

Development and Construction of a Composite Battery Container for Formula Student Vehicle

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DECLARATION OF AUTHORSHIP

Has undersigned, hereby it his declared that this work entitled “Development and Construction of a Composite Battery Container for Formula Student Vehicle” is the original work and that it has not previously in its entirety or in part been submitted at any university or higher education institution for the award of any degree, diploma, or other qualifications. It is also hereby declared that to the best of the knowledge, this work contains no material previously published or written by another person, except where due reference, acknowledgement, and citation is made.

Leiria, June 2026

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RESUMO

Este trabalho apresenta o desenvolvimento e construção de um contentor para os módulos de alta tensão (TSAC - Tractive System Accumulator Container) em material compósito para o protótipo elétrico da *Formula Student* da equipa LART (Leiria Academic Racing Team), com o objetivo de redução de massa em 40% face ao design anterior, cumprindo os regulamentos da competição, os requisitos de integridade estrutural e as restrições de manutenção.

O processo de design seguiu uma metodologia estruturada em três fases: dimensionamento analítico com base na teoria de placas e na Teoria Clássica dos Laminados, validação numérica por análise de elementos finitos, e a caracterização experimental do sistema de materiais selecionado. Foi adotado um compósito de aramida com resina Sika Biresin CR132FR, pela combinação de desempenho estrutural, isolamento elétrico, resistência ao fogo e acessibilidade de fabrico. Um laminado uniforme $[0/90]_{16}$ foi adotado em todos os painéis para simplificar a produção.

O dimensionamento analítico identificou a tensão de corte da cola como o critério dominante para a maioria dos painéis. Os modelos de elementos finitos demonstraram que a formulação de deflexão pequena constitui uma fase conservadora, e revelaram que a formulação de deflexão grande é não conservadora para tensões em condições ortotrópicas. Os modelos em condições reais destacaram as tensões de corte da cola como a limitação crítica da abordagem analítica sob concentrações localizadas, sendo isto resolvido pelo laminado uniforme adotado.

A validação experimental confirmou as propriedades do material através de ensaios de tração, densidade, flexão em três pontos e corte por perímetro. O laminado $[0/90]_{16}$ demonstrou uma resistência à flexão de 281 MPa e uma resistência ao corte de 169 MPa - três vezes superior ao valor analítico assumido. Os ensaios de flamabilidade confirmaram a conformidade UL94 V-0.

O TSAC construído atingiu uma massa total de 9,15 kg, representando um excesso de 0,69 kg face ao objetivo de 8,46 kg. Este resultado vai de encontro às decisões tomadas durante a fase de design para garantir o cumprimento do regulamento, a fiabilidade estrutural e a facilidade de produção, em particular a adoção do laminado uniforme $[0/90]_{16}$.

Palavras-Chave: materiais compósitos, aramida, contentor de bateria, Fórmula Student, Teoria de Laminado Clássico, análise de elementos finitos, design estrutural.

ABSTRACT

This work presents the development and construction of a composite Tractive System Accumulator Container (TSAC) for the LART Formula Student electric race car, targeting a 40% mass reduction relative to the previous metallic enclosure while satisfying competition regulations, structural integrity requirements, and serviceability constraints.

The design followed a structured three-stage methodology: analytical dimensioning based on classical plate theory and Classical Lamination Theory, numerical validation through finite element analysis, and experimental material characterisation. A wet lay-up aramid composite with Sika Biresin CR132FR resin was selected for its combination of structural performance, electrical insulation, fire retardancy, and manufacturing accessibility, with a uniform $[0/90]_{16}$ layup adopted across all panels.

Analytical dimensioning identified adhesive shear as the governing design criterion for most panels. Finite element models confirmed the conservatism of small deflection theory while revealing that the large deflection formulation is non-conservative for stress in orthotropic laminates. Real condition models further highlighted adhesive shear under localised loading as a critical limitation of the analytical approach, resolved by the uniform layup. Full assembly simulation confirmed structural adequacy under X and Z- loading.

Experimental validation confirmed the material properties through tensile, density, three-point bending, and perimeter shear tests. The $[0/90]_{16}$ laminate demonstrated a flexural strength of 281 MPa and a shear strength of 169 MPa providing substantial safety margins beyond the regulatory minimum. Flammability tests confirmed UL94 V-0 compliance for the selected material system.

The completed TSAC achieved a measured total mass of 9.15 kg, representing an excess of 0.69 kg relative to the 8.46 kg target. This result reflects the design decisions taken to ensure regulatory compliance, structural reliability, and manufacturing feasibility - in particular the adoption of the uniform $[0/90]_{16}$ layup. The project additionally established in-house composite manufacturing capabilities within the team, laying the groundwork for future application of composite materials to other structural components.

Keywords: composite materials, aramid fibre, battery container, Formula Student, Classical Lamination Theory, finite element analysis, structural design.

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INTRODUCTION

1.1 Background

The evolution of electric mobility represents one of the most significant shifts in modern transportation - a transition driven not only by environmental necessity, but by decades of technological ambition. It was not until the close of the 20th century that electrification truly emerged as a credible alternative to the internal combustion engine, propelled by growing environmental awareness and rapid advances in battery and power electronics technology. Motorsport, long regarded as the ultimate proving ground for automotive innovation, has been no exception to this shift. Competitions such as Formula E and MotoE have brought fully electric powertrains to the forefront of high-performance racing [1], actively driving breakthroughs in battery energy density, thermal management, and high-performance composite materials - innovations that ultimately find their way into road vehicles.

Central to the performance of any electric vehicle is the transmission system responsible for delivering power from the motor to the wheels. Unlike internal combustion engines, electric motors produce high torque from rest and sustain it across a wide speed range, fundamentally changing the role of the drivetrain and opening new possibilities in transmission and shifting system design. The use of advanced composite materials has further expanded these possibilities, enabling lighter, stiffer, and more efficient structural components that meet the demanding requirements of both motorsport and road applications. It is within this context that the present project is developed, contributing to the advancement of electric drivetrain technology through the work of a dedicated engineering team.

1.2 Formula Student

Formula Student is an international engineering design competition in which university teams design, build, and race a single-seat formula-style car. Beyond the technical challenge, it bridges academic study and professional engineering practice by exposing students to the full product development cycle – from concept and CAD

(Computer-Aided Design) modelling through FEA (Finite Element Analysis), CFD (Computational Fluid Dynamics), and manufacturing, to competition and performance evaluation. The project also develops soft skills such as leadership, communication, and project management, which are essential in an engineering environment [2].



Figure 1.1: Formula Student teams at Hockenheimring, 2024 [3]

The Competition

Teams can compete with Internal Combustion Vehicles (CV), Electric Vehicles (EV), or Driverless Vehicles (DV), across two main segments: static and dynamic events.

Static events assess the team's engineering and business decision-making through three judged events: the Business Plan Presentation (75 points), the Cost and Manufacturing Event (100 points), and the Engineering Design Event (150 points) [4].

Technical reviews are mandatory before any vehicle is cleared for dynamic competition. Each vehicle must pass six system-specific inspections - mechanical, electrical, accumulator, rain test, tilt test, and brake test - earning a sticker for each, all of which are required to compete in subsequent events.

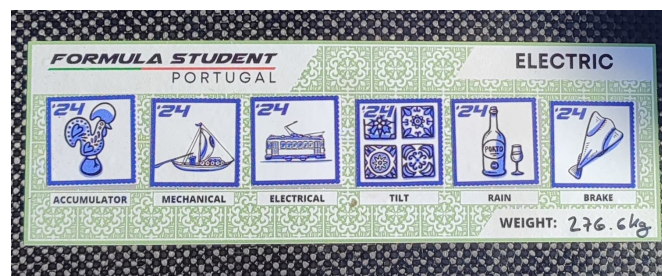


Figure 1.2: Stickers of LART Team in FSPortugal Competition 2024

Dynamic events push the vehicle and driver to their limits across four performance challenges: Acceleration (50 points), Skidpad (50 points), Autocross (100 points),

and Endurance (250 points) - a 22 km race testing reliability and energy management. An additional Efficiency score (75 points) rewards vehicles that perform well while minimising energy consumption.

Meet the Team

The Leiria Academic Racing Team (LART), formerly known as FSIPLeiria, has represented the Polytechnic Institute of Leiria in Formula Student competitions since 2006. Early highlights include a 1st place overall at Silverstone with the T-06, and the T-14 in 2014 - one of the first cars in Europe to implement a fly-by-wire system. In 2022, the T-22 integrated a turbocharger and biofuel, with most components developed in-house.

Transitioning into electric mobility, LART has since embarked on its most ambitious project: an electric race car with autonomous driving capabilities. In 2023, the team secured 1st place in Class 2 at the inaugural Formula Student Portugal competition. The following year brought increased complexity, with a focus on high-voltage systems, improved vehicle dynamics, and the integration of carbon fibre components - laying the groundwork for full autonomous integration in future iterations.

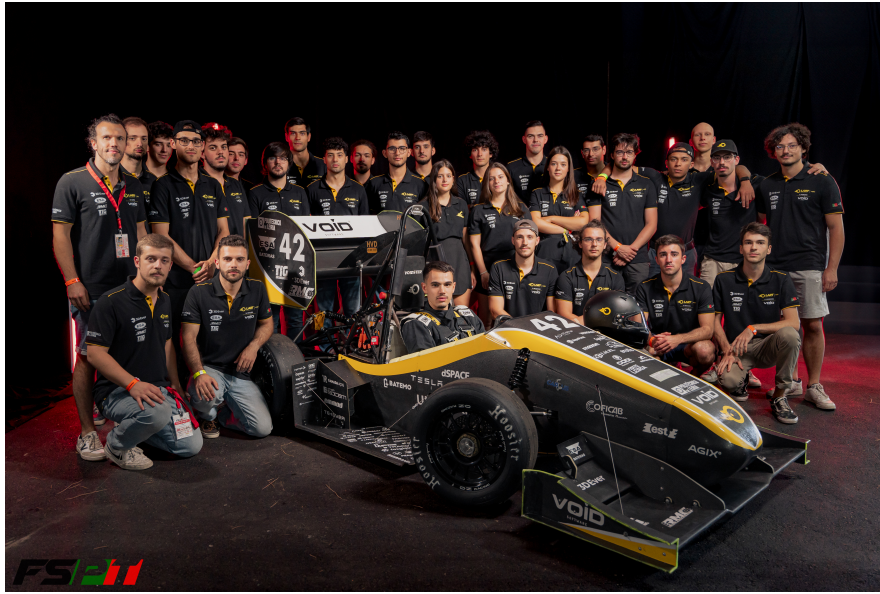


Figure 1.3: LART Team at the 2024 Formula Student Portugal event in Castelo Branco, Portugal [5]

1.3 Goals

The aim of this work is to improve the performance and reliability of the accumulator system through a purely structural approach, addressing the casing design and material selection while treating stiffness, ease of maintenance, and vehicle integration as design constraints. The adoption of composite materials is motivated by their presence in state-of-the-art Formula Student battery systems, their potential for significant weight reduction, and the opportunity to establish in-house composite manufactur-

ing capabilities that can support future development of other structural components. Detailed design targets and specifications are presented in Chapter 3.

The main objectives are:

- Design a battery pack casing that integrates mounting points and complies with both competition regulations and team requirements;
- Select a composite material configuration based on the state of the art and applicable rules;
- Develop an analytical procedure to study and characterise composite wall behaviour under structural loading;
- Build FEM models that replicate the experimental test conditions, enabling direct comparison between numerical and analytical results;
- Validate the material properties and the chosen layup through experimental testing, confirming the predictive accuracy of the models.

Documentation of the full design and validation process is also a key deliverable, supporting future development by the team.

1.4 Thesis Structure

This thesis is organised into eight chapters: Chapter 1 introduces the context of electric mobility and motorsport, presents the LART team and the Formula Student competition, and defines the goals and objectives of the work. Chapter 2 reviews the relevant background literature, covering EV battery pack technology, composite container materials, plate theory, Classical Lamination Theory, failure criteria, bolted joint design, and the experimental methods adopted for material characterisation. Chapter 3 defines the design goals and constraints of the TSAC, including mass targets, load cases, functional requirements, concept selection, the material and adhesive choices. Chapter 4 presents the analytical dimensioning of all TSAC panels, the L-profile and mounting system design, and the Formula Student rules compliance verification. Chapter 5 describes the finite element models developed to validate the analytical approach, covering tensile and three-point bending simulations, individual panel models under analytical and real conditions, and the full battery pack assembly simulation. Chapter 6 presents the experimental validation programme, including tensile, density, flammability, three-point bending, perimeter shear, and adhesive tests, with comparison against analytical and numerical predictions. Chapter 7 documents the manufacturing and assembly process, covering laminate production, waterjet cutting, casing assembly, lid manufacturing, and components fabrication. Chapter 8 draws the overall conclusions of the work and identifies directions for future development.

LITERATURE REVIEW

2.1 EV Battery Packs

2.1.1 Cell Types and Formats

The electrochemical chemistry and physical format of battery cells directly influence the design requirements of the surrounding structure, thermal management, and safety systems.

Among the available chemistries, lithium-ion cells are dominant in EV and motor-sport applications due to their high energy and power density, efficiency, and long life cycle [6]. Their main challenges include sensitivity to fault conditions - such as overcharge, over-discharge, and thermal abuse - which makes robust structural containment and electrical isolation critical design requirements.

Battery cells are also available in three main formats, each with structural implications [7, 8, 9]:

- **Cylindrical** - high mechanical stability and efficient heat dissipation; widely adopted in motorsport and consumer EVs;
- **Prismatic** - rigid rectangular casing that maximises space efficiency, though more complex to integrate structurally;
- **Pouch** - lightweight and flexible, but require external mechanical support and are susceptible to swelling under thermal or electrical stress.

The choice of cell format has a direct impact on module layout, wall loading, and the structural requirements of the accumulator casing - all of which inform the design approach developed in this work.

2.1.2 Car Integration

Battery pack integration directly influences vehicle dynamics, structural design, and overall weight distribution. Positioning the pack close to the vehicle floor lowers the centre of gravity, improving stability and handling - a particularly relevant consideration in motorsport applications.

Integration architectures can be broadly categorised by how cells and modules relate to the vehicle structure [10]. The traditional Module-to-Chassis (MTC) approach offers modularity and ease of maintenance, but its individual module enclosures add weight and reduce space efficiency. More recent approaches address this: Cell-to-Pack (CTP) eliminates the module layer entirely, recovering 15–20% of space, while Cell-to-Chassis (CTC) goes further by incorporating the pack into the vehicle structure, achieving an additional 10% weight reduction [11, 12].

These trends highlight how integration strategy and material selection are increasingly interlinked - a tighter, more structural pack demands careful consideration of the casing's mechanical behaviour.

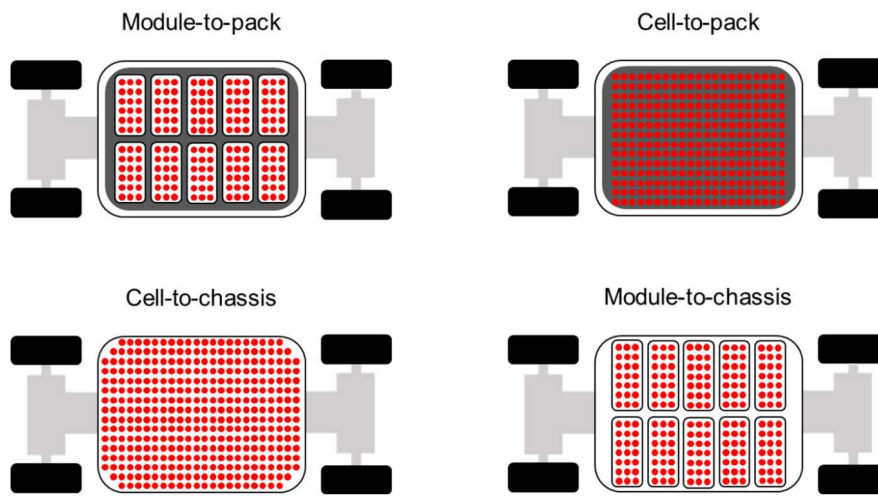


Figure 2.1: Conceptual view of the main EV battery pack design trends [10]

2.1.3 Container Materials

The battery enclosure is a strategic point for mass reduction, as the pack alone accounts for around 40% of a vehicle's total weight and up to 50% of its total cost [13]. Material selection therefore has a direct impact on efficiency, range, and performance.

Steel was historically the dominant choice due to its strength, malleability, and cost-effectiveness, but its high density has driven a shift toward lighter alternatives. Aluminium now accounts for approximately 80% of EV battery enclosures [13], offering a favourable strength-to-weight ratio and excellent thermal conductivity. At up to 50% lighter than steel [14], it significantly improves vehicle range - though at a higher production cost.

Composites and thermoplastics are emerging as the next step, with the potential to reduce enclosure weight by up to 30% [15] and total system weight by up to 40% [16]. Beyond weight savings, they offer design flexibility and improved sustainability, positioning them as a compelling alternative in high-performance applications - and the direction pursued in this work.

2.2 Theory of Plates

Plate theory provides an analytical framework for modelling flat structural elements whose thickness is small compared to their in-plane dimensions, with loads acting predominantly normal to the surface and resisted mainly through bending and twisting actions governed by boundary conditions. In this thesis, plate theory is applied to the battery pack walls, which can be idealized as thin flat plates subjected to transverse loads arising from module and peripheral masses during vehicle acceleration, providing a preliminary wall dimensioning prior to detailed numerical analysis.

2.2.1 Classification and Assumptions of Thin Plate Theory

The mechanical behaviour of plates depends primarily on the ratio between a characteristic in-plane dimension a and the thickness h , as well as on the magnitude of transverse deflections w . Thin plates are defined by $8 \sim 10 \leq a/h \leq 80 \sim 100$. Within this range, stiff thin plates ($w/h \leq 0.2$) carry loads predominantly through bending, and flexible thin plates ($w/h \geq 0.3$) exhibit coupled bending-membrane behaviour. Outside these limits, plates with smaller a/h ratio behave as membranes, while larger ratios correspond to thick plates where transverse shear deformation becomes significant [17].

For classical linear thin plate theory (Kirchhoff–Love theory), the following assumptions are adopted: [17]

- The plate is initially flat, homogeneous, and isotropic.
- The thickness is small compared to the in-plane dimensions.
- Transverse deflections are small relative to the thickness.
- The mid-surface remains unstrained during bending.
- Lines initially normal to the mid-surface remain straight and normal during deformation (Kirchhoff–Love hypothesis), implying negligible transverse shear deformation.

Regarding boundary conditions, the walls of the accumulator casing are not all identical in their support conditions. Depending on their location and attachment to the structure, they may be modelled as fully fixed, simply supported, or free along their edges. The appropriate boundary condition for each wall is addressed in the design chapter, where it directly determines the coefficients used in the analytical formulations.

2.2.2 Governing Equations of Thin Plate Bending

Relating strains to the curvature of the deflected mid-surface and applying Hooke's law for plane stress, illustrated by Fig. 2.2, the normal stresses induced by bending can be expressed in terms of the second derivatives of w . Integration of these stresses leads

to the definition of the plate bending or flexural rigidity, D , (eq. 2.1), which combines the effects of material stiffness and thickness into a single parameter.

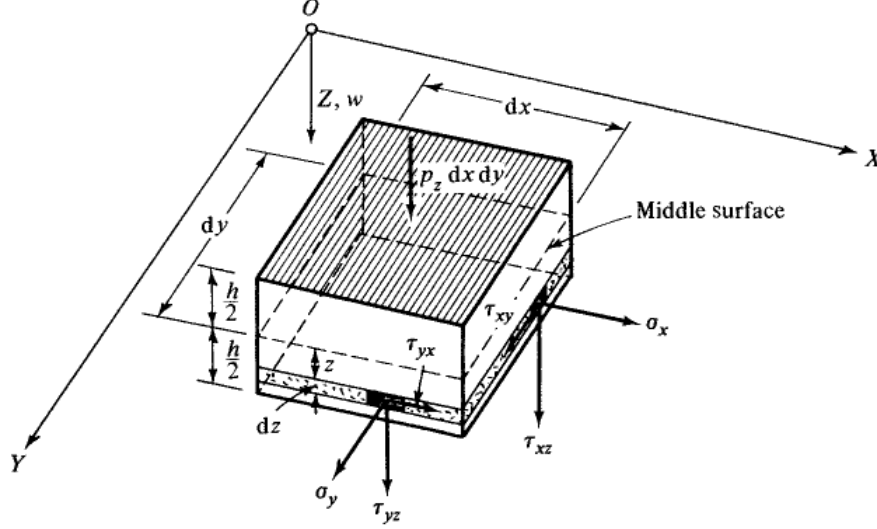


Figure 2.2: Stresses on a plate element [18]

$$D = \frac{Eh^3}{12(1-\nu^2)} \quad (2.1)$$

where E is Young's Modulus and ν stands for the Poisson's ratio of the material.

Enforcing equilibrium of bending moments and transverse shear forces on an infinitesimal plate element results in the governing differential equation of thin plate bending [18]:

$$D \left(\frac{\partial^4 w}{\partial x^4} + 2 \frac{\partial^4 w}{\partial x^2 \partial y^2} + \frac{\partial^4 w}{\partial y^4} \right) = p_z(x, y) \quad (2.2)$$

or, in compact form,

$$D \nabla^4 w(x, y) = p_z(x, y) \quad (2.3)$$

where $p_z(x, y)$ denotes the distributed transverse load acting on the plate.

By introducing a dimensionless coefficient C , dependent on the plate aspect ratio a/b , where a and b are the plate length and width, respectively, and boundary conditions BC , the maximum deflection can be expressed as:

$$w_{\max} = C(a/b, BC) \frac{qL^4}{D} \quad (2.4)$$

where q represents the force per unit area and L the characteristic length of the plate.

2.2.3 Stress Resultants and Bending Moments

Under Kirchhoff–Love thin plate assumptions, the in-plane stresses are related to strains by [17]:

$$\sigma_x = \frac{E}{1-\nu^2} (\varepsilon_x + \nu \varepsilon_y), \quad \sigma_y = \frac{E}{1-\nu^2} (\varepsilon_y + \nu \varepsilon_x), \quad \tau_{xy} = G \gamma_{xy}. \quad (2.5)$$

where σ_x, σ_y and τ_{xy} represents the stress components in X, Y and XY directions, ε_x and ε_y are the normal strain in X and Y directions, respectively, and γ_{xy} is the shear strain in XY direction. Additionally, G is the shear modulus of the plate material.

Because bending induces curvature of the mid-surface, the strains vary linearly through the thickness:

$$\varepsilon_x = -z k_{xx}, \quad \varepsilon_y = -z k_{yy}, \quad \gamma_{xy} = -2z k_{xy}. \quad (2.6)$$

where z is the thickness relating to a referential and k_{xx}, k_{yy} and k_{xy} are the curvatures of the deflected middle surface in X, Y and XY directions.

Instead of working with stresses directly, it is convenient to introduce stress resultants, defined as the through-thickness integrals of stresses. Performing this integration leads to the moment–curvature relations M_x in X direction and M_y in Y direction [17]:

$$\begin{aligned} M_x &= -D \left(\frac{\partial^2 w}{\partial x^2} + \nu \frac{\partial^2 w}{\partial y^2} \right) \\ M_y &= -D \left(\frac{\partial^2 w}{\partial y^2} + \nu \frac{\partial^2 w}{\partial x^2} \right) \end{aligned} \quad (2.7)$$

2.2.4 Small Deflection Formulation

For a uniformly distributed load acting on a rectangular plate, the analytical solution of equation 2.2 leads, after differentiation, to bending moments expressed as:

$$M_x = \frac{\beta_x(\lambda)}{6} q b^2, \quad M_y = \frac{\beta_y(\lambda)}{6} q b^2 \quad (2.8)$$

where:

- $\lambda = a/b$ is the aspect ratio,
- b is the shorter side of the plate,
- $q = \frac{F}{ab}$ is the distributed force over the plate,
- β_x, β_y are coefficients dependent on geometry and boundary conditions.

By combining the elementary bending stress relation with the moment–curvature expressions in equation 2.7, the surface bending stress is obtained:

$$\sigma_{\text{plate}} = \beta(\lambda) \frac{q b^2}{t^2} \quad (2.9)$$

with t representing the plate thickness.

This expression corresponds to the analytical formulation presented in Roark's Formulas for Stress and Strain [19], where tabulated coefficients condense the full dis-

placement solution. In addition, transverse shear forces Q_x and Q_y develop in the plate, obtained from the spatial variation of bending moments:

$$\begin{aligned} Q_x &= \frac{\partial M_x}{\partial x} + \frac{\partial M_{xy}}{\partial y} \\ Q_y &= \frac{\partial M_y}{\partial x} + \frac{\partial M_{xy}}{\partial x} \end{aligned} \quad (2.10)$$

For a rectangular plate under uniformly distributed load, the transverse shear force per unit length is expressed in coefficient form as:

$$Q = \gamma(\lambda)qb \quad (2.11)$$

where $\gamma(\lambda)$ is obtained from tabulated solutions and depends on the aspect ratio and boundary conditions.

2.2.5 Large Deflection Formulation

The small deflection formulation is valid only when transverse deflections remain small relative to the plate thickness. When this condition is not met, geometric nonlinearities arise: the deformation induces stretching of the middle surface, generating in-plane membrane forces that interact with bending moments and increase the effective stiffness of the plate - a phenomenon known as membrane stiffening [19].

In the large deflection regime, the plate behaviour is described by the nonlinear Föppl–von Kármán equations, and the load-deflection relationship is expressed in non-dimensional form as:

$$\frac{qb^4}{Et^4} = A \left(\frac{w}{t} \right) + B \left(\frac{w}{t} \right)^3 \quad (2.12)$$

where w is the maximum plate deflection, t is the plate thickness, and A and B are coefficients dependent on the aspect ratio and boundary conditions. The linear term represents the classical bending response, while the cubic term accounts for membrane stiffening. Since no closed-form solution exists, results are presented in tabulated form and the final wall dimensions are determined from this formulation whenever the small deflection assumption is violated.

2.3 Mechanics of Composite Materials

This section establishes the theoretical framework used to characterise the mechanical behaviour of composite laminates, complementing the plate theory presented previously. Starting from the constituent material properties, micromechanical models derive the effective elastic constants of a single lamina, which are then combined through Classical Lamination Theory to obtain the laminate stiffness - and in particular the equivalent flexural rigidity that bridges into the Roark formulations used for

wall dimensioning. Failure criteria are then presented to assess ply-level load-bearing capacity.

2.3.1 Classical Lamination Theory

A composite material is a combination of at least two constituent materials on a macroscopic scale, producing a result that inherits and often exceeds the properties of its components - including increased stiffness, greater wear resistance, reduced weight, and improved thermal and damping behaviour [20]. A single layer is referred to as a lamina, while a stack of laminae with varying fibre orientations forms a laminate, as illustrated in Fig. 2.3.

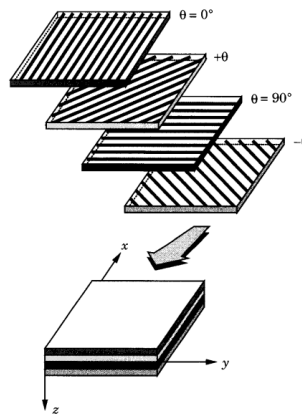


Figure 2.3: A laminate composed of laminae with different fibre orientations [21]

Predicting the mechanical behaviour of such composites requires determining the elastic properties of each lamina from its constituents - fibres and matrix. This is addressed through micromechanics, which adopts the following simplifying assumptions [21]:

1. Fibres are uniformly distributed throughout the matrix.
2. Perfect bonding exists between fibres and matrix.
3. The matrix is free of voids.
4. Applied forces are either parallel or normal to the fibre direction.
5. The lamina is initially stress-free.
6. Both fibres and matrix behave as linearly elastic materials.

2.3.2 Elastic Properties

The effective elastic properties of a composite lamina - longitudinal and transverse Young's moduli, major Poisson's ratio, and in-plane shear modulus - form the basis for the subsequent application of Classical Lamination Theory.

The longitudinal Young's modulus E_1 is estimated using the Modified Rule of Mix-

tures [22], which introduces a geometric correction factor κ for axial tension:

$$E_1 = \kappa E_f V_f + E_m V_m \quad (2.13)$$

with E_f and E_m representing the Modulus of Elasticity of the fibre and matrix, respectively, and V_f and V_m the fraction of fibre and matrix the laminate, respectively.

The transverse Young's modulus E_2 is obtained through a Modified Inverse Rule of Mixtures based on the Halpin-Tsai semi-empirical relation [22]:

$$E_2 = E_m \cdot \frac{(1 + V_f)E_f + V_m E_m}{V_m E_f + (1 + V_f)E_m} \quad (2.14)$$

The major Poisson's ratio is estimated directly from the Rule of Mixtures:

$$\nu_{12} = \nu_f V_f + \nu_m V_m \quad (2.15)$$

where ν_f and ν_m are the Poisson's coefficient of the fibre and matrix, respectively.

The in-plane shear modulus G_{12} is estimated using the Halpin-Tsai model [23], which extends the Rule of Mixtures by introducing ξ , accounting for fibre geometry and packing, and η , representing fibre-matrix bond conditions:

$$G_{12} = G_m \cdot \frac{1 + \xi \eta V_f}{1 - \eta V_f}, \quad \eta = \frac{\left(\frac{G_f}{G_m}\right) - 1}{\left(\frac{G_f}{G_m}\right) + \xi} \quad (2.16)$$

where G_f and G_m are the Shear Modulus of the fibre and matrix, respectively.

Finally, the composite density, ρ_c , is estimated from the volume fractions and mass densities of the constituents:

$$\rho_c = \rho_f V_f + \rho_m V_m \quad (2.17)$$

where ρ_f and ρ_m are the volumetric mass of the fibre and matrix, respectively.

2.3.3 Ultimate Strengths

Defining the ultimate strengths of the lamina is essential for assessing load-bearing capacity and predicting failure. The longitudinal tensile strength assumes all fibres share the applied load and failure occurs at the fibre ultimate strain [23]:

$$(\sigma_1^T)_{ult} = (\sigma_f)_{ult} \left[V_f + \frac{E_m}{E_f} (1 - V_f) \right] \quad (2.18)$$

The compressive strength is estimated through the micro-buckling model proposed by Xu and Reifsnider, based on a Beam-on-Elastic Foundation approach [22]:

$$\sigma_1^C = G_m \left[V_f + \frac{E_m}{E_{f1}} (1 - V_f) \right] \times \left\{ 2(1 + \nu_m) \sqrt{\frac{\pi \sqrt{\pi} \eta r_f}{3 \left(\frac{E_m}{E_{f1}} \right) \left(V_f \frac{E_m}{E_{f1}} + 1 - V_f \right) (1 + V_f \nu_f + \nu_m (1 - V_f))}} + 1 - \xi - \frac{\sin \pi \xi}{2\pi} \right\} \quad (2.19)$$

Since fibres contribute little to shear resistance, the in-plane shear strength of the laminate is taken as the matrix shear strength.

