

Design and Structural Analysis of a Monocoque for a Formula Student Car

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Abstract

This work presents the design and structural analysis of a carbon fibre monocoque chassis for a Formula Student car, developed as a replacement for the steel spaceframe employed in previous FSFEUP prototypes. The primary objective was to establish and validate an analytical tool capable of sizing sandwich composite panels in compliance with the Structural Equivalency Spreadsheet (SES) requirements. The work integrates analytical, experimental, and numerical approaches to ensure accuracy and applicability.

A MATLAB-based framework was created to implement Classical Laminate Theory and several composite failure criteria, enabling predictions of stiffness, strength, and energy absorption. Input data were obtained through an extensive experimental campaign on CFRP laminates, including tensile, compressive, and shear tests, with strains measured using digital image correlation and strain gauges. Sandwich panel performance was further validated through three-point bending and perimeter shear tests, allowing direct comparison with analytical predictions. While stiffness predictions proved reliable, strength estimations were systematically higher than the values obtained experimentally, requiring high safety factors in the sizing process.

The validated framework was then applied to the full chassis design under the geometric and packaging constraints of the FSFEUP 02 prototype. The resulting monocoque achieved a mass of approximately 34.6 kg, compared with 37.0 kg for the steel spaceframe, while tripling torsional stiffness, from 2201 Nm/° to 6685 Nm/°. Although these results confirmed the inherent efficiency of the monocoque concept, the conservative layups adopted led to excessive stiffness and limited mass reduction, highlighting significant potential for optimisation through tailored layup strategies.

Overall, the study demonstrates both the feasibility and advantages of transitioning to a composite monocoque within FSFEUP. Beyond enabling the design of an SES-compliant structure, the developed methodology provides a transferable workflow that combines analytical modelling, experimental characterisation, and numerical simulation. This framework equips the team with a robust foundation for future optimisation, supporting enhanced performance and competitiveness in Formula Student.

Resumo

Este trabalho apresenta o projeto e a análise estrutural de um chassi monocoque em fibra de carbono para um carro Formula Student, desenvolvido como substituição do chassi tubular em aço utilizado em protótipos anteriores da FSFEUP. O principal objetivo consistiu em estabelecer e validar uma ferramenta analítica capaz de dimensionar painéis sanduíche em compósito em conformidade com os requisitos da Structural Equivalency Spreadsheet (SES). O trabalho integra abordagens analíticas, experimentais e numéricas de forma a garantir precisão e aplicabilidade.

Foi desenvolvida uma ferramenta em MATLAB para implementar a Teoria Clássica dos Laminados e diversos critérios de falha de compósitos, possibilitando a previsão da rigidez, resistência e capacidade de absorção de energia. Os dados necessários foram obtidos através de uma extensa campanha de testagem experimental em laminados de CFRP, incluindo ensaios de tração, compressão e corte, com medições de deformação realizadas por correlação de imagem digital e extensometria. O desempenho dos painéis sanduíche foi adicionalmente validado por ensaios de flexão em três pontos e de corte perimetral, permitindo a comparação direta com as previsões analíticas. Embora as previsões de rigidez se tenham revelado fiáveis, as estimativas de resistência mostraram-se sistematicamente superiores ao valor medido experimentalmente, exigindo fatores de segurança elevados no processo de dimensionamento.

A ferramenta validada foi posteriormente aplicada ao projeto integral do chassi, considerando as restrições geométricas e de packaging do protótipo FSFEUP 02. O monocoque resultante atingiu uma massa de aproximadamente 34,6 kg, em comparação com 37,0 kg do chassi tubular em aço, triplicando a rigidez torsional, de 2201 Nm/° para 6685 Nm/°. Embora estes resultados confirmem a eficiência inerente do conceito do chassi monocoque, as configurações de laminado conservadoras adotadas conduziram a uma rigidez excessiva e a uma redução de massa limitada, evidenciando um elevado potencial de otimização através de estratégias de laminado mais ajustadas.

Em suma, o estudo demonstra tanto a viabilidade como as vantagens da transição para um monocoque em compósito no âmbito da FSFEUP. Para além de viabilizar o projeto de uma estrutura compatível com os requisitos da SES, a metodologia desenvolvida fornece um fluxo de trabalho transferível que combina modelação analítica, caracterização experimental e simulação numérica. Este enquadramento dota a equipa de uma base sólida para futuras otimizações, promovendo melhorias de desempenho e competitividade na Formula Student.

UN Sustainable Development Goals

The United Nations Sustainable Development Goals (SDGs) provide a global framework to achieve a better and more sustainable future for all. It includes 17 goals to address the world's most pressing challenges, including poverty, inequality, climate change, environmental degradation, peace, and justice.

This thesis on the design and structural analysis of a carbon fibre monocoque for a Formula Student car supports several SDGs by promoting structural innovation, material efficiency, energy reduction potential, and hands-on engineering education.

SDG 4 Ensure inclusive and equitable quality education and promote lifelong learning opportunities for all

SDG 9 Build resilient infrastructure, promote inclusive and sustainable industrialization and foster innovation

SDG 12 Ensure sustainable consumption and production patterns

SDG 13 Take urgent action to combat climate change and its impacts

SDG	Target	Contribution	Performance Indicators and Metrics
4	4.4	Participation in the Formula Student project supports the development of technical and vocational skills relevant to engineering employment and innovation.	Acquisition of design, simulation, and composite analysis skills through practical application.
9	9.5	Development of a CFRP monocoque fosters innovation in lightweight structural design and application of advanced composite materials.	Use of advanced composite solutions; improvements in torsional stiffness and weight reduction.
12	12.2	Structural optimisation contributes to reducing material usage while maintaining performance targets.	Material volume and mass required to meet stiffness targets.
13	13.3	The lightweight design of the monocoque improves structural efficiency, reducing energy demands in future electric or hybrid vehicles.	Structural mass reduction relative to space-frame designs; potential impact on energy consumption.

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Chapter 1

Introduction

This master's thesis in mechanical engineering is developed within the context of the Formula Student competition, as part of the research and development efforts of the FSFEUP team, the Formula Student team representing the Faculty of Engineering of the University of Porto (FEUP). The focus of this study is the design and structural analysis of a carbon fibre monocoque chassis, a critical component in high-performance race cars. As the team aims to improve its future prototypes, transitioning from traditional space frame chassis to lightweight and high-stiffness composite structures, this research serves as an opportunity to deepen knowledge in advanced materials and structural optimisation. By exploring the engineering principles behind carbon fibre monocoque construction, this work contributes to the team's long-term development strategy, aligning with the latest trends in motorsport and automotive engineering.

1.1 Contextualisation

The Formula Student (FS) competition is one of the most well established educational engineering challenges in the world. It is designed to test students' ability to design, build, and race a single-seater race car while applying real-world engineering principles. Unlike traditional motorsport, Formula Student does not simply reward the fastest car; rather, the competition evaluates a team's ability to develop a well-rounded engineering solution that balances performance, cost-effectiveness and manufacturability [1].

The Formula Student concept originated in 1981 when the Society of Automotive Engineers (SAE) in the United States launched the Formula SAE competition. Since then, the event has expanded globally, with regional editions in Europe, Japan, Brazil, and Australia [2]. The European iteration of the competition began in 1998, when the Institution of Mechanical Engineers (IMechE) partnered with SAE to organise Formula Student UK. The competition has taken place annually, growing significantly in prestige and participation. In 2007, the event moved to the Silverstone Circuit, home of the British Grand Prix, providing students with the opportunity to work in a professional motorsport environment [3].

Today, Formula Student Germany (FSG) and Formula Student UK (FSUK) are among the most competitive and highly regarded events. With over 100 university teams participating each year, the competition continues to evolve, introducing new categories such as autonomous vehicle challenges (FS-AI) and electric vehicle development, reflecting industry trends towards sustainable and high-tech mobility solutions.

1.2. Motivation

The educational impact of Formula Student cannot be overstated. It provides students with hands-on engineering experience, bridging the gap between academic learning and professional industry demands. Many former participants have gone on to work in Formula 1, high-performance automotive companies, and research institutions, making Formula Student a recognised standard for graduate engineers [4].

Within this highly competitive environment, FS FEUP was founded in 2021 to represent the University of Porto in Formula Student, providing students with a platform to apply engineering knowledge to real-world challenges and continuously improve vehicle design and performance.

The team quickly gained momentum, securing its first international awards in 2022 through design and project competitions with the FS FEUP 01. This prototype, featuring a single-motor EV rear-wheel-drive powertrain, a steel space-frame tubular chassis, and carbon fibre suspension, laid the foundation for future developments. The successful completion of the team's first competitive season demonstrated its capabilities in both design and manufacturing. Building on this experience, FS FEUP 02 introduced significant weight reductions, aerodynamic improvements, and autonomous driving features, achieving further recognition in concept and project classes.

1.2 Motivation

FS FEUP aims to establish itself as a top-10 team at Formula Student Germany (FSG). This ambitious objective drives the team towards the development of a carbon fibre monocoque chassis for the FS FEUP 03 prototype, marking a new phase in its pursuit of technological innovation and engineering excellence.

The chassis is a fundamental component of the Formula Student car, directly influencing structural integrity, weight distribution, and overall performance. While FS FEUP is transitioning towards a monocoque design for future prototypes, different teams employ various chassis configurations, each with distinct advantages and limitations. Understanding these design choices is essential to contextualise the motivation behind adopting a carbon fibre monocoque chassis.

Over the past decades, three primary chassis designs have been utilised in motorsport: space-frame, monocoque and hybrid. Each of these designs has distinct structural principles, advantages and drawbacks that influence their suitability for different applications [5].

The space-frame chassis consists of a three-dimensional tubular structure, where individual members are connected at nodes to form a rigid framework. This design relies heavily on the principle of triangulation, where structural loads are primarily carried in tension and compression, minimising bending forces. One of the key advantages of a space-frame chassis is its relatively low cost and ease of fabrication, as it can be constructed using steel or aluminium tubing without requiring highly specialised manufacturing processes. However, space-frame chassis tend to be heavier than their monocoque counterparts due to the use of numerous interconnected members in order for the structure to be compliant with the competition's rulebook. Furthermore, their external bodywork is non-structural, leading to additional weight and potential aerodynamic inefficiencies. While a well-designed space-frame can achieve significant rigidity, its packaging constraints can be a limiting factor, as the tubular members often interfere with the placement of other vehicle components.

The monocoque chassis (from the French *mono*, meaning “single”, and *coque*, meaning “shell”) is a self-supporting structure in which the continuous surface itself carries the structural loads.

Unlike space frames, which rely on discrete tubular members, monocoques employ stressed panels and shells that distribute forces across the entire geometry. This integrated load-bearing surface provides an efficient stiffness-to-weight ratio and allows for greater design freedom compared to traditional framed structures. This approach offers superior stiffness-to-weight ratio, particularly when manufactured from carbon fibre composites. The absence of a separate support frame allows for greater design flexibility and aerodynamic efficiency, as the chassis and bodywork can be seamlessly integrated. Additionally, the improved impact resistance of a monocoque enhances driver safety, a crucial aspect in high-performance racing applications. However, monocoque structures come with notable challenges. The manufacturing process is significantly more complex and expensive, requiring advanced materials, precise moulding techniques, and specialised curing processes such as autoclave treatment. Furthermore, repairs and maintenance are significantly more challenging than in space frames. Damage to a monocoque chassis often requires extensive reconstruction rather than simple part replacement, and in the event of a structural failure, not uncommon in Formula Student, the chassis may be impossible to repair within the tight schedule, potentially compromising the team's ability to compete. Finally, structural openings, such as those required for driver access and component mounting, can reduce torsional stiffness unless reinforced with internal ribs and bulkheads.

