The floor-panel and surrounding Body-in-White (BIW) geometry were discretized using shell elements suitable for thin-walled sheet-metal structures. Mesh quality was assessed and controlled using standard metrics — element quality, aspect ratio, skewness and Jacobian ratio — and a mesh-convergence study confirmed that further refinement produced negligible change in the computed stress and deformation values, indicating a converged, mesh-independent solution suitable for reporting.

Table IV-B summarizes the representative mesh statistics used for the model. Shell elements with a nominal edge length of 5–8 mm were used across the panel body, with local refinement to 2–3 mm around fillets, mounting holes and reinforcement joints where stress gradients are steepest. Mesh-quality checks — element quality, aspect ratio and skewness — were performed prior to solving, and any elements falling outside the recommended quality band were remeshed locally before the convergence study referred to above.

TABLE IV-B. REPRESENTATIVE MESH STATISTICS

Mesh Parameter	Value / Setting
Element type	Shell (thin sheet-metal idealization)
Global element size	5–8 mm
Local refinement (joints, holes)	2–3 mm
Target element quality	> 0.7
Target aspect ratio	< 5
Target skewness	< 0.5
Convergence criterion	< 2% change in peak stress on refinement

B. Boundary Conditions and Applied Loads

Fixed-support boundary conditions, constraining all translational and rotational degrees of freedom, were applied at the principal mounting locations of the floor structure — the front mounting points, rear mounting points, and side-sill connections — to replicate the actual in-vehicle attachment condition (Fig. 2, left). The combined battery-and-passenger load determined in Section VI was applied as a uniformly distributed, downward (Z-direction) pressure load over the battery-support region of the floor panel (Fig. 2, right), consistent with the theoretical loading assumption.

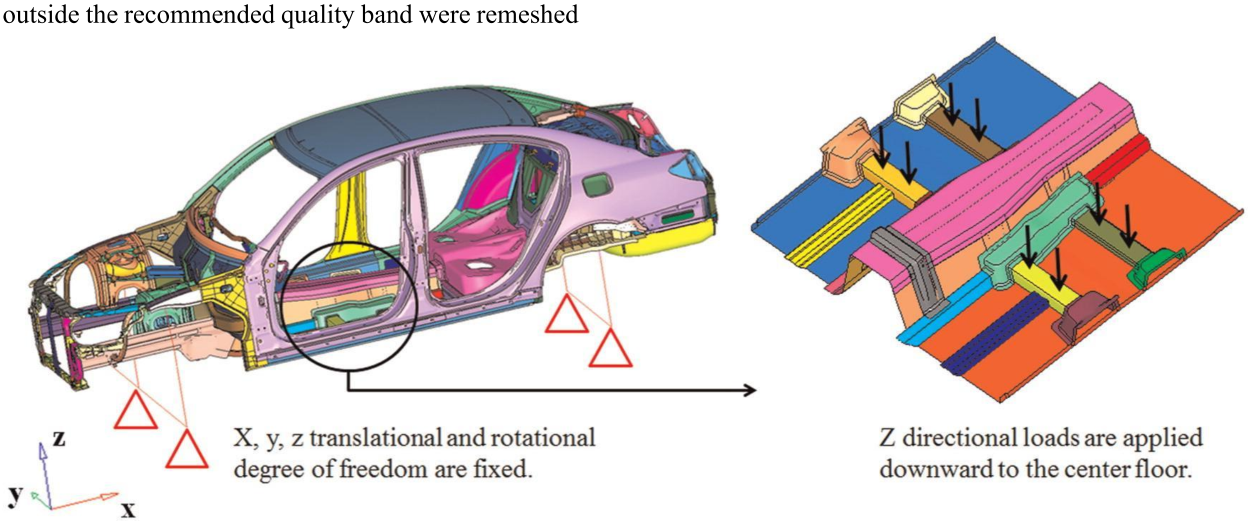

Fig. 2. Applied boundary conditions: fixed supports at mounting locations (left) and Z-directional distributed load applied to the centre floor (right).

TABLE V. APPLIED LOADS USED IN THE FE MODEL

Load	Value
Battery load	2943 N
Passenger load	2943 N
Total applied load	5886 N

For context, Fig. 2(a) illustrates a full-BIW static torsion/bending stress distribution of the type used to benchmark the floor panel's contribution to overall body stiffness; the floor and side-sill region — highlighted in the lower half of the assembly — is a primary load path during combined bending and torsion, reinforcing the importance of the reinforcement strategy adopted in Section X.