2.3.4 Constitutive Matrices of Laminates

The mechanical behaviour of a lamina is described through the generalised Hooke's law, $\{\sigma\} = [C]\{\epsilon\}$, where $[C]$ is the 6×6 stiffness matrix. Under plane stress and on-axis conditions, this reduces to the stress-strain relationship [24]:

$$\begin{bmatrix} \sigma_x \\ \sigma_y \\ \tau_{xy} \end{bmatrix} = \begin{bmatrix} Q_{11} & Q_{12} & 0 \\ Q_{12} & Q_{22} & 0 \\ 0 & 0 & Q_{66} \end{bmatrix} \begin{bmatrix} \epsilon_x \\ \epsilon_y \\ \gamma_{xy} \end{bmatrix} \quad (2.20)$$

where the stiffness coefficients Q_{ij} are derived from the lamina elastic properties [24]:

$$Q_{11} = \frac{E_1}{1 - \nu_{12}\nu_{21}}, \quad Q_{22} = \frac{E_2}{1 - \nu_{12}\nu_{21}}, \quad Q_{12} = \frac{\nu_{12}E_2}{1 - \nu_{12}\nu_{21}}, \quad Q_{66} = G_{12} \quad (2.21)$$

When a lamina is rotated by an angle θ relative to the global axes, the stiffness coefficients transform to $\bar{Q}_{ij}$ following [24]:

$$\bar{Q}_{11} = Q_{11} \cos^4 \theta + Q_{22} \sin^4 \theta + 2(Q_{12} + 2Q_{66}) \sin^2 \theta \cos^2 \theta \quad (2.22)$$

$$\bar{Q}_{12} = (Q_{11} + Q_{22} - 4Q_{66}) \sin^2 \theta \cos^2 \theta + Q_{12} (\cos^4 \theta + \sin^4 \theta) \quad (2.23)$$

$$\bar{Q}_{22} = Q_{11} \sin^4 \theta + Q_{22} \cos^4 \theta + 2(Q_{12} + 2Q_{66}) \sin^2 \theta \cos^2 \theta \quad (2.24)$$

$$\bar{Q}_{16} = (Q_{11} - Q_{12} - 2Q_{66}) \cos^3 \theta \sin \theta - (Q_{22} - Q_{12} - 2Q_{66}) \cos \theta \sin^3 \theta \quad (2.25)$$

$$\bar{Q}_{26} = (Q_{11} - Q_{12} - 2Q_{66}) \cos \theta \sin^3 \theta - (Q_{22} - Q_{12} - 2Q_{66}) \cos^3 \theta \sin \theta \quad (2.26)$$

$$\bar{Q}_{66} = (Q_{11} + Q_{22} - 2Q_{12} - 2Q_{66}) \sin^2 \theta \cos^2 \theta + Q_{66} (\sin^4 \theta + \cos^4 \theta) \quad (2.27)$$

The stiffness contributions of all laminae are then assembled into the laminate **ABD** matrix, which relates resultant in-plane forces $\{N\}$ and bending moments $\{M\}$ to mid-plane strains $\{\epsilon\}$ and curvatures $\{\kappa\}$ [24]:

$$\begin{Bmatrix} N \\ M \end{Bmatrix} = \begin{bmatrix} [A] & [B] \\ [B] & [D] \end{bmatrix} \begin{Bmatrix} \epsilon \\ \kappa \end{Bmatrix} \quad (2.28)$$

where the sub matrices are computed as:

$$\begin{aligned} A_{mn} &= \sum_{j=1}^N \left(\overline{Q}_{mn} \right)_j (h_j - h_{j-1}) \\ B_{mn} &= \frac{1}{2} \sum_{j=1}^N \left(\overline{Q}_{mn} \right)_j (h_j^2 - h_{j-1}^2) \\ D_{mn} &= \frac{1}{3} \sum_{j=1}^N \left(\overline{Q}_{mn} \right)_j (h_j^3 - h_{j-1}^3) \end{aligned} \quad (2.29)$$

Once the mid-plane strains and curvatures are known, the strains at any ply through-thickness position z are recovered as [24]:

$$\begin{aligned} \varepsilon_{xx} &= \varepsilon_{xx}^0 + z k_{xx} \\ \varepsilon_{yy} &= \varepsilon_{yy}^0 + z k_{yy} \\ \gamma_{xy} &= \gamma_{xy}^0 + z k_{xy} \end{aligned} \quad (2.30)$$

and ply stresses are then retrieved using equation 2.20.

2.3.5 Failure Criteria

In composite design, applied stresses or strains are compared against material strength limits using failure criteria. Unlike metals, fibre-reinforced composites are anisotropic and do not exhibit gross yielding, requiring specialised theories. These can be broadly divided into two groups:

- **Criteria not associated with failure modes** - polynomial and tensorial criteria that describe a failure surface as a mathematical function of material strengths, without distinguishing the underlying damage mechanism. Some examples of failure criteria are the Tsai-Wu Failure Theory, Maximum Stress Theory and Maximum Strain Theory.
- **Criteria associated with failure modes** - physically motivated criteria that distinguish between specific damage mechanisms such as fibre rupture, transverse matrix cracking, and shear matrix cracking. Hashin and Puck Theories are within this scope of failure criteria.

Tsai-Wu Failure Theory

The Tsai-Wu criterion predicts failure when the following condition is satisfied for an orthotropic lamina under plane stress [24]:

$$F_1 \sigma_1 + F_2 \sigma_2 + F_{11} \sigma_1^2 + F_{22} \sigma_2^2 + F_{66} \tau_{12}^2 + 2F_{12} \sigma_1 \sigma_2 > 1 \quad (2.31)$$

The strength coefficients are determined from uniaxial tensile, compressive, and

shear tests. Since the interaction term F_{12} requires biaxial testing, a conservative approximation is commonly adopted [24]:

$$\begin{aligned} F_1 &= \frac{1}{S_{Lt}} - \frac{1}{S_{Lc}}, & F_2 &= \frac{1}{S_{Tt}} - \frac{1}{S_{Tc}}, & F_{11} &= \frac{1}{S_{Lt}S_{Lc}}, \\ F_{22} &= \frac{1}{S_{Tt}S_{Tc}}, & F_{66} &= \frac{1}{S_{LT}^2}, & F_{12} &\cong -\frac{1}{2}\sqrt{F_{11}F_{22}} \end{aligned} \quad (2.32)$$

Maximum Stress Theory

A non-interactive criterion that predicts failure when any stress component in the principal material direction exceeds its corresponding strength limit [24]:

$$\begin{aligned} -S_{Lc} &< \sigma_{11} < S_{Lt} \\ -S_{Tc} &< \sigma_{22} < S_{Tt} \\ -S_{LTs} &< \tau_{12} < S_{LTs} \end{aligned} \quad (2.33)$$

Maximum Strain Theory

Analogous to Maximum Stress Theory, failure is predicted when any principal strain component exceeds its allowable limit [24]:

$$\begin{aligned} -\varepsilon_{Lc} &< \varepsilon_{11} < \varepsilon_{Lt} \\ -\varepsilon_{Tc} &< \varepsilon_{22} < \varepsilon_{Tt} \\ -\gamma_{LTs} &< \gamma_{12} < \gamma_{LTs} \end{aligned} \quad (2.34)$$

Both maximum stress and strain theories are non-interactive: they do not account for combined stress states, which can lead to non-conservative strength predictions under multiaxial loading.

Hashin (3D Extension)

Hashin's criterion distinguishes four physically distinct failure modes - fibre tension, fibre compression, matrix tension, and matrix compression - and accounts for the contribution of shear stress in fibre tensile failure and adopts a quadratic formulation for matrix failure, since a linear approach underestimates matrix strength:

$$\text{Fibre tension } (\sigma_1 > 0) : \left(\frac{\sigma_1}{\sigma_{1T}^u} \right)^2 + \frac{\sigma_{12}^2 + \sigma_{13}^2}{(\sigma_{12}^u)^2} = 1$$

$$\text{Fibre compression } (\sigma_1 < 0) : -\sigma_1 = \sigma_{1C}^u$$

$$\text{Matrix tension } (\sigma_2 + \sigma_3 > 0) : \left(\frac{\sigma_2 + \sigma_3}{\sigma_{2T}^u} \right)^2 + \frac{\sigma_{23}^2 - \sigma_2\sigma_3}{(\sigma_{23}^u)^2} + \frac{\sigma_{12}^2 + \sigma_{13}^2}{(\sigma_{12}^u)^2} = 1$$

Matrix compression ($\sigma_2 + \sigma_3 < 0$) :

$$\left[\left(\frac{\sigma_{2C}^u}{2\sigma_{23}^u} \right)^2 - 1 \right] \frac{\sigma_2 + \sigma_3}{\sigma_{2C}^u} + \left(\frac{\sigma_2 + \sigma_3}{2\sigma_{23}^u} \right)^2 + \frac{\sigma_{23}^2 - \sigma_2\sigma_3}{(\sigma_{23}^u)^2} + \frac{\sigma_{12}^2 + \sigma_{13}^2}{(\sigma_{12}^u)^2} = 1$$

Puck

The Puck criterion distinguishes between fibre fracture and inter-fibre fracture (matrix cracking), and notably identifies three modes of matrix failure - A, B, and C - differentiated by the fracture plane orientation and the nature of the driving load, as illustrated in Fig. 2.4. This physically motivated decomposition provides more accurate failure predictions under complex stress states compared to the Hashin formulation.

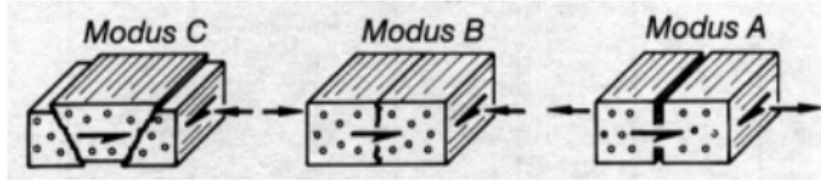


Figure 2.4: Inter-fibre fracture modes A, B and C [25]

The criteria presented above operate at the lamina level. Predicting the strength of a full laminate requires accounting for the progressive nature of damage accumulation, as failure typically initiates in individual plies and propagates through stiffness degradation, delamination, and interlaminar effects before final structural failure occurs [25].

2.4 Bolted Joint Design

Bolted joints are widely used in structural assemblies due to their reliability, ease of disassembly, and predictable mechanical behaviour. In the context of the accumulator casing, they are the primary means of connecting the composite walls and mounting the pack to the vehicle structure as welding is not applicable to composite materials, and bolted connections allow the disassembly required for maintenance and inspection, consistent with the design constraints established earlier. The interaction between bolt stiffness, member stiffness, preload, and external loading governs load distribution - and is the focus of the analytical formulation presented here.

2.4.1 Bolt and Member Stiffness

Within the clamped zone, a bolt behaves as an axial spring under tensile loading. Since the shank and threaded portions have different cross-sectional areas, both are treated as springs in series, giving the bolt stiffness [26]:

$$k_b = \frac{A_t A_d E}{A_d l_t + A_t l_d} \quad (2.35)$$

where A_t and A_d are the tensile stress and shank areas, l_t and l_d are the threaded and unthreaded grip lengths, and E is the bolt material Young's modulus.

The clamped members act as compressive springs opposing bolt elongation. Their compression zone is approximated by a conical frustum with half-apex angle $\alpha = 30^\circ$, a standard assumption for metallic and composite assemblies [26]. The stiffness of each frustum, k_m , is obtained by integrating the elemental compression over the grip length:

$$k_m = \frac{\pi E d \tan \alpha}{\ln \left[\frac{(2t \tan \alpha + D_O - d)(D_O + d)}{(2t \tan \alpha + D_O + d)(D_O - d)} \right]} \quad (2.36)$$

where D_O is the large diameter and d is the small diameter of the bolt frustum.

When the joint stacks multiple layers of different materials - such as composite walls, metal structures, and gaskets - each layer contributes its own frustum stiffness. The total member stiffness is then obtained as a series combination:

$$\frac{1}{k_m} = \frac{1}{k_{m,1}} + \frac{1}{k_{m,2}} + \dots + \frac{1}{k_{m,n}} \quad (2.37)$$

The joint stiffness constant, which governs how external load is shared between bolt and members, is then defined as:

$$C_b = \frac{k_b}{k_b + k_m} \quad (2.38)$$

2.4.2 Load Distribution in a Preloaded Joint

When tightened, the bolt is subjected to an initial preload F_i . For non-permanent joints, this is estimated from the bolt proof strength σ_p [26]:

$$F_i = 0.7 \cdot \sigma_p \cdot A_t \quad (2.39)$$

Upon application of an external tensile load P , the total bolt force and the residual clamping force on the members become:

$$F_b = F_i + C_b P, \quad F_m = F_i - (1 - C_b)P \quad (2.40)$$

Joint separation occurs when F_m reaches zero, so the joint remains closed provided:

$$F_i > (1 - C_b)P \quad (2.41)$$

The resulting tensile stress in the bolt is:

$$\sigma_b = \frac{F_b}{A_t} = \frac{F_i + C_b P}{A_t} \quad (2.42)$$

Since bolts are preloaded close to their proof strength, both yielding and joint separation must be verified to ensure safe static operation.

2.4.3 Shear Loading

When bolted joints are subjected to shear forces - such as those arising from lateral acceleration loads in the chassis mounting - the bolt is loaded transversely rather than axially. In this case, the external shear force V is carried by friction between the clamped members, provided the joint does not slip, and by the bolt shank in bearing and shear if slip occurs.

The friction capacity of the joint is governed by the clamping force and the coefficient of friction μ between the contact surfaces:

$$V_{friction} = \mu \cdot F_i \cdot n_f \quad (2.43)$$

where n_f is the number of friction interfaces. If the friction capacity is exceeded, the bolt shank enters bearing, and the shear stress in the bolt must be verified against its allowable shear strength. For a bolt loaded in single shear:

$$\tau_b = \frac{V}{A_d} \quad (2.44)$$

2.4.4 Gasketed Joints

When a gasket is present, its stiffness typically dominates the member compliance. The gasket pressure per bolt under combined preload and external loading is [26]:

$$p = [F_i - (1 - C_b)P] \frac{N}{A_g} \quad (2.45)$$

where A_g is the total gasket area and N is the number of bolts. This pressure must be verified against the recommended minimum seating pressure for the gasket material [27].

2.5 Experimental Tests

The following test methods provide the experimental basis for material characterisation and layup validation. They are introduced here as standardised procedures and applied in the design and validation chapters, once the material system and concept are established.

2.5.1 Tensile Tests

Tensile testing, conducted in accordance with ASTM D3039 [28], is the standardised method for determining the in-plane tensile properties of polymer matrix composites reinforced with continuous or discontinuous high-modulus fibres. The key

parameters obtained are the ultimate tensile strength (UTS), ultimate tensile strain, tensile chord modulus of elasticity, and Poisson's ratio.

Specimens consist of rectangular coupons with constant cross-section, with tabs bonded at the ends to improve load transfer and prevent grip-induced failure, as illustrated in Fig. 2.5. A minimum of five specimens per condition is required, tested at a crosshead speed of 2 mm/min. Strain is measured using extensometers with a 10–50 mm gauge length or strain gauges of at least 6 mm. For Poisson's ratio measurement, gauges must have a minimum length of 3 mm and resistance above 350 Ω , aligned parallel to the load direction at the specimen midsection.

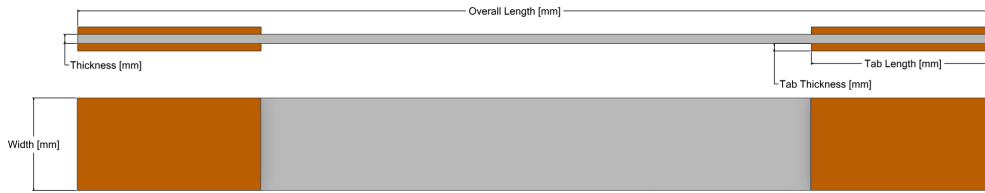


Figure 2.5: Tensile test specimen geometry

Specimen geometry varies with fibre orientation, as summarised in Table 2.1.

Table 2.1: Tensile specimen geometry recommendations [28]

Fiber Orientation	Width [mm]	Length [mm]	Thickness [mm]	Tab Length [mm]	Tab Bevel Angle [°]
0° unidirectional	15	250	1.0	56	7 or 90
90° unidirectional	25	175	2.0	25	90
Balanced and symmetric	25	250	2.5	emery cloth	–
Random-discontinuous	25	250	2.5	emery cloth	–

2.5.2 Three-Point Bending Test

The three-point bending (3PB) test, standardised under ASTM D7264 [29], characterises the flexural properties of polymer matrix composites, providing flexural strength, flexural modulus, and strain at failure. A specimen is supported at both ends and loaded centrally, as shown in Fig. 2.6, producing a well-defined bending moment and shear distribution directly relevant to the loading conditions of flat plate components such as the accumulator walls.

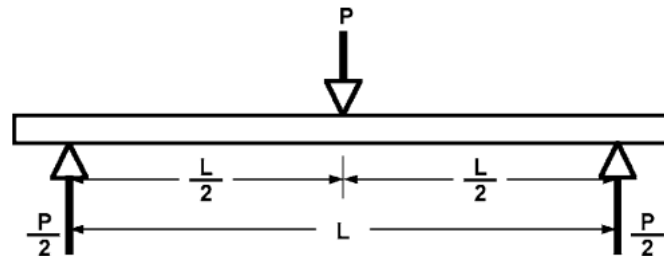


Figure 2.6: Three-point bending load diagram [29].

Specimen geometry and test setup requirements are summarised in Table 2.2. Rule constraints may limit strict adherence to the standard geometry, and experimental out-

comes may differ slightly from fully standardised values as a result, according to rule T 3.6.1.

Table 2.2: Geometric and specimen requirements [29].

Parameter	Requirement
Geometry	Rectangular plate, $L/h = 32 : 1$ $h = 4 \text{ mm}$, $b = 13 \text{ mm}$, $L = 1.2 \times \text{span}$
Trials	≥ 5 per condition
Supports & loading nose	Cylindrical, radius 3 mm, hardness 60–62 HRC
Crosshead speed	1.0 mm/min

As an alternative, representative panel tests can be used to prove equivalency. For general design purposes, panels of 275×500 mm are adopted, while accumulator-specific validation may use a reduced 150×275 mm panel when the core thickness does not exceed 5 mm. In this configuration, a minimum support span of 200 mm and a loading nose radius of at least 5 mm are required.

For results to be valid, failure must initiate at the outer surface - either by tensile cracking or compressive ply buckling - without prior interlaminar shear failure or crushing at the supports.

The flexural stress at the outer surface is:

$$\sigma = \frac{3PL}{2bh^2} \quad (2.46)$$

where P is the applied force [N], L the support span [mm], b the specimen width [mm], and h the specimen thickness [mm]. Flexural strength corresponds to σ evaluated at maximum load prior to failure. The flexural strain at the outer surface is:

$$\varepsilon = \frac{6\delta h}{L^2} \quad (2.47)$$

where δ is the mid-span deflection [mm]. Finally, the flexural chord modulus, evaluated over the 0.001–0.003 strain range to avoid initial nonlinear effects, is:

$$E_f^{\text{chord}} = \frac{\Delta\sigma}{\Delta\varepsilon} = \frac{\Delta P}{\Delta\delta} \cdot \frac{L^3}{4bh^3} \quad (2.48)$$

2.5.3 Perimeter Shear Strength Test

The perimeter shear strength test, conducted in accordance with ASTM D732–02 [30], determines the apparent shear strength of polymer matrix composites using a punch tool, as illustrated in Fig. 2.7. Specimen requirements are summarised in Table 2.3.

It should be noted that the Formula Student rules [4] prescribe different test parameters from the ASTM standard. Specifically, a 25 mm diameter flat punch must be pulled through a flat laminate sample of at least 100×100 mm (by T 3.6.11), supported by a fixture with a 32 mm hole aligned coaxially with the punch. The sample must not

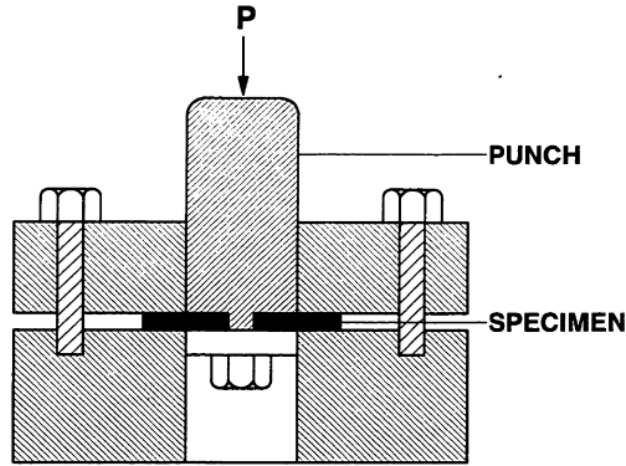


Figure 2.7: Illustrative Perimeter shear test setup [31]

Table 2.3: Perimeter shear test specimen requirements [30].

Parameter	Requirement
Shape	50 mm square plates or 50 mm diameter disks
Thickness	1.27–12.7 mm
Central hole	11 mm (punch shaft alignment)
Specimens	≥ 5 per condition
Crosshead speed	1.3 ± 0.3 mm/min

be clamped to the fixture (by T 3.6.12). These rule-defined conditions take precedence in this work and may produce results that differ from fully standardised ASTM values.

The shear strength is defined as the maximum stress required to fully shear the specimen:

$$\tau = \frac{P}{\pi dt} \quad (2.49)$$

where P is the maximum load [N], d the punch diameter [mm], and t the specimen thickness [mm].

2.5.4 Density Test

The density test is carried out in accordance with ASTM D792-07 [32], using the Archimedes immersion method, illustrated in Fig. 2.8. Setup, specimen, and conditioning requirements are summarised in Table 2.4.

The procedure consists of three weighings: the specimen in air (a), the specimen fully immersed in distilled water (b), and the suspension wire and sinker immersed to the same depth without the specimen (w). Visible air bubbles are removed before the immersed reading is taken, immersion depth is kept constant throughout, and measurements are performed promptly to limit water absorption. At least two determinations are performed per sample. The specimen density at temperature T is:

$$\rho_{\text{specimen}}(T) = \frac{a}{a + w - b} \rho_{\text{water}}(T) \quad (2.50)$$

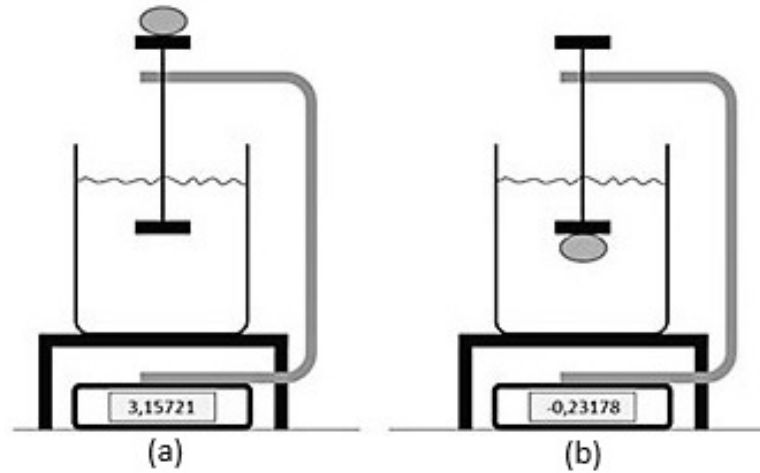


Figure 2.8: Density test apparatus [33]

where a is the apparent mass in air, b the apparent mass when immersed, and w the apparent mass of the immersed wire and sinker alone.

Table 2.4: ASTM D792 - Summary of setup, specimens and conditioning by Method A.

Aspect	Description
Apparatus	Analytical balance (0.1 mg precision; 0.05% accuracy) with pan-straddle support, fine suspension wire, immersion vessel, and thermometer.
Materials	Deionised/distilled water, substantially air-free.
Specimens	Single piece with smooth surfaces and edges; typical mass 1–5 g (up to ~50 g).
Conditioning	$23 \pm 2^\circ\text{C}$ and $50 \pm 5\%$ Relative Humidity for ≥ 40 h (per D618).
Immersion	Constant immersion depth; weighings performed promptly to avoid liquid absorption.

2.5.5 Flammability Test

As required by rule EV 5.5.3, all materials used in the accumulator container must be fire retardant. Compliance can typically be demonstrated through manufacturer datasheets; where this is not available, experimental verification is required, as defined in rule T 1.2.1.

Experimental verification follows the UL94 vertical burning test [34], which evaluates the flammability of polymeric materials by exposing vertically oriented specimens to a controlled flame and assessing their burning behaviour. The rules require a V-0 rating, whose acceptance criteria are summarised in Table 2.5. The experimental setup is illustrated in Fig. 2.9.

Table 2.5: *UL94 vertical burning test – V-0 rating criteria*

Requirements
After-flame time ≤ 10 s per specimen after each flame application
Total after-flame time ≤ 50 s (10 applications, 5 specimens)
After-glow time ≤ 30 s after second flame application
No flaming drips that ignite cotton
Specimen must not burn up to the clamp

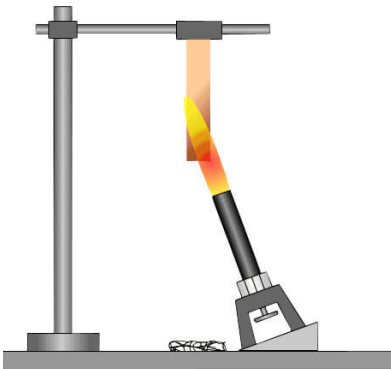


Figure 2.9: *Illustration of the setup for the vertical burning test.*

GOALS AND CONSTRAINTS

3.1 TSAC Overview

The Tractive System Accumulator Container (TSAC) is the structural enclosure responsible for housing the battery system of the vehicle. It contains the energy storage cells, battery management system (BMS), high-voltage wiring, and auxiliary components required for safe operation.

Beyond simple containment, the TSAC must fulfil multiple functions: structural integrity, electrical isolation, thermal management, and compliance with Formula Student regulations. It must also ensure safe operation under dynamic loading conditions while remaining accessible for maintenance and inspection. These requirements directly drive the design goals, load cases, and material selection developed throughout this chapter.

3.2 Design Goals

3.2.1 Mass Target

In Formula Student, vehicle performance is strongly influenced by mass, as it directly affects acceleration, braking, and cornering behaviour. Weight reduction is therefore a primary design objective with direct impact on lap time and overall competitiveness.

Performance simulations conducted in commercial lap time simulation software, parametrised with the current vehicle design targets, indicate that a mass reduction of 50 kg yields an improvement of approximately 2 seconds per lap - a significant gain in a competition where margins are small. The previous prototype had a total mass of 283 kg, against a current target of 231 kg, representing a reduction of 52 kg. This excess weight was primarily attributed to conservative safety factors and the widespread use of metallic materials in components where composites would offer a superior strength-to-weight ratio.

As shown in Fig. 3.1, the powertrain is the dominant contributor to vehicle mass, followed by the chassis and dynamics systems. Within the accumulator system, the

cells account for nearly half of the total mass of 68.6 kg (Table 3.1). While the number of cells is constrained by energy and voltage requirements, structural components such as the TSAC offer significant potential for mass reduction.

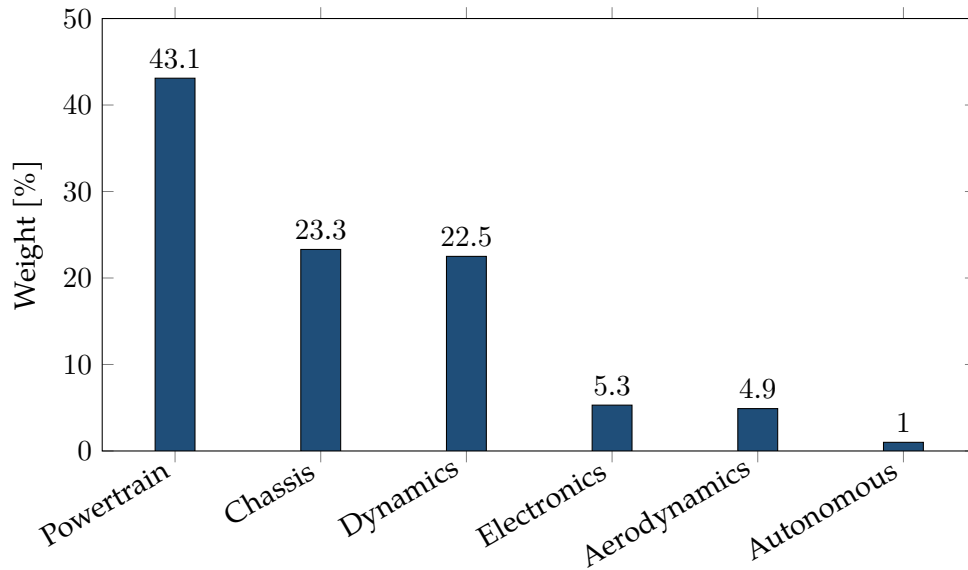


Figure 3.1: System weight distribution by department

Table 3.1: Accumulator weight distribution

Component	Weight [kg]
Cells	33.6
Segment parts	16.5
TSAC	14.1
Accessories	4.4
Total	68.6

Based on these observations, team goals and lap times, a 25% reduction in total accumulator mass is targeted, with a stronger focus on the TSAC structure. A reduction of approximately 40% is defined for the TSAC, decreasing its mass from 14.1 kg to 8.46 kg. This target is aligned with strategies reported in the literature (Chapter 2.1.3) and guides the material selection and structural design throughout this work.

3.2.2 Load Cases

The TSAC is dimensioned according to Formula Student load requirements, using equivalent acceleration-based load cases. Safety factors are defined using the Pugsley method [35], which determines the appropriate factor of safety based on the quality of applied loads, material knowledge, maintenance standards, and consequence of failure.

According to regulations, the structure must withstand:

- 20G in the vertical direction
- 40G in the longitudinal and lateral directions

These accelerations are applied to the internal mass to obtain the forces acting on the walls, joints, and mountings. The same conditions are considered for adhesive joints, including temperature effects.

The most critical parameter is the mass of a single module, as any underestimation is amplified across the full battery pack. The nominal module mass is estimated at 5.9 kg; however, a factor of 1.2 is applied to account for uncertainties in adhesive mass, electronics, and manufacturing variations. This factor is derived from the Pugsley method, considering very good control of applied loads - given that the regulatory load cases already represent extreme peak accelerations - together with very good material quality and maintenance standards. With no direct danger to people from structural failure and no significant economic impact, the Pugsley multipliers yield $N_1 = N_2 = 1$, resulting in a combined safety factor of $\text{FoS} = 1.2 \times 1 = 1.2$ and a design module mass of 7.1 kg.

Different components are dimensioned according to their function:

- **Structural components:** $\text{FoS} = 1$, as required by regulations under the prescribed load cases;
- **Adhesive joints:** reduced strength at elevated temperature and additional safety margins are considered;
- **Gasket:** a compliance factor of 2.05 is adopted to ensure sealing under compression;
- **Bolted connections:** $\text{FoS} = 1$, since preload can be controlled during assembly.

3.2.3 Functional Requirements

The design targets a centre of gravity located approximately 250 mm above the ground with a 50/50% front-to-rear weight distribution, consistent with the vehicle's dynamic objectives. Given that the battery pack represents a significant portion of total vehicle mass, its placement is critical. The TSAC is therefore positioned as low as possible and near the rear centre of the vehicle, contributing to both CoG reduction and balanced load distribution (Fig. 3.2).

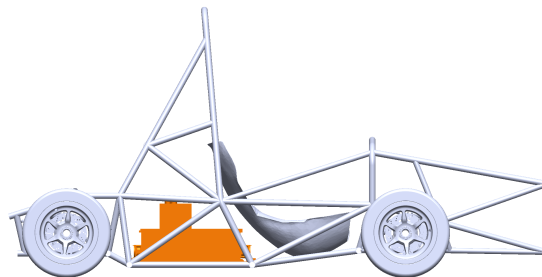


Figure 3.2: *Accumulator position relative to the chassis*

A specific serviceability objective was also defined: the accumulator must be removable from the vehicle with minimal effort, improving safety during charging and

facilitating technical inspections. The previous design allocated bolts in shear, requiring the battery to be manoeuvred through multiple positions to clear the mounting brackets — a process that was both difficult to align and time-consuming. The new concept addresses this by loading the mounting bolts in tension, allowing the TSAC to be lifted vertically without lateral movement. A target removal time of 5 minutes was set, compared to the 15 minutes required in the previous design.