A hybrid chassis combines elements of both space frame and monocoque designs, typically featuring a monocoque cockpit section with a space frame rear for powertrain and suspension mounting. This approach offers a compromise between manufacturing complexity and performance improvements. It allows teams to benefit from the increased stiffness of a monocoque while maintaining the cost-effectiveness of a tubular frame. However, the transition between the two structures introduces challenges in terms of load distribution and structural integration.

Each of these chassis configurations presents a unique set of trade-offs, influencing the decisions made by Formula Student teams based on their design priorities and available resources. While space-frame chassis remain a cost-effective and accessible solution, monocoques provide significant performance advantages in terms of weight reduction, stiffness, and aerodynamics.

The top-performing teams at Formula Student Germany (FSG) consistently utilise monocoque chassis constructed from composite sandwich panels, typically featuring carbon fibre reinforced polymer facesheets with an aluminium honeycomb core. This combination offers an exceptional strength-to-weight ratio while ensuring high torsional rigidity and optimised aerodynamic efficiency, allowing these teams to excel in dynamic events.

To achieve FS FEUP's long-term objective of becoming a top-10 team at FSG, it is essential to transition from a steel space-frame chassis to a carbon fibre monocoque. This shift will enable the team to develop a lighter, stiffer, and more competitive prototype, aligning with the engineering standards set by the most successful teams in the competition.

1.3 Objectives and Constraints

The primary objective of this work is to develop a comprehensive understanding of sandwich panel sizing methodologies applicable to the design of a composite monocoque chassis for Formula Student applications. This study is motivated by FSFEUP's strategic transition from a space frame chassis to a composite monocoque, aiming to identify an optimised structural solution that is compliant with the Formula Student Structural Equivalency Spreadsheet (SES) requirements.

The project focuses on conducting detailed structural analysis and optimisation of the chassis concept under the specific geometric and packaging constraints of the FSFEUP 02 prototype. This

1.4. Thesis Layout

ensures direct comparability between the proposed monocoque and the steel space frame developed during the same design cycle. While maximum performance is not the primary focus, the chassis is required to meet key performance targets, specifically maintaining a mass below 35 kg and achieving a torsional stiffness of at least 2000 Nm/°, ensuring adequate structural rigidity for good handling and overall vehicle performance.

To simplify integration with existing vehicle components and validate the design as a proof of concept, the chassis geometry will adhere to the constraints defined by the FSFEUP 02 prototype. This includes maintaining the same suspension hardpoints, as well as retaining the nose cone geometry and driver position. The design will also replicate the main hoop and bracing structure used in FSFEUP 02, while accommodating an inboard tractive system configuration.

Key objectives include:

- Applying analytical and numerical methods to size composite sandwich panels for the monocoque structure.
- Ensuring full compliance with Formula Student SES and related safety requirements.
- Performing a comparative assessment between the optimised composite monocoque and the existing FSFEUP 02 space frame in terms of torsional stiffness and mass.

This approach provides a framework for informed decision-making regarding the feasibility and performance potential of a composite monocoque within the team's design and competition constraints.

1.4 Thesis Layout

This thesis is structured to present the development, validation, and application of analytical tools for the structural sizing of composite sandwich panels in the context of a Formula Student monocoque chassis.

Chapter 2 provides a review of the state-of-the-art in composite monocoque design for Formula Student applications, including material selection, geometric design approaches, and sandwich panel sizing methodologies adopted by other teams.

Chapter 3 details the analytical framework developed for panel sizing. It describes the regulatory background, the theoretical formulations based on Classical Laminate Theory and composite failure models, and their implementation in a MATLAB tool capable of predicting flexural rigidity, three-point bending strength, energy absorption and perimeter shear resistance.

Chapter 4 presents the validation of the developed tools. This includes the mechanical characterisation of the selected CFRP material, the manufacture of representative sandwich panels, and the comparison between analytical predictions and experimental test results for both three-point bending and perimeter shear.

Chapter 5 applies the validated sizing methodology to the conceptual development of an SES-compliant carbon fibre monocoque. The process considers FSFEUP 02 geometric constraints, defines the layups for each chassis section, and evaluates the resulting structure in terms of torsional stiffness and mass, enabling direct comparison with the steel space frame of the same design cycle.

Finally, Chapter 6 summarises the key findings of the work, discusses its limitations, and outlines recommendations for future development.

Chapter 2

State-of-Art and Literature Review

This chapter provides a review of the existing literature relevant to the design, analysis, and manufacturing of monocoque chassis, with a focus on Formula Student applications. It begins with an overview of monocoque chassis materials, discussing their structural principles, advantages, and material evolution, from steel and aluminium to advanced composites such as carbon fibre-reinforced polymer (CFRP). The chapter then examines analytical and numerical modelling approaches employed for structural validation and optimisation. Finally, it reviews the experimental tests commonly used to assess the structural performance of CFRP monocoques.

2.1 Monocoque Chassis Materials

Throughout automotive history, monocoque chassis have been constructed from various materials, each offering distinct mechanical properties and manufacturing considerations.

Steel was one of the earliest materials used in monocoque construction, with the 1922 Lancia Lambda being the first production car to adopt such techniques [6]. However, only a thousand units were produced. The first mass-produced unibody car was the 1934 Citroën Traction Avant [7]. While steel monocoques provided good crash resistance and affordability, their high density limited their application in lightweight racing prototypes, prompting the transition to lighter materials.

Aluminium alloy monocoque chassis offer significant advantages over traditional steel constructions, primarily due to the lower density of aluminium, which enables substantial weight reduction without compromising too much on structural rigidity. This weight savings enhances vehicle performance, fuel efficiency, and handling dynamics. The pioneering application of an aluminium alloy monocoque in motorsports was the Lotus 25 Formula 1 car, introduced in 1962 [8]. In the realm of production vehicles, the Honda NSX, launched in 1990, was the first mass-produced car to feature an all-aluminium monocoque body [9]. An example of a Formula Student team that has successfully implemented an all-aluminium monocoque chassis in competition is the Cardiff University Formula Student team. Their aluminium honeycomb monocoque design demonstrated a practical and cost-effective alternative to traditional space-frame and composite monocoque configurations, showcasing the material's potential for achieving a balance between weight reduction, manufacturing simplicity, and structural performance [10].

2.2. Design Techniques for Composite Monocoque Chassis

Magnesium alloy monocoques have seen limited application in motorsport, with one of the most notable examples being the Honda RA302, a highly innovative Formula 1 car that featured a magnesium-skinned monocoque [11]. This design aimed to take advantage of magnesium's exceptionally low density, which offered significant weight savings compared to aluminium and steel. However, despite its potential, magnesium monocoques were not widely adopted in racing or production cars due to several critical drawbacks. Magnesium is highly flammable, especially when exposed to intense heat or friction, posing significant safety risks in the event of a crash. Additionally, the material is prone to corrosion and requires specialized coatings and treatments to ensure durability.

Fibreglass monocoques, such as the one implemented in the Lotus Elite [12], marked the introduction of composite materials into automotive design. However, despite their advantages in weight reduction and manufacturability, they exhibited limited mechanical strength, making them less suitable for high-stress applications. In the context of Formula Student, a conceptual design for a fibreglass monocoque was developed by the AAU Racing team for their G8 race car. While the new structure improved stiffness compared to the original space frame chassis, it also resulted in increased mass, offsetting some of the potential performance benefits [13].

The most prevalent material in modern motorsports monocoque chassis is carbon fibre reinforced polymers (CFRP). CFRP is extensively used due to its high stiffness-to-weight ratio, excellent energy absorption, and aerodynamic flexibility. Unlike metal counterparts, carbon fibre composites allow engineers to tailor mechanical properties through laminate orientation and layup configurations, enabling precise control over strength, rigidity, and deformation characteristics [14]. This makes CFRP the material of choice in the highest levels of motorsport, including Formula 1, Le Mans Hypercar (LMH), GT3, and IndyCar, as well as Formula Student, where weight savings and chassis stiffness are critical for performance [15]. All top ten teams from FSG24 employed CFRP monocoque chassis designs, with the top three positions secured by AMZ, FS Team Tallinn, and EForce Prague Formula [16]. Beyond motorsport, CFRP is also widely adopted in high-performance road cars from manufacturers such as McLaren, Ferrari, Lamborghini, Koenigsegg and many more, reflecting its superior structural efficiency and lightweight advantages in automotive engineering.

In the context of CFRP monocoque chassis, designs based on sandwich panels have distinct advantages over monolithic laminates. While monolithic CFRP constructions can achieve high in-plane stiffness, their out-of-plane bending resistance is limited by the compact placement of plies. By contrast, sandwich panels introduce a lightweight core that separates the skins, which substantially increases the second moment of inertia of the section. This geometric effect enhances flexural rigidity without a proportional mass penalty, enabling structures that are simultaneously stiffer and lighter [17]. For this reason, sandwich-based designs have become the prevailing solution in Formula Student and other motorsport applications, where maximising torsional stiffness while minimising weight is critical to performance.

2.2 Design Techniques for Composite Monocoque Chassis

To support the development of this project's composite monocoque, a comparative study was conducted on multiple publicly available Formula Student theses and technical reports. These documents provided valuable insights into the methodologies employed by other teams in the design and analysis of composite chassis structures. Particular attention was given to five key areas: geometric design methods, composite system selection, sandwich core material selection, adhesive selection, and sandwich panel sizing methods. The information collected from these

sources is synthesised in the subsections below, with the aim of identifying best practices, common challenges, and effective solutions that can inform and justify the design choices made in this dissertation.

2.2.1 Geometric Design Methods

The geometric design of a composite monocoque chassis in Formula Student vehicles is inherently multidisciplinary, requiring a balance between structural integrity, subsystem packaging, ergonomics, manufacturability, and aerodynamic performance. Most teams adopt an iterative design process based on CAD, often coupled with FEA tools, to develop a chassis that complies with competition rules while optimising performance and manufacturability. The use of parametric sketches and cross-sectional profiles was a common strategy to maintain geometric consistency and expedite iterative design changes.

A common strategy among teams is to adopt an iterative, subsystem-driven approach to monocoque geometric design, beginning with human-factor constraints and refining the layout through progressive integration of suspension, powertrain, and manufacturing requirements. For instance, Cal Poly Formula SAE defined their initial geometry based on 95th percentile male templates and optimised it via torsional stiffness sensitivity studies, packaging constraints, and continuous feedback from subsystem leads, resulting in over 70 documented revisions [18]. FSIPLeiria followed a similar path, starting from rulebook templates and ergonomic assessments, and validating driver positioning with a mock-up rig before progressing to detailed CAD work [19]. Squadra Corse PoliTO also employed a rules-compliant and manufacturability-focused design cycle, prioritising simplified geometry with minimal curvature to streamline mould preparation and enable faster prototyping [20]. In contrast, Metropolia Motorsport introduced a parametric CAD model from the outset, enabling more efficient iterations in response to evolving packaging constraints and ergonomic inputs. This model was complemented by physical validation of driver position using a rig, and final recommendations encouraged merging the initially segmented chassis geometry to ease tooling complexity [21]. Despite variations in emphasis and tooling strategies, these teams exemplify a shared methodology grounded in ergonomic foundations, rule compliance, and iterative geometry refinement.