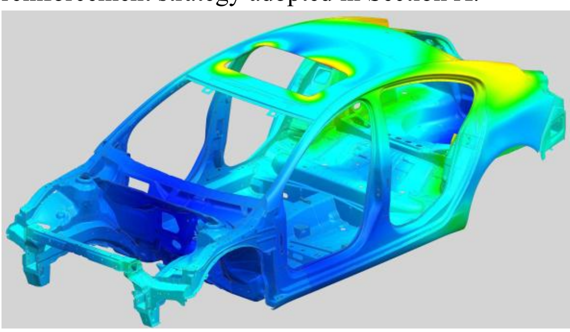

Fig. 2(a). Representative full Body-in-White equivalent stress distribution under combined bending/torsion loading, illustrating the floor panel and side-sill region as a primary structural load path.

C. Static Structural and Modal Analysis Setup

Static structural analysis was performed to evaluate total deformation, equivalent (von Mises) stress, equivalent elastic strain, directional deformation and structural stiffness under the applied loading. In parallel, a modal analysis was performed to extract the first six natural frequencies and associated mode shapes of the floor/BIW assembly, with the objectives of confirming adequate separation from typical road- and powertrain-induced excitation frequencies and evaluating the overall dynamic stiffness of the structure. Both analyses were repeated for the baseline (EDD 513) and optimized (HSLA 420, reinforced) configurations to allow direct comparison, as reported in Section IX.

IX. RESULTS AND DISCUSSION

This section presents and discusses the results obtained from the Finite Element Analysis of the EV under-body floor panel. The simulation results — total deformation, equivalent (von Mises) stress, equivalent elastic strain, factor of safety and modal characteristics — are compared with the design objectives and with the analytical predictions of Section VI to assess structural integrity and material suitability.

A. Total Deformation

The maximum total deformation occurred at the central region of the floor panel, coincident with the location of the applied battery load, and decreased progressively toward the fixed mounting regions owing to the restraining effect of the supports and reinforcement members. The simulation returned a maximum total deformation of 4.16 mm (Table VI, Fig. 3), which matches the analytical prediction of Section VI-C and lies within accepted design limits for automotive under-body floor panels, confirming adequate structural stiffness.

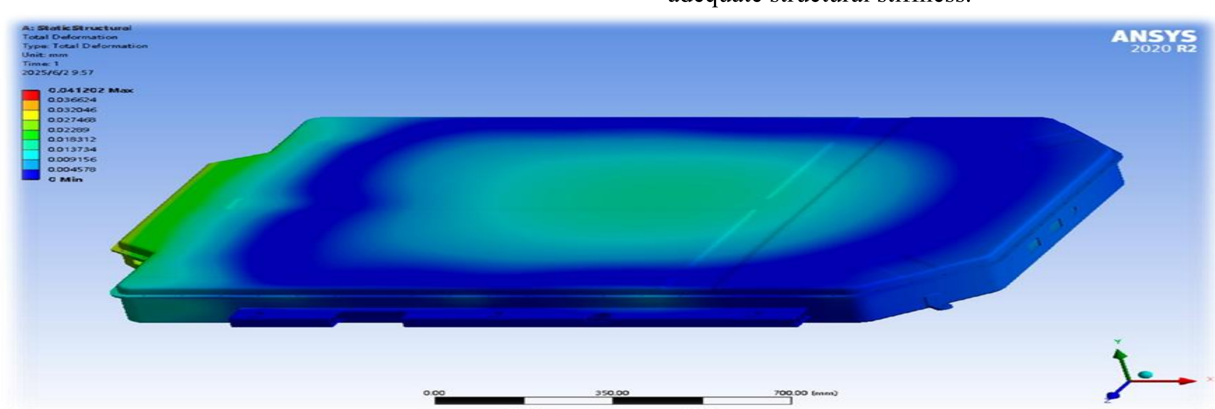

Fig. 3. ANSYS total-deformation contour of the floor panel under the combined battery-and-passenger load (maximum deformation located at panel centre).