Beyond these quantitative targets, the TSAC must also comply with the safety and regulatory requirements established in the literature review, namely electrical insulation, fire resistance, thermal management, and structural protection of internal components. Together with the weight reduction targets, these constraints directly drive the material selection and structural configuration developed in the following sections.

3.3 Concept Development

3.3.1 Concept Selection

Four concepts were evaluated based on the defined requirements, as illustrated in Fig. 3.3. The selection process considered weight, thermal performance, manufacturability, integration, and packaging, with criteria weighted according to their relative importance from 0 to 5, being 5 the best configuration (Table 3.2).

Despite weight being an important parameter, greater emphasis was placed on reliability-related aspects, particularly thermal behaviour and manufacturing feasibility. Minimising thermal gradients was identified as critical for safety, while manufacturing constraints – due to limited resources – strongly influence achievable tolerances and build quality. Integration, packaging, and battery centre of gravity were also weighted heavily, as the accumulator's size and position directly affects compatibility with the chassis and surrounding systems.

Concept 3 achieved the highest score and was selected as the most balanced solution, offering the strongest combination of system integration and accessibility for maintenance. The latter is a particularly important advantage given the 5-minute removal target, which requires a layout that minimises disassembly complexity.

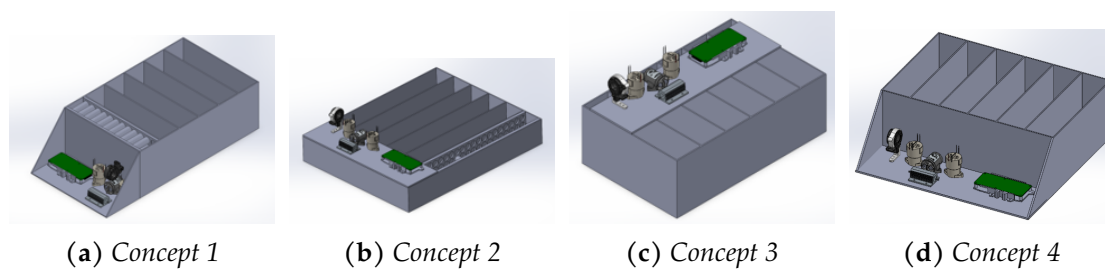


Figure 3.3: Concepts developed according to targets and constraints

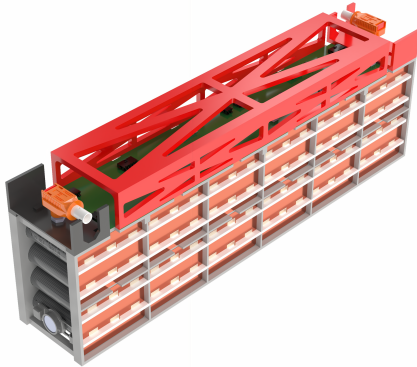
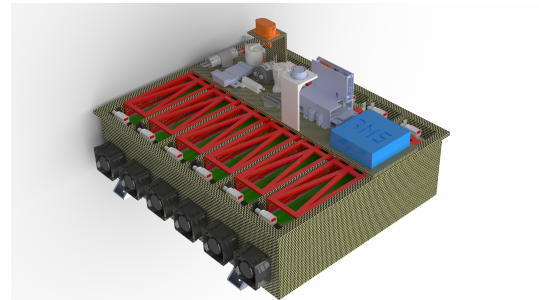
Table 3.2: *Concept evaluation matrix*

Parameter	Weight	Concept 1	Concept 2	Concept 3	Concept 4
Weight	5	5	3	5	5
Cooling	10	5	2	5	3
Manufacturing	25	4	2	3	4
Packaging	15	3	5	5	3
Integration	20	3	3	5	5
Battery CoG	15	3	5	3	4
Future Proof	10	4	2	4	4
Total	100	3.65	3.15	4.10	4.00

3.3.2 Concept 3 Description

The TSAC is mounted to the chassis through aluminium supports bonded to the transverse walls and secured with bolted connections, allowing the pack to be removed within the target time while maintaining a rigid and repeatable attachment. Internally, partition walls separate the modules from one another, preventing contact between them and contributing to the overall structural integrity of the enclosure. The assembly is sealed by a removable lid bolted along the perimeter, balancing weatherproofing with the accessibility required for inspection and maintenance.

The choice of cylindrical cells was driven by their reliability, commercial availability, and favourable thermal characteristics. Arranged into six modules to satisfy the required capacity and voltage targets (Fig. 3.4), this concept also promotes airflow between cell rows and positions the battery mass low within the vehicle, benefiting the centre of gravity. Rather than being mechanically fastened to the casing, the modules are laterally constrained by the stiffness of the side walls and vertically supported by the floor panel and a stopper integrated into the lid - an approach that simplifies assembly without compromising positional stability under dynamic loading. Thermal management is provided by externally mounted cooling fans, which drive a pressure-induced airflow through the enclosure and maintain the cells within their optimal operating temperature range.

**Figure 3.4:** *Accumulator module***Figure 3.5:** *TSAC representation*

In addition to the battery modules and BMS, the TSAC houses high-voltage com-

ponents including DC contactors, fuses, current sensors, HV connectors, and signal wiring (Fig. 3.5). Their layout is defined to ensure accessibility, proper clearances, and compliance with Formula Student EV regulations.

The described concept establishes the structural layout and functional requirements of the TSAC, which directly influence material selection. The need for lightweight structures with adequate stiffness, and compatibility with adhesive bonding drives the use of composite materials for the enclosure, combined with metallic components in critical load-transfer regions.

3.4 Materials

Material selection must comply with Formula Student regulations. According to EV5.5.5, the TSAC may be manufactured from steel or aluminium with specified minimum thickness requirements. Alternatively, composite materials may be used if equivalent mechanical performance to the steel baseline is demonstrated, as stated in EV5.5.6. All materials must additionally be flame-retardant in accordance with EV5.5.3 and T1.2.1. For conductive materials, proper insulation or grounding must be implemented per EV5.4.1 and EV3.1.1; for insulating materials, an electrical resistance of at least 2 MΩ must be ensured.

Based on these requirements and common practices in composite casing design, the typical material combinations are summarised in Table 3.3.

Table 3.3: Common materials and combinations for composite casings

Materials	Material Combinations
Carbon Fibre	1) Carbon Fibre + Aluminium + Nomex/Glass Fibre (+ Core)
Aramid	2) Aramid (+ Core)
Glass Fibre	3) Glass Fibre (+ Core)
Core	
Nomex Sheet	
Aluminium sheet	

The selection between configurations is driven by weight, manufacturability, available thickness, and cost.

Option 1: Carbon Fibre + Aluminium + Insulation

This configuration provides high stiffness through carbon fibre and enables electrical grounding via aluminium. However, it introduces significant manufacturing challenges including delamination risk at bonded interfaces and air entrapment during assembly. The aluminium layer increases weight while contributing minimally to structural performance, and the insulating layers impose strict tolerance requirements. This option was not selected.

Option 2: Aramid (+ Core)

Aramid fibres provide inherent electrical insulation while offering competitive mechanical properties, simplifying the structural design. A single material system improves manufacturability and allows better control of laminate thickness and dimen-

sional tolerances. This option was selected due to its balance between structural performance, weight efficiency, and manufacturing feasibility.

Option 3: Glass Fibre (+ Core)

Glass fibre offers a cost-effective solution with inherent insulation, but its lower strength-to-weight ratio results in significantly heavier structures, making it unsuitable for a weight-critical application.

With the material system defined, the manufacturing process further constrains the available options. Prepreg solutions are difficult to source and require autoclave processing, which is not accessible within the available resources. A wet layup process is therefore adopted, with Sika Biresin® CR132 FR selected as the resin system for its flame-retardant properties and compatibility with this process. Although foam cores compliant with the required standards are available, they require greater wall thickness than the design constraints allow and are therefore discarded. The final solution is a solid aramid laminate, using Kevlar® 49 plain weave fabric at 170 g/m².

3.4.1 Aramid Properties Definition

The fibre and matrix properties were taken from the respective manufacturer technical datasheets [36] [37], and the lamina properties subsequently estimated through the micromechanical framework outlined in Chapter 2.3.2, for a fibre volume fraction of 50%. Where data was unavailable, values were sourced from literature: the matrix Poisson's ratio was taken as 0.37 [38], fibre shear modulus and compressive strength were obtained from experimental characterisation data for Kevlar 49 [22], and the in-plane shear strength was approximated using the matrix tensile strength as a matrix-dominated property.

Table 3.4: *Constituent material properties (aramid composite)*

Property	Fibre	Matrix
Elastic Modulus E [GPa]	112.4	3.6
Poisson's Ratio ν [-]	0.36	0.37
Shear Modulus G [GPa]	2.9	1.31
Density ρ [kg m ⁻³]	1440	1240
Tensile Strength S_T [MPa]	3 000	52
Compressive Strength S_C [MPa]	517	124

Experimental tests were conducted on laminates manufactured using the selected materials to validate and refine these estimates - detailed in Chapter 6.1. Tensile tests provided the Young's modulus, Poisson's ratio, mean tensile strength, and an average ply thickness of 0.31 mm (dividing the total thickness by number of plies), while a density test verified the volumetric mass of the laminate in chapter 6.2.

The resulting design values are summarised in Table 3.5. Experimentally obtained properties are used wherever available; remaining parameters are estimated analytically. It is worth noting that the significant discrepancy between analytical and experimental elastic modulus and strength values is expected: the micromechanical mod-

els assume a unidirectional fibre arrangement, while the actual material uses a plain weave architecture. The crimp and interlacing inherent to woven fabrics reduce both stiffness and strength relative to the unidirectional ideal, which explains the reduction from 53 448 MPa to 19 796 MPa in E_1 and from 1 548 MPa to 373 MPa in tensile strength. Experimental values are therefore representative of the actual laminate behaviour and are adopted as design values.

Table 3.5: *Lamina properties*

Property	Analytical	Experimental	Design Values
Elastic Properties			
E_1 [MPa]	53 448	19 796	19 796
E_2 [MPa]	53 448	19 796	19 796
E_3 [MPa]	9 958	–	9 958
ν_{12} [–]	0.37	0.09	0.09
ν_{13} [–]	0.37	–	0.37
ν_{23} [–]	0.37	–	0.37
G_{12} [MPa]	1 926	–	1 926
G_{13} [MPa]	3 942	–	3 942
G_{23} [MPa]	3 942	–	3 942
Strength Properties			
S_{T1} [MPa]	1 548	373	373
S_{T2} [MPa]	1 548	373	373
S_{T3} [MPa]	26	–	26
S_{C1} [MPa]	278	–	278
S_{C2} [MPa]	278	–	278
S_{C3} [MPa]	62	–	62
S_{S12} [MPa]	52	–	52
S_{S13} [MPa]	52	–	52
S_{S23} [MPa]	52	–	52
Physical Property			
ρ [kg m ^{−3}]	1340	1243	1243

3.4.2 Adhesive Selection

The adhesive selected is Permabond TA4230 [39], chosen for its compliance with flammability requirements, adequate shear strength, and availability. The manufacturer specifies a shear strength of 12–16 MPa for carbon fibre substrates, from which a representative value of 14 MPa is adopted. Temperature effects are considered in accordance with T 3.2.8: at 60°C the shear strength reduces to approximately 10.14 MPa, and a further 50% knockdown factor is applied, yielding a design shear strength of $\tau_{\text{design}} = 5.07$ MPa.

Experimental shear tests conducted on the selected adhesive, detailed in Chapter 6.6, returned a mean value of 8.6 MPa. Although this is consistent with the datasheet range, applying the temperature reduction and regulatory knockdown factors would

result in an overly conservative design value. The datasheet-derived strength is therefore retained as the basis for dimensioning, as it corresponds to the characterised substrate and provides a more representative design reference.

3.4.3 Gasket Selection

The gasket prevents fluid ingress into the TSAC through bolted joints and connectors, and is critical for passing the rain test. Due to UL94-V0 flammability requirements, material options are limited; EPDM [40] and PTFE [41] were identified as viable candidates.

PTFE was selected due to its availability and stable mechanical and thermal properties, including low water absorption and thermal stability. Although less compressible than EPDM, adequate sealing is achieved through controlled surface flatness and bolt preload. The stiffness ratio of the joint is accounted for in the bolted joint analysis, where the gasket minimum seating pressure is verified against the design clamping force.

ANALYTICAL IMPLEMENTATION

The objective of the analytical phase is to determine the minimum-thickness laminate configuration that satisfies strength and serviceability requirements for each TSAC wall under the governing load cases. While geometric parameters such as panel length and height are dictated by the module architecture, laminate thickness and stacking sequence are treated as design variables and determined simultaneously through a failure-index-driven procedure.

The dimensioning follows a three-stage approach. First, isotropic thin-plate formulations are applied to identify governing geometric parameters and establish baseline thickness requirements. Subsequently, Classical Lamination Theory is used to derive an equivalent flexural rigidity, allowing composite behaviour to be incorporated while retaining closed-form solutions. Finally, ply-level stresses are estimated from the global curvature field and assessed against the Tsai–Wu failure criterion. The panel nomenclature adopted throughout is illustrated in Figs. 4.1 and 4.2.

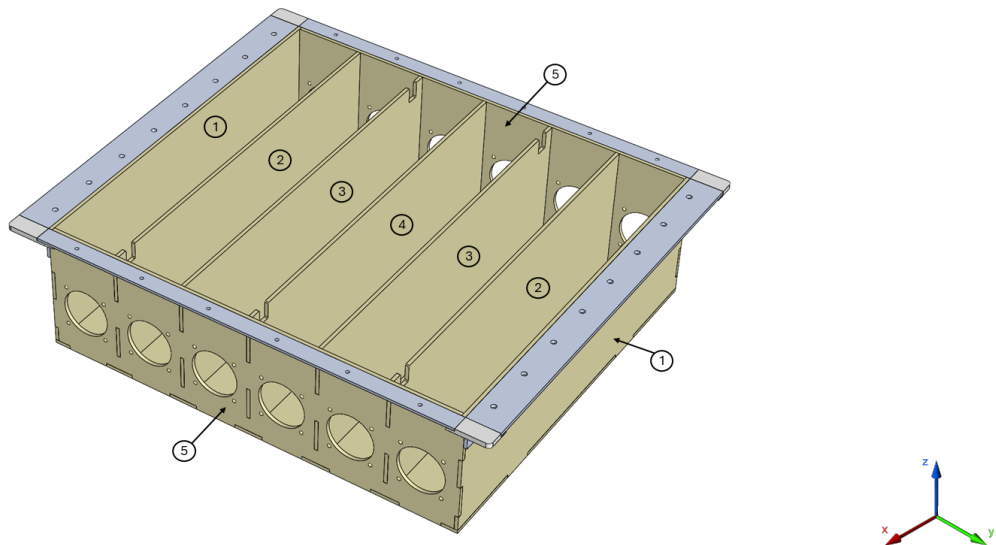


Figure 4.1: TSAC internal walls (2,3,4) and outer walls (1,5)

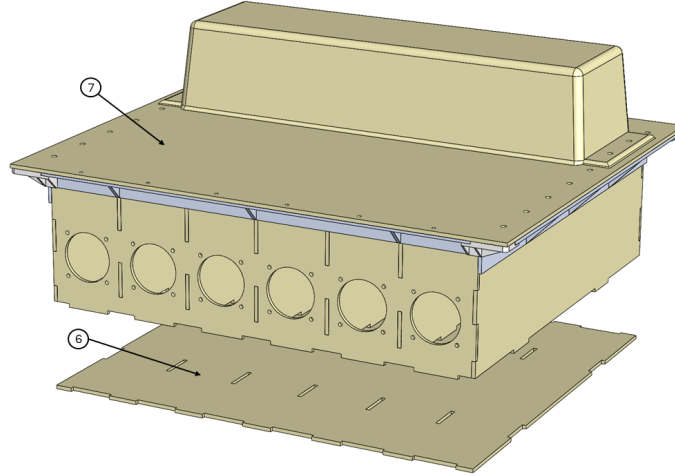


Figure 4.2: TSAC cover (7) and floor (6)

4.1 Wall Dimensioning

4.1.1 Internal Walls

Since the modules are not mechanically fastened within their compartments, their movement is constrained only by the geometric tolerances of the enclosure. Under dynamic loading, the modules can therefore translate freely until contact with the surrounding walls is established.

The internal walls are bonded along the bottom and lateral edges, which are modelled as clamped. The top edge is considered simply supported, as the lid structure limits translation while permitting rotation. Under the lateral design acceleration of 40G, the modules accumulate against the outermost constrained wall. Wall 2 (Fig. 4.1) therefore experiences the highest cumulative load, corresponding to the combined mass of five modules. The intermediate walls carry progressively smaller loads, while the outer wall absorbs the total boundary reaction.



Figure 4.3: TSAC internal walls (Fig. 4.1, walls 2, 3, 4)

Regarding boundary conditions, Roark's Case 10 (Fig. 4.4d) represents the exact edge configuration. However, it may underestimate stresses and overestimate dis-

placements since it assumes greater freedom of movement at the upper edge than is physically present - the adjacent modules and lid partially constrain this edge. Roark's Case 9 (Fig. 4.4c), which models a plate with a supported upper edge allowing rotation but preventing translation under uniform load, was therefore selected as the most representative analytical case.

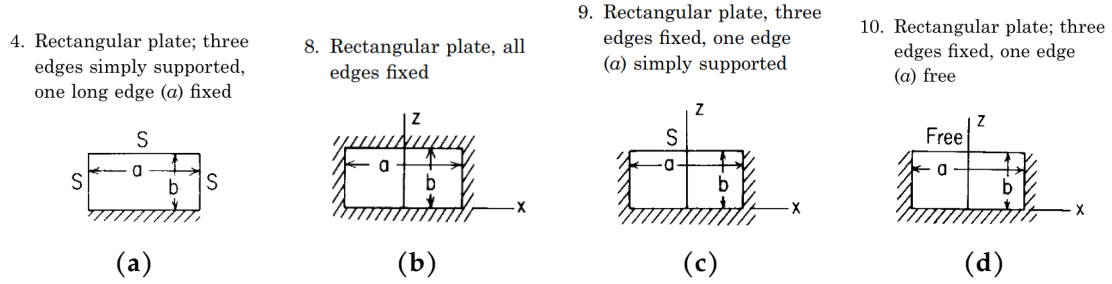


Figure 4.4: Boundary condition cases from Roark's formulations [19]: (a) Case 4, (b) Case 8, (c) Case 9, (d) Case 10.

Based on the geometric parameters in Table 4.1, the laminate was dimensioned using a spreadsheet-based iterative approach, with number of plies and ply orientation as the design variables. The Tsai–Wu failure criterion was selected for its physically realistic failure envelope and consistency with experimental data reported in the literature.

Table 4.1: Geometric parameters - internal walls

Length a	457.2 mm
Width b	153.6 mm
Aspect ratio $\lambda = a/b$	≈ 3

Once a suitable layout was identified, adhesive shear stresses were verified using Eq. 2.11. The results showed that adhesive shear governs the design in all internal wall cases, requiring laminate thickness to be increased beyond the value dictated by plate stress alone.

For cases where Roark's tables do not directly provide a displacement coefficient, Cases 8a and 4a were used as bounding references, representing fully clamped and simply supported conditions respectively. Applying the small deflection formulation yielded $w/h > 0.5$ in all cases, indicating that the small deflection assumption is not valid. The large deflection formulation (Annex B) was therefore applied using Eq. 2.12, with an aspect ratio of $\lambda = 3$ and a held-but-not-fixed boundary condition. This formulation provides both displacement and the decomposed bending and membrane stress components, from which the Tsai–Wu index and adhesive shear stress are evaluated. The final dimensioning results are presented in Table 4.2.

Adapting the laminate thickness to the load demand of each wall allows the number of plies to be reduced where loads are smaller. Based on an estimated wall area of 70 226 mm² and a ply areal density derived from Table 3.5, each ply contributes approximately 27 g. The differentiated layout strategy therefore yields a mass saving of

Table 4.2: Internal wall dimensioning results

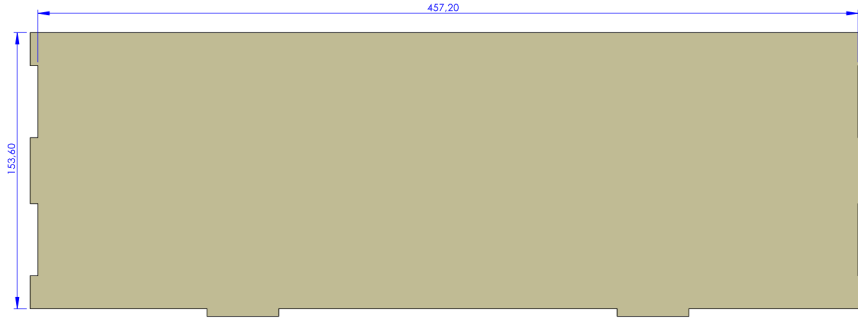
Parameter	Wall 2	Wall 3	Wall 4
Load [N]	13 891	11 113	8 335
Thickness [mm]	4.03	3.10	2.48
Layup	[0/90] ₁₃	[0/90] ₁₀	[0/90] ₈
Max stress [MPa]	98	84	76
Glue shear [MPa]	4.71	4.90	4.59
Displacement [mm]	3.81	3.84	3.97
Governing criterion	Glue shear	Glue shear	Glue shear

approximately 300 g relative to designing all walls for the worst-case load, bringing the total internal wall mass to approximately 1 461 g.

4.1.2 Outer Walls

Y Direction

The dimensioning of the outer wall in the Y direction (Fig. 4.5) follows the same procedure described in Section 4.1.1. This wall must resist the total combined load from all six modules under 40G lateral acceleration. The bottom and lateral edges are bonded and modelled as clamped; the upper edge is bolted and treated as simply supported. Roark's Case 9a was selected accordingly, with the same geometric dimensions as the internal walls (Table 4.1).

**Figure 4.5:** TSAC outer wall - Y direction (Fig. 4.1, wall 1)

An alternative fully clamped condition using Case 8a was also evaluated to assess sensitivity to boundary condition assumptions; however, adhesive shear was found to govern the design in both cases. Large deflection behaviour was observed, with $0.4 < w/h < 0.8$, and the final results were obtained from this formulation, as summarised in Table 4.3.

X Direction

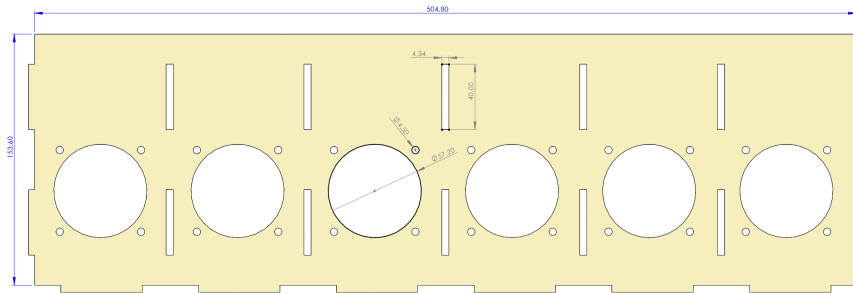
The outer wall in the X direction (Wall 5, Fig. 4.6) presents a more complex configuration due to the presence of openings for the cooling system and associated attach-

Table 4.3: Outer wall dimensioning results - Y direction

Parameter	Wall 1
Load [N]	13 891
Thickness [mm]	4.65
Layup	[0/90] ₁₅
Max stress [MPa]	89
Glue shear [MPa]	4.90
Displacement [mm]	3.24
Governing criterion	Glue shear

ments. These features introduce stress concentrations and alter the load path relative to a solid panel.

A uniform load was assumed as a simplification, since the design was expected to be governed by shear rather than bending. In practice, the contact between the modules and the wall is concentrated near the lateral edges (Fig. 4.7), which locally increases adhesive shear stress. To better represent this behaviour, the wall was analysed by dividing it into bays between adjacent internal walls. The governing bay corresponds to the region between Walls 2 and 4, where load sharing between bays reduces effective stiffness and amplifies adhesive stress. The bay geometric parameters are summarised in Table 4.4.

**Figure 4.6:** TSAC outer wall - X direction (Fig. 4.1, wall 5)**Table 4.4:** Geometric parameters - outer wall, X direction

Length a	80.35 mm
Width b	153.6 mm
$\lambda = a/b$	≈ 0.5

Figure 4.7: Module-to-wall interface - X direction

The initial design based on large deflection theory yielded a deflection ratio of

$7.2 < w/h < 15.7$, well beyond the valid range of the tabulated coefficients. Although bending and shear stresses remained within allowable limits, the predicted displacement was excessive. Two additional plies were therefore introduced, reducing the deflection ratio to $2.3 < w/h < 5$ - a range for which reliable coefficients are available. The final results are presented in Table 4.5.

Table 4.5: Outer wall dimensioning results - X direction

Parameter	Wall 5
Load [N]	2778
Thickness [mm]	2.48
Layup	[0/90] ₈
Max stress [MPa]	101
Glue shear [MPa]	3.21
Displacement [mm]	4.81
Governing criterion	Excessive deflection

Following the bending and shear verification, stress concentrations around the openings were assessed using Roark's Case 6b1 (Fig. 4.8). The results, summarised in Table 4.6, confirm that the current laminate provides adequate resistance and no modification to the layup is required.

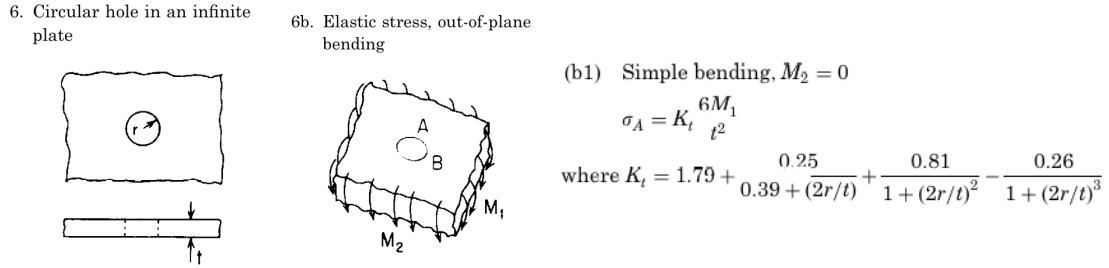


Figure 4.8: Stress concentration - Roark's Case 6 [19]

Table 4.6: Stress concentration assessment - outer wall, X direction (Wall 5)

Parameter	Wall 5
Hole radius [mm]	28.6
Moment [N mm/mm]	107
Stress concentration factor	1.80
Peak stress [MPa]	188
Factor of safety	1.40

4.1.3 Floor

The floor panel (Fig. 4.9) supports the total module weight in the downward direction. Since the load acts normal to the floor plane, the governing design acceleration is 20G. The weight of peripheral components is neglected, as the internal walls are expected to carry the majority of the vertical reaction, behaving analogously to continuous beams spanning the height of the enclosure.

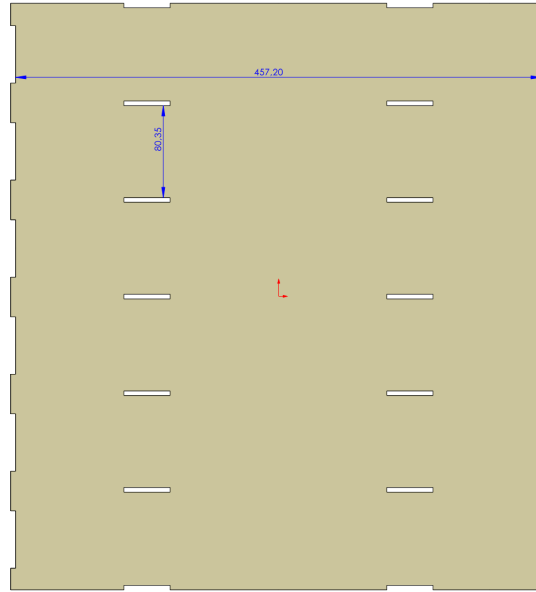


Figure 4.9: TSAC floor panel (Fig. 4.2, Wall 6)

A single governing bay with shared load was analysed, as this condition produces the highest plate stress. Since all edges are bonded, Roark's Case 8a (fully clamped plate) was applied. Adhesive shear was estimated conservatively using Case 1a (simply supported edges), which provides a shear reaction coefficient as a function of plate geometry. The geometric parameters are given in Table 4.7.

The small deflection formulation yields $w/h = 1.2$, confirming that large deflection theory must be applied. Although the large deflection formulation would allow a reduction by one ply while maintaining a factor of safety above unity, the resulting displacement was found to be excessive. The final design is therefore based on a more conservative configuration, as summarised in Table 4.8.

Table 4.7: Floor geometric parameters

Length a	457.2 mm
Width b	80.35 mm
λ	≈ 5.7

1. Rectangular plate; all edges simply supported

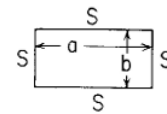


Figure 4.10: Roark's Case 1 [19]

Table 4.8: Floor panel dimensioning results

Parameter	Floor (Wall 6)
Load [N]	1 389
Thickness [mm]	1.24
Layup	$[0/90]_4$
Max stress [MPa]	49
Glue shear [MPa]	1.23
Displacement [mm]	1.54
Governing criterion	Plate stress

4.1.4 Cover

The cover (Fig. 4.11) is modelled as a flat rectangular plate, neglecting the geometric bump associated with peripheral components. This simplification is considered acceptable for preliminary dimensioning, as it allows closed-form solutions to be applied while capturing the overall load behaviour. The applied load combines the peripheral mass of 3 kg with the inertial load from all six modules acting in the upward direction under the design acceleration, scaled by the factor of 1.2 defined in Chapter 3. The cover is modelled as fully clamped, consistent with the restraint provided by the bolted perimeter joint.

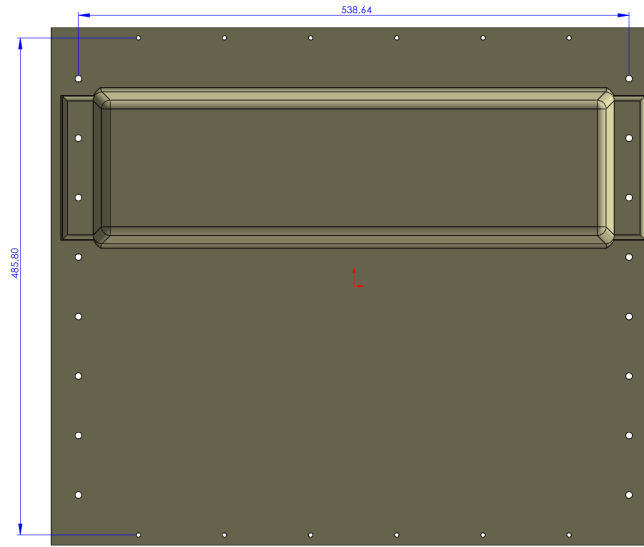


Figure 4.11: TSAC cover (Fig. 4.2, Wall 7)

The initial layup produced a large deflection coefficient of 634, exceeding the maximum tabulated value of 250 (Annex B) and indicating an inadmissible displacement level. Three additional plies were introduced, reducing the coefficient to within the valid range and allowing the large deflection formulation to yield converged results. The geometric parameters are in the table 4.9 and final results are summarised in the Table 4.10.