Monash Motorsport distinguished itself from other teams by integrating finite element analysis (FEA) directly into the geometric design phase, using simulation not only to validate but to actively shape the chassis layout. Rather than relying solely on ergonomic and packaging constraints to define the geometry, FEA was used to guide performance optimisation through structural load path analysis and stiffness driven shape refinement. High-level design constraints—such as suspension hardpoints, packaging boundaries, and safety envelope—were still considered, but the geometry was iteratively updated in conjunction with FEM-based sizing tools to align structural efficiency with overall vehicle objectives [22]. This simulation informed workflow contrasts with the more conventional CAD first approaches observed in other teams, where FEA typically enters the process only after the geometry has been established.

In some cases, the aerodynamic envelope also informed geometry decisions. While some teams like Cal Poly Formula SAE prioritised the undertray design and evaluated geometry alternatives through CFD [23], others such as Oxford Brookes Racing acknowledged aerodynamic limitations but prioritised structural optimisation instead [15]. Raised noses, rear diffusers, and unified shapes were common features adopted to support efficient airflow without compromising compliance or manufacturability.

To ensure manufacturability, fillets are introduced, draft angles are reviewed, and the surface

2.2. Design Techniques for Composite Monocoque Chassis

complexity is deliberately minimised to avoid excessive curvature and improve laminate quality [24].

Serviceability and accessibility are also critical considerations. Several teams explicitly re-designed bulkheads and rear bays to facilitate faster assembly and easier maintenance access to components such as the accumulator and inverters [21]. These changes often led to modified cut-outs and removable sections, which were balanced against the need to preserve structural stiffness and simplify manufacturing.

The geometric design process typically involves multiple feedback loops, both internal to the chassis department, such as those driven by FEM simulations for stiffness and CFD analyses for aerodynamic performance, and external, through continuous coordination with other subsystems to ensure compatibility with packaging, ergonomics, and load path requirements. A generalised scheme illustrating these feedback loops is presented in Figure 2.1. While not all teams follow every loop shown, the diagram provides an overview of the typical interactions and design iterations observed across various methodologies.

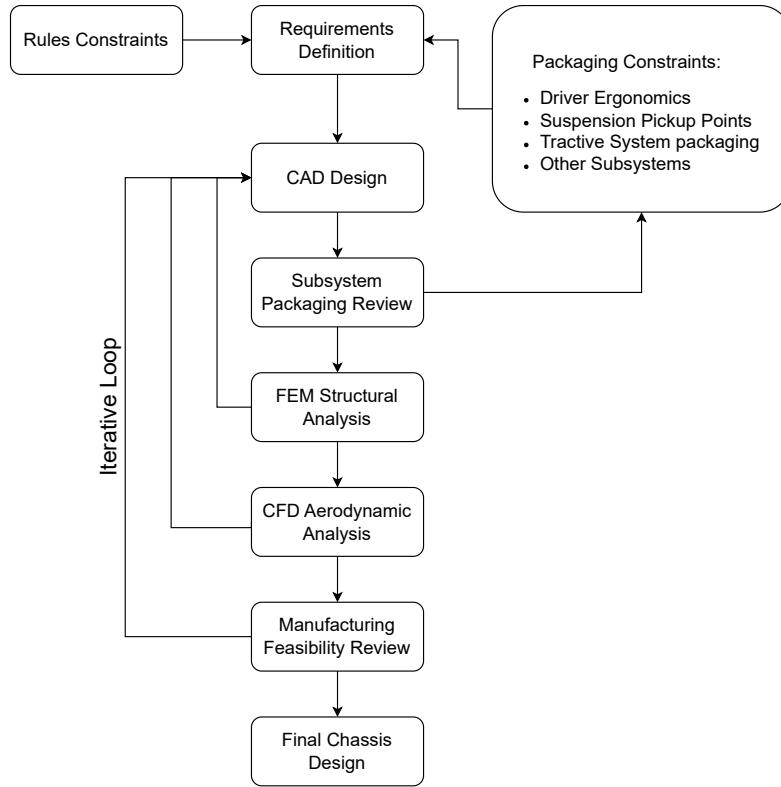


Figure 2.1: Generalised schematic of the feedback loops typically involved in the design of a composite monocoque chassis.

Overall, the geometric design of composite monocoques in Formula Student follows a pragmatic approach. Initial models are driven by ergonomic and regulatory constraints, refined through subsystem feedback, and optimised for structural stiffness and manufacturability. While some variation exists in prioritisation - whether aerodynamics, stiffness, or serviceability - the methodologies consistently demonstrate a systems-level integration philosophy and a strong reliance on iterative CAD development.

2.2.2 Composite System Selection

Carbon fibre-reinforced polymers (CFRPs) were the most commonly adopted material system across teams, primarily due to their high specific stiffness and strength, combined with a well-established track record in motorsport applications. Several teams compared fibre types using property-to-density ratios to identify the most suitable reinforcement for different chassis zones. For instance, Metropolia Motorsport Formula Student [21] performed a comparative evaluation of carbon, glass, aramid, basalt, and flax fibres, ultimately selecting carbon due to its superior stiffness and compatibility with existing laminate manufacturing expertise.

The majority of teams opted for prepreg systems, citing benefits such as improved workability and resin uniformity. Chalmers Formula Student [25] and Cal Poly Formula SAE [18] both relied on prepregs to ensure consistent mechanical properties and to simplify the layup process, especially for large components. FSIPLeiría [19] also highlighted the practicality of using woven prepregs for conforming to complex geometries.

Regarding fibre modulus, KTH Formula Student [26] proposed the use of both high modulus (HM) and high strength (HS) fibres depending on the structural role. Stiffness-critical areas benefited from HM fibres to meet regulatory requirements, while energy absorption zones like the Side Impact Structure (SIS) used HS fibres. However, due to cost and availability, most teams selected intermediate-grade carbon fibres, which offer a balance between performance and affordability.

In some cases, material sponsorship played a decisive role in fibre selection. MIT Motorsports [24], for instance, considered the use of Toray fibres with the strategic aim of establishing a future partnership. The team selected the 2510 system, citing its low-temperature cure cycle and compatibility with out-of-autoclave processing as key advantages.

Across all reviewed teams, fibre selection was typically driven by a combination of structural performance requirements, manufacturing feasibility, and material availability. Stiffness was generally the most influential factor, particularly in regions subjected to bending and torsional loads, followed by strength considerations in energy-absorbing zones. Cost-effectiveness and accessibility were also critical, especially for student teams operating under tight budget constraints. Workability and geometric conformity—especially around complex moulds—often favoured woven fabrics over unidirectional plies in certain regions. Consequently, most teams converged on intermediate-modulus (IM) carbon fibres in prepreg form. Twill weaves were the most commonly used reinforcement, frequently employed on outer plies due to their excellent drapability. Several teams also combined UD materials with fabrics in areas requiring directional reinforcement, balancing manufacturability with mechanical performance.

2.2.3 Sandwich Core Material Selection

The selection of core materials is primarily driven by considerations of shear stiffness, compressive strength, and compatibility with prepreg cure temperatures. Most teams converged on aluminium honeycomb as the optimal core material due to its significantly higher mechanical performance compared to foam alternatives. For instance, Cal Poly Formula SAE [18], FSIPLeiría [19], and Metropolia Motorsport Formula Student [21] all compared aluminium honeycomb to polymeric cores like PET and Divinycell, concluding that the former provides an order-of-magnitude improvement in shear stiffness at equivalent densities.

However, teams acknowledged that aluminium honeycomb offers far less flexibility in terms of design, as it cannot be readily shaped into complex geometries like foam cores, which can be

2.2. Design Techniques for Composite Monocoque Chassis

machined into virtually any contour. While this limits its adaptability, aluminium honeycomb avoids the long machining times associated with foams and provides superior specific stiffness and strength, making it an attractive option when performance takes precedence over geometric versatility. Metropolia Motorsport Formula Student [21] noted the need to account for its anisotropic properties, especially in shear, and adapted their simulations accordingly. A key concern was ensuring that the core would not degrade at the elevated cure temperatures of CFRP prepregs, ruling out several common foam cores [26].

Depending on structural and geometry requirements, different cell sizes and thicknesses were adopted across panels [21].

2.2.4 Adhesive Selection

Though often briefly mentioned, adhesive films play a critical role in ensuring proper bonding between face sheets and the core. KTH Formula Student [26] highlighted the need to account for the additional weight introduced by adhesive layers, especially in lightweight structures such as a monocoque chassis. Similarly, Cal Poly Formula SAE [18] evaluated whether it was feasible to not use adhesive films altogether but ultimately decided against it due to concerns over structural integrity. In principle, the skins could be co-cured directly with the honeycomb core; however, this practice is generally avoided because under autoclave pressure the resin may infiltrate the core cells. Such infiltration degrades the mechanical properties of the honeycomb, increases the mass of the monocoque, and can generate resin voids within the fibres, thereby compromising the structural integrity of the chassis. Moreover, curing the skins and core together limits the processing pressure that can be applied, since excessive pressure risks crushing the lightweight honeycomb. The application of an adhesive film not only ensures reliable bonding but also provides a physical barrier that prevents resin ingress into the cells.

In most cases, adhesive selection was closely tied to the cure cycle of the chosen prepreg system, as the adhesive had to withstand the same thermal and pressure conditions. Teams selected compatible films designed for co-curing with CFRP, prioritising reliable bonding over cost or weight minimisation.

2.2.5 Sandwich Panel Sizing Methods

Across high-performance sectors, sizing of sandwich panels follows the same physics but differs in what is optimised and how performance is verified. Classical sandwich theory partitions roles between facesheets and core: facesheets carry in-plane forces and bending, whilst the honeycomb core carries transverse shear and stabilises the faces against local instabilities [27]. Within this common framework, three tightly coupled sizing decisions dominate: number of plies, ply orientation schedule, and core thickness.

First, the number of plies directly governs facesheet thickness and thus bending stiffness, compressive capacity, and wrinkling resistance. Increasing ply count enhances load-carrying capacity of the faces, but the marginal stiffness benefit diminishes once transverse shear of the core dominates panel deflections [27]. Weight-optimal studies show that efficient designs arise when multiple constraints (face compressive strain, core shear stress, wrinkling) become active simultaneously, beyond which increasing core thickness is usually more effective for stiffness-to-mass improvements [28].

Second, the ply orientation schedule determines how in-plane and bending stiffness is distributed, and controls stability of the faces under compression. Balanced and quasi-symmetric

layups with 0° , 45° , -45° and 90° plies are commonly adopted to tailor stiffness while limiting coupling effects. Optimisation studies consistently show that placing 0° plies toward the outer sublaminates is most effective for axial stiffness and buckling loads, while $\pm 45^\circ$ plies govern shear and torsion and 90° plies stabilise transverse directions [29]. Increasing the fraction of 0° plies in outer layers can also raise the wrinkling stress, although improvements level-off without concurrent increases in core stiffness [30].