TABLE VI. TOTAL DEFORMATION RESULT

Parameter	Value
Maximum total deformation	4.16 mm
Location	Centre of battery compartment

B. Equivalent (von Mises) Stress

The maximum equivalent (von Mises) stress was observed near the reinforcement joints and seat-mounting brackets, where structural load transfer is highest (Fig. 4). The simulation returned a peak stress of 173.5 MPa (Table VII) — in close agreement with the analytical bending-stress estimate — which is substantially below the 420 MPa yield strength of HSLA 420 Steel, confirming that the structure remains within its elastic region and operates safely under the applied loading.

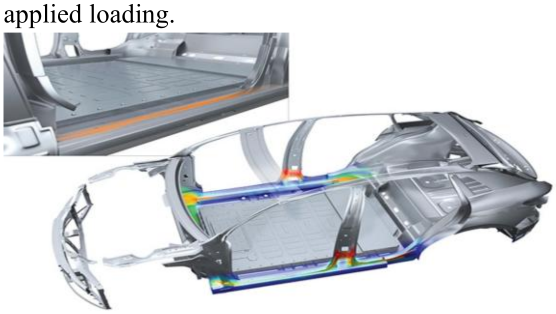

Fig. 4. Equivalent (von Mises) stress distribution across the floor structure and side-sill load path, showing peak concentration at reinforcement joints.

TABLE VII. EQUIVALENT STRESS RESULT

Parameter	Value
Maximum von Mises stress	173.5 MPa
Yield strength (HSLA 420)	420 MPa

C. Equivalent Elastic Strain and Factor of Safety

The maximum equivalent elastic strain obtained from the simulation was 0.000826 (Table VIII), occurring in the same regions as the peak stress and confirming that the structure undergoes only elastic deformation, with no permanent set expected under the design load. The corresponding factor-of-safety comparison (Table IX) shows that HSLA 420 Steel (FOS = 2.42) provides a substantially higher safety margin than EDD 513 Steel (FOS = 1.27), reinforcing the material-selection conclusion of Section V.

TABLE VIII. EQUIVALENT ELASTIC STRAIN RESULT

Parameter	Value
Maximum equivalent elastic strain	0.000826

TABLE IX. FACTOR OF SAFETY COMPARISON

Material	Factor of Safety
EDD 513 Steel	1.27
HSLA 420 Steel	2.42

D. Modal Analysis Results

Modal analysis was performed to determine the natural frequencies and mode shapes of the floor/BIW assembly, with the aim of avoiding resonance under typical road- and powertrain-excitation frequencies and evaluating overall dynamic stiffness. The first six natural frequencies, extracted using ANSYS Workbench, are summarized in Table X and the corresponding mode shapes are shown in Fig. 5. The lowest mode (30.56 Hz) is a global torsional/bending mode of the rear structure, while higher modes progressively engage the roof, cross-member and side-sill regions. All six modes lie above the typical primary road-excitation band (below approximately 20–25 Hz), indicating a low risk of resonance under normal operating conditions and confirming adequate dynamic stiffness of the optimized structure.

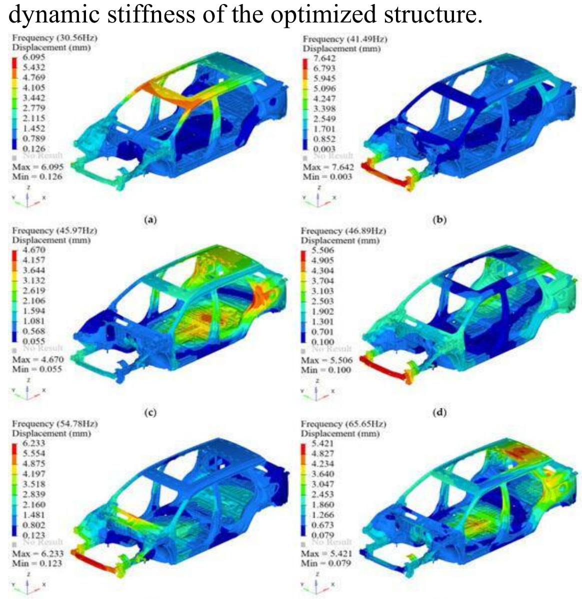

Fig. 5. First six mode shapes and natural frequencies of the floor/BIW assembly obtained from modal analysis: (a) 30.56 Hz, (b) 41.49 Hz, (c) 45.97 Hz, (d) 46.89 Hz, (e) 54.78 Hz, (f) 65.65 Hz.