Table 4.9: Cover geometric parameters

Length a	504.8 mm
Width b	457.2 mm
$\lambda = a/b$	≈ 1.1

Table 4.10: Cover panel dimensioning results

Parameter	Wall 7
Load [N]	9 041
Thickness [mm]	4.34
Layup	$[0/90]_{14}$
Max stress [MPa]	35
Displacement [mm]	6.68
Governing criterion	Plate stress

Peripheral Component Shelf

An internal shelf plate is used to support peripheral components (Fig. 4.12). This plate is bolted along its edges, clamped between the lower structure and the cover, and carries both the peripheral mass and a fraction of the module load transferred through the module stopper contacts.

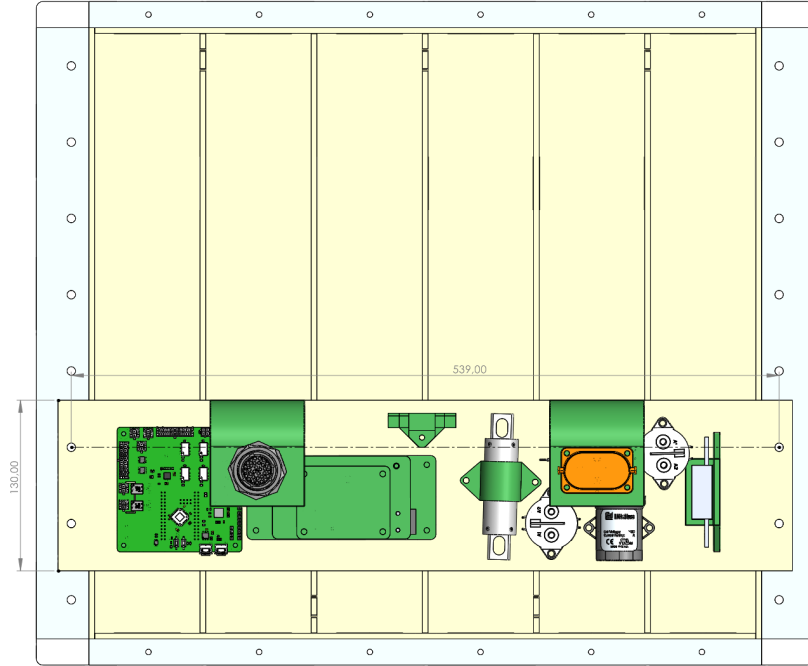


Figure 4.12: TSAC peripheral component shelf

The total load was distributed uniformly along the span and analysed using beam theory to determine the maximum bending moment (Fig. 4.13), from which the laminate was dimensioned. Plate displacement was subsequently evaluated using the resulting thickness and the corresponding flexural stiffness. Input parameters and results are presented in Tables 4.11 and 4.12.

4.2 Rules Compliance Verification

The Accumulator Structural Equivalency Spreadsheet (ASES) is used to verify that each TSAC panel demonstrates structural behaviour equivalent to the defined steel baseline, which sets the minimum required performance. Using experimental data from three-point bending and perimeter shear tests, equivalency factors for bending strength, flexural rigidity, and shear resistance are computed. A panel is compliant when all three factors satisfy $\text{FoS} \geq 1$; otherwise, the layup must be improved.

This requirement operates independently of the applied load cases and may govern the design even where structural loading alone would permit a thinner laminate. The bending, flexural, and shear factors are calculated using Eqs. 4.1–4.3, where t_1 and t_2 are the skin thicknesses, b the core thickness, $y_{\max}$ the ultimate three-point bending

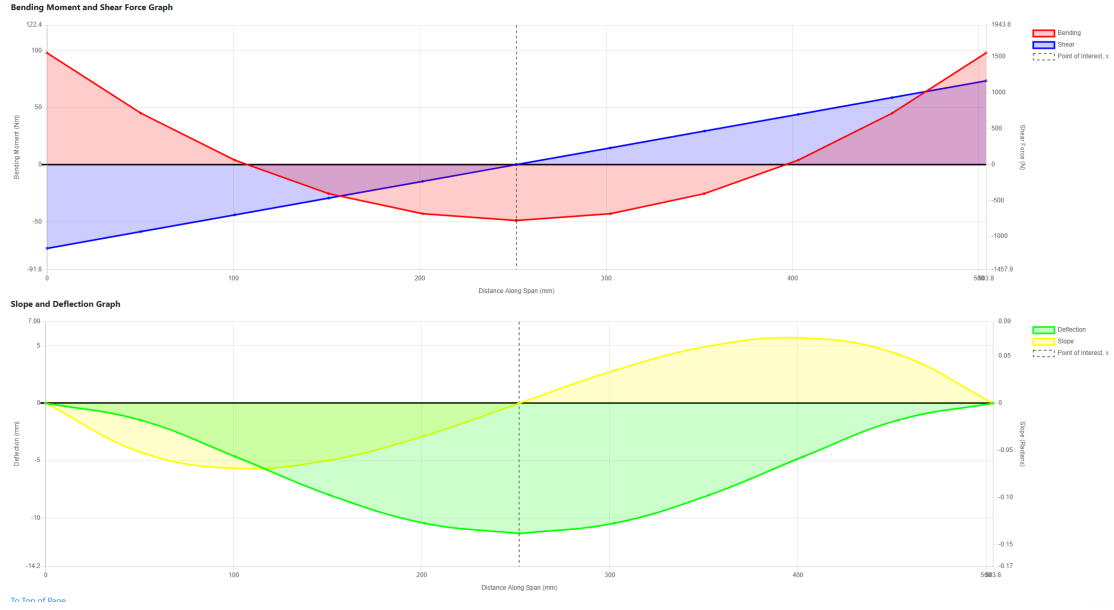


Figure 4.13: Shelf bending moment and displacement distribution [42]

Table 4.11: Shelf load and geometric parameters

Parameter	Value
Module mass [kg]	7.08
Number of modules	6
Peripheral mass [kg]	3.6
Total force [N]	9 041
Length a [mm]	503.6
Length b [mm]	130.0
Distributed force [N]	2 334
Distributed load [N/mm]	4.63
Maximum moment [N m]	97.93
Plate moment [N mm/mm]	753

Table 4.12: Shelf plate dimensioning results

Parameter	Shelf
Load [N]	9 041
Thickness [mm]	4.34
Layup	[0/90] ₁₄
Max stress [MPa]	-240
Displacement [mm]	11.34
Governing criterion	Plate stress

load, L and h the geometric parameters, and (x_1, y_1) , (x_2, y_2) the coordinates used to define the elastic slope. The shear factor uses the peak perimeter shear load $F_{max\ shear}$. The parameters t_b , σ_{yield} , and E_b are the steel baseline constants defined per panel.

$$FoS_{bending} = \frac{(t_1 + t_2)(b + t_1 + t_2) y_{max} L}{h [(b + t_1 + t_2)^3 - b^3]} \cdot \frac{3 \times 10^6}{2 \cdot t_b \cdot \sigma_{yield}} \quad (4.1)$$

$$FoS_{flexural} = \left(\frac{y_2 - y_1}{x_2 - x_1} \right) \left(\frac{L^3}{h} \right) \frac{10^6}{4 \cdot E_b \cdot t_b^3} \quad (4.2)$$

$$FoS_{shear} = \frac{F_{max\ shear}}{5475 \cdot \pi \cdot t_b} \quad (4.3)$$

Each panel was first evaluated using the layup obtained from the load-driven design. The minimum laminate thickness required to achieve $FoS \geq 1$ for each criterion was then determined independently, and the corresponding governing layups are summarised in Table 4.13.

Table 4.13: Minimum laminate requirements and governing layup per panel

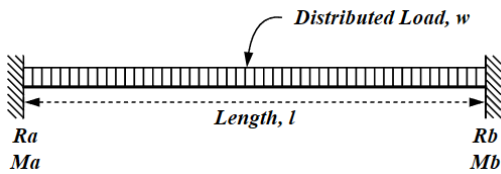
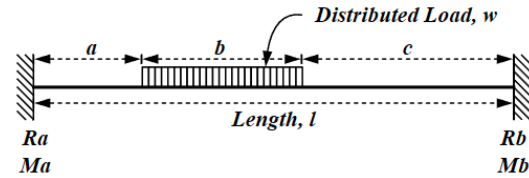
Parameter	Floor (6)	Outer X (5)	Outer Y (1)	Int. wall (2)	Int. wall (3)	Int. wall (4)	Cover (7)
t_{design} [mm]	1.24	2.48	4.65	4.03	3.10	2.48	4.34
Layup _{design}	[0/90] ₄	[0/90] ₈	[0/90] ₁₅	[0/90] ₁₃	[0/90] ₁₀	[0/90] ₈	[0/90] ₁₄
$t_{min, bending}$ [mm]	4.34	3.10	3.10	3.10	3.10	3.10	3.10
FoS _{bending}	1.04	1.04	1.04	1.04	1.04	1.04	1.04
Layup _{bending}	[0/90] ₁₄	[0/90] ₁₀	[0/90] ₁₀	[0/90] ₁₀	[0/90] ₁₀	[0/90] ₁₀	[0/90] ₁₀
$t_{min, shear}$ [mm]	5.27	4.03	4.03	4.03	4.03	4.03	4.03
FoS _{shear}	1.00	1.07	1.07	1.07	1.07	1.07	1.07
Layup _{shear}	[0/90] ₁₇	[0/90] ₁₃	[0/90] ₁₃	[0/90] ₁₃	[0/90] ₁₃	[0/90] ₁₃	[0/90] ₁₃
$t_{min, flexural}$ [mm]	2.79	2.17	2.17	2.17	2.17	2.17	2.17
FoS _{flexural}	1.04	1.12	1.12	1.12	1.12	1.12	1.12
Layup _{flexural}	[0/90] ₉	[0/90] ₇	[0/90] ₇	[0/90] ₇	[0/90] ₇	[0/90] ₇	[0/90] ₇
Governing layup	[0/90] ₁₇	[0/90] ₁₃	[0/90] ₁₅	[0/90] ₁₃	[0/90] ₁₃	[0/90] ₁₃	[0/90] ₁₄
Governing source	Shear rule	Shear rule	Design load	Design load	Shear rule	Shear rule	Design load

The final layup for each panel is taken as the most demanding between the load-driven and rule-based requirements. The shear equivalency criterion is generally the most restrictive, imposing a minimum of [0/90]₁₃ across most panels. The outer wall in the Y direction and the cover are governed by the structural load cases, while the floor panel is governed by the shear requirement, resulting in the thickest laminate at [0/90]₁₇.

4.3 L-Profile Design

The cover is clamped to the outer wall through an L-profile. One flange is bonded to the outer lateral wall, while the other is bolted to the cover with a gasket in between to ensure water sealing.

The axial bolt force was estimated using beam theory. A fully fixed beam under uniformly distributed load was considered in the Y direction (Fig. 4.14), and a partially distributed load in the X direction (Fig. 4.15), reflecting the different contact conditions in each case. The edge moments obtained represent the critical regions at the upper and lower edges, where the structure is clamped by bolts and supported by the adhesive joint respectively. Neglecting the contribution of the side bonded edges - which would increase overall stiffness - yields a conservative estimate.

**Figure 4.14:** Fully fixed beam with UDL [42]**Figure 4.15:** Fully fixed beam with partial UDL [42]

For shear force evaluation, the total wall load was distributed along the perimeter and the resultant corresponding to the upper edge length was used for design. This provides a conservative average load distribution without capturing local peak effects.

The bolted flange was dimensioned through a parametric study varying bolt diameter (M4 and M6) and strength class (8.8, 10.9, and 12.9). Preload was set at 70% of

the proof load, consistent with standard practice for non-permanent joints. Load distribution between bolt and clamped members was evaluated using the stiffness ratio approach. Since the gasket stiffness is significantly lower than the remaining components, joint compliance is dominated by the gasket, yielding $C \approx 1$ - meaning most of the external load is transferred directly to the bolt.

The number of bolts is governed by the minimum gasket seating pressure [27], with an additional safety factor of $\text{FoS} = 2.05$. The L-profile is manufactured in Aluminium 5005, whose ductility allows the use of gussets to limit deformation. The two primary design variables - flange width and thickness - were adjusted iteratively until all failure criteria were satisfied with a factor of safety marginally above unity.

4.3.1 Y Direction

Table 4.14 summarises the load distribution and geometric parameters for each bolt configuration, and Table 4.15 presents the corresponding safety factors across all evaluated failure modes.

Table 4.14: Load distribution and L-profile geometry - Y direction

	M4 8.8	M4 10.9	M4 12.9	M6 8.8	M6 10.9	M6 12.9
Axial force per bay [N]	1524	2345	2735	3048	5268	5927
Shear force per bay [N]	624	891	1040	1248	2080	2080
Maximum preload [N]	3933	5531	6638	9005	12663	15196
Number of bolts	10	7	6	5	3	3
e/D_{FoS}	2.33	2.17	2.17	1.55	1.50	1.33
L-profile width [mm]	28	26	26	28	27	24
L-profile thickness [mm]	1.4	2.0	2.3	1.9	3.1	3.1

Table 4.15: Safety factors for all configurations - Y direction

	M4 8.8	M4 10.9	M4 12.9	M6 8.8	M6 10.9	M6 12.9
Bolt tensile	1.03	1.00	1.01	1.07	1.01	1.03
Bolt shear	8.08	8.83	7.83	8.70	5.44	8.86
Contact pressure	2.00	1.40	1.20	2.00	1.40	1.20
Friction (slip)	3.15	3.10	3.19	3.61	3.04	3.65
L-profile bearing	1.04	1.04	1.02	1.04	1.02	1.02
L-profile tear-out	11.06	10.13	9.98	6.87	6.42	5.49
Glue joint	10.40	9.66	9.66	10.40	10.03	8.91

The design is governed by bolt tensile stress and flange bearing, consistent with the stiffness ratio result showing that most external load is carried directly by the bolts. Tear-out and adhesive failure exhibit substantially higher safety margins and are non-critical. Friction capacity is sufficient to prevent slip in all configurations, confirming that shear transfer is dominated by bolt tension rather than friction.

Increasing bolt diameter from M4 to M6 reduces the bolt count without significantly improving the governing safety factors, since the limiting failure modes re-

main unchanged. Higher strength classes provide only marginal benefit, as they tend to shift the critical condition toward contact pressure and bearing. From a practical standpoint, fewer bolts simplify assembly and reduce weight, making M6 configurations preferable. The **M6 8.8 configuration with five bolts** was selected as the most suitable solution, providing the best balance between structural performance and assembly practicality.

4.3.2 X Direction

Tables 4.16 and 4.17 present the load distribution, geometric parameters, and safety factors for the X direction configurations.

Table 4.16: Load distribution and L-profile geometry - X direction

	M4 8.8	M4 10.9	M4 12.9	M6 8.8	M6 10.9	M6 12.9
Axial force per bay [N]	1685	1416	2831	3725	3725	3725
Shear force per bay [N]	239	239	477	477	477	477
Maximum preload [N]	3933	5531	6638	9005	12663	15196
Number of bolts	2	2	1	1	1	1
e/D	2.6	3.1	3.1	1.6	1.6	1.6
L-profile width [mm]	21	25	25	19	19	19
L-profile thickness [mm]	0.6	0.6	1.1	0.7	0.7	0.7

Table 4.17: Safety factors for all configurations - X direction

	M4 8.8	M4 10.9	M4 12.9	M6 8.8	M6 10.9	M6 12.9
Bolt tensile	1.0	1.1	1.0	1.0	1.1	1.1
Bolt shear	21.1	33.0	21.1	32.7	34.2	72.2
Contact pressure	1.9	1.6	1.1	1.9	1.5	1.3
Friction (slip)	8.2	11.6	7.0	9.4	13.3	15.9
L-profile bearing	1.2	1.2	1.1	1.0	1.0	1.0
L-profile tear-out	4.6	5.8	5.3	2.1	2.1	2.1
Glue joint	17.9	21.3	21.3	16.2	16.2	16.2

The governing failure modes are consistent with the Y direction - bolt tensile stress and flange bearing remain critical, while shear-related modes and friction are non-critical across all configurations. The key difference is that the number of bolts per bay is constrained by the module width, excluding configurations that require more bolts than the available span permits. Flange dimensions are reduced relative to the Y direction, resulting in lower bearing and tear-out margins, though both remain acceptable. The **M4 12.9 configuration** was the most balanced solution for this direction, minimising bolt count while satisfying all strength requirements.

4.4 Mountings Dimensioning

4.4.1 Mounting Concept

The TSAC is attached to the vehicle chassis through a bolted mounting system, designed to satisfy structural, regulatory, and serviceability requirements while allowing the battery pack to be removed from underneath the vehicle by jack after unbolting.

Rule EV 5.5.15 requires a minimum of two attachment points. An initial four-corner configuration was considered; however, the relatively large longitudinal span was expected to produce excessive bending of the TSAC base structure. Two additional mounting points were therefore introduced along the longer edges, resulting in six mountings in total. This configuration improves load distribution, reduces bending stresses, and increases overall structural stiffness. Mountings 1, 3, 4, and 6 are referred to as corner mountings, while mountings 2 and 5 are the middle mountings, as illustrated in Fig. 4.16.

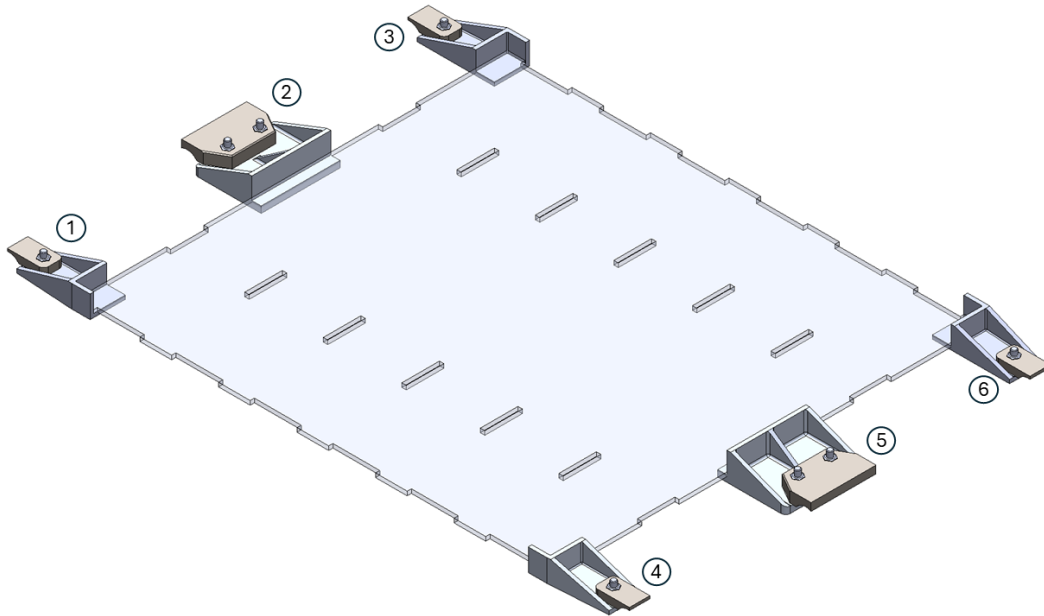


Figure 4.16: TSAC-to-chassis mounting points (numbered)

4.4.2 Load Case Definition

The TSAC base was modelled as a beam on multiple supports under a distributed load, providing a straightforward estimation of the reaction forces at each mounting while maintaining analytical accuracy. In multi-support configurations, intermediate supports carry higher reaction forces than corner supports. Based on standard solutions for beams on multiple supports (Fig. 4.17), distribution coefficients of 0.375 and 1.25 were applied to corner and middle mountings respectively.

In the X direction, the load is assumed to be uniformly distributed across all mountings, as they act in series along this axis. In the Y and Z directions, the multi-support

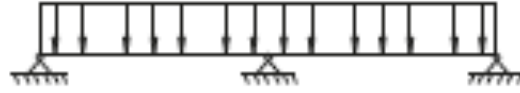


Figure 4.17: Beam on multiple supports with distributed load [43]

distribution is applied. Since mountings are present on both sides of the pack, the total load is first divided equally between each side before the distribution coefficients are applied. The resulting force distributions are illustrated in Fig. 4.18.

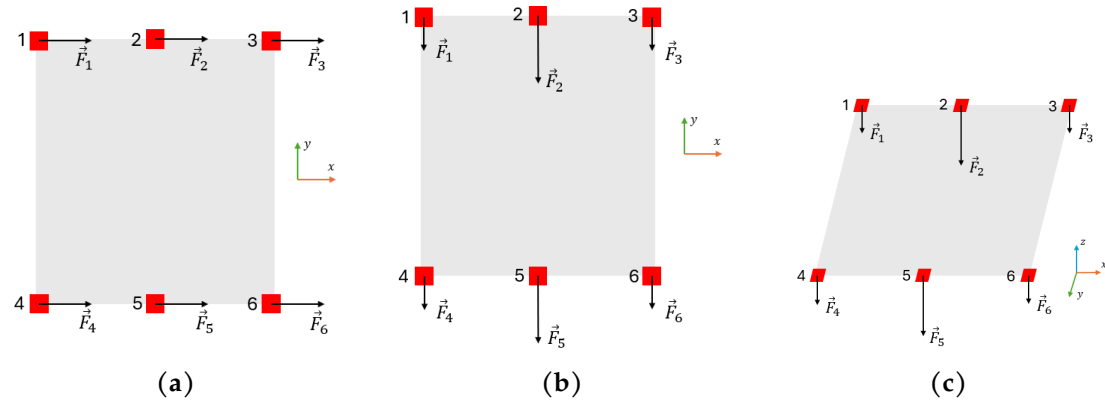


Figure 4.18: Load distribution in (a) X direction, (b) Y direction, and (c) Z direction

The total battery pack mass is estimated at 60 kg, including modules, peripherals, and enclosure. The resulting forces at each mounting point are summarised in Table 4.18.

Table 4.18: Forces per mounting point [N]

Mounting	F_x	F_y	F_z
1, 3, 4, 6	3924	2207	1104
2, 5	3924	7358	3679

4.4.3 Bolt Dimensioning

Preliminary evaluation showed that smaller diameter bolts, even at higher strength classes, produced insufficient safety factors under the shear-dominated loading conditions. M6 class 8.8 bolts were selected, consistent with the fastening solution used for the cover-to-wall connection, limiting the number of distinct fastener types in the assembly.

Middle mountings (2 and 5) use two bolts per point to improve load distribution and reduce the force carried by each fastener. Corner mountings use a single bolt; rotational freedom at these locations is mitigated by ensuring adequate shear resistance through bolt preload friction and adhesive contribution, preventing significant rotation and avoiding prying effects on the adhesive layer.

Due to the high stiffness of the mounting components relative to the gasket, only approximately 10% of the external load is transferred to the bolt - a result consistent

with the stiffness ratio approach. This low load fraction makes the bolt response insensitive to variations in joint stiffness, confirming the suitability of the selected configuration without iterative refinement. Preload was calculated using Eq. 2.39, and the corresponding tightening torques were determined assuming a static friction coefficient of $\mu = 0.61$ for dry aluminium-to-steel contact [44], yielding 4.82 Nm for middle mountings and 5.41 Nm for corner mountings.

The bolt analysis results are summarised in Table 4.19.

Table 4.19: Bolt analysis results - TSAC mountings

Parameter	Middle Mounting			Corner Mounting		
	X	Y	Z	X	Y	Z
Preload F_i [N]	9005					
FoS (Shear friction)	1.49	2.80	–	2.49	1.40	–
Load factor C	0.09	0.09	0.09	0.10	0.10	0.10
Tensile stress [MPa]	448	448	456	448	448	453
FoS (Tensile)	1.43	1.43	1.40	1.43	1.43	1.41
Shear stress [MPa]	130	69	–	78	139	–
FoS (Shear)	2.95	5.53	–	4.92	2.77	–

4.4.4 Mounting Dimensioning

The mounting geometry was developed considering regulatory constraints, material selection, geometric limitations, and multiple failure criteria.

Rule T 10.1.4 imposes a minimum edge distance ratio of $e/D = 1.5$, and rule EV 5.5.15 defines minimum thickness requirements per material, establishing the baseline geometry. Aluminium 7075-T6 was selected for the TSAC-side mountings due to its high strength-to-weight ratio, while AISI P20 steel was used for the chassis-side brackets to ensure weld compatibility. A full steel solution was considered initially but was discarded due to the weight penalty associated with the increased thickness required under bending-dominated loading.

The mounting length on the TSAC side is constrained by the requirement to remove the pack from underneath the vehicle. Aluminium mountings are therefore extended to connect to welded steel brackets on the chassis, ensuring adequate load transfer across the interface.

Load transfer to the chassis brackets was analysed using a rectangular fillet weld model under bending [45], with applied loads modelled as partially distributed along the mounting length, as illustrated in Fig. 4.19, and evaluated using standard cantilever solutions under partial UDL.

Structural verification results are presented in Tables 4.20, 4.21, and 4.22, covering local failure modes at the TSAC-side mountings, adhesive joint performance, and weld integrity at the chassis-side brackets respectively.

All failure criteria are satisfied across both mounting types. The most critical conditions are the edge distance ratio (e/D) and local bending at the corner mountings,

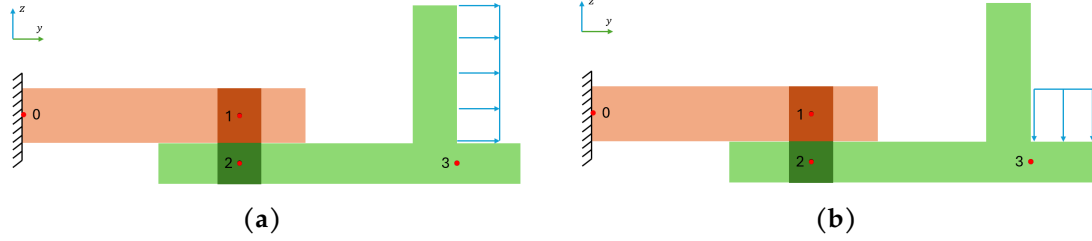


Figure 4.19: Mounting load distribution in (a) Y direction and (b) Z direction

Table 4.20: Local failure analysis - TSAC-side mountings

Parameter	Middle Mounting			Corner Mounting		
	X	Y	Z	X	Y	Z
Hole diameter [mm]		6.4			6.4	
Thickness [mm]		7			4	
Free edge distance [mm]	19	12	–	17	2	–
FoS (Tear-out)	18.92	11.05	–	9.28	3.16	–
FoS (e/D)	2.01	1.25	–	1.77	1.25	–
FoS (Bearing)	7.06	3.77	–	2.02	1.08	–
FoS (Contact pressure)		11.85			11.85	
FoS (Open ratio)	–	127	54	–	232	91
FoS (Bending - point 3)	–	2.36	–	–	2.49	–
FoS (Bending - point 2)	24.97	2.48	1.16	23.57	2.97	1.35

Table 4.21: Adhesive joint safety factors - mounting points (total mass = 60 kg)

Mounting	Area [mm ²]	FoS X	FoS Y	FoS Z
1, 3, 4, 6	1720	2.22	2.22	4.44
2, 5	2805	3.62	1.93	3.86

Table 4.22: Local failure and weld analysis - chassis-side brackets

Parameter	Middle Mounting			Corner Mounting		
	X	Y	Z	X	Y	Z
Hole diameter [mm]		6.4			6.4	
Thickness [mm]		12			7	
Free edge distance [mm]	12.3	12	–	11	11	–
FoS (Tear-out)	13.86	7.15	–	6.93	3.70	–
FoS (e/D)	1.28	1.25	–	1.15	1.15	–
FoS (Bearing)	4.57	2.44	–	2.67	2.23	–
FoS (Contact pressure)	2.32	2.32	–	2.32	2.32	–
FoS (Bending - point 0)	21.59	2.95	1.16	8.53	3.38	1.10
FoS (Weld)	5.74	3.53	1.38	1.16	3.71	1.21

where the minimum factor of safety approaches unity. Adhesive joints show lower margins in the Y direction for the middle mountings, though still within acceptable limits. The estimated total mass of the mounting system, excluding weld seams, is 1 259 g.

4.5 Analytical Design Conclusions

The analytical dimensioning defined laminate configurations for each TSAC panel based on load-driven and rule-based constraints. The resulting layups are summarised in Table 4.23.

Table 4.23: Summary of laminate configurations per panel

Component	Thickness [mm]	Layup
Internal wall 2	4.03	[0/90] ₁₃
Internal wall 3	3.10	[0/90] ₁₀
Internal wall 4	2.48	[0/90] ₈
Outer wall (Y)	4.65	[0/90] ₁₅
Outer wall (X)	2.48	[0/90] ₈
Floor	5.27	[0/90] ₁₇
Cover	4.34	[0/90] ₁₄
Shelf	4.34	[0/90] ₁₄

Laminate thickness is primarily governed by adhesive shear stress and rule-based shear requirements, establishing a baseline of [0/90]₁₃ for most panels, as seen in table 4.13. Certain components require additional plies due to bending and deflection constraints, while the floor is governed by the shear equivalency rule, resulting in the thickest configuration at [0/90]₁₇.

To simplify manufacturing and reduce the number of configurations requiring experimental validation, a unified [0/90]₁₆ layup is adopted. This configuration exceeds the minimum requirement of [0/90]₁₃ for all panels, approaches the outer wall Y requirement of [0/90]₁₅, and marginally reduces the conservatism of the floor design, where the [0/90]₁₇ requirement is driven by matrix-dominated shear assumptions. The [0/90]₁₆ configuration is therefore selected as the reference layup for subsequent numerical and experimental validation.

Joint and Assembly

The bolted cover-to-wall connection was refined considering both structural performance and practical constraints. In the Y direction, the analytically selected configuration of five M6 bolts per side was found to produce an excessively large bolt pitch, introducing uncertainty in gasket sealing continuity. The number of fasteners was therefore increased to eight per side, improving contact pressure uniformity and reducing the risk of local separation under dynamic loading. The associated mass penalty is approximately 150 g - negligible relative to the overall mass budget - and is considered an acceptable trade-off for improved sealing reliability.

Regarding L-profile selection, standard available profiles were used as the basis for the final geometry. In the Y direction, a 40 mm × 40 mm × 3 mm aluminium profile was selected, exceeding minimum requirements and increasing safety margins. In the X direction, a 20 mm × 20 mm × 3 mm profile was used; although narrower than analytically required, compliance is maintained by reducing the bolt preload from

6.6 kN to 5.9 kN, preserving adequate shear resistance ($\text{FoS} \approx 6.2$) and gasket contact pressure.

Design Requirements Compliance

The design satisfies the main system requirements established in Chapter 3:

- **Electrical insulation** - provided by the aramid composite material; grounding is achieved through metallic fasteners;
- **Fire resistance** - ensured by UL94-V0 compliant resin and gasket materials;
- **Thermal management** - airflow is promoted through openings in the longitudinal walls, within the maximum allowable cut-out area of 25%;
- **Serviceability** - the six-point bolted mounting concept allows removal of the TSAC from underneath the vehicle, compatible with the 5-minute target.

Mass Evaluation

The mass implications of the three laminate configurations - optimised analytical, minimum rule-based, and unified $[0/90]_{16}$ - are compared in Table 4.24.