Third, core thickness is the dominant driver of bending stiffness. A thicker core increases the separation between the faces, which makes the panel substantially stiffer in bending while also improving shear rigidity. At the same time, greater core depth helps to reduce shear deflections and delays shear crimping. However, core thickening alone does not guarantee higher wrinkling strength, since this mode also depends on the stiffness of both faces and core as well as the quality of the face-core bond [31]. Recent experimental studies confirm that coordinated increases in both facesheet thickness and core depth yield the most efficient stiffness-to-mass gains, while unbalanced changes shift the governing failure mode [32].

Together, these three parameters form the primary design levers in sandwich panel sizing across aerospace, motorsport, and marine applications, with industry-specific optimisation strategies ensuring that stiffness, strength, and safety requirements are met efficiently.

Different industries wrap these sizing choices in different optimisation and verification loops. In aerospace, global FE models are used to extract panel loads, and panel-level sizing focuses on stacking sequence and core depth. Analytical checks are applied for buckling, wrinkling, dimpling, and core shear, and coupon or element tests are used to validate design limits [27]. In motorsport, analytical and FE screening is used to meet torsional stiffness targets while respecting manufacturing rules such as ply drops and symmetry. Final layups and core densities are then constrained by FIA test requirements for impact performance [33]. In marine design, FE is used to limit core shear strains under slamming loads, and durability concerns such as water ingress or corrosion guide the minimum core thickness and facesheet balance. These choices are then validated by bending and local load tests [34]. In all cases, the optimum is achieved by tuning ply number, orientation, and core thickness so that stiffness targets are met and failure modes remain within design limits.

In summary, aerospace, motorsport, and marine design all apply the same underlying principles of sandwich sizing, but each industry emphasises different optimisation targets and verification strategies. Formula Student teams adopt similar methodologies, though tailored to the regulatory framework of the competition and the practical constraints of student led projects.

The structural sizing of composite sandwich panels for monocoque chassis applications typically involves a combination of analytical, numerical and experimental methods. These methods aim to ensure compliance with Formula Student rules, such as minimum flexural rigidity and energy absorption targets, while optimising the stiffness to mass ratio of each structural region.

A subset of teams adopt purely analytical approaches based on Classical Laminate Theory (CLT) combined with failure criteria such as Maximum Stress, Tsai-Wu, or Hashin. For example, FSIPLeiria [19] used MATLAB implementations of CLT and Tsai-Wu theory to analyse each panel under three-point bending loads, estimating strain and stress in individual plies and validating rule compliance through predicted failure modes and stiffness. Similarly, Metropolia Motorsport [21] built a custom MATLAB tool that automatically calculated bending stiffness and ply-level stresses for a wide range of layup and core combinations, filtering the most promising candidates for further analysis. These methods allowed rapid iteration and were particularly valuable in the early stages of design, where broad trade-offs needed to be evaluated efficiently.

2.3. Composite Monocoque Performance Assessment

Other teams rely more heavily on numerical methods. Chalmers Formula Student [25] conducted finite element simulations in ANSYS to assess both global panel deformation and local failure mechanisms under three-point bending. This approach allowed them to simulate geometric details like core shear deformation and laminate curvature more accurately than analytical models. Cal Poly Formula SAE [18] developed a full chassis model in ANSYS ACP, including shell-based facesheets and solid core representations, using the simulation results to guide their design toward compliance with SES (Structural Equivalency Spreadsheet) requirements.

The majority of teams, however, employ a hybrid approach. For example, Squadra Corse PoliTO [20] first used analytical tools to identify a set of feasible layups based on stiffness, mass, and manufacturing constraints. These preliminary candidates were then refined through FEA models, which incorporated details such as boundary conditions, ply drop-offs, and stress concentrations. MIT Motorsports [24], despite initial plans to validate panel performance with FEM, opted to focus on spreadsheet-based models due to uncertainty in material properties, prioritising correlation with historical test data.

Some teams also adopted formal optimisation strategies. KTH Formula Student [26] applied composite sizing tools from Altair Hyperworks, using OptiStruct for free-size optimisation under torsional loading. The output from the optimiser was then post-processed to derive rule compliant tested layups. This iterative loop between optimisation, rule checking, and manufacturing feasibility provided a rigorous framework for panel development.

In summary, most teams used a combination of analytical and numerical methods in a feedback-driven workflow, allowing them to balance rule compliance, structural efficiency, and manufacturing constraints. Analytical tools were preferred for initial screening and mass estimation, while FEA was employed to refine performance predictions and validate local effects. This blended methodology proved to be the most effective approach for achieving optimised sandwich panel designs under the tight constraints typical of student engineering projects. A generalised representation of this iterative sizing loop is presented in Figure 2.2. Not all teams follow every step, but the diagram aims to capture the range of techniques observed across the reviewed methodologies.

2.3 Composite Monocoque Performance Assessment

The structural validation of a composite sandwich monocoque relies on a combination of standardised tests that address both local and global load-bearing capabilities. Formula Student regulations, through the Structural Equivalency Spreadsheet (SES), explicitly require teams to demonstrate that their sandwich panels meet prescribed benchmarks for bending and shear behaviour. In parallel, the overall torsional stiffness of the assembled chassis must be established to ensure that the structure provides the rigidity required for effective suspension performance.

Three-point bending tests provide insight into the flexural stiffness of representative panels, while perimeter shear tests verify their ability to sustain concentrated shear loads introduced at structural interfaces. Together, these two methods form the regulatory foundation for panel-level validation. At the chassis scale, torsional stiffness assessment complements these tests by quantifying the global rigidity of the monocoque, directly linking panel performance to overall vehicle dynamics.

2.3.1 Three-Point Bending

Three-point bending (3PB) tests are the principal method used to characterise the flexural response of sandwich panels. In this setup, a specimen is supported at two ends and loaded

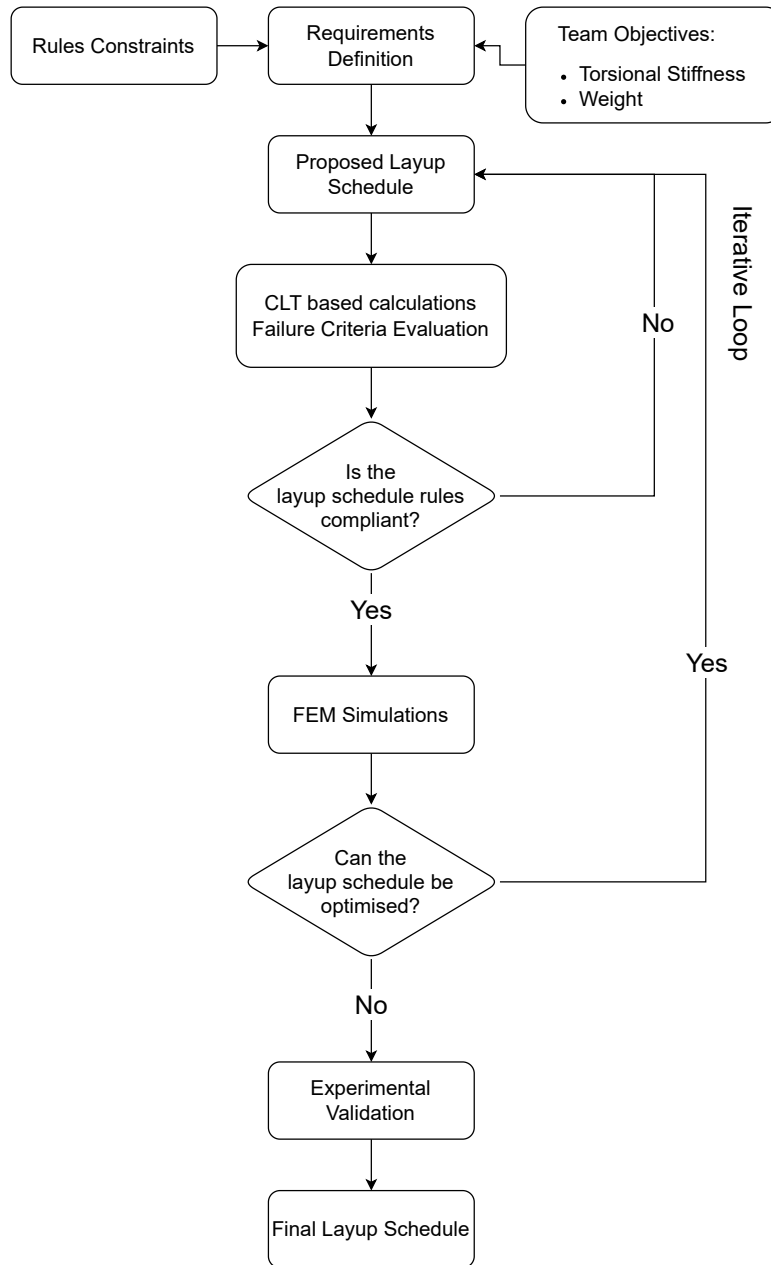


Figure 2.2: Generalised schematic of the feedback loops typically involved in the sizing of composite sandwich panels.

2.3. Composite Monocoque Performance Assessment

centrally, which subjects the structure to a well-defined bending moment. The resulting load-displacement curve provides a direct measurement of flexural rigidity and allows identification of critical failure modes such as core shear, facesheet wrinkling, indentation and delamination [17, 35].

For Formula Student applications, three-point bending tests are essential because they bridge panel level behaviour with chassis level requirements. The SES rules explicitly mandate bending tests on representative sandwich specimens to demonstrate equivalency with metallic baselines, thereby ensuring that the monocoque can safely sustain bending loads. Beyond regulatory compliance, these tests provide direct experimental validation of analytical and finite element sizing tools: the measured flexural modulus and ultimate strength can be compared against predictions from sandwich theory and finite element models. This feedback loop is vital for accurate and reliable sizing, preventing under-designed laminates that might compromise driver safety, and ensuring that design tools are both theoretically consistent and experimentally grounded.

Finally, panel flexural rigidity is directly linked to the torsional stiffness of the full chassis. Since the torsional response of a monocoque derives from the combined bending stiffness of its constituent panels arranged in a closed section, ensuring that each panel achieves its predicted flexural properties is fundamental to achieving the global torsional stiffness target [36]. In this way, three-point bending tests provide the essential foundation for both rule compliance and performance assurance in composite sandwich monocoques.

2.3.2 Perimeter Shear

Perimeter shear tests are designed to evaluate the ability of sandwich structures to transfer concentrated loads through their core and facesheets when subjected to out-of-plane forces. In this configuration, a punch applies a localised load to the panel while the edges are supported, introducing circumferential shear stresses that drive damage in the core and at the face-core interface. The resulting force-displacement response characterises how effectively the sandwich structure can resist localised shear failure, delamination, or core crushing [37].

In the context of Formula Student, this form of testing is particularly important because the monocoque must carry highly concentrated loads where external structures interface with the chassis. Suspension pick-up points, roll hoop attachments, bulkhead interfaces, and the impact attenuator all introduce out-of-plane loading into the composite panels. If these regions are not adequately reinforced, the concentrated loads can trigger premature face-core separation or core shear failure, undermining the structural integrity of the entire chassis. To prevent this, the SES rules explicitly require perimeter shear tests to demonstrate that the chosen sandwich panel layups can withstand such loading scenarios, ensuring both compliance and safety.