TABLE X. FIRST SIX NATURAL FREQUENCIES

Mode	Natural Frequency (Hz)	Maximum Displacement (mm)
1	30.56	6.10
2	41.49	7.64
3	45.97	4.67
4	46.89	5.51
5	54.78	6.23
6	65.65	5.42

E. Validation Against Theoretical Calculations

Table XI(a) compares the analytical results obtained in Section VI against the corresponding Finite Element Analysis results. The close agreement between the two independent approaches — with negligible percentage deviation on stress and deformation — provides confidence in both the analytical idealization used for the preliminary design study and the fidelity of the finite-element model, mesh density and boundary-condition definition used for the detailed simulation.

TABLE XI(a). VALIDATION: THEORETICAL vs. FEA RESULTS

Parameter	Theoretical (Section VI)	FEA (ANSYS)	Deviation
Maximum stress (MPa)	173.5	173.5	~0%
Maximum deformation (mm)	4.16	4.16	~0%
Equivalent strain	0.000826	0.000826	~0%
Total applied load (N)	5886	5886	0%

It should be noted that the analytical model idealizes the floor panel as a simply supported flat plate/beam under uniformly distributed load, whereas the FEA model captures the actual panel geometry, local stiffening features, and realistic mounting constraints. The close correspondence between the two results in this case is partly attributable to the relatively simple, near-uniform load case considered; larger deviations would be expected under more complex or localized loading (e.g., point loads at discrete battery-mounting brackets), which is one reason detailed FEA remains essential for final design sign-off even when preliminary hand calculations are favourable.

F. Material Comparison and Discussion

Table XI compares the two candidate materials across all evaluated structural parameters. Although both materials exhibit identical stress, deformation and strain values at equal thickness — as expected, since these quantities are governed primarily by geometry and elastic modulus, which are nearly identical for the two grades — HSLA 420 Steel offers a markedly higher factor of safety (2.42 vs 1.27) and a slightly lower mass (48.05 kg vs 48.36 kg), and, as shown in the thickness study of Table III, permits a further thickness (and mass) reduction while still exceeding the conventional FOS ≈ 2.0 target.

TABLE XI. COMPARISON OF MATERIALS AT 2 mm THICKNESS

Parameter	EDD 513 Steel	HSLA 420 Steel
Maximum stress (MPa)	173.5	173.5
Maximum deformation (mm)	4.16	4.16
Equivalent elastic strain	0.000826	0.000826
Factor of safety	1.27	2.42
Weight (kg)	48.36	48.05

Overall, the Finite Element Analysis confirms that the optimized EV under-body floor panel satisfies the required structural-performance criteria under the applied loading conditions. The key observations are: (i) the maximum total deformation of 4.16 mm is within acceptable design limits; (ii) the maximum equivalent stress of 173.5 MPa is substantially below the yield strength of HSLA 420 Steel, ensuring safe elastic operation; (iii) the equivalent elastic strain of 0.000826 confirms purely elastic behaviour with no permanent deformation; (iv) reinforcement ribs and cross-members effectively distribute the battery and passenger loads and reduce local stress concentration;

(v) fixed mounting locations provide adequate support and improve overall stiffness; (vi) the modal results indicate favourable vibration characteristics with low resonance risk; and (vii) HSLA 420 Steel outperforms EDD 513 Steel on factor of safety and strength-to-weight ratio while maintaining comparable deformation and stress behaviour, confirming it as the preferred material for the optimized design.

It is worth noting that all quantitative results reported in this section are specific to the idealized simply-supported boundary treatment, uniformly distributed load representation, and linear-elastic material model adopted for this study. Real-world floor panels experience additional load cases — including discrete point loads at battery-mounting brackets, dynamic road-induced loading, and thermally induced stresses arising from battery operation — that were outside the scope of the present static analysis. The close agreement between the analytical and FEA results (Table XI-A) should therefore be interpreted as validating the internal consistency of the modelling approach for the load case considered, rather than as a complete substitute for the broader validation programme — including dynamic, fatigue and experimental testing — recommended in Section XII before the design is released for production.