Table 4.24: Mass comparison for TSAC wall configurations

Component	Qty	Area [mm ²]	Design Mass [g]	Min. Layup Mass [g]	Final Mass [g]
Floor (6)	1	233 144	359	1527	1437
Outer Y (1)	2	61 490	380	616	758
Outer X (5)	2	71 066	822	822	876
Int. (2)	2	70 827	710	710	874
Int. (3)	2	70 827	546	710	874
Int. (4)	1	70 827	218	355	437
Cover (7)	1	412 364	2225	2225	2542
Shelf	1	72 696	392	392	448
Total	–	–	5652	7357	8246

The optimised analytical configuration yields a structural wall mass of 5.65 kg. Applying minimum rule-based laminates increases this to 7.36 kg, and the unified $[0/90]_{16}$ configuration results in 8.25 kg. Including the aluminium mounting brackets (567 g) and the cover attachment system (1 229 g), the total estimated TSAC mass is approximately **10.05 kg**, exceeding the initial target of 8.46 kg defined in Chapter 3. This difference is primarily driven by the shear equivalency rule, which imposed a minimum layup of $[0/90]_{17}$ on the floor panel, and by the decision to adopt a uniform $[0/90]_{16}$ layup across all panels to ensure manufacturing consistency, experimental validation coverage, and structural safety margins. While the mass target was not met, the design prioritised regulatory compliance, structural reliability, and in-house manufacturability - constraints that collectively justify the additional mass as an acceptable trade-off for a first-generation composite TSAC.

NUMERICAL IMPLEMENTATION

Finite element models were developed to complement and verify the analytical wall dimensioning, addressing the limitations inherent to closed-form solutions - namely idealised boundary conditions, homogenised material behaviour, and the inability to capture local effects such as stress concentrations and contact interactions. Two modelling stages are considered: models replicating the analytical assumptions to quantify the deviation introduced by applying isotropic plate theory to orthotropic laminates, and models incorporating realistic boundary conditions to assess the actual structural response of each panel. All simulations were performed in Siemens NX Nastran.

5.1 Tensile Simulation

A tensile simulation was performed to verify that the material model produces results consistent with the input properties before applying it to more complex structural analyses. Shell elements were adopted under a plane stress assumption, appropriate given that laminate thickness is small relative to in-plane dimensions and no through-thickness stresses are expected. Linear quadrilateral elements (CQUAD4) were selected, providing sufficient accuracy for a uniaxial loading condition with a near-uniform stress field; a mesh convergence study was therefore not required.

The specimen geometry follows ASTM D3039 [28] at 250×25 mm, with material properties from Table 3.5 and the full $[0/90]_{16}$ stacking sequence. One end was fully constrained and a tensile load of 15 kN applied at the opposite end, within the elastic regime. Stress and strain were extracted as average values at the fixed end and compared against the analytical prediction using Hooke's law, as shown in Figs. 5.1 and 5.2 and summarised in Table 5.1.

Table 5.1: *Tensile simulation - analytical vs. FEM*

Parameter	Analytical	FEM	Deviation [%]
Stress σ [MPa]	120.97	121.04	+0.06
Strain ε [-]	0.00606	0.00611	+0.84
Elastic modulus E [MPa]	19 958	19 953	-0.02

All three parameters show deviations below 1%, confirming that the material model

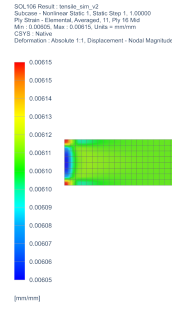


Figure 5.1: Strain distribution - tensile coupon

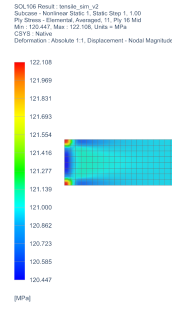


Figure 5.2: Stress distribution - tensile coupon

is consistent with the input properties. The small residual discrepancy in strain is attributed to the limited numerical precision available when extracting the elastic modulus from the software output for this simplified problem, and has no practical impact on the results. The model is therefore considered validated for application to the wall simulations that follow.

5.2 Three-Point Bending

Two models were developed for the three-point bending analysis: a shell-based model, implemented for its simplicity and low computational cost, and a solid laminate model using 3D elements, providing through-thickness stress information and more realistic contact and boundary condition representation. Both models use a rectangular laminate of 275×150 mm, supported by two cylindrical rollers and loaded by a central punch, following the configuration defined by rule T 3.6.1. The laminate thickness of 4.96 mm corresponds to the $[0/90]_{16}$ stacking sequence adopted throughout the design.

5.2.1 Shell Model

The laminate was modelled using quadratic shell elements (CQUAD4), while the supports and punch were represented using CHEXA8 solid elements to ensure adequate geometric representation and contact stability. An enforced displacement of 15 mm was applied under a multi-step nonlinear analysis with a friction coefficient of 0.1. To achieve convergence, in-plane motion of the laminate was constrained in the X and Y directions, since the structural response of interest is in the Z direction (Fig. 5.3). The supports were fixed throughout.

The solution converged successfully. Due to the homogenised laminate representation (PSHELL), ply-level stresses were not directly available and were reconstructed in post-processing for the Tsai–Wu failure assessment. Failure is governed by the outer plies, consistent with bending-dominated behaviour (Fig. 5.4). The top surface is critical, with a failure index of 4.9, while the bottom surface reaches 4.38 - the difference reflecting the lower compressive strength of the material relative to its tensile strength.

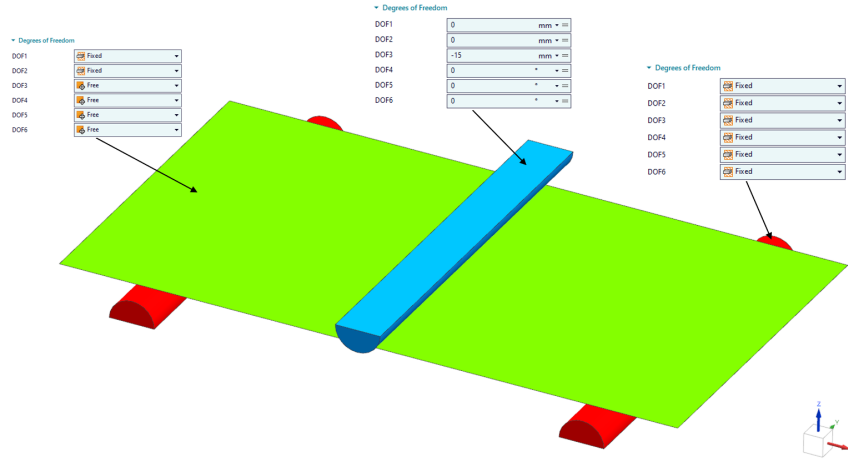


Figure 5.3: *Shell model boundary conditions - three-point bending*

These results, however, are strongly influenced by the constrained boundary conditions imposed for convergence, which limit the physical representativeness of the shell model. A solid model was therefore developed to provide a more reliable assessment with direct ply-level output.

SOL 402 Result : Sheet_3pb_fem1_non-linear-analysis
 Subcase : Nonlinear Statics 1, Increment 20, 1.00s
 Stress [PK2] - Element-Nodal, Averaged, XX
 Shell Section : Top and Bottom
 Min : -668.56, Max : 723.22, Units = MPa
 CSYS : Absolute Rectangular
 Deformation : Absolute 1:1, Displacement - Nodal Magnitude

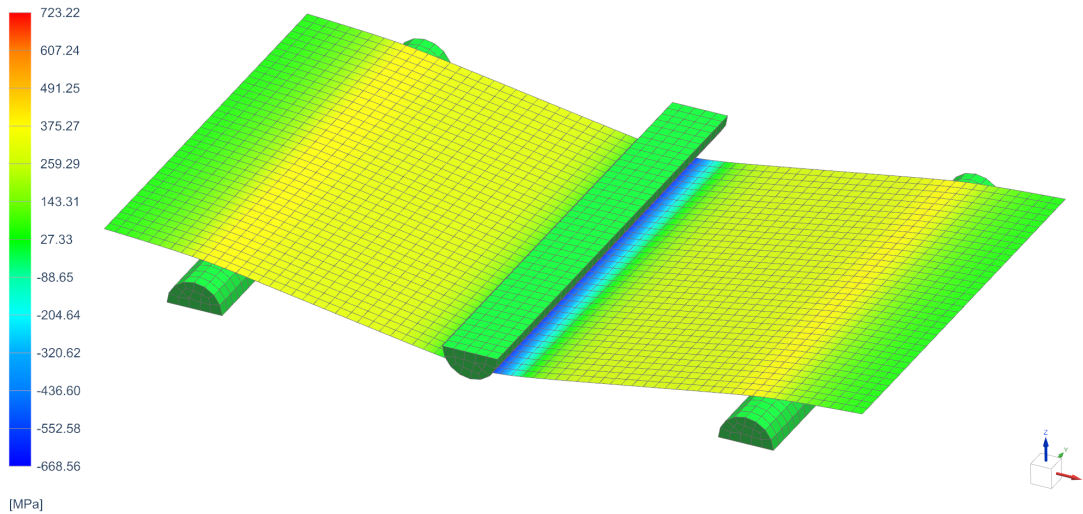


Figure 5.4: *Stress distribution in XX direction - shell model, top and bottom surfaces*

5.2.2 Solid Model

The solid model represents the laminate using 3D elements with the same stacking sequence (Fig. 5.5), with mesh parameters consistent with the shell model. Unlike the shell model, the laminate was free to slide in the X and Y directions, enabling more realistic contact behaviour - although the increased number of degrees of freedom in-

troduces greater contact instability and significantly higher computational cost. Friction was set to zero to mitigate additional convergence difficulties arising from this complexity (Fig. 5.6).

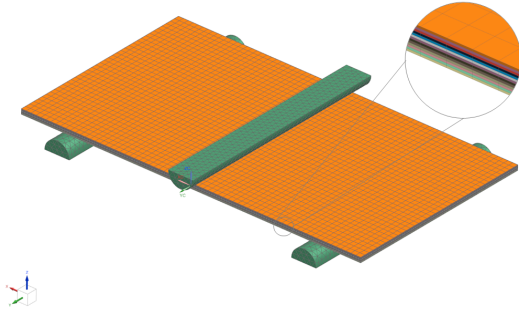


Figure 5.5: Solid model mesh - three-point bending

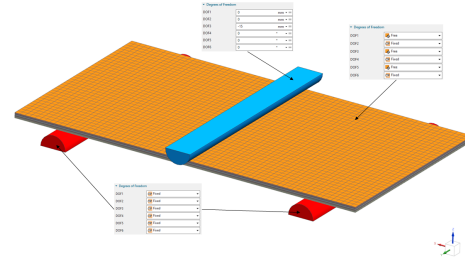


Figure 5.6: Solid model boundary conditions - three-point bending

Ply-level stresses are extracted directly, confirming that the highest stresses occur at the outer plies in both tension and compression, consistent with the shell model and the analytical prediction (Figs. 5.7 and 5.8).

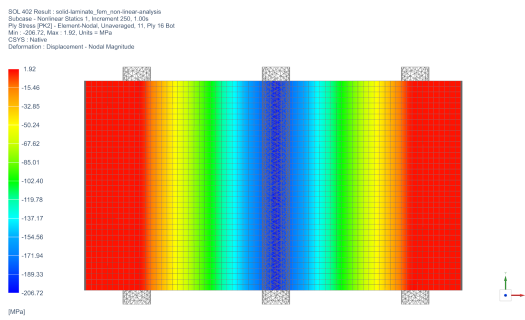


Figure 5.7: Stress in XX direction - top surface

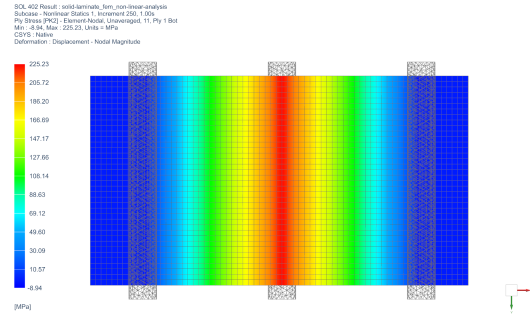


Figure 5.8: Stress in XX direction - bottom surface

5.2.3 Conclusions

The three model approaches are compared in Table 5.2 at a displacement of 15 mm.

Table 5.2: Comparison between analytical, shell, and solid models - three-point bending at $\delta = 15$ mm

Parameter	Analytical	Shell	Δ [%]	Solid	Δ [%]
Contact force [N]	2740	2058	-24.9	2821	+3.0
Stress - bottom [MPa]	223	723	+224	225	+1.0
Stress - top [MPa]	-223	-668	+200	-243	+9.0
Tsai-Wu index [-]	0.68	4.92	-	0.77	-
Factor of safety [-]	1.21	0.45	-63	1.14	-5.7

The solid model shows close agreement with the analytical solution in contact force (+3%) and stress (+1% at the bottom ply), and provides direct ply-level failure assessment without post-processing. The Tsai-Wu index of 0.77 at 15 mm is consistent with

the analytical prediction of 0.68, both indicating that failure has not yet occurred at this displacement level.

The shell model produces significant discrepancies in both stress (+200 and +224%) and failure index (4.92 vs. 0.68), driven primarily by the constrained in-plane boundary conditions required for convergence, which artificially increase laminate stiffness. The homogenised formulation also prevents direct ply-level evaluation, requiring additional post-processing that introduces further uncertainty. The shell model is therefore not suitable for this analysis.

Given a runtime of 2 minutes for the shell model against 25 hours for the solid model, the solid approach carries a significant computational cost - however, given its accuracy and the critical nature of the failure assessment, it is the appropriate tool for this application. For rapid design iteration, the analytical solution provides a reliable and consistent alternative.

5.3 Perimeter Shear Strength

The perimeter shear strength model was developed using a 100×100 mm plate geometry, following the dimensions prescribed by rule T 3.6.11 rather than the ASTM D732-02 standard [30], ensuring direct compatibility between the numerical model and the experimental setup. The laminate corresponds to the $[0/90]_{16}$ configuration, totalling 4.96 mm thickness.

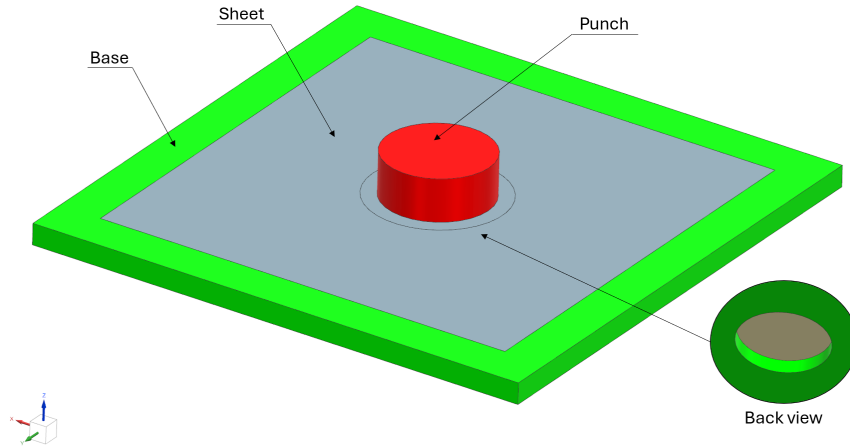


Figure 5.9: *Perimeter shear strength FEM model*

Since interlaminar shear is the governing failure mode, solid elements are required to capture through-thickness stress gradients. Three modelling strategies were pursued in sequence with progressively simplified contact conditions, none of which produced physically meaningful results.

Full contact model. The laminate was discretised with CQUAD8 shell elements and the punch and support base with CTETRA10 solid elements (Fig. 5.9), with an enforced displacement of 2 mm applied to the punch. The laminate was constrained

against out-of-plane rotation, though this constraint is ineffective for solid elements since the solver does not apply moments to solid elements. Despite iterative adjustment of contact parameters, stabilisation, and relaxation settings, the nonlinear solver failed to converge and no results were obtained.

Simplified contact model. The support base was removed and idealised constraints applied directly to the laminate edges to isolate the punch-to-laminate contact as the primary source of instability. The solution converged, confirming that the high node count at the laminate-support interface was a contributing factor. However, the results were not usable: stresses were artificially elevated at the punch edge due to contact singularities, and the reaction force was redistributed to the constrained edges rather than the punch interface, preventing any meaningful failure prediction.

Contact-free model. All contact definitions were removed and the load applied as a surface pressure with the laminate fully constrained at its boundary. Analytically, using Eq. 2.49, the applied load of 20 257 N should produce a failure index approaching unity for a matrix-dominated shear strength of 52 MPa, a punch diameter of 25 mm, and a plate thickness of 4.96 mm. Instead, interlaminar shear stresses and reaction forces diverged by factors of hundreds, consistent with the artificial stiffness imposed by the rigid boundary conditions. The results are not physically meaningful.

Cohesive element formulations [46] were also explored as an alternative means of modelling the through-thickness failure mechanism. However, these introduced additional complexity in both setup and material definitions - requiring explicit interface properties between fibre layers - and either failed to converge or produced results as unreliable as the previous approaches.

The fundamental difficulty lies in the highly localised punch penetration failure mode, which involves severe contact nonlinearity and material instability at the point of failure - conditions that implicit nonlinear solvers handle poorly. Explicit solvers are better suited to this class of problem but fall outside the scope and computational resources of the present work. Numerical simulation of the perimeter shear test is therefore not pursued further, and experimental results are used directly for the ASES equivalency verification.

5.4 Walls Simulation

Finite element models were developed to complement the analytical dimensioning presented in the previous chapter. Two levels of modelling are considered: first, models replicating the analytical assumptions as closely as possible, used to quantify the deviation introduced by applying isotropic plate theory to orthotropic composite laminates; and second, models incorporating realistic boundary conditions and load introduction, used to assess the actual structural response of each panel. Throughout both stages, shell elements (CQUAD8) were used, providing an accurate and efficient representation of bending-dominated behaviour in thin-walled structures.

5.4.1 Analytical Condition Models

To assess the applicability of the closed-form analytical solutions to composite panels, FEM models were constructed to mirror the analytical assumptions: uniform pressure loading, idealised boundary conditions (S for simply supported, C for clamped), and the same geometry defined in the design phase. A uniform element size of 2 mm was adopted across all panels.

Among the developed configurations of the internal walls, the $[0/90]_{10}$ layup was identified as the governing case, presenting the highest load-to-thickness ratio of 3585 N/mm, where for the $[0/90]_{13}$ and $[0/90]_8$ layups, settle on 3447 and 3361 N/mm, respectively. The remaining internal wall configurations share identical geometry, loading, and boundary conditions, differing only in laminate thickness, and are therefore represented by this case to avoid repetition.

Stresses were evaluated at the plate centre, consistent with the analytical formulation. Adhesive shear stress was estimated indirectly by extracting the resultant reaction force over the bonded region and dividing by the bonded area, defined as the product of laminate thickness and element length.

Table 5.3 summarises the geometry, loading, and structural response for all panels under analytical conditions, acquired in chapter 4.1.

Table 5.3: Summary of analytical condition FEM models for TSAC panels

Category	Parameter	Internal	Outer Y	Outer X	Floor	Cover
Geometry	a [mm]	457.2	457.2	153.6	457.2	504.8
	b [mm]	153.6	153.6	80.4	80.35	457.2
	Layup	$[0/90]_{10}$	$[0/90]_{15}$	$[0/90]_8$	$[0/90]_4$	$[0/90]_{14}$
	BC	SSSS	SSSS	SSSS	SSSS	CCCC
	Elements [-]	17 633	17 633	3 080	9 160	57 708
	Load [N]	11 113	16 669	2 778	1 389	9 041
Response	Displacement [mm]	9.49	4.24	1.98	2.65	23.01
	Glue shear [MPa]	2.01	2.01	1.90	0.61	–
Stress	Top ply [MPa]	-85.2	-60.1	-50.5	-65.8	-71.7
	Bottom ply [MPa]	183.6	125.7	109.7	25.6	71.7

Displacement

Table 5.4 compares analytical large and small deflection predictions against non-linear FEM results.

The large deflection formulation shows inconsistent correlation with FEM, with deviations ranging from -71% to $+143\%$. This scatter is physically attributable to the Poisson's ratio assumed in the Roark tables ($\nu = 0.316$) versus the effective laminate value of $\nu \approx 0.09$. Since membrane stiffening in large deflection theory is governed by biaxial coupling, the analytical formulation overestimates stiffening and consequently under-predicts displacement relative to FEM. This was confirmed through a controlled

Table 5.4: Displacement comparison - analytical models vs. nonlinear FEM

Panel	FEM [mm]	Large def. [mm]	Δ [%]	Small def. [mm]	Δ [%]
Internal	9.49	3.84	−60	19.78	+108
Outer Y	4.24	3.24	−24	7.31	+72
Outer X	1.98	4.81	+143	3.42	+72
Floor	2.65	1.54	−42	5.89	+123
Cover	23.01	6.68	−71	64.60	+181

isotropic test case, where a 31% discrepancy persisted with composite effects fully eliminated, isolating Poisson’s ratio as the governing parameter.

The small deflection formulation, while technically invalid for these panels since all cases exceed the $w/h > 0.3$ threshold, consistently over-predicts displacement across all panels. From a design standpoint this is a conservative outcome, and small deflection theory can therefore serve as a reliable upper-bound displacement estimator even beyond its formal validity range.

Plate Stress

Table 5.5 compares analytical stress predictions against FEM for the bottom ply, which represents the critical tensile face.

Table 5.5: Bottom ply stress comparison - analytical models vs. FEM

Panel	FEM [MPa]	LD [MPa]	Δ [%]	SD [MPa]	Δ [%]
Internal	183.6	84	−54	277	+51
Outer Y	125.7	89	−29	185	+47
Outer X	109.7	101	−8	248	+126
Floor	25.6	49	+91	119	+365
Cover	71.7	35	−51	164	+129

The large deflection stress formulation under-predicts FEM stress in four out of five panels, with deviations ranging from −54% to +91%. The exception is the Floor panel, where the analytical value exceeds FEM by 91%, likely driven by the panel’s low b dimension relative to its high aspect ratio. Critically, the large deflection formulation is non-conservative for most panels, meaning that a design based solely on this criterion would underestimate the actual laminate stress.

The small deflection formulation, by contrast, over-predicts stress in all panels, making it the conservative and therefore more appropriate criterion for structural dimensioning.

Adhesive Shear

Adhesive shear predictions are consistently conservative, over-predicting FEM values by 41% to 59%. This conservatism arises because the analytical formulation assumes a perfectly rigid boundary, whereas FEM redistributes load and reduces peak

Table 5.6: *Adhesive shear comparison - analytical models vs. FEM*

Panel	Analytical [MPa]	FEM [MPa]	Δ [%]
Internal	4.90	2.01	-59
Outer Y	4.90	2.01	-59
Outer X	3.21	1.90	-41
Floor	1.23	0.61	-50

reactions. The analytical adhesive shear methodology is therefore a reliable and safe basis for joint dimensioning under idealised conditions.

5.4.2 Real Condition Models

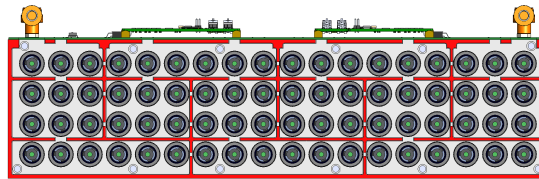
To better represent the actual structural response, more detailed models were developed incorporating realistic load introduction, geometric features, and boundary conditions. Although the full TSAC assembly is not modelled, these panel-level models provide a significantly improved representation of in-service behaviour compared to the analytical condition models.

Relevant geometric features - such as cut-outs in the outer wall in the X direction - are explicitly included to evaluate their influence on stress distribution and potential stress concentrations. Each panel presents distinct structural behaviour due to differences in geometry, load introduction, and support conditions, and is therefore evaluated individually.

All models were discretised using CQUAD8 shell elements. Mesh convergence was assessed based on displacement and maximum principal stress in the outer plies, which govern bending behaviour, being ply 1 the layer subjected to tensile stress. To improve robustness and reduce local numerical noise, results are reported as the average of the top 1% of elements by stress magnitude. A nonlinear solver was employed throughout, providing improved accuracy for the material and deformation levels encountered.

Internal Wall

The internal wall model incorporates the actual contact area between the module and the wall surface (Fig. 5.10), as well as a cut-out corresponding to the maintenance plug, which may introduce local stress concentrations.

**Figure 5.10:** *Module contact area on the internal wall*

The wall fitments are modelled as clamped, reflecting their mechanical constraint

within the structure. The remaining edges are defined with fixed translations and free rotations, approximating the behaviour of an adhesive joint. The load is applied uniformly over the module contact area (Fig. 5.11).

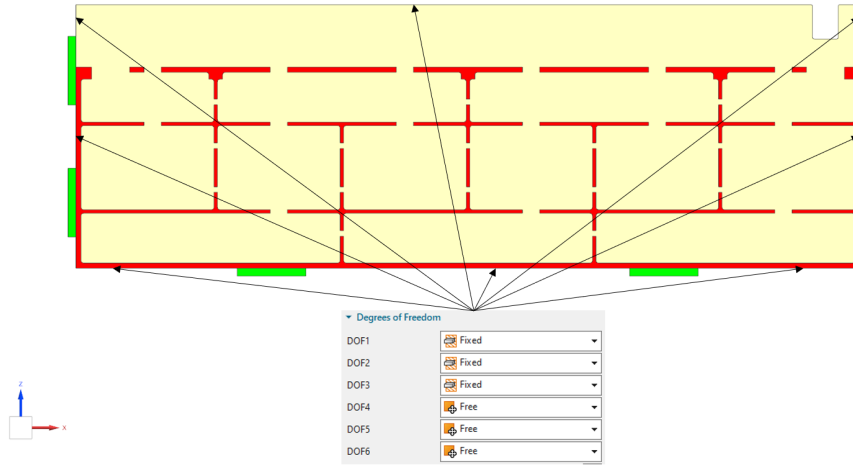


Figure 5.11: Load distribution and boundary conditions - internal wall

A mesh of 78 309 elements was selected based on convergence analysis (Table 5.7), providing stable stress and displacement results with negligible variation between refinement levels.

Table 5.7: Mesh convergence - internal wall

Elements	Ply 1 [MPa]		Ply 10 [MPa]		u_z [mm]	
	Value	Δ [%]	Value	Δ [%]	Value	Δ [%]
12 748	100.3	—	200.3	—	9.88	—
16 252	100.0	-0.2	200.0	-0.1	9.88	-0.01
24 723	99.9	-0.1	199.4	-0.3	9.87	-0.02
38 510	100.2	+0.3	198.1	-0.6	9.87	-0.05
78 309	100.0	-0.3	195.8	-1.2	9.85	-0.15

Outer Wall - X Direction

Taking advantage of geometric symmetry, only a representative portion of the outer wall in the X direction was modelled, corresponding to a load-sharing condition between adjacent internal walls. The total load of 2 778 N is applied through two contact strips, each carrying 1 389 N, consistent with the module interface geometry.

The top edge, representing the bolted connection to the cover, is modelled as clamped, as are the lateral supports. The remaining edges are constrained in translation with free rotations, approximating adhesive behaviour (Fig. 5.12).

A mesh of 22 361 elements at 0.7 mm element size was selected based on convergence results (Table 5.8), ensuring sufficient accuracy for both stress and displacement.

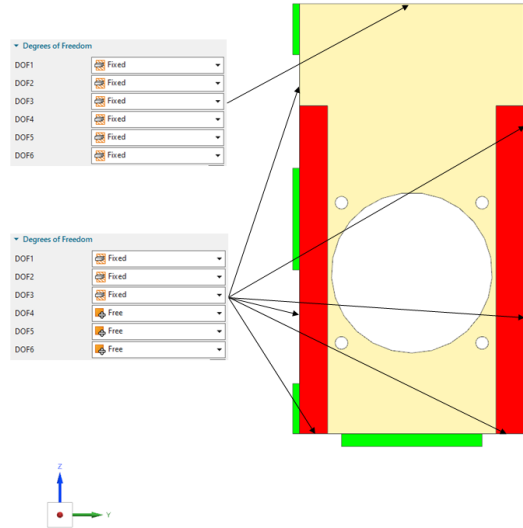


Figure 5.12: Boundary conditions - outer wall, X direction

Table 5.8: Mesh convergence - outer wall, X direction

Elements	Ply 1 [MPa]		Ply 8 [MPa]		u_z [mm]	
	Value	Δ [%]	Value	Δ [%]	Value	Δ [%]
752	27.0	—	47.7	—	0.409	—
1 687	30.3	+12.2	48.0	+0.7	0.436	+6.6
3 254	30.0	-1.1	41.9	-12.7	0.435	-0.23
11 437	30.6	+2.3	36.3	-13.4	0.433	-0.46
22 361	30.9	+0.8	34.6	-4.7	0.430	-0.46

Outer Wall - Y Direction

The outer wall in the Y direction is modelled analogously to the internal wall, with the exception that no cut-outs are present. The total load of 13 891 N is applied over the module contact area. The fitment regions are modelled as clamped, while the remaining edges are constrained in translation with free rotations (Fig. 5.13).

A mesh of 23 792 elements at 2 mm element size was selected based on convergence results (Table 5.9).

Table 5.9: Mesh convergence - outer wall, Y direction

Elements	Ply 1 [MPa]		Ply 15 [MPa]		u_z [mm]	
	Value	Δ [%]	Value	Δ [%]	Value	Δ [%]
11 325	46.5	—	67.0	—	2.255	—
14 811	46.3	-0.4	67.0	0.0	2.252	-0.13
23 792	46.2	0.0	66.8	-0.3	2.251	-0.04
37 503	46.5	+0.6	66.2	-0.9	2.253	+0.09

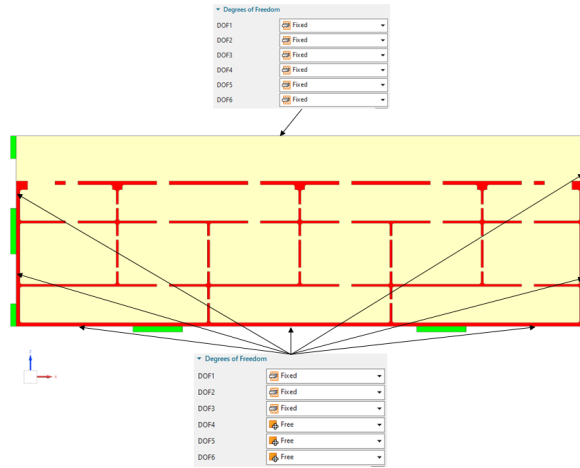


Figure 5.13: Load area and boundary conditions - outer wall, Y direction

Floor

Rather than modelling the entire floor panel, a representative section was analysed corresponding to the most critical load transfer region between adjacent internal walls. The load is applied through two narrow strips (Fig. 5.14), each carrying 694.5 N, representing half the module load under 20G vertical acceleration and reflecting the actual contact geometry. All edges are constrained in translation with free rotations, approximating adhesive behaviour and allowing realistic panel deformation.

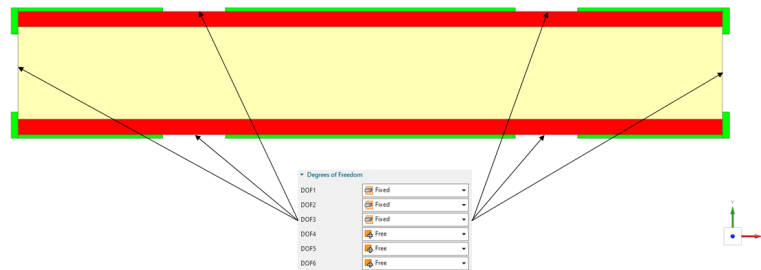


Figure 5.14: Boundary conditions - floor panel

A mesh of 20 092 elements was selected based on convergence analysis (Table 5.10).