Ultimately, perimeter shear testing ensures that the panels forming the CFRP monocoque are not only efficient in bending but also robust against concentrated out-of-plane loads. In this way, it complements three-point bending tests by addressing the localised loading conditions that are unavoidable in Formula Student chassis design. Together, these validation methods provide confidence that the sizing methodology produces a structure that is both lightweight and safe under realistic operational conditions.

2.3.3 Torsional Stiffness

Torsional stiffness is the principal metric by which chassis performance is assessed in motorsport. It describes the resistance of the structure to twisting about its longitudinal axis, a deformation

mode that directly governs how effectively the suspension can operate. If the chassis lacks sufficient torsional rigidity, part of the suspension deflection is absorbed by structural compliance, leading to unpredictable handling, poor vehicle balance, and inconsistent driver feedback. For this reason, torsional stiffness is the single most important structural property of a race car chassis, as it forms the foundation for suspension tuning and overall dynamic behaviour [5].

Experimentally, torsional stiffness is typically measured by rigidly fixing one axle of the chassis and applying a known torque to the opposite end. The induced angular displacement is recorded, often via dial gauges or displacement sensors placed at the suspension hard points. The stiffness is then calculated as the applied torque divided by the measured angle of twist. This setup, while simplified, provides a direct measure of global rigidity and allows teams to compare different chassis iterations against performance targets and suspension roll stiffness [38].

Numerically, torsional stiffness is most often evaluated through finite element simulations. A torsional load case is applied by constraining one end of the model and twisting the other, replicating the experimental boundary conditions. Simulation enables detailed analysis of sectional stiffness distributions, identification of weak regions, and evaluation of the effect of local reinforcements. It also enables the comparison between preliminary designs without the need to manufacture and test each configuration, thereby saving considerable time and cost while guiding design optimisation more efficiently.

In practice, the combination of numerical prediction and experimental measurement provides both confidence and efficiency. Simulations enable rapid refinement of the design at early stages while experiments ensure that safety and compliance are demonstrated in a physical chassis. Together, they ensure that the monocoque not only satisfies Formula Student requirements but also delivers the stiffness foundation required for competitive vehicle dynamics.

Taken collectively, these assessment methods ensure that both the local behaviour of the sandwich panels and the global performance of the assembled chassis are fully characterised. Bending and shear tests provide the regulatory evidence needed to certify the strength and stiffness of the composite panels, while torsional stiffness evaluation connects these results to the overall dynamics of the vehicle. By combining panel-level and chassis-level validation, teams can demonstrate compliance with Formula Student requirements and gain confidence that the monocoque delivers the balance of safety, efficiency, and performance demanded in competition.

Chapter 3

Monocoque Sizing Methods

This chapter presents the analytical framework developed to support the structural sizing and validation of the composite sandwich panels used in the monocoque chassis of the FSFEUP Formula Student car. These methods were designed to ensure compliance with the Formula Student rules while enabling an efficient and optimised use of carbon fibre-reinforced polymer (CFRP) materials. Given the diverse structural requirements imposed on different regions of the chassis, such as flexural rigidity, perimeter shear strength, and energy absorption, a robust set of analytical tools is required to guide the selection of material layups, core configurations, and overall panel architecture. The methodologies described in this chapter are primarily based on Classical Laminate Theory and composite failure modelling approaches. The final implementation was carried out in MATLAB, allowing for fast and systematic evaluation of alternative design configurations and supporting the generation of documentation required for structural equivalency validation.

3.1 Formula Student Regulations

The use of carbon fibre reinforced polymer (CFRP) in the primary structure of a Formula Student vehicle is subject to strict regulatory oversight. These rules are designed to ensure that composite alternatives provide performance and safety equivalent to those of traditional steel tube designs. The following list outlines the specific requirements imposed by the Formula Student rulebook for monocoque chassis and tractive system accumulator container (TSAC) structures, with emphasis on flexural rigidity, shear strength, failure testing, and energy absorption performance.

T 3.3.1 Alternative materials may be used for all parts of the primary structure and the TSAC with the following exceptions:

- The main hoop and the main hoop bracing must be steel
- The front hoop must be metal
- Any welded structures of the primary structure must be steel
- However, the front hoop may be an aluminium welded structure

T 3.3.2 If any other materials than steel tubing are used in the primary structure or the TSAC, physical testing is required to show equivalency to the minimum material properties for steel in T 3.2.

3.1. Formula Student Regulations

T 3.4.1 If composite structures are used in the primary structure and/or the TSAC:

- The Flexural Rigidity (EI) of that structure must be calculated with the tools and formulas in the SES and/or Accumulator Structural Equivalency Spreadsheet (ASES).
- The EI must be calculated as the EI of a flat panel about its neutral axis.
- The structure used for flat panel calculations must have the same composition as the structure used in the primary structure or the TSAC.
- The actual geometry and curvature of the panel may only be considered if the flat panel EI equivalency exceeds 60% of the minimum requirements (T 3.2).
- Actual geometry calculations are not permitted for the side impact structure, front bulkhead and TSAC.

T 3.4.2 If composite materials are used in the primary structure or the TSAC, the SES and/or ASES must include:

- Material type(s)
- Cloth weights
- Resin type
- Fiber orientation
- Number of layers
- Core material
- Lay-up technique
- 3-point-bend test and shear test data (the AIP is exempt from these tests, but must comply with T 3.16.4)

T 3.4.3 For any laminate in the primary structure and/or the TSAC, the maximum weight content of parallel fibers, relative to the weight of all fibers in the laminate, is 50%. All fibers laid within any orientation $\pm 10^\circ$ count as parallel in this case.

T 3.4.4 If an asymmetrical lay-up is used in the primary structure, the thinner skin must have a thickness of at least 40% of the thicker skin or 1 mm whichever is lower.

T 3.5.1 If composite materials are used for any part of the primary structure (except for the AIP) or the TSAC, the team must:

- Build a representative test panel which must measure exactly 275 mm $\times$ 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 T 3.5.5. 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 3.5.6 and T 3.5.7 do not apply.

- T 3.5.2 If a panel represents side impact structure it must be proven that it has at least the same properties as two steel tubes meeting the requirements for side impact structure tubes for buckling modulus, yield strength and absorbed energy.
- T 3.5.3 Composite structures with different core thicknesses but otherwise identical construction may use material properties derived from a single test panel. The panel with the thicker core must be tested and the structure using derived material properties must not use a core thickness of less than 66% of the tested panel.
- T 3.5.4 When a laminate is not quasi-isotropic, i.e. has equal strength and stiffness in the 0°, 90° and ±45° direction, the results from the 3 point bending test will be assigned to the 0° lay-up direction. To show equivalence in the SES, the 0° lay-up direction must be used and oriented accordingly in the chassis.
- T 3.5.6 The distance between the two test panel supports must be at least 400 mm.
- T 3.5.7 The load applicator used to test any panel or tube must be metallic and have a radius of 50 mm.
- T 3.5.8 The load applicator must overhang the test piece to prevent edge loading.
- T 3.5.9 There must be no material between the load applicator and the test piece.
- T 3.5.10 Perimeter shear tests must be completed which measure the force required to push or pull a 25 mm diameter flat punch through a flat laminate sample. The sample must be at least 100 mm × 100 mm. Core and skin thicknesses must be identical to those used in the actual primary structure and be manufactured using the same materials and processes. If an asymmetrical lay-up is used, the thinner skin must face the punch.
- T 3.5.11 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 3.11.4 Composite front hoop bracing structures and their attachments cannot be counted towards the front bulkhead support structures and vice-versa for the structural equivalency documentation.
- T 3.12.1 Any alternative material used for the front bulkhead must have a perimeter shear strength equivalent to a 1.5 mm thick steel plate.
- T 3.12.2 If the front bulkhead is part of a composite structure and is modeled as an “L” shape, the EI of the front bulkhead about the vertical and lateral axes must be equivalent to a steel tube meeting the requirements for the front bulkhead. The length of the section perpendicular to the bulkhead may be a maximum of 25 mm measured from the rearmost face of the bulkhead.
- T 3.13.2 If the front bulkhead support is part of a composite structure, it must have equivalent EI to the sum of the EI of the six baseline steel tubes that it replaces and it must not be counted towards the composite front hoop bracing structures for the structural equivalency documentation, i.e. T 3.11.4.

3.1. Formula Student Regulations

T 3.13.3 The EI of the vertical side of the front bulkhead support structure must be equivalent to at least the EI of one baseline steel tube that it replaces.

T 3.13.4 The perimeter shear strength of the monocoque laminate in the front bulkhead support structure must be at least 4 kN.

T 3.14.3 If the impact structure is part of a composite structure, the following is required:

- The region of the structure up to a height of 320 mm above the lowest inside chassis point between the front and main hoop must have an EI equal to the three baseline steel tubes that it replaces. An example for side impact structure is given in figure 8.
- The vertical impact structure must have an EI equivalent to two baseline steel tubes and half the horizontal floor must have an EI equivalent to one baseline steel tube.
- The vertical impact structure must have an absorbed energy equivalent to two baseline steel tubes, exceeding 65 J.
- The perimeter shear strength must be at least 7.5 kN.

Although the full set of rules provides comprehensive design guidance, many of them converge toward a common objective: demonstrating that each composite panel can match or exceed the mechanical performance of a steel-based reference configuration. For each structural region of the car, the regulations define the number and geometry of baseline steel tubes being replaced, as well as the specific performance criteria that must be met through analytical evaluation and physical testing. These requirements are consolidated in Table 3.1, which summarises the structural equivalency targets for each monocoque panel according to the applicable FS rules.

Table 3.1: Summary of FS Regulatory Constraints for Composite Chassis Sections.

Car Section	Baseline Tubes (n × Ø × t) [mm]	Min. Perimeter Shear [kN]	Min. Absorbed Energy [J]	Required Tests
Front Hoop Bracing	1 × 25.4 × 1.6	-	-	3PB
Front Bulkhead Support	6 × 25.4 × 1.2	4	-	3PB Perimeter Shear
Front Bulkhead	2 × 25.4 × 1.6	equivalent to 1.5 mm thick steel plate	-	3PB Perimeter Shear
Side Impact (Vertical)	2 × 25.4 × 1.6	7.5	65	3PB Perimeter Shear
Side Impact (Horizontal)	1 × 25.4 × 1.6	-	-	3PB
TS Side Impact (Vertical)	2 × 25.4 × 1.6	7.5	65	3PB Perimeter Shear
TS Side Impact (Horizontal)	1 × 25.4 × 1.6	-	-	3PB
TS Rear Impact Structure	2 × 25.4 × 1.6	7.5	65	3PB Perimeter Shear
Main Hoop Bracing Support	2 × 25.4 × 1.2	-	-	3PB

The regulatory constraints outlined in Table 3.1 impose clear design implications for the use of composite materials in the primary structure. Minimum flexural rigidity requirements limit

the extent to which laminate thickness can be reduced for weight saving, while perimeter shear strength is primarily governed by the thickness and quality of the skins, which must be sufficient to withstand out-of-plane loading. Energy absorption thresholds add a further layer of complexity by forcing a balance between stiffness and crashworthiness: while stiffer fibres increase flexural rigidity, they tend to reduce the laminate’s ability to absorb energy in an impact. The symmetry and minimum skin thickness rules also constrain optimisation freedom, preventing unbalanced or highly asymmetric laminates that would otherwise compromise structural integrity. Given these strict requirements, the analytical tool developed in this work was conceived to provide rapid estimation of the mechanical properties demanded by the SES, enabling efficient iterative design at an early stage and reducing the number of costly physical tests required for validation.