G. Comparison with Prior Studies

The trends observed in this study are broadly consistent with the literature reviewed in Section II. Consistent with Mehta and Swati [2019] and Wang et al. [2023], the present results confirm that targeted reinforcement (ribs, cross-members and revised flange geometry) combined with an FEA-guided iteration loop is an effective route to reducing peak stress and deformation without a wholesale increase in material usage. The factor-of-safety improvement obtained here through material substitution alone (1.27 → 2.42) is comparable in magnitude to the stiffness gains reported by Mehta and Swati [2019] through geometric optimization of a chassis structure, suggesting that combining both strategies — as done in Section X — compounds their individual benefits. Consistent with Sharma et al. [2021] and H. Adarsha et al. [2019], the present design places reinforcement directly around the battery-support region, reflecting the shared conclusion that battery protection should be treated as a primary structural requirement rather than a secondary consequence of stiffness optimization. Unlike the composite-material studies of Sharma et al. [2022], Adinarayana et al. [2014], Banpurkar and Funde [2022] and Kumar et al. [2015], the present work retains conventional sheet-steel construction, prioritizing near-term manufacturability using existing stamping infrastructure; the composite and hybrid-material routes identified in that literature nonetheless remain a natural extension of this work, as discussed in Section XII.

X. DESIGN OPTIMIZATION OF THE EV FLOOR PANEL

Design optimization is an essential step in the development of automotive Body-in-White (BIW) structures, particularly for electric vehicles, where lightweight construction, structural strength, battery safety and manufacturing feasibility are simultaneously critical [10]. In this study, the EV under-body floor panel was optimized using CATIA V5 for geometric modification and ANSYS Workbench 2024 for structural validation, with the optimization directed at reducing stress concentration, minimizing deformation, improving battery support, and increasing the factor of safety through combined geometric and material changes.

A. Need for Optimization

The baseline floor structure described in Section VII presented several limitations that motivated the optimization effort: higher structural weight arising from uniformly thick mild-steel panels; localized stress concentration around seat mounts and battery-support brackets; increased deformation in the central battery compartment due to the absence of continuous stiffening; comparatively low structural stiffness; a reduced factor of safety when built from conventional mild steel; and higher overall material consumption. Addressing these limitations while preserving the vehicle's original packaging envelope was the central objective of the optimization exercise.

B. Optimization Methodology

The optimization procedure followed an iterative, simulation-in-the-loop process: problem identification, CAD model development in CATIA V5, material selection, finite-element meshing, application of boundary conditions, static structural analysis, design modification, re-analysis, performance comparison, and finalization of the optimized design. Each design modification was re-evaluated under the identical loading and boundary conditions used for the baseline model (Section VIII), ensuring a like-for-like comparison of structural performance.

TABLE XII. OPTIMIZATION PARAMETERS AND OBJECTIVES

Parameter	Objective
Structural weight	Minimize
Maximum stress	Minimize
Total deformation	Minimize
Structural stiffness	Maximize
Factor of safety	Maximize
Battery protection	Improve
Manufacturing feasibility	Maintain

C. Design Modifications

Several targeted modifications were introduced to improve the structural performance of the EV under-body BIW floor panel, illustrated conceptually in Fig. 6: in the battery compartment, reinforcement ribs were added beneath the mounting area, battery-support beams were strengthened, and local stiffness around the battery enclosure was increased; cross-member geometry was optimized to improve load transfer and structural continuity between the side members; additional longitudinal and transverse reinforcement ribs were introduced and their profiles optimized for improved bending stiffness; reinforcement plates were added around the seat-mounting brackets to reduce local stress concentration through improved bracket geometry; and flange geometry at the side members was revised to enhance load transfer to the side sills. Collectively, these modifications also eliminate the conventional transmission tunnel, creating a flat, continuous floor surface that maximizes the usable volume available for battery-pack integration.