Table 5.10: Mesh convergence - floor panel

Elements	Ply 1 [MPa]		Ply 4 [MPa]		u_z [mm]	
	Value	Δ [%]	Value	Δ [%]	Value	Δ [%]
706	6.99	—	25.29	—	-0.299	—
1 434	6.10	-12.7	23.67	-6.4	-0.312	-4.3
2 804	5.10	-16.5	22.75	-3.9	-0.315	-1.0
4 496	4.53	-11.1	20.10	-11.6	-0.316	-0.3
9 846	4.03	-11.0	20.56	+2.3	-0.316	0.0
13 928	4.87	+20.7	20.95	+1.9	-0.316	0.0
20 092	4.68	-3.8	21.12	+0.8	-0.316	0.0

Cover

The cover model extends the analytical configuration by incorporating the geometric bump associated with peripheral components, which alters the load path and introduces additional stress redistribution. Module loads are applied over discrete regions corresponding to the supporting walls (Fig. 5.15), with edges modelled as clamped to reflect the restraint provided by the bolted perimeter joint under dynamic loading.

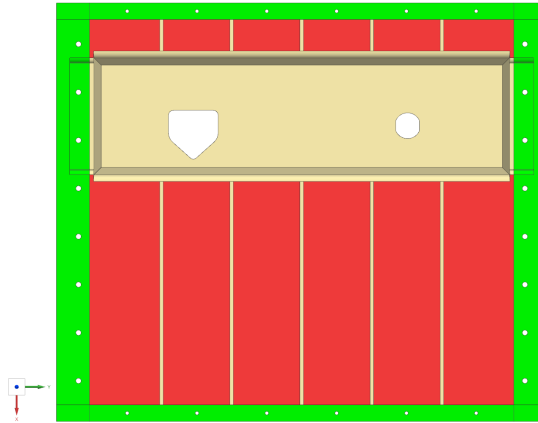


Figure 5.15: Boundary conditions - cover panel

The load from peripheral components is not applied through explicit local supports but is conservatively included within the module load, yielding a total applied load of 9 041 N. A mesh of 63 164 elements at 3 mm element size was selected based on convergence analysis (Table 5.11).

Table 5.11: Mesh convergence - cover panel

Elements	Ply 1 [MPa]		Ply 14 [MPa]		u_z [mm]	
	Value	Δ [%]	Value	Δ [%]	Value	Δ [%]
20 340	46.9	–	48.3	–	8.117	–
24 598	54.4	+16.1	56.6	+17.0	8.491	+4.61
31 556	58.8	+8.1	61.1	+8.0	8.686	+2.30
43 912	60.6	+3.0	62.9	+3.0	8.743	+0.66
63 164	61.1	+0.9	63.5	+0.9	8.760	+0.24

5.4.3 Discussion

Displacement

Table 5.12 compares the analytical formulations against the real condition FEM results.

The small deflection formulation is conservative across all panels, over-predicting real FEM displacement by 100% to 1752%. This strong conservatism is explained by two effects: the absence of membrane stiffening in small deflection theory, and the additional restraint provided by realistic boundary conditions relative to the idealised

Table 5.12: *Displacement comparison - analytical vs. real condition FEM*

Panel	FEM [mm]	LD [mm]	Δ [%]	SD [mm]	Δ [%]
Internal	9.88	3.84	−61	19.78	+100
Outer Y	2.26	3.24	+43	7.31	+223
Outer X	0.45	4.81	+971	3.42	+662
Floor	0.32	1.54	+384	5.89	+1752
Cover	8.86	6.68	−25	64.60	+630

analytical case. Although technically outside its formal validity range, small deflection theory provides a reliable upper-bound displacement estimate for all panels.

The large deflection formulation shows mixed results. For the Outer Y, Outer X, and Floor panels, it over-estimates displacement by 43% to 971%, confirming that real boundary conditions provide substantially more restraint than the analytical model assumes. For the Internal wall and Cover, however, the analytical formulation under-predicts displacement by 61% and 25% respectively. In both cases this is attributed to localised load introduction: the module contact area concentrates loading in a manner that the equivalent uniform pressure assumption cannot capture, producing higher local deflections than the analytical model predicts.

Plate Stress

Plate stress results are summarised in Table 5.13.

Table 5.13: *Bottom ply stress comparison - analytical vs. real condition FEM*

Panel	FEM [MPa]	LD [MPa]	Δ [%]	SD [MPa]	Δ [%]
Internal	103	84	−19	277	+169
Outer Y	47	89	+88	185	+290
Outer X	59	101	+72	248	+322
Floor	10	49	+383	119	+1073
Cover	64	35	−45	164	+158

The small deflection formulation remains consistently conservative across all panels, over-predicting FEM stress by 158% to 1073%, confirming its suitability as the governing design criterion. The large deflection formulation shows mixed results: most panels are within acceptable range, though the Internal wall bottom ply is marginally non-conservative at 19%, which is attributed to the localised load concentration at the module contact area.

Adhesive Shear

Adhesive shear results reveal the most significant limitation of the analytical methodology, as shown in Table 5.14.

While the analytical formulation was conservative relative to the simplified FEM models, the real condition results reveal that the Outer X and Floor panels exceed the

Table 5.14: *Adhesive shear comparison - analytical vs. real condition FEM*

Panel	Analytical [MPa]	FEM [MPa]	Δ [%]
Internal	4.90	2.25	−54
Outer Y	4.90	0.57	−88
Outer X	3.21	7.32	+128
Floor	1.23	2.17	+76

analytical prediction by +128% and +76% respectively. This reversal is explained by the localised load introduction in the real condition models: rather than a uniform pressure, load is applied over narrow contact strips, concentrating reactions at the bonded edges. The narrow aspect ratio of these panels further amplifies the edge reaction forces, resulting in adhesive shear stresses that the analytical formulation cannot capture. This identifies adhesive shear under localised loading as a critical limitation of the analytical approach, requiring FEM assessment for reliable joint dimensioning.

5.4.4 16-Ply Uniform Layup - Final Design Assessment

Due to constraints related to experimental validation and manufacturability, a uniform $[0/90]_{16}$ layup was adopted across all panels. Table 5.15 compares the structural response of this final design against the individually optimised real condition FEM results.

Table 5.15: *Real condition FEM vs. uniform 16-ply final design FEM*

Panel	Displacement [mm]		Glue Shear [MPa]		Bottom Ply [MPa]	
	Optimised	16-ply	Optimised	16-ply	Optimised	16-ply
Internal	9.88	2.43	2.25	1.74	103	129
Outer Y	2.26	1.87	0.57	0.33	47	42
Outer X	0.45	0.06	7.32	3.01	59	16.7
Floor	0.32	0.01	2.17	0.35	10	1.3
Cover	8.86	5.99	–	–	64	49

The uniform 16-ply design consistently reduces displacement and adhesive shear relative to the individually optimised models across all panels. This is particularly significant for the Outer X panel, where adhesive shear was identified as non-conservative under real loading conditions (+128% relative to analytical). The additional thickness of the uniform layup reduces this to 3.01 MPa, bringing it within acceptable margins and resolving the critical limitation identified in the real condition analysis.

Laminate stress is similarly reduced in most panels. The exception is the Internal wall, where the bottom ply stress increases from 103 MPa to 129 MPa. This is attributed to the additional bending stiffness of the thicker laminate attracting higher bending moments under the localised module contact load.

5.5 Battery Pack

5.5.1 Setup

To simulate the full TSAC assembly efficiently, the composite walls were modelled as shell elements with a composite laminate definition, while metallic components - chassis mountings, TSAC mountings, and L-profiles - were represented with CTETRA10 solid elements. Material assignments follow the design specifications: AISI P20 steel for chassis brackets, aluminium 7075-T6 for TSAC mountings, and aluminium A356 (properties equivalent to A5005) for the L-profiles.

Element sizing was consistent with the mesh convergence study conducted in Section 5.4.2, with minor adjustments to some wall elements while remaining within the established accuracy margins. Bolt connections were introduced as connector elements at each hole location, allowing axial and shear forces to be extracted under dynamic loading, though without preload representation. All bonded interfaces - internal walls to outer walls and floor, and L-profiles to outer walls - were connected using gluing capabilities within the nonlinear solver, verified using an eigenvalue analysis to identify and eliminate unrestrained rigid bodies. Chassis mounting holes were fully constrained to represent the welded bracket interface. Loads were applied per the design load cases, with symmetry exploited where applicable to reduce computational cost.

5.5.2 Results

The three load directions are evaluated in terms of displacement, ply stress, adhesive shear, and bolt stress. Results for each direction are summarised in Tables 5.16–5.19, with figures highlighting the governing failure locations.

X Direction - 40G

In the X direction, module load is applied directly to the outer X wall contact strips, consistent with the real condition model (Fig. 5.12). Module mass has negligible influence in this direction and was not modelled explicitly.

Table 5.16: Structural results summary - X direction, 40G

Component	Parameter	Value	Location
TSAC walls	Max displacement [mm]	1.39	Peripheral bump
	Max ply stress [MPa]	78.4	Internal wall fitments
Adhesive	Max shear [MPa]	0.97	Middle mounting, Y+
Mounting bolts	Combined stress [MPa]	127.90	Middle mounting, Y+
	Combined stress [MPa]	93.20	Corner mounting, Y+
Cover bolts	Combined stress [MPa]	16.17	M4, corner X+/Y+
	Combined stress [MPa]	5.34	M6, corner X-/Y+

Maximum displacement occurs at the peripheral bump (Fig. 5.16). Peak ply stress is located at the internal wall fitment edges (Fig. 5.17), consistent with the stress concentration expected at bonded boundaries. All results remain within allowable limits.

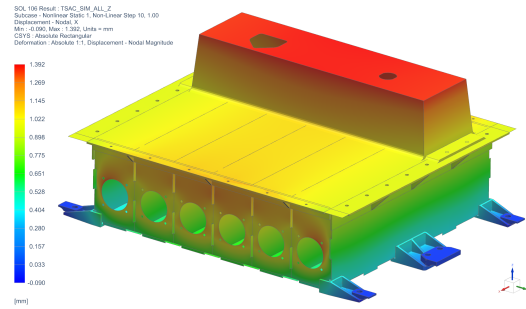


Figure 5.16: Displacement - X direction load

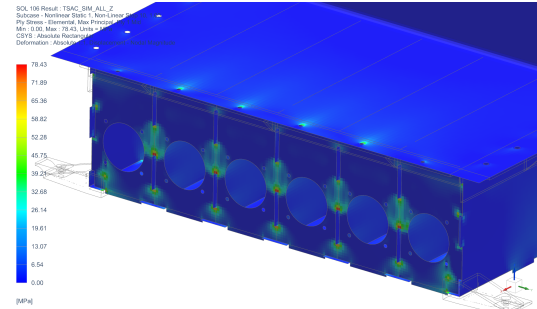


Figure 5.17: Worst ply stress - X direction load

Y Direction - 40G

In the Y direction, module inertial load is transmitted through the internal walls rather than directly to the outer Y wall. Connector elements were added between adjacent internal walls (Fig. 5.18) to represent the stiffness contribution of the module stack without explicitly modelling the cells.

Table 5.17: Structural results summary - Y direction, 40G

Component	Parameter	Value	Location
TSAC walls	Max displacement [mm]	0.34	Internal wall, upper mid-edge
	Max ply stress [MPa]	28.4	Constrained regions
Adhesive	Max shear [MPa]	0.66	Corner X+/Y+
Mounting bolts	Combined stress [MPa]	78.24	Middle mounting, Y-
	Combined stress [MPa]	55.25	Corner mounting, Y-
Cover bolts	Combined stress [MPa]	4.20	M4, corner X+/Y+
	Combined stress [MPa]	3.42	M6, corner X+/Y+

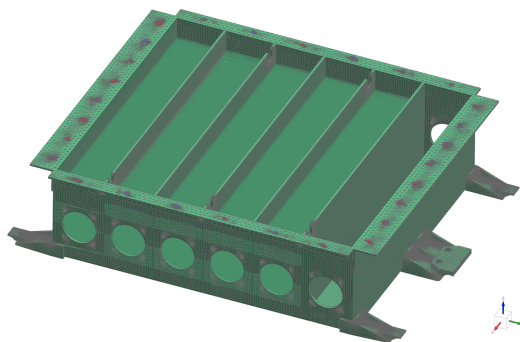


Figure 5.18: Internal wall connectors

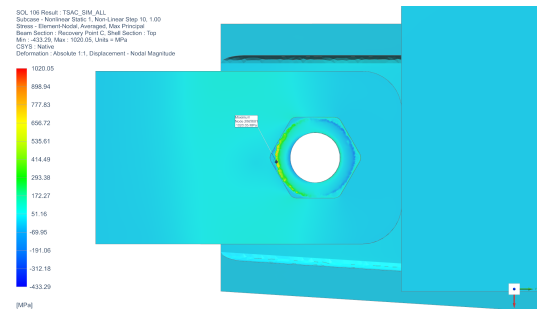


Figure 5.19: Corner mounting local stress - Y direction

A peak stress of 1020 MPa is observed locally at the corner mounting (Fig. 5.19), attributed to the point-based bolt connector representation - in practice, stress is distributed across the full insert contact area and this value is not physically representa-

tive. Combined bolt stresses of 78.24 MPa (middle) and 55.25 MPa (corner) are within allowable limits. Adhesive shear of 0.66 MPa and ply stress of 28.4 MPa both represent substantial margins relative to the respective design allowable of 5.07 MPa and 373 MPa. All criteria are satisfied.

Z Direction - 20G

The Z direction presents two distinct load cases due to the asymmetric response of the TSAC in this axis.

In the Z– direction, module weight loads the floor through the internal wall contact strips. Displacement is dominated by the cover in the X+ direction due to the large loaded area and limited local stiffness. A Tsai–Wu failure index of 0.779 is observed at the outer X wall corner, localised to the constrained edge; the remainder of the wall remains below 0.2. Stress concentration around the fan openings reaches a failure index of 0.294 (61 MPa), within acceptable margins (Fig. 5.20). Mounting bolt stresses should be interpreted with caution as constraint-driven peaks dominate; secondary stress values at the chassis mountings average 166 MPa. The L-profile region peaks at 34 MPa, within yield limits. All structural criteria are satisfied.

Table 5.18: *Structural results summary - Z– direction, 20G*

Component	Parameter	Value	Location
TSAC walls	Max displacement [mm]	–1.77	Middle internal wall
	Max ply stress [MPa]	72.39	Outer wall, side edge X+
Adhesive	Max shear [MPa]	0.25	Corner mounting, X–
Mounting bolts	Combined stress [MPa]	510.2	Middle mounting
	Combined stress [MPa]	516.9	Corner mounting, X–/Y+
Cover bolts	Combined stress [MPa]	146	M4, corner X–/Y–
	Combined stress [MPa]	64.56	M6, corner X–/Y–

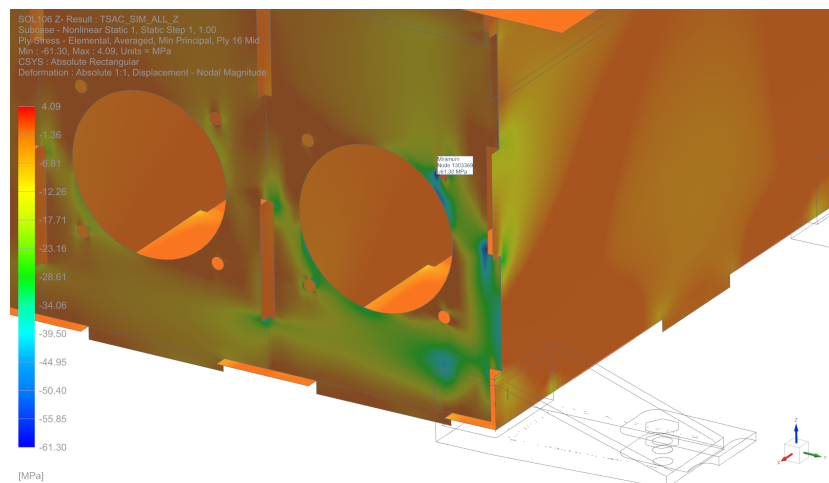


Figure 5.20: *Stress concentration at outer X wall fan openings - Z– direction*

In the Z+ direction, the inertial load acts upward against the cover, applied over the

stopper contact area. This load case presents modelling limitations: the modules act as rigid blocks that distribute load differently from a surface pressure, and replicating the Y-direction connector approach is not valid here as it would artificially stiffen the cover by sharing load with the floor - which does not occur physically. Modelling the modules explicitly as solid bodies would improve accuracy but is beyond the computational scope of this work.

As a consequence, several results are not reliable for design assessment. Cover displacement reaches 16.88 mm (Fig. 5.21), the M4 bolt holes show ply stresses up to 571 MPa (Fig. 5.22), and cover bolt stresses indicate failure. These are direct consequences of the simplified loading assumption and do not represent the physical response. The only results considered reliable from this load case are adhesive shear stress (1.85 MPa, well within allowable) and M6 mounting bolt stresses, which depend on the global reaction at the outer walls rather than the local load introduction.

Table 5.19: Structural results summary - Z+ direction, 20G

Component	Parameter	Value	Location	Reliable
TSAC walls	Max displacement [mm]	16.88	Cover	No
	Max ply stress [MPa]	571	M4 hole, X+	No
Adhesive	Max shear [MPa]	1.85	Middle, X+	Yes
Mounting bolts	Combined stress [MPa]	524	Middle mounting	No
	Combined stress [MPa]	690	Corner X+/Y+	No
Cover bolts	Combined stress [MPa]	3461	M4, middle X+	No
	Combined stress [MPa]	485	M6, peripherals X-	Partial

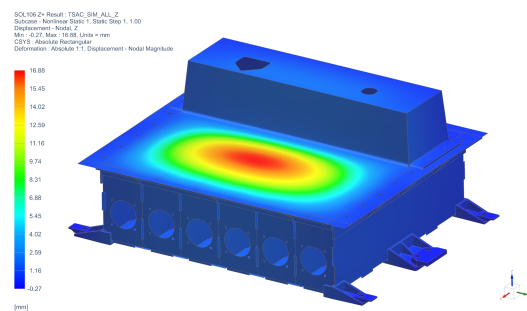


Figure 5.21: Cover displacement - Z+ direction

SOL106 Z+ Result: TSAC_SIM_ALL_Z
Subcase - Nonlinear Static 1, Static Step 1, 1.00
Stress, Von-Mises
Element 736081, Node 2353223
Min: 41.19, Max: 3461.49, Units = MPa
CSYS: Native

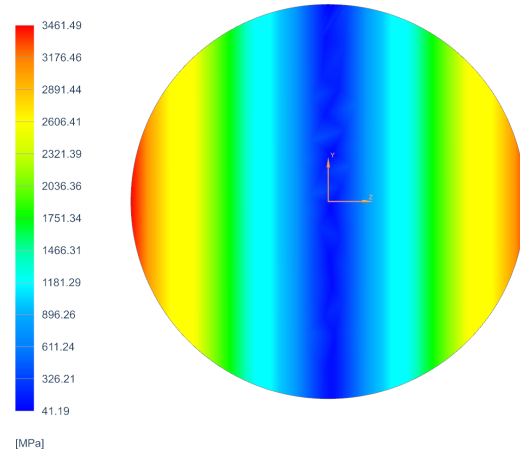


Figure 5.22: M4 bolt stress - Z+ direction

5.6 Numerical Implementation Conclusions

The numerical investigation covered four areas: material model validation, three-point bending simulation, individual wall assessment, and full battery pack simulation under design load cases.

The tensile simulation confirmed that the material model reproduces analytical predictions with deviations below 1%, validating its application to the structural models that follow. The three-point bending study established that the solid FEM model is the appropriate tool for failure assessment, showing close agreement with analytical stress predictions (+1%) and contact force (+3%), while the shell model is unsuitable due to artificial stiffness introduced by convergence constraints. The perimeter shear test could not be reliably simulated using implicit methods, and experimental results are used directly for the ASES equivalency verification.

Regarding the wall simulations, the small deflection formulation consistently over-predicts both displacement and stress, making it a conservative and reliable criterion for preliminary dimensioning despite being formally outside its validity range. The large deflection formulation is non-conservative for stress in most panels and should not be used as the sole dimensioning criterion. The primary source of discrepancy is the Poisson's ratio assumed in the Roark tables ($\nu = 0.316$) versus the effective laminate value ($\nu \approx 0.09$), which governs membrane stiffening. Adhesive shear is identified as the most critical analytical limitation: while conservative under idealised conditions, the Outer X and Floor panels exceed the analytical prediction by +128% and +76% respectively under realistic load introduction. The adoption of the uniform $[0/90]_{16}$ layup resolves this, reducing Outer X adhesive shear from 7.32 MPa to 3.01 MPa. The only remaining concern is the Internal wall, where the thicker laminate raises bottom ply stress from 103 MPa to 129 MPa under localised loading, which must be verified against the laminate strength allowable.

The full battery pack simulation confirmed structural adequacy under X and Z-loading, with all ply stresses, adhesive shear values, and bolt stresses within allowable limits. Under Y loading, the governing parameters are similarly within margins, with the localised mounting bolt peak of 1020 MPa identified as a modelling issue rather than a physical result. The Z+ direction could not be reliably assessed due to the simplified load application on the cover - a known limitation of the modelling approach - with only adhesive shear (1.85 MPa) and M6 mounting bolt stresses considered representative. This load case requires further investigation, ideally through explicit module modelling or an experimental approach.

VALIDATION

6.1 Tensile Tests

Composite laminates frequently exhibit discrepancies between analytically estimated and experimentally measured properties, since micromechanical models assume idealised conditions not fully representative of manufactured laminates. Uniaxial tensile tests were therefore conducted to determine the elastic modulus, tensile strength, and Poisson's ratio of the selected laminate, providing the experimentally grounded material properties used throughout the design.

6.1.1 Manufacturing

Specimens were produced from a single flat plate (Fig. 6.1) laminated on a glass surface to minimise porosity. The resin was vacuum-degassed prior to application and worked into the fabric using a brush and silicone squeegee to ensure full fibre wet-out without pooling. Peel-ply and breather layers were used to facilitate demoulding, vent trapped air, and absorb excess resin. The plate was cured at 40°C under approximately 0.9 bar vacuum for 8 hours. Full material and process details are given in Table 6.2.

After full cure, six specimens were cut from the plate using a table saw, with cuts parallel to the fibre direction. Thickness and width were measured at three locations along the gauge section (Fig. 6.2), with resulting dimensions close to the target values, as summarised in Table 6.1.

Table 6.1: *Specimen dimensions*

Specimen	Thickness [mm]				Width [mm]				Length [mm]
	A	B	C	Mean	A	B	C	Mean	
1	3.07	3.06	3.03	3.05	25.48	25.21	25.00	25.23	250
2	2.99	3.03	3.09	3.04	25.06	25.23	25.20	25.16	249
3	3.22	3.33	3.33	3.29	25.82	25.98	25.65	25.82	250
4	3.11	3.14	3.30	3.18	25.81	25.74	25.59	25.71	250
5	2.99	3.05	3.12	3.05	26.02	25.68	25.80	25.83	249
6	3.12	3.16	3.32	3.20	25.06	25.65	25.45	25.42	251

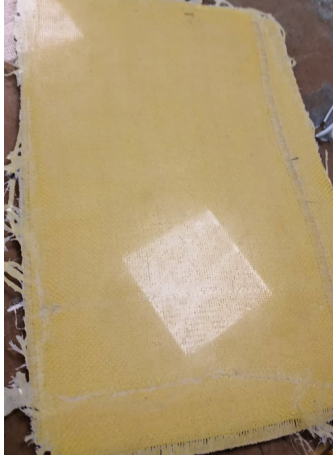


Figure 6.1: Specimen plate

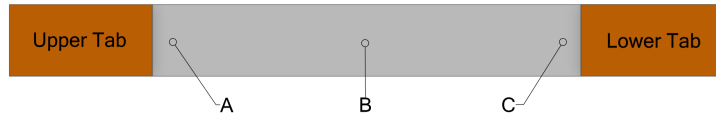


Figure 6.2: Measurement locations for mean thickness and width

Table 6.2: Material and specimen details

Parameter	Description
Material type	Kevlar 49, 22 tex, plain weave, 170 g/m ²
Resin / Hardener	Sika Biresin CR132FR / CH132-2
Lay-up	[0/90] ₁₀
Process	Wet lay-up + vacuum bag, 8 h at 40°C
Specimen geometry	Rectangular, 250 × 25 × 3 mm
Quantity	6
Tabs	Emery cloth, default adhesive
Cutting method	Table saw, parallel to fibre direction
Conditioning	None

6.1.2 Specimen Preparation

Each specimen was marked at mid-span for strain gauge placement, and emery cloth tabs (50 × 25 × 1 mm) were bonded to both surfaces at each end (Fig. 6.3). However, preliminary tests on dummy coupons revealed that the emery cloth produced consistent grip slippage, likely due to insufficient adhesion between the cloth and the aramid surface. Tabs were therefore removed for all subsequent tests, and specimens were gripped directly, yielding more reliable results.

A biaxial strain gauge (0°/90°) was bonded at the specimen midsection, enabling simultaneous acquisition of longitudinal and transverse strains from a single gauge. Although the standard recommends three gauges per specimen to detect bending, only one gauge per coupon was installed due to limited availability. The bonding surface was cleaned with alcohol and the gauge aligned parallel to the fibre directions using adhesive tape as a positioning aid. Instant adhesive was applied under light pressure to ensure full seating, excess adhesive was removed, and varnish was applied over the connections to prevent short circuits. Thin wires were subsequently soldered to the gauge terminals.

Strain signals were acquired using a custom LabVIEW programme, handling real-time visualisation, calibration, filtering, and data logging. No environmental conditioning was performed; all tests were conducted at room temperature of 22°C.

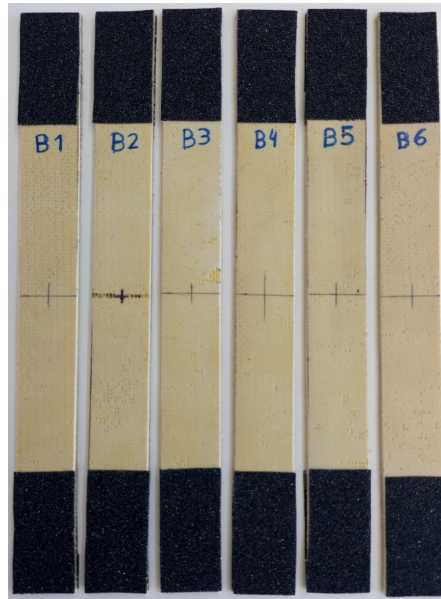


Figure 6.3: *Specimens with emery cloth tabs*

6.1.3 Setup Preparation

The test setup is summarised in Table 6.3. Each specimen was gripped over the full tab region using mechanical wedge-action grips, with alignment verified against a square to minimise bending. The clip-on extensometer was ideally positioned at mid-span; however, since the bonded strain gauge occupied the centre of the gauge section, the extensometer was offset slightly toward the upper tab. Given the laminate's high in-plane stiffness and uniform cross-section, this offset is not expected to introduce significant error. The extensometer was lightly secured with rubber bands to prevent slippage, and the strain gauge channel was zeroed and calibrated immediately before each test.

Table 6.3: *Machine and test setup parameters*

Parameter	Description
Testing machine	Instron 5584, 50 kN load cell
Grips	Mechanical wedge-action tensile grips
Alignment check	Verified with a square
Gauge length	50 mm (extensometer, offset from strain gauge)
Crosshead speed	2 mm/min
Calibration	Default machine calibration
Data acquisition	50 Hz



Figure 6.4: *Machine setup*

6.1.4 Results

Force-displacement curves for all specimens are presented in Fig. 6.5, and the extracted mechanical properties are summarised in Table 6.4.

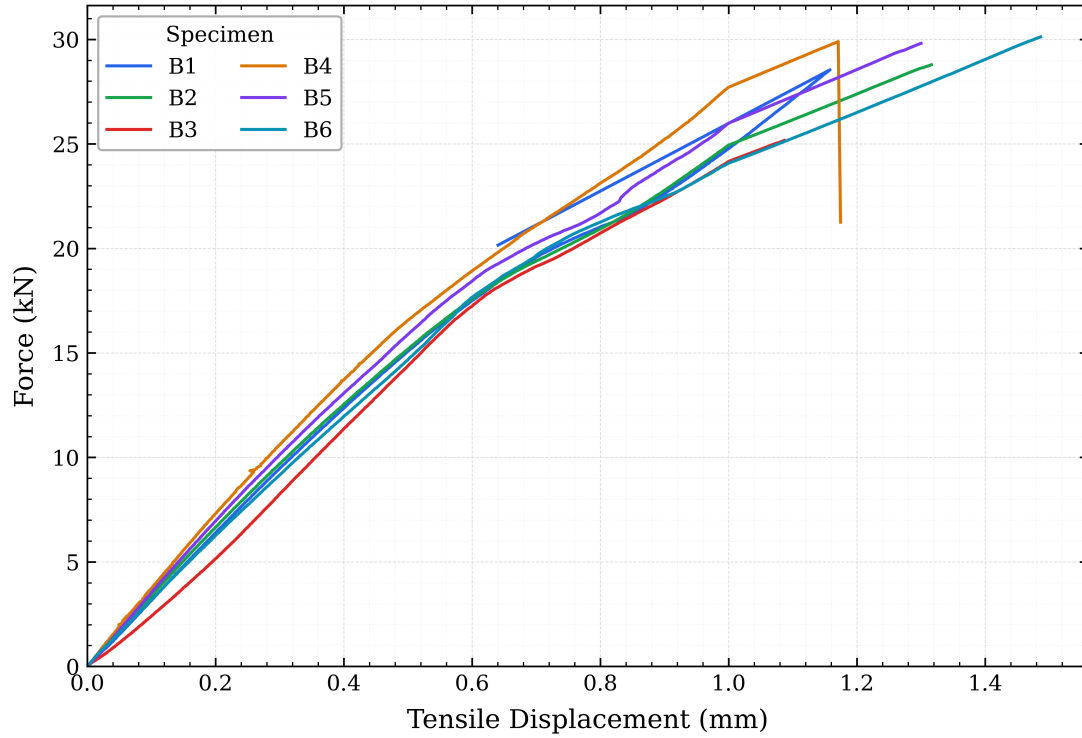


Figure 6.5: Force vs. tensile displacement for specimens B1–B6

Table 6.4: Tensile test results - all specimens

Property / Specimen	1	2	3	4	5	6	Mean	Std. Dev.	CV [%]
Elastic modulus [GPa]	19.53	19.46	20.40	20.55	20.44	19.00	19.90	0.65	3.3
Tensile strength [MPa]	371	377	296	366	379	370	360	31.6	8.8
Displacement [mm]	1.16	1.32	1.09	1.17	1.30	1.49	1.26	0.14	11.5
Strain [%]	2.32	2.64	2.18	2.35	2.60	2.98	2.51	0.29	11.5
Poisson's ratio	0.069	0.123	0.043	0.045	0.095	0.108	0.080	0.033	41.4

Specimen 3 exhibits a concave stress-strain response relative to the remaining specimens, consistent with grip slippage during the test. It was therefore identified as an outlier and excluded from the design values. The primary effect of its removal is on tensile strength, whose coefficient of variation drops from 8.8% to 1.4%; the remaining properties are largely unaffected. The filtered results are presented in Table 6.5.

Table 6.5: Tensile test results - specimen 3 excluded (grip slippage)

Property / Specimen	1	2	4	5	6	Mean	Std. Dev.	CV [%]
Elastic modulus [GPa]	19.53	19.46	20.55	20.44	19.00	19.80	0.67	3.4
Tensile strength [MPa]	371	377	366	379	370	373	5.3	1.4
Displacement [mm]	1.16	1.32	1.17	1.30	1.49	1.29	0.13	10.4
Strain [%]	2.32	2.64	2.35	2.60	2.98	2.58	0.27	10.3
Poisson's ratio	0.069	0.123	0.045	0.095	0.108	0.090	0.031	35.5

The elastic modulus shows excellent repeatability ($CV = 3.4\%$), confirming consistent fibre volume fraction and laminate quality across the plate. Tensile strength is similarly repeatable after outlier removal ($CV = 1.4\%$). Poisson's ratio exhibits the highest variability ($CV = 35.5\%$), which is characteristic of woven aramid laminates where the lateral response is sensitive to local weave geometry and resin distribution, and is considered acceptable for the level of precision required in the design.