Together, these constraints form the basis for the analytical framework and sizing methodology presented in this work. The tool was specifically designed to quantify the performance parameters mandated by the regulations and to assess them against the steel-based reference targets. In doing so, it supports the preliminary evaluation and optimisation of composite sandwich panels prior to physical testing, ensuring that the selected designs are both rules-compliant and structurally efficient. More broadly, it provides FSFEUP with a structured means of assessing whether the transition from a steel spaceframe to a composite monocoque is justified in terms of regulatory compliance, structural performance, and overall feasibility. The following sections present the methodology and theoretical framework used to design and validate the composite panels in accordance with these regulations.

3.2 Methodology

Given the structural requirements imposed by the Formula Student rules, a systematic methodology was developed to guide the sizing and validation of each composite sandwich panel in the monocoque chassis. This methodology ensures that all panels meet the required performance targets in terms of flexural rigidity, perimeter shear strength, resistance to bending and impact loads, and is illustrated in Figure 3.1.

The workflow begins with the definition of input parameters for each panel, including the fibre and matrix mechanical properties, ply thickness, layup orientations, core thickness, and panel dimensions. Based on these inputs, the laminate stiffness matrices are computed using Classical Laminate Theory (CLT), allowing the calculation of the global flexural rigidity (EI) about the appropriate axis.

The next step involves evaluating the panel’s perimeter shear strength, derived from the in-plane shear properties of the skins and the perimeter geometry. The failure load under three-point bending is then estimated by applying appropriate failure criteria, namely Tsai-Hill, Tsai-Wu, and Hashin, to the stress state in the outermost plies. For panels in regions subject to crash or impact regulations, an energy absorption estimate is also determined from the panel’s mechanical and geometric characteristics.

Finally, the predicted values for each panel are compared to the regulatory minimums, and safety factors are calculated. Compliance is verified when all relevant safety factors exceed unity. This methodology provides a structured framework to evaluate and iterate on laminate designs to ensure full conformity with the structural equivalency rules.

3.3. Panel Property Definition and Input Parameters

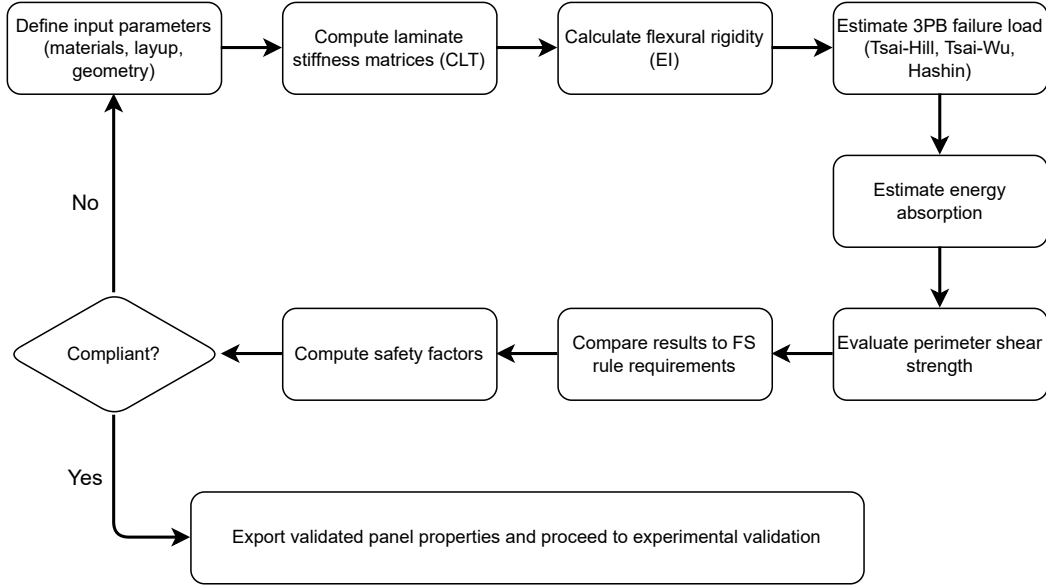


Figure 3.1: Workflow developed for CFRP sandwich panel sizing and compliance evaluation.

3.3 Panel Property Definition and Input Parameters

The panel sizing workflow begins with the specification of the geometric, material, and layup parameters that characterise each composite sandwich panel. These inputs provide the foundation for subsequent calculations of stiffness, strength, and compliance, and are tailored to the specific structural region of the chassis being analysed.

In accordance with Formula Student regulations, all panels must be analysed as flat rectangular sections; therefore, dimensions must be extracted carefully from the CAD design to ensure they represent a valid flattened approximation of the actual geometry. The structural layup is defined by the number and orientation of plies within each CFRP skin, while the core, typically a lightweight foam or honeycomb, is characterised by its thickness and shear modulus.

In parallel, the mechanical properties of the fibre and matrix system must be specified. These include the ply thickness (associated with the selected prepreg), Young’s moduli in the longitudinal and transverse directions (E_1 , E_2), in-plane shear modulus (G_{12}), Poisson’s ratio (ν_{12}), and strength values such as tensile, compressive, and shear strengths in the relevant directions.

Table 3.2 summarises the typical input parameters required for panel definition and analysis.

Table 3.2: Typical input parameters for sandwich panel analysis.

Parameter Category	Input Description	Symbol / Unit
Geometry	Panel length	L [mm]
	Panel width	W [mm]
Layup	Number of plies	n [-]
	Ply orientations	θ_i [°]
	Core thickness	t_{core} [mm]
Material Properties	Ply thickness	t_{ply} [mm]
	Longitudinal direction modulus	E_1 [MPa]
	Transverse modulus	E_2 [MPa]
	In-plane shear modulus	G_{12} [MPa]
	Poisson's ratio	ν_{12} [-]
	In-plane longitudinal tensile strength	X_t [MPa]
	In-plane longitudinal compressive strength	X_c [MPa]
	In-plane transverse tensile strength	Y_t [MPa]
	In-plane transverse compressive strength	Y_c [MPa]
	In-plane shear strength	S [MPa]

These values are either measured experimentally from characterisation tests (Chapter 4) or obtained from manufacturer datasheets. Once all parameters are defined, they are used to construct the laminate stiffness matrices and to evaluate the structural performance of each panel through the subsequent steps in the analysis workflow. The methodology proceeds by computing these stiffness matrices using Classical Laminate Theory, as detailed in the following section.

A dedicated MATLAB interface was developed to streamline the input of these variables and propagate them through all computational modules. Full code listings are available in Appendix A.1.

3.4 Laminate Stiffness Matrices

To evaluate the bending stiffness and in-plane behaviour of each composite skin, Classical Laminate Theory (CLT) is employed to compute the laminate stiffness matrices. CLT offers a reliable and computationally efficient method for predicting the elastic response of layered composite laminates, taking into account the anisotropic nature and orientation of each ply. It is particularly well suited to flat panel geometries, as assumed in the SES regulatory framework, and provides the foundation for estimating panel flexural rigidity and stress distributions.

The key output of this stage is the construction of the laminate's $[A]$, $[B]$, and $[D]$ matrices, which respectively represent the in-plane, coupling, and bending stiffness contributions of the laminate. For symmetric stacking sequences, the $[B]$ matrix is zero, simplifying the analysis and eliminating extension-bending coupling.

3.4.1 Theoretical Background

Classical Laminate Theory (CLT) provides a mathematical framework for predicting the mechanical behaviour of multi-layered composite laminates under in-plane and out-of-plane loading. It extends classical plate theory by incorporating the anisotropic and layered nature of composite materials, making it an essential tool in the analysis and design of composite structures.

CLT relies on the following key assumptions [39]:

3.4. Laminate Stiffness Matrices

- Each lamina behaves as a homogeneous, linearly elastic orthotropic material.
- The laminate is thin compared to its in-plane dimensions.
- Transverse normal stress σ_z is negligible.
- Perfect bonding exists between layers, meaning there is no slip or separation.
- The displacement field follows Kirchhoff-Love plate theory: straight lines normal to the mid-surface before deformation remain straight and normal after deformation, and experience no extension.

For a single lamina in its principal material axes (1-2), the stress-strain relationship under plane stress is given by:

$$\begin{bmatrix} \sigma_1 \\ \sigma_2 \\ \tau_{12} \end{bmatrix} = \begin{bmatrix} Q_{11} & Q_{12} & Q_{16} \\ Q_{12} & Q_{22} & Q_{26} \\ Q_{16} & Q_{26} & Q_{66} \end{bmatrix} \begin{bmatrix} \varepsilon_1 \\ \varepsilon_2 \\ \gamma_{12} \end{bmatrix} \quad (3.1)$$

The reduced stiffness coefficients Q_{ij} for an orthotropic lamina are defined as:

$$Q_{11} = \frac{E_1}{1 - \nu_{12}\nu_{21}} \quad (3.2)$$

$$Q_{22} = \frac{E_2}{1 - \nu_{12}\nu_{21}} \quad (3.3)$$

$$Q_{12} = \frac{\nu_{12}E_2}{1 - \nu_{12}\nu_{21}} \quad (3.4)$$

$$Q_{66} = G_{12} \quad (3.5)$$

where E_1 , E_2 are the Young's moduli in the longitudinal and transverse directions, ν_{12} and ν_{21} are the major and minor Poisson's ratios, and G_{12} is the in-plane shear modulus.

To account for ply orientation within the laminate, the local reduced stiffness matrix $[Q]$ must be transformed to the global laminate coordinate system. The resulting matrix, $\bar{Q}$, incorporates the ply angle θ and is given by:

$$\bar{Q} = T_\sigma^{-1} \cdot Q' \cdot T_\varepsilon \quad (3.6)$$

where T_σ and T_ε are the transformation matrices for stress and strain, respectively, and Q' is the stiffness matrix in the rotated coordinate system.