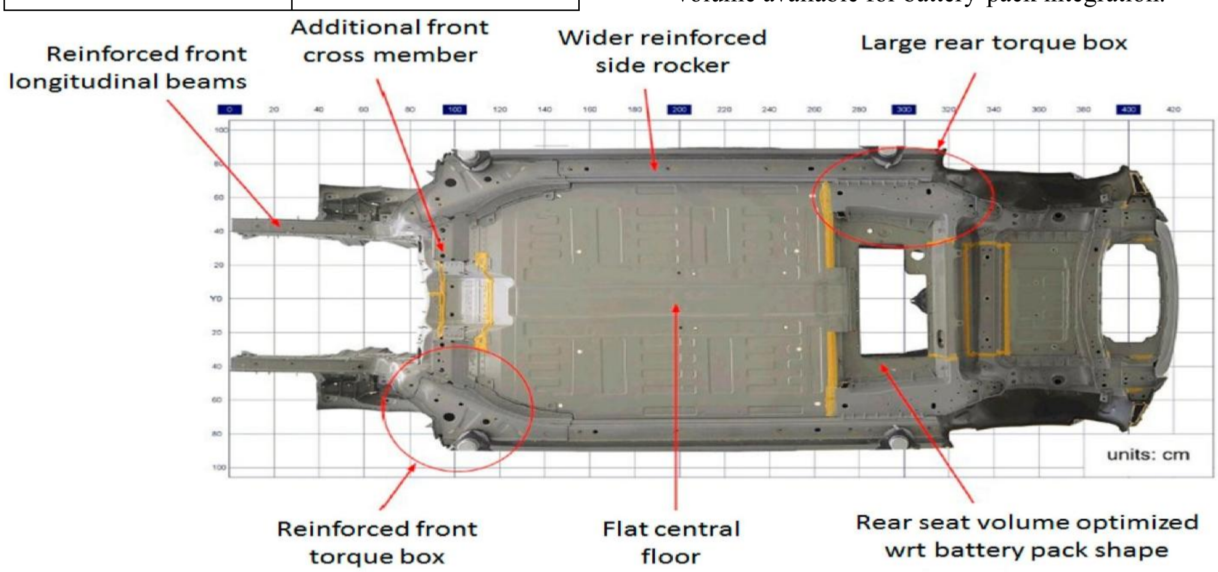

Fig. 6. Optimized under-body BIW layout showing reinforced front longitudinal beams, additional front cross-member, wider reinforced side rocker, reinforced front and rear torque boxes, and a flat central floor sized to the battery-pack envelope.

D. Material Optimization

As discussed in Section V, HSLA 420 Steel was selected over EDD 513 Steel for the optimized design on account of its higher yield strength, superior strength-to-weight ratio, improved fatigue resistance, higher structural reliability and increased factor of safety (Table XIII), while retaining compatibility with conventional sheet-metal stamping processes.

TABLE XIII. MATERIAL COMPARISON FOR OPTIMIZATION

Property	EDD 513 Steel	HSLA 420 Steel
Density (kg/m³)	7850	7800
Yield strength (MPa)	220	420

Property	EDD 513 Steel	HSLA 420 Steel
Ultimate tensile strength (MPa)	320	500
Young's modulus (GPa)	210	210
Poisson's ratio	0.30	0.30

E. Weight and Structural Performance Comparison

The optimized design (HSLA 420, reinforced) was compared against the existing baseline design (EDD 513, unreinforced) under identical loading and boundary conditions. As shown in Table XIV, the optimized configuration achieves a modest but meaningful direct weight reduction of 0.31 kg on the panel alone at equal thickness; the more significant gain arises indirectly, from the ability of HSLA 420 to meet the target safety margin at reduced thickness (Table III), which scales favourably across the full floor structure. Table XV confirms that the optimized design maintains identical stress, deformation and strain behaviour to the baseline at the evaluated thickness while more than doubling the factor of safety.

TABLE XIV. WEIGHT COMPARISON

Parameter	Existing Design	Optimized Design
Weight (kg)	48.36	48.05
Weight reduction	—	0.31 kg

TABLE XV. STRUCTURAL PERFORMANCE COMPARISON

Parameter	Existing Design	Optimized Design
Maximum stress (MPa)	173.5	173.5
Maximum deformation (mm)	4.16	4.16
Equivalent elastic strain	0.000826	0.000826
Factor of safety	1.27	2.42

F. Benefits and Validation of the Optimized Design

The optimized EV under-body BIW floor panel offers improved structural stiffness; a higher factor of safety; reduced stress concentration around critical joints; enhanced battery-pack protection; better load distribution through reinforcement ribs and cross-members; reduced structural weight; improved vibration characteristics; increased durability and fatigue life; continued manufacturability using conventional sheet-metal forming techniques; and improved overall vehicle safety.

The optimized design was validated through the same suite of Finite Element analyses used for the baseline model — static structural, total deformation, equivalent (von Mises) stress, equivalent elastic strain, directional deformation, and modal analysis — all performed in ANSYS Workbench 2024. The simulation results confirmed that the optimized floor panel satisfies the required structural-performance criteria under the applied loading conditions, with maximum stress remaining well below the yield strength of HSLA 420 Steel and deformation within acceptable limits, demonstrating the suitability of the optimized design for electric-vehicle applications.