6.2 Density Test

The volumetric mass of the aramid laminate was determined experimentally using the Archimedes immersion method, following the procedure described in Section 2.5.4. The objective was to obtain a more accurate estimate of the laminate density than the analytical prediction, for use in the battery pack casing mass evaluation.

Five samples (Fig. 6.6a) were cut from the same plate used for the tensile specimens (Fig. 6.1), ensuring material consistency. Each specimen was first weighed in air to obtain a , then fully immersed in distilled water to record the displaced fluid reading b (Figs. 6.6b and 6.6c). Air bubbles were carefully removed using a sharp object during immersion. The wire and sinker factor w was not applied, as the balance had been zeroed with the sinker already immersed. Density was calculated using Eq. 2.50, and the results are presented in Table 6.6.

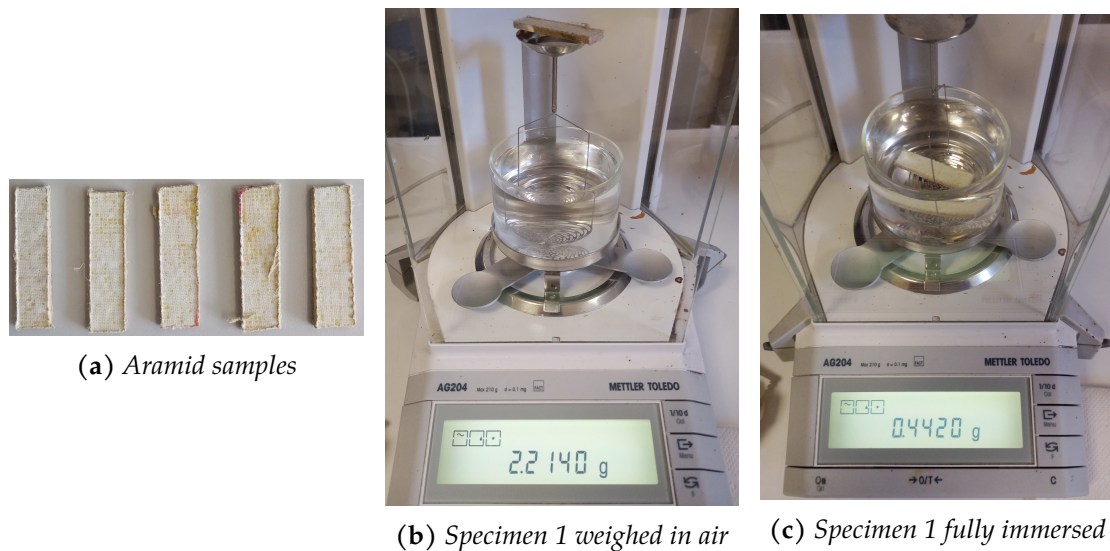


Figure 6.6: Archimedes immersion method for density measurement

The coefficient of variation of 1.1% indicates highly consistent measurements across all specimens, confirming the uniformity of the manufactured laminate. A density of 1243 kg/m^3 is therefore adopted for all subsequent mass estimations, in place of the analytically predicted value of 1340 kg/m^3 .

Table 6.6: Density measurement results - Archimedes method

Specimen	Mass in air [g]	Mass submerged [g]	Density [kg/m ³]
1	2.214	0.442	1246
2	2.056	0.407	1243
3	2.283	0.480	1263
4	2.180	0.405	1225
5	1.847	0.362	1240
Mean		1243 kg/m ³	
Standard deviation		13.7 kg/m ³	
CV		1.1%	

Distilled water properties	
Temperature [°C]	24
Density [kg/m ³]	997.3

6.3 Flammability Tests

Flammability tests were conducted to verify compliance with the UL94 V-0 rating required by the competition regulations, using specimens manufactured from the selected material system. Specimen details are summarised in Table 6.7.

Table 6.7: Material and specimen details - flammability tests

Parameter	Description
Material type	Kevlar 49, 22 tex, plain weave, 170 g/m ²
Resin / Hardener	Sika Biresin CR132FR / CH132-2
Lay-up	[0/90] ₁₀
Process	Wet lay-up + vacuum bag, 8 h at 40°C
Specimen geometry	Rectangular, 125 × 13 × 3.3 mm
Quantity	4
Cutting method	Table saw, parallel to fibre direction
Conditioning	None

Tests were conducted inside a laboratory fume cupboard. A Bunsen burner operating on an 80/20 butane/propane mixture was calibrated to produce a 20 mm flame height and positioned at 45° relative to the specimen to prevent contamination from molten material or debris. Each specimen was clamped with its lower edge 10 mm above the flame tip and 300 mm above a cotton indicator placed on the bench, as illustrated in Fig. 6.7. Minor adaptations to the setup were required due to available equipment, though all critical distances and procedural steps were strictly maintained.

Each specimen was exposed to the flame for 10 s, after which the burner was removed and the after-flame duration recorded. Once the flame self-extinguished, the burner was reapplied for a second 10 s exposure, followed by a second after-flame measurement. Two operators were required - one to control the stopwatch and one to position the burner accurately. Results are presented in Table 6.8 and Fig. 6.8.

All specimens exhibited zero after-flame time on the first exposure, demonstrating strong self-extinguishing behaviour. Second-exposure results were less consistent:

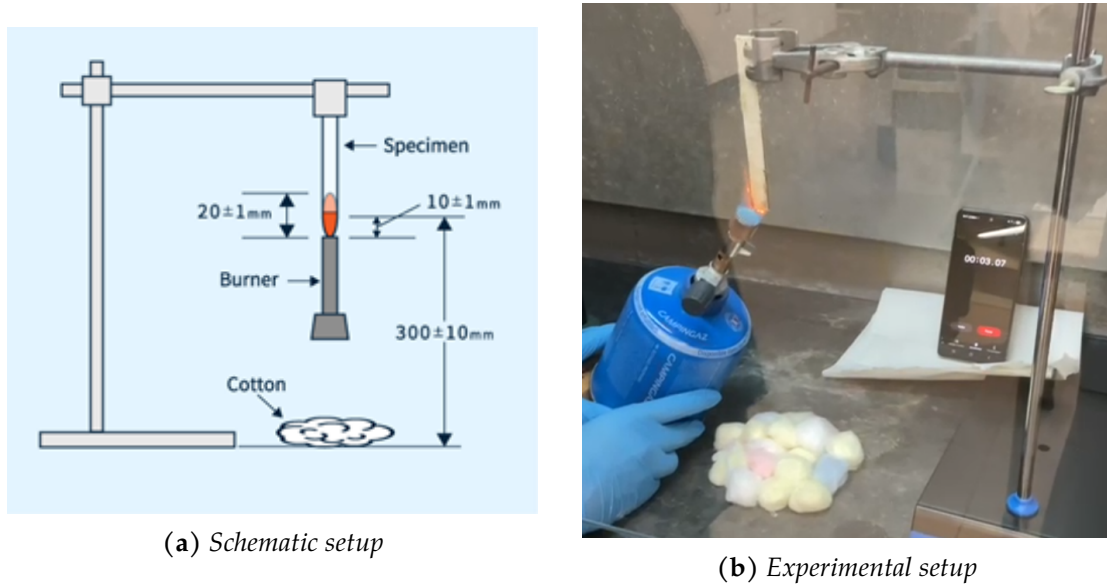


Figure 6.7: Flammability test setup

Table 6.8: Flammability test results

Specimen	1 st burn [s]	2 nd burn [s]	After-glow [s]	Cotton ignition	Flame to clamp
1	0	9	0	No	No
2	0	16	0	No	No
3	0	2	0	No	No
4	0	0	0	No	No
Total	0	27	0	–	–

specimen 4 extinguished immediately while specimen 2 sustained burning for 16 s. This variability is attributed to fibre protrusions left by the table saw cut, which ignite more readily than a clean edge; water-jet cutting would likely eliminate this effect.

In no case did the flame reach the clamp, cotton ignition occurred, or after-glow was detected. All UL94 V-0 criteria are satisfied: individual after-flame times remain below 10 s for first exposure and below the allowable limit for the second, the total after-flame time of 27 s falls within the 40 s threshold, and no dripping or cotton ignition was observed. The material system is therefore classified as UL94 V-0 compliant, and this rating is considered applicable to plates of equal or greater thickness manufactured using the same process.

6.4 Three-Point Bending Tests

Three-point bending tests were conducted to evaluate the flexural behaviour of the $[0/90]_{16}$ laminate, characterising stiffness, strength, and failure mechanisms. The results serve to validate the analytical and numerical models developed in the preceding chapters.

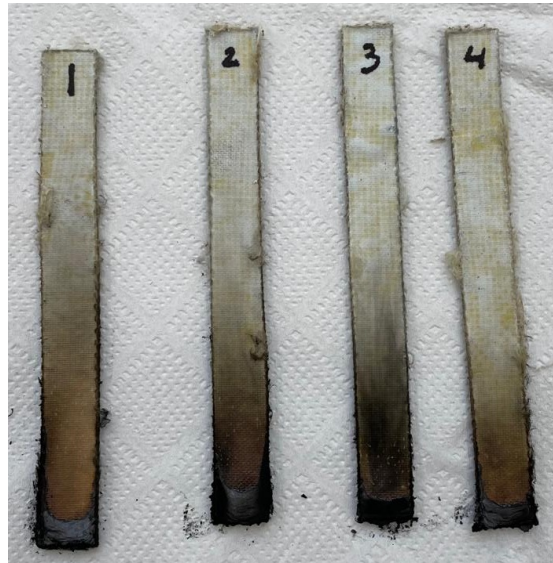


Figure 6.8: Specimens after flammability testing

6.4.1 Manufacturing and Specimen Preparation

Specimens were manufactured following the same wet lay-up procedure adopted for the tensile specimens (Section 6.1), and cut from a single laminate plate to minimise material variability. The plate was cured for 8 hours at 40°C under vacuum bagging, targeting a fibre volume fraction of approximately 60%; deviations are expected due to resin absorption by the peel-ply and breather layers during processing. Three specimens were extracted using a table saw with cuts parallel to the fibre direction, marked in accordance with rule T 3.6.6, and visually inspected - no detectable voids or defects were observed. No environmental conditioning was performed; all specimens were tested under identical ambient laboratory conditions. Manufacturing details are summarised in Table 6.9.

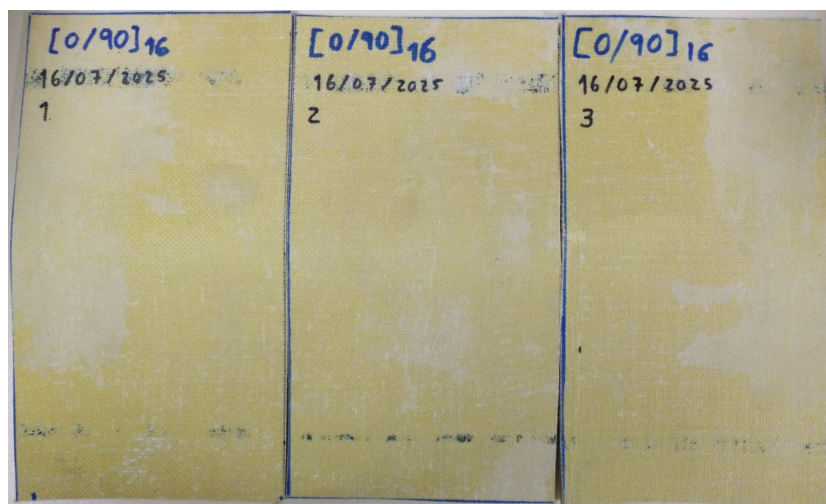


Figure 6.9: Three-point bending specimens

Table 6.9: Specimen manufacturing details - three-point bending

Parameter	Description
Material type	Kevlar 49, 22 tex, plain weave, 170 g/m ²
Resin / Hardener	Sika Biresin CR132FR / CH132-2
Lay-up	[0/90] ₁₆
Process	Wet lay-up + vacuum bagging, 8 h at 40°C
Target V_f	~60% (deviations expected due to process losses)
Specimen geometry	Rectangular, ~275 × 150 × 4 mm
Quantity	3
Cutting method	Table saw, parallel to fibre direction
Conditioning	None

6.4.2 Setup Preparation

The test setup is shown in Fig. 6.10 and summarised in Table 6.10. A cylindrical punch of 50 mm radius applied the central load, while two cylindrical rollers of 15 mm radius acted as supports at a span of 200 mm, following rule T 3.6.1. In the physical arrangement, the punch was positioned below the specimen and the supports above - mechanically equivalent to the conventional configuration provided alignment is maintained, which was verified using a square.

A crosshead speed of 2 mm/min was adopted rather than the standard 1 mm/min, reducing testing time while remaining within the acceptable quasi-static range defined by ASTM D7264 [29].



Figure 6.10: Three-point bending experimental setup



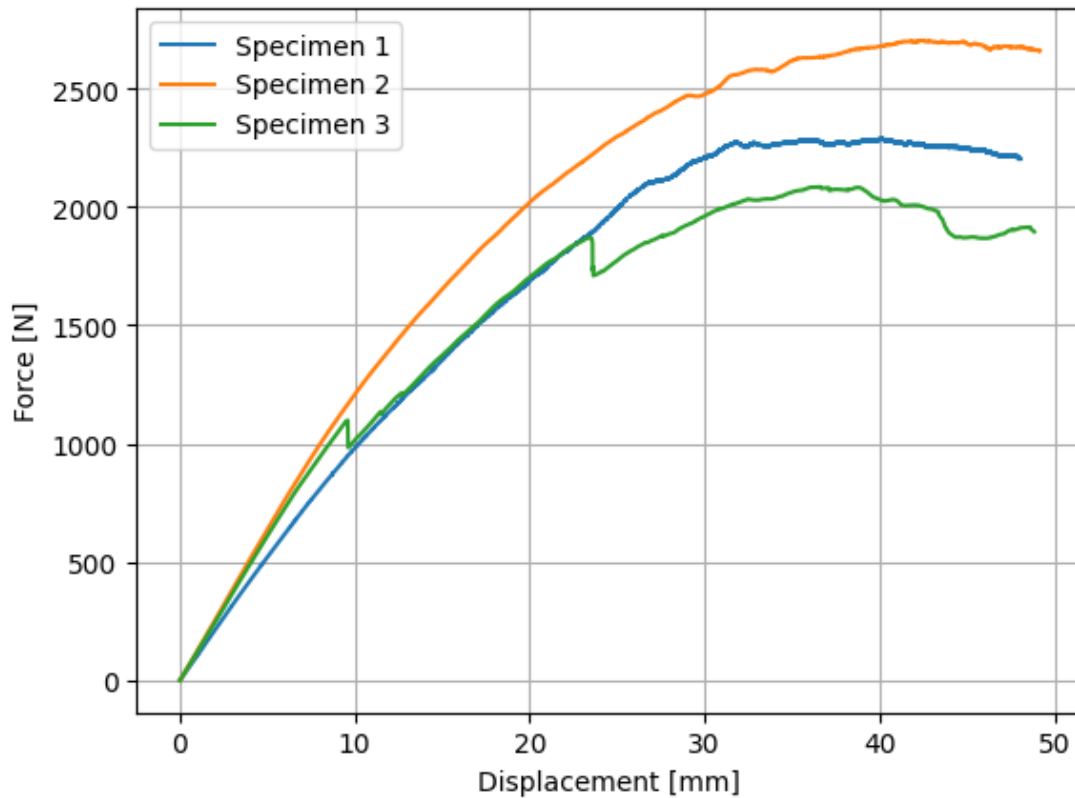
Figure 6.11: Support span

Table 6.10: *Three-point bending setup parameters*

Parameter	Description
Testing machine	Instron 5584, 50 kN load cell
Punch	Cylinder, 50 mm radius
Supports	Cylinders, 15 mm radius
Span	200 mm
Alignment check	Verified with a square
Crosshead speed	2 mm/min
Calibration	Default machine calibration
Data acquisition	10 Hz

6.4.3 Results

Force-displacement curves are presented in Fig. 6.12 and extracted properties summarised in Table 6.11. The flexural modulus was computed from the slope between 1 mm and 2 mm displacement, as the standard strain range of 0.001–0.003 proved too small to extract reliably given the specimen geometry and data resolution.

**Figure 6.12:** *Force-displacement curves - $[0/90]_{16}$ laminate*

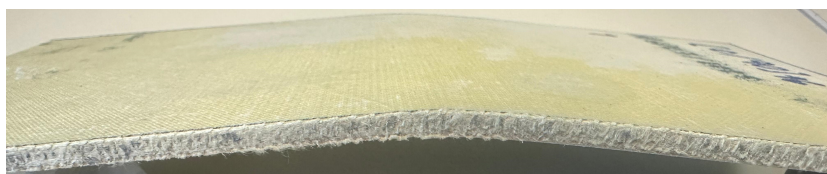
All specimens show a consistent linear initial response, confirmed by the low variability in bending stiffness ($CV = 9.1\%$), indicating that the elastic regime is governed by global laminate properties. Flexural strength shows considerably higher scatter ($CV = 22.2\%$), consistent with failure being controlled by local effects such as fibre misalignment, resin-rich zones, or manufacturing defects inherent to the wet lay-up pro-

Table 6.11: *Three-point bending results - all specimens*

Property / Specimen	1	2	3	Mean	Std. Dev.	CV [%]
Max force [N]	2286	2703	2085	2358	315.5	13.4
Displacement at max [mm]	40.16	42.40	36.61	39.72	2.92	7.4
Maximum strain	0.0245	0.0251	0.0238	0.0245	0.0006	2.5
Flexural strength [MPa]	275.6	346.0	221.4	281	62.5	22.2
Flexural modulus [GPa]	21.21	27.76	20.01	23.00	4.2	18.1
Bending stiffness [N/mm]	107.2	128.2	122.7	119.4	10.9	9.1

cess. The elevated dispersion in flexural modulus ($CV = 18.1\%$) is largely attributable to the squared thickness dependence in the modulus formula, which amplifies small geometric variations between specimens.

Specimen 3 exhibited progressive damage with multiple load drops in the force-displacement curve, accompanied by visible delamination (Fig. 6.13), and local load peaks at approximately 9.6 mm and 23 mm of displacement - consistent with sequential matrix cracking and interlaminar failure. Specimen 2 (Fig. 6.14) reached the highest load of 2703 N with a smooth response and no visible delamination, suggesting a lower defect density in that region of the plate.

**Figure 6.13:** *Delamination observed in specimen 3 after testing***Figure 6.14:** *Specimen 2 after testing*

6.4.4 Comparison with Numerical Predictions

The comparison relies solely on the solid FEM model, as the shell model produced results inconsistent with both analytical predictions and physical expectations, as discussed in Section 5.2. Since the simulation could not be continued beyond 15 mm displacement, the Tsai–Wu index was not directly observed to reach unity; however, given that the solid model yields an index of 0.90 at 15 mm, failure is expected to occur shortly beyond this point. The force at 15 mm displacement is therefore taken as the reference load for comparison, and results are summarised in Table 6.12.

Table 6.12: *FEM and experimental three-point bending comparison*

Parameter	FEM	Experimental (mean)	Δ [%]
Force at 15 mm [N]	2821	1460	−48
Flexural strength [MPa]	229	281	+23
Bending stiffness [N/mm]	183.2	119.4	−35

Contact force and bending stiffness are over-predicted by 48% and 35% respectively, consistent with the idealised assumptions of the model - perfect bonding, no manufacturing defects, and no progressive damage. The experimental specimens exhibited matrix cracking and delamination well before ultimate failure, reducing effective stiffness and peak load in a manner the FEM cannot replicate. The high variability in experimental modulus ($CV = 18.1\%$) further reflects sensitivity to local manufacturing quality absent from the numerical model.

Flexural strength shows the best agreement, with the FEM under-predicting the experimental mean by 23% - a conservative and favourable outcome for design. The solid model is therefore considered a suitable tool for laminate dimensioning, providing a conservative lower bound on flexural capacity. The analytical solution is in close agreement with the solid FEM and remains adequate for preliminary design, while the shell model is not appropriate for this analysis due to the artificial stiffness introduced by the convergence constraints discussed previously.

6.5 Perimeter Shear Strength Tests

Perimeter shear strength tests were conducted to demonstrate compliance with competition regulations and to verify that the selected laminate meets the shear strength requirements governing several TSAC panels identified in the analytical dimensioning.

6.5.1 Manufacturing and Specimen Preparation

The perimeter shear strength specimens were extracted from the same plate as the three-point bending specimens (Section 6.4), manufactured under identical conditions. Three specimens were cut using a table saw with cuts parallel to the fibre di-

rection, marked in accordance with rule T 3.6.6, and visually inspected - no detectable voids or defects were observed. No environmental conditioning was performed. Manufacturing details are summarised in Table 6.13 and specimens are shown in Fig. 6.15.

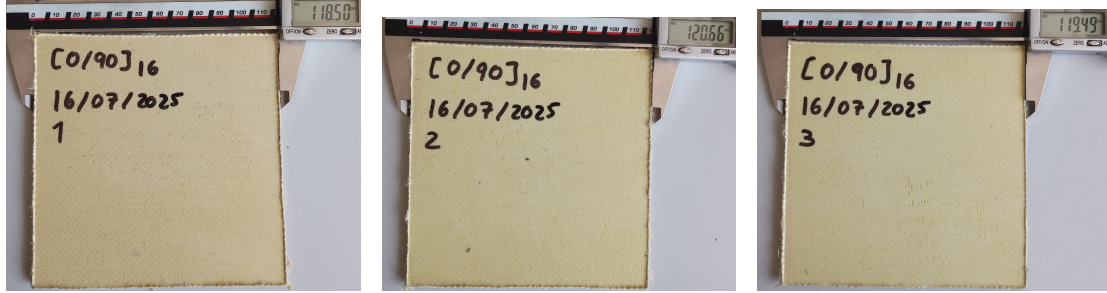


Figure 6.15: Perimeter shear strength specimens

Table 6.13: Specimen manufacturing details - perimeter shear strength

Parameter	Description
Material type	Kevlar 49, 22 tex, plain weave, 170 g/m ²
Resin / Hardener	Sika Biresin CR132FR / CH132-2
Lay-up	[0/90] ₁₆
Process	Wet lay-up + vacuum bagging, 8 h at 40°C
Target V_f	~60% (deviations expected due to process losses)
Specimen geometry	Square, ~120 × 120 × 4 mm
Quantity	3
Cutting method	Table saw, parallel to fibre direction
Conditioning	None

6.5.2 Setup Preparation

The test configuration follows the dimensions prescribed by rules T 3.6.11 and T 3.6.12 rather than the ASTM D732–02 geometry [30], ensuring direct compatibility between the experimental setup and the regulatory compliance requirements. A flat-head cylindrical punch of 25 mm diameter is pushed through the specimen, which rests on a steel plate with a coaxial 32 mm slot. The specimen is not clamped, as required by the rules. The setup is symmetric with respect to fibre orientation, so specimen alignment is not a critical parameter. The crosshead speed of 1.3 mm/min follows ASTM D732–02, as this parameter is retained from the standard. The setup is illustrated in Fig. 6.16 and summarised in Table 6.14.

6.5.3 Results

Force-displacement curves for all three specimens are presented in Fig. 6.17. Tests were stopped at approximately 50 kN, corresponding to the reliable operating limit of the load cell. No specimen reached ultimate failure within this range.

A minor offset in the initial portion of the curves is attributed to contact settlement between the punch and specimen surface. Beyond this, all three specimens ex-

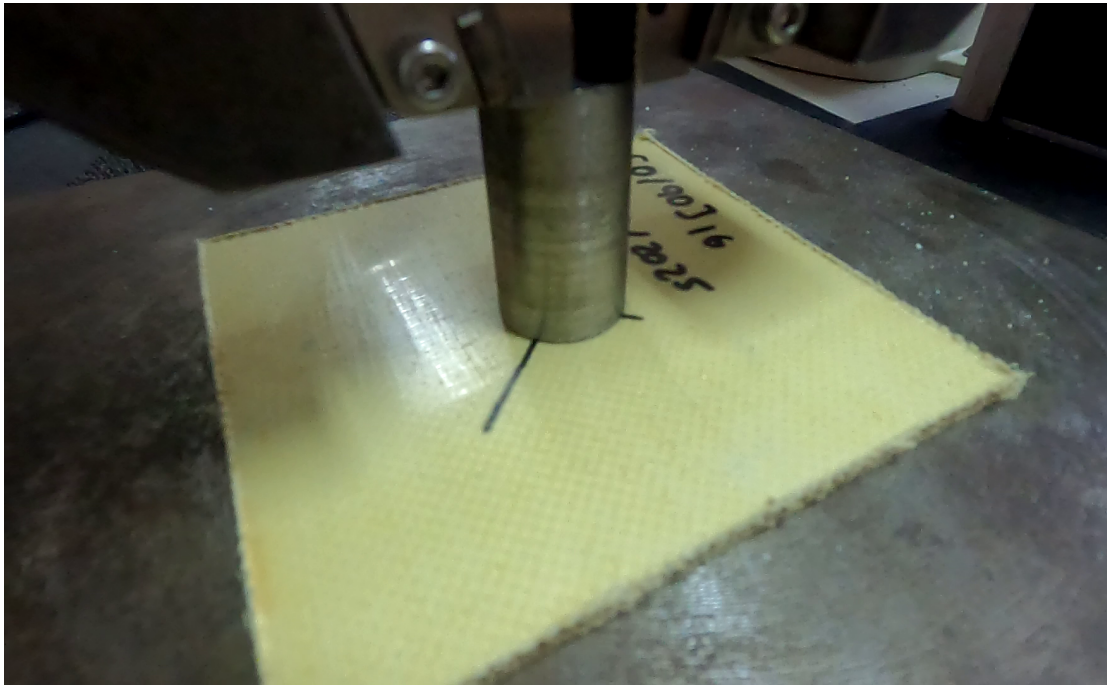


Figure 6.16: *Perimeter shear strength experimental setup*

Table 6.14: *Perimeter shear test setup parameters*

Parameter	Description
Testing machine	Instron 5584, 50 kN load cell
Punch	Flat-head cylinder, 25 mm diameter
Support plate	Slotted, 32 mm diameter hole
Alignment	Punch centred to slot by marking
Crosshead speed	1.3 mm/min
Calibration	Default machine calibration
Data acquisition	50 Hz

hibit a consistent response: an initial nonlinear region followed by a load plateau at approximately 1 mm displacement, marking the onset of matrix shear yielding and interlaminar delamination. During this plateau, the laminate continues to deform at near-constant load, indicating stable damage propagation with significant energy absorption - characteristic of aramid composites, which are known for their high damage tolerance. A subsequent load recovery is then observed as stress redistributes among intact fibres, followed by minor fluctuations associated with progressive localised failure.

Post-test inspection of specimen 1 (Fig. 6.18) confirms local deformation around the punch perimeter and visible fibre damage on the back face, with no complete rupture - consistent with the laminate retaining load-carrying capacity beyond the test limit.

The maximum force recorded for each specimen is given in Table 6.15, noting that these values represent the load at test termination rather than ultimate failure. Shear strength was subsequently calculated using Eq. 2.49 and compared against the theo-

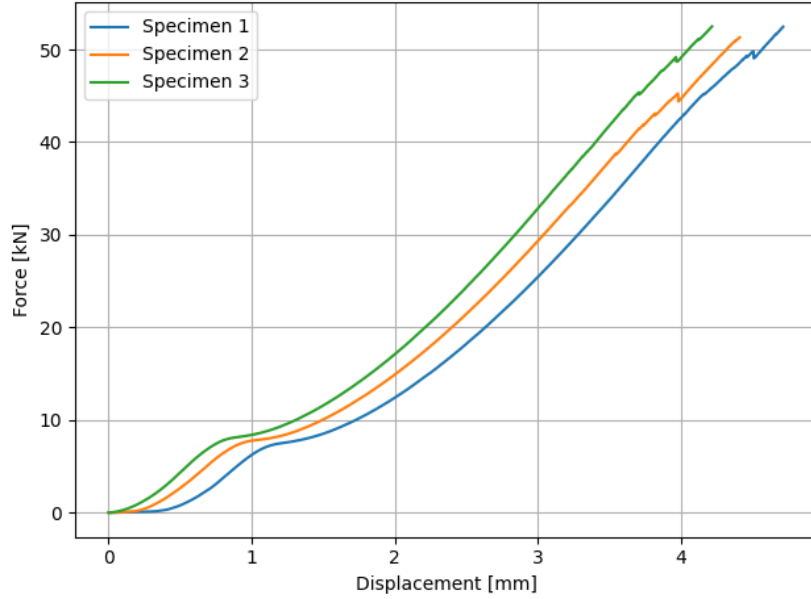


Figure 6.17: Force-displacement response - perimeter shear strength

retical value in Table 6.16.

Table 6.15: Perimeter shear test results

Parameter / Specimen	1	2	3
Max force [N]	52 456	51 307	52 496

Table 6.16: Shear strength - theoretical vs. experimental

Parameter	Theoretical	Experimental	Unit
Punch diameter	25	25	mm
Failure load	20 251	51 307	N
Thickness	4.96	3.90 / 3.87 / 3.84	mm
Shear strength	52	168 / 169 / 170	MPa
Mean shear strength	52	169	MPa
Relative error	–	+225	%

The experimental shear strength of 169 MPa is approximately three times the theoretical value of 52 MPa used in the analytical dimensioning. As shown in Table 4.13, several panels - including the floor and outer walls - are governed by the shear equivalency rule, and this result suggests that the layup requirements for those panels could be reduced. However, given that other panels remain governed by structural load cases, that the design constraint of a single $[0/90]_{16}$ validation plate was imposed from the outset, and that the total TSAC mass already satisfies the target, no modification to the current design is warranted. The result nonetheless confirms that the adopted laminate provides a substantial shear strength margin beyond the regulatory minimum.

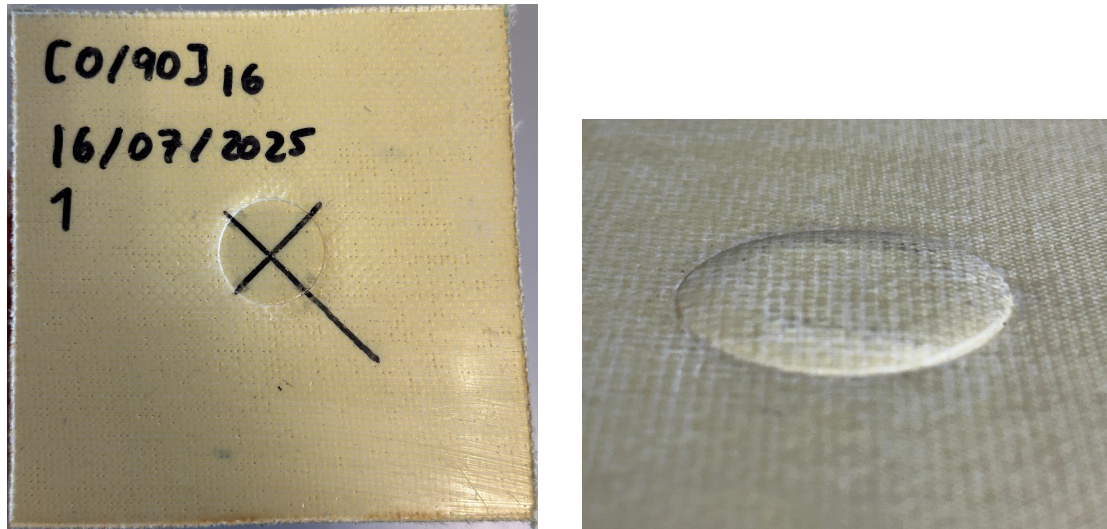


Figure 6.18: *Specimen 1 after perimeter shear test - front and back faces*

6.6 Adhesive Tests

Adhesive shear tests were conducted to evaluate the influence of bond-line gap on joint performance, and to define appropriate assembly tolerances for the TSAC panel connections. Rather than following a standard lap shear procedure, a custom approach was developed reflecting the actual joint geometry - where panels are bonded edge-to-edge with a controlled adhesive gap defined by the waterjet cutting tolerance.