In practice, the transformed reduced stiffness components $\bar{Q}_{ij}$ are explicitly given as:

$$\bar{Q}_{11} = Q_{11} \cos^4 \theta + 2(Q_{12} + 2Q_{66}) \cos^2 \theta \sin^2 \theta + Q_{22} \sin^4 \theta \quad (3.7)$$

$$\bar{Q}_{22} = Q_{11} \sin^4 \theta + 2(Q_{12} + 2Q_{66}) \cos^2 \theta \sin^2 \theta + Q_{22} \cos^4 \theta \quad (3.8)$$

$$\bar{Q}_{12} = (Q_{11} + Q_{22} - 4Q_{66}) \cos^2 \theta \sin^2 \theta + Q_{12}(\cos^4 \theta + \sin^4 \theta) \quad (3.9)$$

$$\bar{Q}_{66} = (Q_{11} + Q_{22} - 2Q_{12} - 2Q_{66}) \cos^2 \theta \sin^2 \theta + Q_{66}(\cos^4 \theta + \sin^4 \theta) \quad (3.10)$$

$$\bar{Q}_{16} = (Q_{11} - Q_{12} - 2Q_{66}) \cos^3 \theta \sin \theta - (Q_{22} - Q_{12} - 2Q_{66}) \cos \theta \sin^3 \theta \quad (3.11)$$

$$\bar{Q}_{26} = (Q_{11} - Q_{12} - 2Q_{66}) \cos \theta \sin^3 \theta - (Q_{22} - Q_{12} - 2Q_{66}) \cos^3 \theta \sin \theta \quad (3.12)$$

These expressions are applied for each ply according to its orientation angle $\theta^{(k)}$.

For a laminate composed of n plies, the relationship between mid-plane strains ε_0 , curvatures κ , and the resultant forces and moments is given by:

$$\begin{bmatrix} N \\ M \end{bmatrix} = \begin{bmatrix} A & B \\ B & D \end{bmatrix} \begin{bmatrix} \varepsilon_0 \\ \kappa \end{bmatrix} \quad (3.13)$$

where:

- $N = [N_x \ N_y \ N_{xy}]^T$ are the in-plane force resultants per unit width;
- $M = [M_x \ M_y \ M_{xy}]^T$ are the bending moment resultants per unit width;
- $\varepsilon_0 = [\varepsilon_x^0 \ \varepsilon_y^0 \ \gamma_{xy}^0]^T$ are the mid-plane strains;
- $\kappa = [\kappa_x \ \kappa_y \ \kappa_{xy}]^T$ are the laminate curvatures.

Each ply in the laminate is modelled as an orthotropic layer with material properties defined in its local coordinate system. Using coordinate transformation laws, the stiffness matrix of each ply is rotated to the global laminate axes. These transformed stiffnesses are then integrated through the laminate thickness to compute the global laminate stiffness matrices:

- The $[A]$ matrix relates in-plane forces to in-plane strains;
- The $[B]$ matrix captures coupling between in-plane forces and curvatures;
- The $[D]$ matrix relates moments to curvatures.

These matrices are given by:

$$A_{ij} = \sum_{k=1}^N \bar{Q}_{ij}^{(k)} (z_k - z_{k-1}) \quad (3.14)$$

$$B_{ij} = \frac{1}{2} \sum_{k=1}^N \bar{Q}_{ij}^{(k)} (z_k^2 - z_{k-1}^2) \quad (3.15)$$

$$D_{ij} = \frac{1}{3} \sum_{k=1}^N \bar{Q}_{ij}^{(k)} (z_k^3 - z_{k-1}^3) \quad (3.16)$$

where $\bar{Q}_{ij}^{(k)}$ are the transformed stiffness coefficients for the k -th ply, and z_k, z_{k-1} are the distances from the mid-plane to the top and bottom of the k -th ply.

3.4. Laminate Stiffness Matrices

Once the laminate stiffness matrices are known, they can be used in either direction: to compute forces and moments from given mid-plane strains and curvatures (direct form), or, conversely, to determine the mid-plane response from known loads (inverse form). The inverse relationship is particularly useful in analytical simulations where external loads are applied, and the resulting strain and curvature states are needed for stress recovery. The inverse form of CLT is expressed as:

$$\begin{bmatrix} \varepsilon_0 \\ \kappa \end{bmatrix} = \begin{bmatrix} A & B \\ B & D \end{bmatrix}^{-1} \begin{bmatrix} N \\ M \end{bmatrix} = \begin{bmatrix} \alpha & \beta \\ \beta^T & \delta \end{bmatrix} \begin{bmatrix} N \\ M \end{bmatrix} \quad (3.17)$$

where the sub-matrices are defined as:

$$[\alpha] = [A]^{-1} + [A]^{-1}[B]([D] - [B][A]^{-1}[B])^{-1}[B][A]^{-1} \quad (3.18)$$

$$[\beta] = -[A]^{-1}[B]([D] - [B][A]^{-1}[B])^{-1} \quad (3.19)$$

$$[\delta] = ([D] - [B][A]^{-1}[B])^{-1} \quad (3.20)$$

These matrices allow the complete recovery of laminate mid-plane deformation from known applied loads and moments, forming the backbone of CLT-based analysis tools. In the case of a symmetric laminate, the coupling matrix $[B]$ is zero due to the symmetric stacking of plies about the mid-plane. As a result, there is no interaction between in-plane deformation and bending behaviour, meaning that the membrane and flexural responses are entirely decoupled. This allows the inverse laminate relations to simplify considerably:

$$[\alpha] = [A]^{-1} = [a] \quad (3.21)$$

$$[\beta] = 0 \quad (3.22)$$

$$[\delta] = [D]^{-1} = [d] \quad (3.23)$$

These simplifications streamline the strain–curvature calculations and are valid for all symmetric laminates, which are commonly used in engineering practice to avoid coupling effects.

CLT is widely used for preliminary design and optimisation of laminated composite structures, especially in aerospace and automotive industries. However, it does not account for:

- Transverse shear deformation (neglected due to Kirchhoff-Love assumptions).
- Nonlinear behaviour (e.g., large deformations, plasticity, or progressive damage).
- Interlaminar stresses, which require 3D or layerwise theories.

Therefore, while CLT provides a powerful analytical tool, its results should be interpreted with awareness of these assumptions and validated using more advanced models or experimental data when necessary. Despite these simplifications, CLT remains the method of choice in the early design and equivalency assessment stages due to its speed and effectiveness for flat laminate configurations.

3.4.2 Implementation

The sandwich panel is modelled as a single laminate composed of the two CFRP facesheets separated by an intermediate "empty" core layer. In CLT, the core provides spacing between the facesheets through its thickness but, for in-plane behaviour, its stiffness is neglected, i.e. $[Q]_{\text{core}} \approx 0$. This ensures the panel bending stiffness is obtained directly from the laminate $[A]$, $[B]$, and $[D]$ matrices over the full thickness, avoiding per-skin calculations and parallel-axis corrections. For each panel:

1. The stacking sequence, ply orientations and core thickness are extracted.
2. The local $[Q]$ matrix is computed for each ply based on its mechanical properties.
3. Each $[Q]$ matrix is transformed to the global coordinate system according to the ply orientation.
4. The $[A]$, $[B]$, and $[D]$ matrices are assembled through numerical integration across the laminate thickness.

All theoretical formulations presented here serve as the foundation for the numerical implementation tools developed in MATLAB and used throughout the panel validation framework. Appendices A.2 and A.3 provide an overview of the code structure.

3.5 Flexural Rigidity Evaluation

The flexural rigidity of the composite sandwich panel is obtained directly from Classical Laminate Theory (CLT) by relating the panel's curvature to an applied bending moment. This approach captures the full anisotropic and layered nature of the panel, including any membrane-bending coupling, and is valid for both symmetric and asymmetric layups. This EI value is then compared with that of the baseline steel structure to verify compliance with the SES equivalency requirements.

3.5.1 Theoretical Background

As seen before, CLT expresses the relationship between the applied force resultants and moment resultants and in-plane deformation and mid-plane curvatures through the expression:

$$\begin{bmatrix} \varepsilon_0 \\ \kappa \end{bmatrix} = \begin{bmatrix} \alpha & \beta \\ \beta^T & \delta \end{bmatrix} \begin{bmatrix} N \\ M \end{bmatrix} \quad (3.24)$$

Which in expanded form is:

$$\begin{bmatrix} \varepsilon_x^0 \\ \varepsilon_y^0 \\ \gamma_{xy}^0 \\ \kappa_x \\ \kappa_y \\ \kappa_{xy} \end{bmatrix} = \begin{bmatrix} \alpha_{11} & \alpha_{12} & \alpha_{16} & \beta_{11} & \beta_{12} & \beta_{16} \\ \alpha_{12} & \alpha_{22} & \alpha_{26} & \beta_{21} & \beta_{22} & \beta_{26} \\ \alpha_{16} & \alpha_{26} & \alpha_{66} & \beta_{61} & \beta_{62} & \beta_{66} \\ \beta_{11} & \beta_{21} & \beta_{61} & \delta_{11} & \delta_{12} & \delta_{16} \\ \beta_{12} & \beta_{22} & \beta_{62} & \delta_{12} & \delta_{22} & \delta_{26} \\ \beta_{16} & \beta_{26} & \beta_{66} & \delta_{16} & \delta_{26} & \delta_{66} \end{bmatrix} \begin{bmatrix} N_x \\ N_y \\ N_{xy} \\ M_x \\ M_y \\ M_{xy} \end{bmatrix} \quad (3.25)$$

Assuming pure bending about the x -axis ($N_x = N_y = N_{xy} = M_y = M_{xy} = 0$), the curvature is given by:

3.5. Flexural Rigidity Evaluation

$$\kappa_x = \delta_{11} M_x \quad (3.26)$$

Classical beam theory gives curvature in terms of the total bending moment M :

$$\kappa_x = \frac{M}{EI} \quad (3.27)$$

Since M_x is the bending moment per unit width $M_x = \frac{M}{b}$. Equating the two expressions for κ_x yields:

$$\frac{M}{EI} = \delta_{11} \frac{M}{b} \Rightarrow EI = \frac{b}{\delta_{11}} \quad (3.28)$$

3.5.2 Implementation

For each panel:

1. The $[A]$, $[B]$, and $[D]$ matrices are assembled and the panel width b is recorded.
2. The bending compliance matrix $[\delta]$ is computed from $[\delta] = ([D] - [B][A]^{-1}[B])^{-1}$.
3. The flexural rigidity of the panel is obtained from $EI_{\text{panel}} = b/\delta_{11}$.

This formulation assumes that the core offers negligible bending resistance and functions solely to stabilise the skins, transfer shear loads, and maintain the structural separation between facesheets. This separation elevates the overall stiffness, enabling lightweight panels to meet the flexural performance targets.

To assess compliance with the SES equivalency requirements, the computed EI_{panel} of the composite sandwich panel is compared to that of the baseline steel structure. For each relevant section, the flexural rigidity of the steel tubes is calculated using:

$$EI_{\text{steel}} = n \cdot E_{\text{steel}} \cdot I_{\text{steel}} = n \cdot E_{\text{steel}} \cdot \frac{\pi}{64} (D^4 - d^4),$$

where:

- E_{steel} is the Young's modulus of the steel material;
- D is the outer diameter of the tube;
- $d = D - 2t$ is the inner diameter, with t being the wall thickness;
- n is the number of tubes contributing to the section stiffness.

Finally, the structural equivalence is verified by computing the flexural rigidity safety factor. A value of $SF_{EI} > 1.0$ confirms that the composite panel meets or exceeds the flexural stiffness of the steel baseline. This comparison provides a clear and quantitative validation of the design under the ruleset requirements.

The calculations described here have been implemented in MATLAB for efficient evaluation across multiple panel configurations. The relevant code is available in Appendix A.4.