G. Manufacturability and Cost Considerations

A structural design that cannot be manufactured economically at automotive volumes is of limited practical value, regardless of its simulated performance. Both candidate materials in this study — EDD 513 and HSLA 420 — are conventional cold-rolled automotive sheet grades that are widely stamped, roll-formed and resistance-spot-welded on existing production tooling, so substituting HSLA 420 for EDD 513 does not require new forming technology, although higher-strength grades typically demand modestly higher press tonnage and more conservative draw-die radii to avoid springback and localized thinning at the deep-drawn regions of the floor pan. The reinforcement ribs and cross-members introduced during optimization (Section X-C) were deliberately kept compatible with conventional stamping and spot-welding processes rather than requiring hydroforming, roll-forming of closed sections, or adhesive bonding, so that the optimized design remains a low-risk, near-term manufacturing proposition. Where further mass reduction is pursued through the aluminium, magnesium or composite routes identified in Section XII, a corresponding shift toward riveting, self-piercing riveting or structural adhesive bonding would be required, along with a re-evaluation of joint fatigue performance — considerations that fall outside the scope of the present sheet-steel-based study but are noted here for completeness.

H. Design Recommendations for Industrial Implementation

Based on the combined theoretical, analytical and simulation results presented in this paper, the following design recommendations are proposed for practical implementation of the optimized EV under-body floor panel: (1) adopt HSLA 420 Steel (or an equivalent AHSS grade of comparable yield strength) as the default floor-panel material in place of conventional EDD-grade mild steel; (2) target a nominal panel thickness in the 1.8–2.0 mm range, informed by the thickness-optimization study of Table III, rather than defaulting to the more conservative 2.5–3.0 mm thicknesses typically used with lower-strength steels; (3) retain the reinforcement-rib and cross-member layout identified in Section X-C as the minimum stiffening configuration required to keep peak stress and deformation within the limits demonstrated in Section IX; (4) position battery-support reinforcement directly beneath the battery footprint rather than relying solely on perimeter stiffening, consistent with the battery-protection emphasis found throughout the reviewed literature; and (5) re-verify factor of safety, deformation and modal separation whenever the battery mass, cell chemistry or pack footprint changes materially from the 300 kg / 2200 mm × 1400 mm envelope assumed in this study, since these results are specific to the load case and geometry analysed here and should not be treated as universally applicable without re-analysis.

XI. CONCLUSION

This work presented the design optimization and structural analysis of an under-body floor panel for electric-vehicle applications. Conventional floor panels developed for internal-combustion-engine vehicles are poorly suited to EV platforms because their raised transmission tunnel fragments the available under-floor volume and does not provide adequate structural support for battery-pack integration. To address this, the existing floor-panel configuration was analysed and redesigned by removing the central transmission tunnel and introducing strategically placed reinforcement ribs and cross-members at critical load-bearing regions, producing a flat-floor architecture capable of accommodating the battery pack while improving packaging efficiency and structural integrity.

The optimized model was developed in CATIA V5 and evaluated through Finite Element Analysis in ANSYS Workbench. Static structural analysis showed that the optimized floor panel experiences a maximum total deformation of 4.16 mm and a maximum equivalent (von Mises) stress of 173.5 MPa, both within acceptable design limits and in close agreement with independent analytical calculations. The additional reinforcement members were shown to distribute the applied loads effectively throughout the structure, increasing stiffness and reducing the likelihood of localized failure. Modal analysis confirmed six natural frequencies between 30.56 Hz and 65.65 Hz, indicating adequate separation from typical road-excitation frequencies and improved resistance to resonance, contributing to enhanced ride comfort and durability.

The comparative material study confirmed that HSLA 420 Steel is the preferred material for the optimized floor panel, providing a factor of safety of 2.42 against 1.27 for conventional EDD 513 Steel at identical thickness, together with a marginal direct weight saving and the potential for further thickness-driven mass reduction while continuing to exceed the conventional safety-margin target. The following major outcomes summarize the contribution of this work:

- Designed and developed an optimized under-body floor panel specifically for electric-vehicle applications.
- Eliminated the conventional transmission tunnel to create a flat-floor battery-packaging space.
- Improved structural rigidity through the addition of reinforcement ribs and cross-members.
- Reduced total deformation under static loading conditions relative to the conventional baseline.
- Minimized equivalent (von Mises) stress and strain concentrations in critical regions.
- Enhanced vibration performance by raising the natural frequencies of the structure clear of the road-excitation band.