6.6.1 Manufacturing and Specimen Preparation

Specimens were manufactured from remaining laminate plates from previous tests, ensuring material consistency. Each specimen consists of a rectangular insert bonded within a slotted plate of the same material, with the slot width defining the adhesive gap - the primary variable in this study (Fig. 6.19). Five gap configurations were tested, ranging from 4.30 mm to 5.10 mm slot width, with three specimens per configuration, giving a total of 15 tests. Laminate thickness was kept consistent across all samples to isolate the effect of the adhesive gap.

Specimen geometry and water-jet cutting accuracy are summarised in Table 6.17, showing a manufacturing precision of ± 0.04 mm. The effective adhesive gaps, derived from coupling measurements, are given in Table 6.18. Surface preparation consisted of sanding and solvent cleaning prior to bonding; the adhesive was cured at room temperature for 24 hours.

6.6.2 Test Setup

Tests were conducted on the Instron 5584 with a 50 kN load cell. One side of the specimen was clamped in wedge-action grips while the opposite side was supported by a custom jig designed to maintain alignment and prevent excess adhesive accumu-



Figure 6.19: Adhesive lap shear specimens

Table 6.17: Specimen geometry and manufacturing deviations

Parameter	Value	Unit
Thickness range	3.65–4.19	mm
CAD slot dimensions	5.10 / 4.90 / 4.70 / 4.50 / 4.30	mm
Mean deviation (waterjet)	0.06 / 0.01 / –0.07 / 0.06 / 0.00	mm
Manufacturing precision	±0.04	mm

Table 6.18: Effective adhesive gap per configuration

Parameter	G1	G2	G3	G4	G5
Coupling dimension [mm]	5.04	4.95	4.75	4.36	4.33
Mean adhesive gap [mm]	0.70	0.62	0.53	0.16	0.30

lation at the interface, ensuring failure occurred within the intended bond-line. The parameters are summarised in Table 6.19 and the setup is shown in Fig. 6.20.

Table 6.19: Adhesive lap shear test parameters

Parameter	Description
Testing machine	Instron 5584, 50 kN load cell
Grips	Mechanical wedge-action tensile grips
Alignment	Bottom face in contact with jig
Crosshead speed	1.3 mm/min
Calibration	Default machine calibration
Data acquisition	50 Hz



Figure 6.20: Specimen in test jig

6.6.3 Results

The relationship between adhesive gap and peak load is presented in Fig. 6.21, revealing a strong and non-linear influence of bond-line thickness on joint performance.

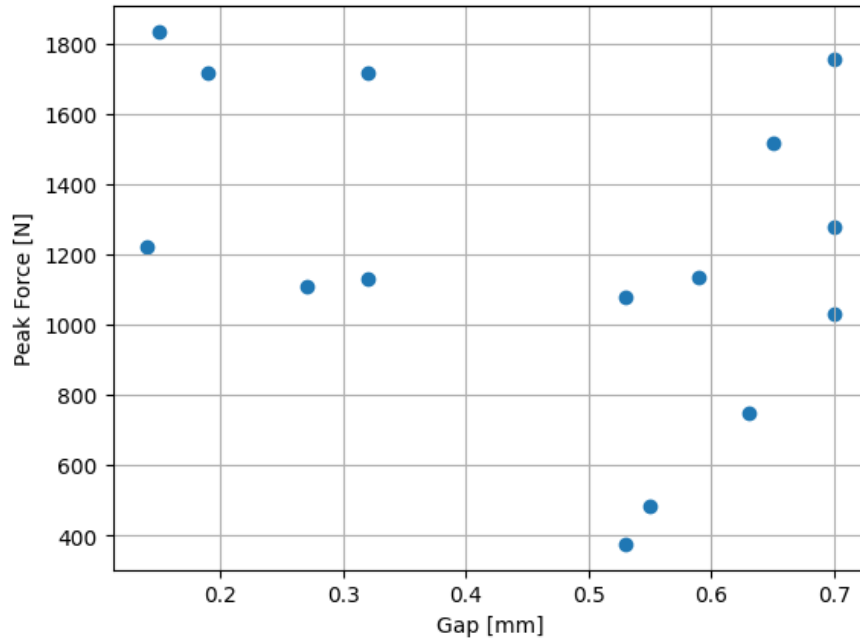


Figure 6.21: Peak force as a function of adhesive gap

Three distinct regimes are observed. For small gaps (0.14–0.32 mm), specimens exhibit the highest load-carrying capacity, reaching up to 1834 N, consistent with efficient load transfer and minimal shear deformation in the adhesive layer. For intermediate gaps (0.50–0.60 mm), peak force drops significantly to as low as 375 N, associated with increased sensitivity to peel and prying effects and higher stress concentrations in the thicker bond-line - consistent with the manufacturer's recommended maximum gap of approximately 0.5 mm [39]. For larger gaps (0.63–0.70 mm), a partial recovery to 1756 N is observed, suggesting a change in failure mechanism where the thicker adhesive layer promotes stress redistribution prior to failure.

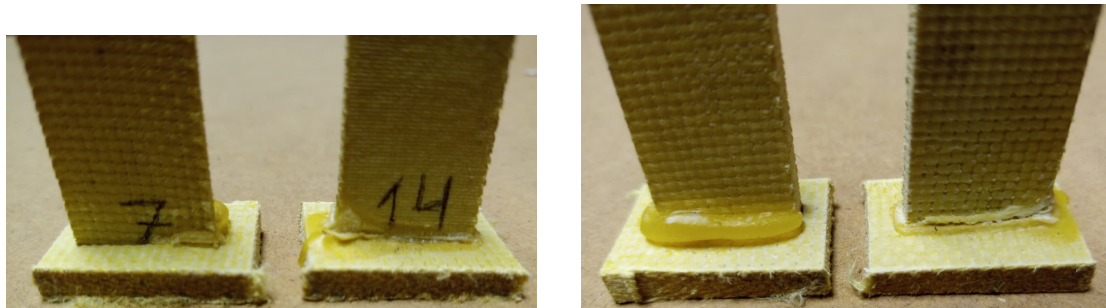


Figure 6.22: Failure behaviour - low peak load specimen 7 vs. high peak load specimen 14

The maximum measured load of 1834 N corresponds to an effective shear strength of approximately 8.6 MPa, compared to the 14 MPa specified in the datasheet for car-

bon fibre substrates [39]. This reduction of approximately 39% is attributed to differences between aramid and carbon fibre surface characteristics, combined with manufacturing and alignment variability inherent to the custom specimen geometry.

Based on these results, an adhesive gap of approximately 0.25 mm with a tolerance of ± 0.04 mm is recommended for panel assembly, corresponding to the small-gap regime where joint strength is maximised. This tolerance is achievable with the water-jet cutting precision demonstrated during specimen preparation.

PRODUCTION

7.1 Laminate Manufacturing

The TSAC walls were produced as flat composite plates using the wet lay-up process with vacuum bagging, consistent with the procedure established during the validation phase. Due to the available glass surface area, the panels were distributed across several separate plates, all manufactured with the $[0/90]_{16}$ layup. A vacuum pressure of approximately 1 bar was achieved throughout, ensuring consistent consolidation. Demoulding was performed carefully to avoid delamination at the plate edges.

One manufacturing limitation was observed in the floor panel, where a localised resin pool formed during layup (Fig. 7.1). Given the large in-plane dimensions and high ply count, controlling resin content uniformly is more demanding than for smaller walls - insufficient resin risks delamination, while excess produces pooling and unnecessary mass. The pool is localised and not expected to significantly affect structural performance, but is noted for completeness.

Following full cure, all plates were sent for waterjet cutting to achieve the precise dimensional tolerances required for the slotted joint geometry, producing the wall fitments, slots, and edge profiles modelled in the design phase and consistent with the adhesive gap targets established experimentally in Chapter 6.6. Representative examples are shown in Figs. 7.2 and 7.3.

7.2 Casing Assembly

Prior to bonding, all slot and edge regions were degreased to ensure adequate adhesive adhesion. The assembly sequence was defined to maintain alignment and prevent distortion during cure.

The internal walls were bonded first (Fig. 7.4), establishing the compartment geometry. During this stage, the outer walls in both directions were temporarily dry-fitted as alignment guides to prevent bending or misalignment. Adhesive was applied



Figure 7.1: Floor panel after cure

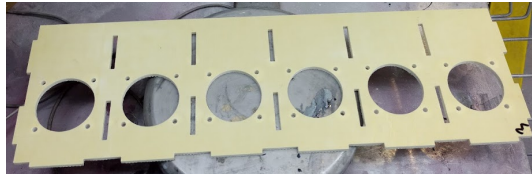


Figure 7.2: Waterjet cut outer wall - X direction



Figure 7.3: Waterjet cut internal wall

to the floor slots and wall bottom edges using a pointed applicator, and a custom 3D-printed spatula was used to remove excess while preserving a 1 mm fillet radius at each interface.

The outer Y walls were then bonded sequentially, followed by the outer X walls. Light pressure was applied at each joint during cure, and a localised heat source was used to reduce the cure cycle between successive bonding steps.

7.3 Cover Manufacturing

The cover was produced over a bespoke MDF mould machined in-house using a Fusion 360 CAM toolpath. Given the cover dimensions, the mould was produced in two halves, bonded together, and the joint rectified with filler paste and sanding (Fig. 7.5). The MDF surface was painted as a sacrificial sealing layer to prevent resin absorption (Fig. 7.6) - varnish was considered but rejected, as the additional thickness would have compromised the small shelf radii. Release agent was applied after the paint reached full cure, and bolt hole positions were pre-machined into the mould for positional accuracy.

A fibre placement template was prepared in advance (Fig. 7.7) to plan the ply cutting pattern and incorporate relief cuts at double-curvature regions, preventing ply

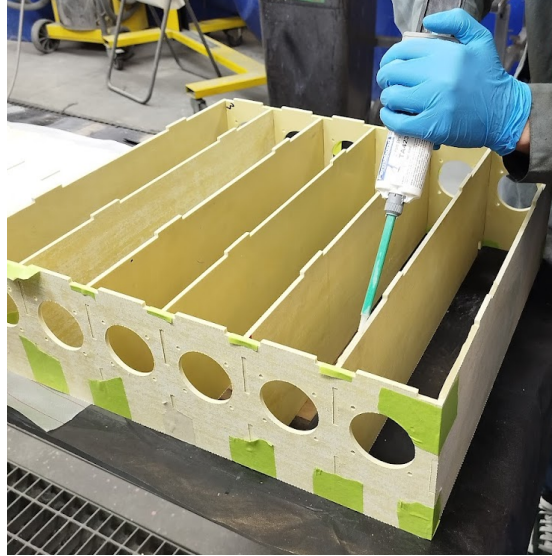


Figure 7.4: *Adhesive application to internal walls*

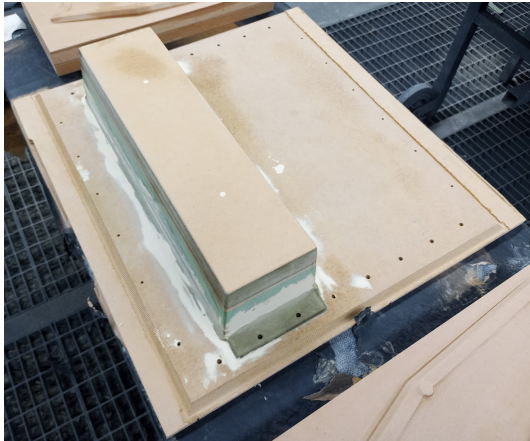


Figure 7.5: *Mould assembly and sanding*

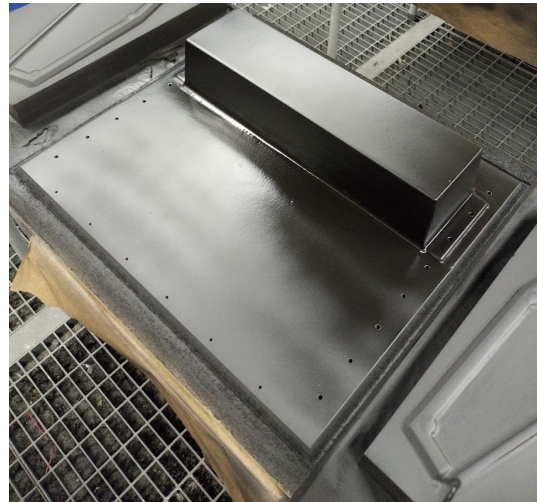


Figure 7.6: *Mould surface painting*

superposition and associated surface irregularities. All fibres were oriented parallel to the longer mould edges at a consistent 0° reference angle (Fig. 7.8). The resin was prepared in smaller batches to extend working time within the 60-minute pot life at approximately 20°C . After completing the 16-ply layup, peel-ply and breather layers were applied and the assembly sealed in a vacuum bag. Demoulding required particular care, as the negative draft angle proved insufficient for single-piece extraction without risk of damage to the component.

Straight edges were trimmed using a table saw referenced to the mould datum edges. High-voltage connector openings were cut by waterjet (Fig. 7.9) due to their complex geometry and central location on the peripheral bump, where table saw access was impractical. Bolt holes were drilled using laminate marks as positional references.



Figure 7.7: Fibre cutting template



Figure 7.8: Cover lamination in progress

7.4 Mountings, L-Profile, and Gasket

The gasket was custom waterjet-cut to match the cover perimeter joint geometry and bolt hole pattern, with additional gaskets produced for the high-voltage connector penetrations (Fig. 7.10).



Figure 7.9: Waterjet cut HV connector openings

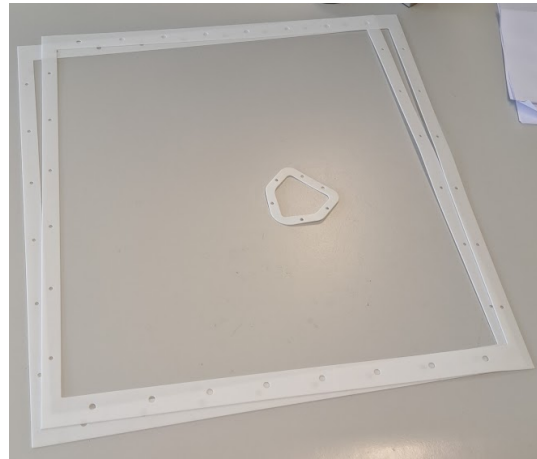


Figure 7.10: Custom-cut cover and HV connector gaskets

The mounting blocks were machined from aluminium stock using Siemens CAM toolpaths and finished on a surface grinder. Bonding faces were abraded and degreased prior to adhesive application. Each mounting was pressed against the wall with both faces in simultaneous contact to ensure perpendicularity; middle mountings were positioned using boundary marks scribed on the wall beforehand.

The L-profile bonding faces were abraded (Fig. 7.11) and degreased. To ensure correct seating, the TSAC was inverted and the L-profile pressed simultaneously against a flat reference surface and the outer wall, aligning it flush with the wall edge so that the cover seats flat under assembly (Fig. 7.12). Bolt positions were transferred from the cover holes, and the shelf attachment holes were drilled through the shelf and L-profile simultaneously to guarantee positional accuracy. Non-structural corner caps manufactured from UL94-V0 compliant PETG were fitted at the L-profile ends to pre-

vent water ingress at the corners.

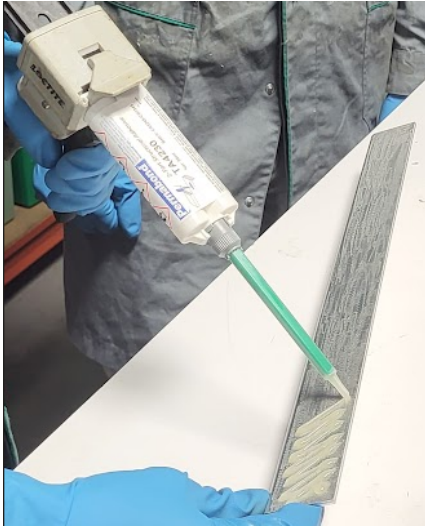


Figure 7.11: *L-profile surface preparation*



Figure 7.12: *L-profile bonding with alignment reference*

7.5 Final Assembly

The completed TSAC is shown in Figs. 7.13 and 7.14. The measured mass of the assembled container is 9.15 kg, excluding fans, representing an excess of 0.69 kg relative to the 8.46 kg target. This difference is consistent with the analytical mass estimate and reflects the design decisions taken to ensure regulatory compliance, structural reliability, and manufacturing feasibility - in particular the adoption of the uniform $[0/90]_{16}$ layup and the shear-rule-driven floor thickness. For a first-generation composite TSAC manufactured entirely in-house, the result demonstrates that the aramid composite approach is viable and competitive, with clear pathways to close the remaining mass gap in future iterations.

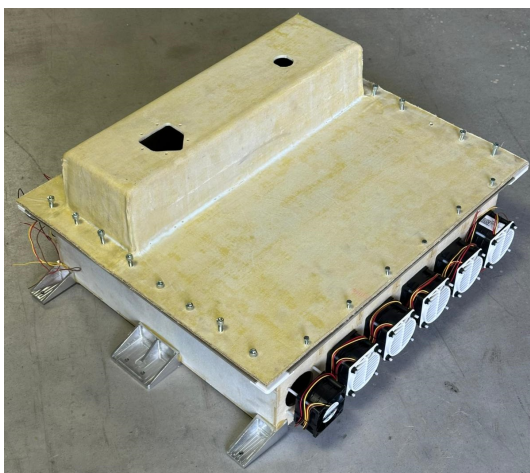


Figure 7.13: *TSAC - front view*

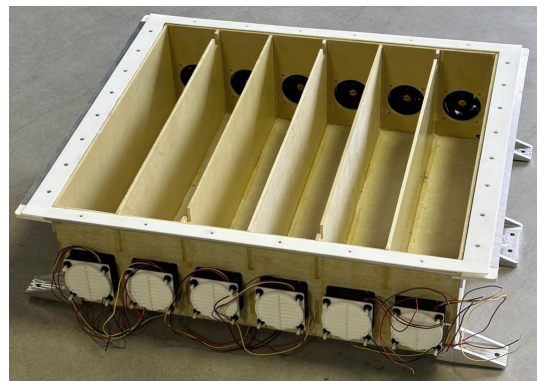


Figure 7.14: *TSAC with cover removed*

CONCLUSION

This work addressed the structural design, analytical dimensioning, numerical validation, experimental characterisation, and physical construction of a composite TSAC for the LART Formula Student electric race car. The primary objective - replacing the previous aluminium enclosure with an aramid composite solution while satisfying all regulatory and structural requirements - was largely achieved, with the mass target closely approached at a measured 9.15 kg against the 8.46 kg goal.

The selected material system - Kevlar 49 plain weave with Sika Biresin CR132FR resin, processed by wet lay-up with vacuum bagging - proved appropriate for the application, providing inherent electrical insulation, UL94 V-0 fire retardancy, and adequate structural performance. Experimental characterisation confirmed an elastic modulus of 19.8 GPa, a tensile strength of 373 MPa, and a density of 1243 kg/m³, all adopted as design values. The discrepancy between these and the micromechanical predictions is consistent with the stiffness and strength knockdown expected from a woven architecture relative to the unidirectional assumption.

The analytical dimensioning, based on Roark's closed-form plate solutions extended to composite laminates through Classical Lamination Theory, successfully identified laminate requirements for all panels. Adhesive shear stress governed the design for most walls, with the floor additionally controlled by the Formula Student shear equivalency rule. The small deflection formulation proved a reliable conservative estimator for both displacement and stress across all panels. The large deflection formulation, while physically more appropriate for the observed deflection ratios, was found non-conservative for stress in orthotropic laminates due to the significant difference between the Roark-assumed Poisson's ratio and the effective laminate value - a systematic limitation that must be accounted for when applying Roark's tabulated solutions to composite panels. The final uniform [0/90]₁₆ layup, adopted to simplify manufacturing and validation, achieved an estimated TSAC structural wall mass of 8.25 kg, with a total system mass of 10.05 kg - exceeding the 8.46 kg target primarily due to the shear-rule-driven floor thickness and the adoption of the uniform layup for manufacturing consistency.

The finite element investigation extended the analytical findings by capturing ef-

fects not accessible through closed-form solutions. Realistic boundary conditions provided substantially more restraint than the analytical idealisations, reducing displacements across most panels. The most significant finding was that adhesive shear under localised contact loading is non-conservative analytically for the Outer X and Floor panels - FEM values exceeded predictions by 128% and 76% respectively - a limitation resolved by the uniform 16-ply layup. At assembly level, structural adequacy was confirmed under X and Z- loading, with the internal wall fitment bonds and corner mounting regions identified as the governing failure locations.

The experimental programme validated the material model and the three-point bending simulation, confirming that the solid FEM model provides a conservative and physically representative prediction of flexural strength. Perimeter shear testing returned an experimental shear strength of 169 MPa - three times the analytical assumption - confirming substantial margin beyond the regulatory minimum and suggesting that rule-driven panels could in principle be reduced, though this was not applied given the overall mass compliance. Flammability, density, and adhesive characterisation tests completed the experimental database required for design validation.

The physical construction confirmed the feasibility of the design and manufacturing approach within the team's available resources. Waterjet cutting achieved the dimensional tolerances required for the slotted joint geometry, and the defined assembly sequence ensured correct wall alignment and adhesive gap control throughout, resulting in a completed enclosure consistent with the design intent.

Overall, the project achieved its primary objectives: a validated composite TSAC compliant with all applicable regulations and manufactured in-house, with a measured mass of 9.15 kg. While the 8.46 kg target was not met, the result demonstrates the viability of the aramid composite approach for this application, with clear pathways to close the remaining mass gap in future iterations. The project additionally established a composite manufacturing baseline within the team that provides a foundation for future structural applications.

8.1 Future Work

Several aspects of the present work identified limitations or opportunities that warrant further investigation:

- **Adhesive characterisation on aramid substrates** - The adhesive shear tests returned a mean strength of 8.6 MPa against a datasheet value of 14 MPa for carbon fibre substrates. A dedicated characterisation programme on aramid-to-aramid bonded joints would reduce the uncertainty in the current design shear strength assumption and may allow the applied knockdown factors to be revised.
- **Large deflection correction for composite laminates** - The non-conservatism of the Roark large deflection formulation for orthotropic laminates, driven by the Poisson's ratio mismatch, represents a broader methodological gap. A correction

factor or modified formulation accounting for the effective laminate ν would improve the accuracy of this widely used design tool for composite plates.

- **Competition validation** - The TSAC has been designed and built but not yet subjected to the full technical inspection and dynamic event cycle at a Formula Student competition. Real-world validation under competition conditions - including rain test, tilt test, and endurance loading - will provide the most complete assessment of the design's performance and durability.
- **Progressive damage modelling** - The current failure assessment uses first-ply failure criteria without accounting for post-first-ply damage accumulation. Incorporating a stiffness degradation model would provide a more accurate prediction of laminate failure progression, particularly relevant for the woven aramid system, which exhibits significant post-damage load-carrying capacity as observed in the three-point bending tests.

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FORMULA STUDENT RULES 2025

T 1.2.1: Fire Retardant - a material meeting one of the following standards:

- UL94 V-0 for the minimum used material thickness
- FAR 25.853(a)(1)(i)
- For foams only: UL94 HF-1 and UL94 VTM-0

Equivalent standards are only accepted, if the team shows equivalence and this is approved by the officials before the competition. (Compliance of alternative materials to above standards can be shown through testing with appropriate documentation, e.g. in the SES/ASES and/or at Technical Inspection).

T 3.2.8: Where bonded joints are applicable and governed by the rules, a 50% reduction must be applied to all nominal and tested adhesive values. Bonded joints in the primary structure must be documented in the SES.

T 3.6.1: If composite materials are used for any part of the primary structure ... or the TSAC the team must:

- Build a representative test panel which must measure exactly 275 mm×500 mm that has the same design, laminate and fabrication method as used for the respective part of the primary structure represented as a flat panel. The sides of the test panel must not be laminated (core material must be visible).
- Perform a 3-point bending test on this panel

The data from these tests and pictures of the test samples and test setup must be included in the SES. In the pictures, the following must be identifiable: distance between the two supports, dimensions of the load applicator and test sample marking as per T3.6.6. The test results must be used to derive strength and stiffness properties used in the SES formula for all laminate panels. Representative test panels for parts of the TSAC may use smaller dimensions, provided that the panel core thickness is 5 mm or smaller. This representative test panel must then measure 150 mm×275 mm. In this case, the distance between the two test panel supports must be at least 200 mm and

the load applicator must have a radius of at least 5 mm ...

T 3.6.6: The test samples must be presented at Technical Inspection. All samples must be marked with the following non-removable (e.g.: permanent marker or engraving, but no sticker) information: laminated structure acronym and date of testing.

T 3.6.11: Perimeter shear tests must be completed which measure the force required to push or pull a 25mm diameter flat punch through a flat laminate sample. The sample must be at least 100 mm × 100 mm...

T 3.6.12: The test fixture must support the entire sample, except for a 32 mm hole aligned co-axially with the punch. The sample must not be clamped to the fixture.

T 10.1.4: Any bolted joint in the primary structure and mounting of the TSAC to the chassis using either tabs or brackets, must have an edge distance ratio “e/D” of 1.5 or greater. “D” equals the hole diameter and “e” equals the distance from the hole centerline to the nearest free edge of the tab or bracket...

EV 3.1.1: TS enclosures, see EV 1.1.2, must consist of either:

- a grounded solid layer made of at least 0.5 mm thick electrically conductive material, aluminium or better, see EV 3.1.3 or
- be made out of electrically insulating materials providing an isolation resistance of at least 2 MΩ, measured at a voltage of 500 V, between the inside and the outside of the enclosure. The TS enclosure must be rigid and must prevent possible mechanical penetrations. Protruding electrically conductive parts, such as fasteners or connectors, must follow EV3.1.2

The TSAC might use at least 0.9 mm thick steel layer as the grounded layer.

EV 5.4.1: If the TSAC is made from an electrically conductive material, the insulation barrier must be adequately protected against conductive penetrations.

EV 5.5.3: All TSAC materials as well as all structural parts used to comply with EV 5.5.8 must be fire retardant, see T 1.2.1. All calculations must be conducted for an ambient temperature of 60 °C except for metallic materials and continuous fiber-reinforced laminates.

EV 5.5.5: TSACs must be constructed of steel or aluminium. With the following requirements:

- The bottom of the TSAC must be at least 1.25 mm thick if made from steel or 3.2 mm if made from aluminium.

- The internal and external vertical walls, covers, and lids must be at least 0.9 mm thick if made from steel or 2.3 mm if made from aluminium.

Alternative materials are allowed with proof of equivalency per T3.4 or for composite materials per EV 5.5.6. When alternative materials are used, test samples must be presented at Technical Inspection.

EV 5.5.6: Composite TSACs must satisfy the following requirements:

- Data obtained from the laminate perimeter shear strength test and three-point bending test, see T 3.6, should be used to prove adequate strength is provided.
- Each attachment point requires steel backing plates with a minimum thickness of 2 mm. Alternate materials may be used for backing plates if equivalency is approved.
- The calculations and physical test results must be included in the ASES.

EV 5.5.15: The TSAC must be mounted using at least 2 attachment points... Any brackets used must be made of either 1.6 mm thick steel or 4 mm thick aluminium and must have gussets to support bending loads.

EV 5.5.16: Holes, both internal and external, in the TSAC, are only allowed for the wiring harness, ventilation, cooling, or fasteners. Holes can extend to the edge of a wall. The TSAC must still be compliant with all other rules, especially the ones concerning its structural requirements. The total cutout area must be below 25% of the area of the respective single wall. External holes must be sealed according to EV4.5.

RECTANGULAR PLATES UDL UNDER LARGE
DEFLECTION [19]

Rectangular plates under uniform load producing large deflection

a/b	Edges and point of max σ	Coef.	qb^4/Et^4											
			0	12.5	25	50	75	100	125	150	175	200	250	
1	Held, not fixed	y/t	0	0.430	0.650	0.930	1.13	1.26	1.37	1.47	1.56	1.63	1.77	
	At center of plate	$\frac{\sigma_d b^2/Et^2}{\sigma b^2/Et^2}$	0 0	0.70 3.80	1.60 5.80	3.00 8.70	4.00 10.90	5.00 12.80	6.10 14.30	7.00 15.60	7.95 17.00	8.60 18.20	10.20 20.50	
1	Held and riveted	y/t	0	0.406	0.600	0.840	1.00	1.13	1.23	1.31	1.40	1.46	1.58	
	At center of plate	$\frac{\sigma_d b^2/Et^2}{\sigma b^2/Et^2}$	0 0	0.609 3.19	1.380 5.18	2.68 7.77	3.80 9.72	4.78 11.34	5.75 12.80	6.54 14.10	7.55 15.40	8.10 16.40	9.53 18.40	
1	Held and fixed	y/t	0	0.165	0.32	0.59	0.80	0.95	1.08	1.19	1.28	1.38	1.54	
	At center of long edges	$\frac{\sigma_d b^2/Et^2}{\sigma b^2/Et^2}$	0 0	0.070 3.80	0.22 6.90	0.75 14.70	1.35 21.0	2.00 26.50	2.70 31.50	3.30 36.20	4.00 40.70	4.60 45.00	5.90 53.50	
	At center of plate	$\frac{\sigma_d b^2/Et^2}{\sigma b^2/Et^2}$	0 0	0.075 1.80	0.30 3.50	0.95 6.60	1.65 9.20	2.40 11.60	3.10 13.0	3.80 14.50	4.50 15.80	5.20 17.10	6.50 19.40	
	Held, not fixed	y/t	0	0.625	0.879	1.18	1.37	1.53	1.68	1.77	1.88	1.96	2.12	
1.5	At center of plate	$\frac{\sigma_d b^2/Et^2}{\sigma b^2/Et^2}$	0 0	1.06 4.48	2.11 6.81	3.78 9.92	5.18 12.25	6.41 14.22	7.65 16.0	8.60 17.50	9.55 18.90	10.60 20.30	12.30 22.80	
2 to ∞	Held, not fixed	y/t	0	0.696	0.946	1.24	1.44	1.60	1.72	1.84	1.94	2.03	2.20	
	At center of plate	$\frac{\sigma_d b^2/Et^2}{\sigma b^2/Et^2}$	0 0	1.29 4.87	2.40 7.16	4.15 10.30	5.61 12.60	6.91 14.60	8.10 16.40	9.21 18.00	10.10 19.40	10.90 20.90	12.20 23.60	
1.5 to ∞	Held and fixed	y/t	0	0.28	0.51	0.825	1.07	1.24	1.40	1.50	1.63	1.72	1.86	
	At center of long edges	$\frac{\sigma_d b^2/Et^2}{\sigma b^2/Et^2}$	0 0	0.20 5.75	0.66 11.12	1.90 20.30	3.20 27.8	4.35 35.0	5.40 41.0	6.50 47.0	7.50 52.50	8.50 57.60	10.30 67.00	