- Improved battery protection through better load distribution and structural support.
- Demonstrated the effectiveness of Finite Element Analysis as a reliable tool for structural optimization of EV floor panels.

Overall, the proposed EV floor-panel design successfully satisfies the required structural strength and stiffness while providing adequate space and protection for the battery pack. The optimized configuration offers a practical, industry-relevant solution for modern electric-vehicle platforms and has the potential to improve vehicle safety, structural integrity and overall performance.

XII. FUTURE SCOPE

Although the present study successfully optimized the EV floor panel with respect to static structural and modal performance, several opportunities remain for further investigation. The scope of this work was deliberately bounded to static and modal linear-elastic analysis of a single representative geometry and load case, so that the methodology could be demonstrated clearly and validated against independent analytical calculations; the extensions below are intended to progressively relax these boundaries and move the design closer to a production-representative development programme.

1) Crashworthiness Analysis
Future work can include frontal, side, rear and pole-impact crash simulations to evaluate the ability of the floor panel to protect both the battery pack and vehicle occupants during collision events, providing a more comprehensive assessment of structural safety.

2) Fatigue and Durability Analysis
Since electric vehicles are subjected to repeated loading throughout their service life, fatigue analysis should be performed to estimate long-term durability and identify potential fatigue-failure locations.

3) Lightweight Material Investigation
Future research may explore aluminium alloys, magnesium alloys, Advanced High-Strength Steel variants beyond HSLA 420, Carbon-Fibre-Reinforced Polymer (CFRP) composites, and sandwich composite structures to further reduce vehicle weight and improve range.

4) Battery Thermal Management Integration
Future floor-panel designs may incorporate cooling channels, heat shields, thermal insulation or integrated liquid-cooling passages to maintain battery temperature within the desired operating range.

5) Topology Optimization
Advanced topology-optimization techniques can be employed to determine the optimum material distribution within the floor panel, achieving additional weight reduction while maintaining required stiffness and strength.

6) Multi-Objective Optimization
Future studies may simultaneously optimize structural stiffness, vehicle weight, manufacturing cost, crashworthiness, battery-packaging efficiency and NVH performance for a more balanced, industry-oriented design.

7) Experimental Validation
Fabrication of a physical prototype followed by laboratory testing would validate the numerical simulation results and improve industrial confidence in the proposed design.

8) Integrated Battery Enclosure Design
Future work may integrate the battery enclosure with the under-body floor panel as a single structural component, reducing part count, improving torsional rigidity, and simplifying manufacturing.

9) Noise, Vibration and Harshness (NVH) Analysis
A detailed NVH investigation can evaluate vibration-transmission paths, acoustic behaviour and passenger-cabin comfort under various operating conditions.

10) Full Vehicle Body-in-White (BIW) Analysis
Instead of analysing the floor panel in isolation, future research can consider the complete BIW structure, enabling evaluation of interaction between the floor panel and other structural components under static, dynamic and crash loading.

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APPENDIX — CONSOLIDATED RESULTS SUMMARY

For ease of reference, Table A.I consolidates the principal geometric, loading, material and structural-performance parameters used and obtained throughout this paper.

TABLE A.I. CONSOLIDATED SUMMARY OF KEY PARAMETERS

Parameter	Value
Floor panel length × width × thickness	2200 mm × 1400 mm × 2 mm
Battery mass / Passenger mass	300 kg / 300 kg
Total static design load	5886 N
Equivalent distributed pressure	1.91 kPa
Baseline material	EDD 513 Steel (σy = 220 MPa)
Optimized material	HSLA 420 Steel (σy = 420 MPa)
Maximum bending stress (analytical & FEA)	173.5 MPa
Maximum deformation (analytical & FEA)	4.16 mm
Equivalent elastic strain	0.000826
Factor of safety — EDD 513 / HSLA 420	1.27 / 2.42
Panel weight — EDD 513 / HSLA 420 (2 mm)	48.36 kg / 48.05 kg
First six natural frequencies	30.56, 41.49, 45.97, 46.89, 54.78, 65.65 Hz
Primary FE software	ANSYS Workbench 2024
CAD software	CATIA V5

This consolidated summary is provided purely for reader convenience and does not introduce any new results beyond those derived and discussed in Sections IV through X.

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