3.6 Failure Load Prediction under Three-point Bending

In accordance with Formula Student rules, composite sandwich panels must be physically tested under three-point bending to validate their structural performance. In parallel, analytical predictions of the failure load provide valuable insight during the design stage and support the SES documentation by demonstrating theoretical compliance. This section describes the methodology used to estimate the failure load of a panel subjected to three-point bending, combining Classical Laminate Theory (CLT) with established composite failure criteria.

3.6.1 Stress Estimation from Bending

The prediction of laminate failure under three-point bending (3PB) relies on computing the stresses induced in the outermost plies of the sandwich panel. These stresses result from the curvature developed under the applied load, which is distributed over a span defined by Formula Student testing rules.

In a standard 3PB configuration, the panel is simply supported and loaded at its mid-span by a central punch. The maximum bending moment M_x occurs directly beneath the punch and is given by:

$$M_x = \frac{FL}{4b} \quad (3.29)$$

where:

- F is the applied force at the mid-span [N],
- L is the support span of the test fixture [mm], typically $L = 400$ mm for Formula Student SES testing,
- b is the panel's width [mm], typically $b = 275$ mm for Formula Student SES testing.

In this formulation, M_x denotes the bending moment per unit width, obtained from the total mid-span moment $M = \frac{FL}{4}$ divided by the panel width b . Assuming pure bending about the x -axis ($N_x = N_y = N_{xy} = M_y = M_{xy} = 0$), the strain vector is given by:

$$\begin{bmatrix} \varepsilon_x^0 \\ \varepsilon_y^0 \\ \gamma_{xy}^0 \\ \kappa_x \\ \kappa_y \\ \kappa_{xy} \end{bmatrix} = \begin{bmatrix} \alpha_{11} & \alpha_{12} & \alpha_{16} & \beta_{11} & \beta_{12} & \beta_{16} \\ \alpha_{12} & \alpha_{22} & \alpha_{26} & \beta_{21} & \beta_{22} & \beta_{26} \\ \alpha_{16} & \alpha_{26} & \alpha_{66} & \beta_{61} & \beta_{62} & \beta_{66} \\ \beta_{11} & \beta_{21} & \beta_{61} & \delta_{11} & \delta_{12} & \delta_{16} \\ \beta_{12} & \beta_{22} & \beta_{62} & \delta_{12} & \delta_{22} & \delta_{26} \\ \beta_{16} & \beta_{26} & \beta_{66} & \delta_{16} & \delta_{26} & \delta_{66} \end{bmatrix} \begin{bmatrix} N_x \\ N_y \\ N_{xy} \\ M_x \\ M_y \\ M_{xy} \end{bmatrix} = \begin{bmatrix} \beta_{11} M_x \\ \beta_{21} M_x \\ \beta_{61} M_x \\ \delta_{11} M_x \\ \delta_{12} M_x \\ \delta_{16} M_x \end{bmatrix} \quad (3.30)$$

The resulting strain is then used to compute the normal strain at any location through the panel thickness. For a given ply located at a distance z from the neutral axis of the sandwich panel, the global strain vector is:

$$\begin{bmatrix} \varepsilon_x(z) \\ \varepsilon_y(z) \\ \gamma_{xy}(z) \end{bmatrix} = \begin{bmatrix} \varepsilon_x^0 \\ \varepsilon_y^0 \\ \gamma_{xy}^0 \end{bmatrix} + z \cdot \begin{bmatrix} \kappa_x \\ \kappa_y \\ \kappa_{xy} \end{bmatrix} = \begin{bmatrix} \beta_{11} M_x \\ \beta_{21} M_x \\ \beta_{61} M_x \end{bmatrix} + z \cdot \begin{bmatrix} \delta_{11} M_x \\ \delta_{12} M_x \\ \delta_{16} M_x \end{bmatrix} \quad (3.31)$$

The coordinate z denotes the distance from the panel mid-plane to the mid-surface of a given ply. The mid-plane is taken as $z = 0$, with $z > 0$ directed towards the bottom facesheet and

3.6. Failure Load Prediction under Three-point Bending

$z < 0$ towards the top facesheet. Positive z values therefore correspond to tensile loading (bottom facesheet), and negative z values to compressive loading (top facesheet).

This global strain must be transformed into the local material coordinate system (1-2) of the ply, based on its orientation angle θ_i . The transformation is performed using the standard strain transformation matrix $[T_\varepsilon]$:

$$\begin{bmatrix} \varepsilon_1 \\ \varepsilon_2 \\ \gamma_{12} \end{bmatrix} = \begin{bmatrix} \cos^2 \theta & \sin^2 \theta & \sin \theta \cos \theta \\ \sin^2 \theta & \cos^2 \theta & -\sin \theta \cos \theta \\ -2 \sin \theta \cos \theta & 2 \sin \theta \cos \theta & \cos^2 \theta - \sin^2 \theta \end{bmatrix} \begin{bmatrix} \varepsilon_x \\ \varepsilon_y \\ \gamma_{xy} \end{bmatrix} \quad (3.32)$$

$$(3.33)$$

Once the local strain vector is obtained, the local stress vector can be computed using the lamina's reduced stiffness matrix $[Q]$:

$$\begin{bmatrix} \sigma_1 \\ \sigma_2 \\ \tau_{12} \end{bmatrix} = \begin{bmatrix} Q_{11} & Q_{12} & Q_{16} \\ Q_{12} & Q_{22} & Q_{26} \\ Q_{16} & Q_{26} & Q_{66} \end{bmatrix} \begin{bmatrix} \varepsilon_1 \\ \varepsilon_2 \\ \gamma_{12} \end{bmatrix} \quad (3.34)$$

This procedure is repeated for each ply in both facesheets by evaluating the strain at its centroidal z -position and transforming it according to its orientation angle θ_i . The resulting stress state can then be used as input for the failure criteria presented in the following section. This enables prediction of the panel's failure load under the 3PB test.

3.6.2 Failure Criteria

To predict failure initiation, three commonly used composite failure theories were implemented: Maximum Stress, Tsai-Hill, Tsai-Wu, and Hashin. Each of these evaluates a scalar failure index based on the local stress state in the ply, and failure is assumed when this index exceeds unity.

3.6.2.1 Maximum Stress Criterion

The Maximum Stress Criterion is one of the most straightforward failure theories applied to composite laminates. It evaluates the stress components in each ply, obtained via CLT, and directly compares them to the corresponding uniaxial strengths of the material in its principal directions. Failure is deemed to occur when any individual stress component exceeds its allowable limit in tension, compression, or shear, independently of the other components. The criterion can be expressed as [39]:

$$\sigma_1 < \begin{cases} X_t, & \text{if } \sigma_1 > 0 \quad (\text{tension}) \\ X_c, & \text{if } \sigma_1 < 0 \quad (\text{compression}) \end{cases} \quad (3.35)$$

$$\sigma_2 < \begin{cases} Y_t, & \text{if } \sigma_2 > 0 \quad (\text{tension}) \\ Y_c, & \text{if } \sigma_2 < 0 \quad (\text{compression}) \end{cases} \quad (3.36)$$

$$|\tau_{12}| < S \quad (3.37)$$

Here, X_t and X_c represent the longitudinal tensile and compressive strengths, respectively; Y_t and Y_c are the transverse tensile and compressive strengths; and S is the in-plane shear strength of the ply.

The Maximum Stress Criterion offers simplicity and ease of implementation, making it suitable for preliminary design assessments. However, it neglects interaction effects between stress components, which can lead to non-conservative or overly conservative predictions depending on the loading state. Consequently, while it can provide a rapid first estimate of ply failure, it is often complemented or replaced by more advanced criteria, such as Tsai–Hill, Tsai–Wu, or Hashin, in detailed laminate analyses.

3.6.2.2 Tsai-Hill Criterion

This criterion is an extension of the von Mises theory for anisotropic materials and is suitable for fibre-dominated failure modes. For a plane stress state, the failure index F_{TH} is given by [39]:

$$F_{TH} = \left(\frac{\sigma_1}{X}\right)^2 - \frac{\sigma_1\sigma_2}{X^2} + \left(\frac{\sigma_2}{Y}\right)^2 + \left(\frac{\tau_{12}}{S}\right)^2 \quad (3.38)$$

where X , Y , and S represent the material strengths in the longitudinal, transverse, and in-plane shear directions, respectively. Importantly, X and Y are selected based on the nature of the loading:

- If $\sigma_1 > 0$, $X = X_t$ (longitudinal tensile strength); else, $X = X_c$ (longitudinal compressive strength).
- If $\sigma_2 > 0$, $Y = Y_t$ (transverse tensile strength); else, $Y = Y_c$ (transverse compressive strength).

This approach allows the criterion to account for the different tensile and compressive capacities of composite plies, improving prediction accuracy under complex loading states.

3.6.2.3 Tsai-Wu Criterion

The Tsai–Wu criterion is a more general failure theory that incorporates stress interaction terms and distinguishes between tensile and compressive strengths in both principal material directions. The failure index F_{TW} is given by [39]:

$$F_{TW} = \frac{\sigma_1^2}{X_t X_c} + \frac{\sigma_2^2}{Y_t Y_c} - \sqrt{\frac{1}{X_t X_c} \cdot \frac{1}{Y_t Y_c}} \sigma_1 \sigma_2 + \left(\frac{1}{X_t} - \frac{1}{X_c}\right) \sigma_1 + \left(\frac{1}{Y_t} - \frac{1}{Y_c}\right) \sigma_2 + \frac{\tau_{12}^2}{S^2} \quad (3.39)$$

While the Tsai–Wu criterion typically yields accurate predictions under various loading conditions, it is known to produce less realistic results under biaxial compression. However, such loading scenarios are not considered within the scope of this tool, and as such, the criterion remains suitable for the intended application.

3.6.2.4 Hashin Criterion

The Hashin failure model distinguishes between different failure mechanisms, allowing the prediction of fibre-dominated and matrix-dominated failures separately. The full three-dimensional formulation of the criteria is given as follows:

3.6. Failure Load Prediction under Three-point Bending

Tensile fibre failure for $\sigma_1 \geq 0$:

$$\left(\frac{\sigma_1}{X_t}\right)^2 + \frac{\tau_{12}^2 + \tau_{13}^2}{S_{12}^2} \geq 1 \quad (3.40)$$

Compressive fibre failure for $\sigma_1 < 0$:

$$\left(\frac{\sigma_1}{X_c}\right)^2 \geq 1 \quad (3.41)$$

Tensile matrix failure for $\sigma_2 + \sigma_3 > 0$:

$$\left(\frac{\sigma_2 + \sigma_3}{Y_t}\right)^2 + \frac{\tau_{23}^2 - \sigma_2\sigma_3}{S_{23}^2} + \frac{\tau_{12}^2 + \tau_{13}^2}{S_{12}^2} \geq 1 \quad (3.42)$$

Compressive matrix failure for $\sigma_2 + \sigma_3 < 0$:

$$\left[\left(\frac{Y_c}{2S_{23}}\right)^2 - 1\right] \frac{\sigma_2 + \sigma_3}{Y_c} + \frac{(\sigma_2 + \sigma_3)^2}{4S_{23}^2} + \frac{\tau_{23}^2 - \sigma_2\sigma_3}{S_{23}^2} + \frac{\tau_{12}^2 + \tau_{13}^2}{S_{12}^2} \geq 1 \quad (3.43)$$

Under a plane stress assumption, the formulation is simplified because $\sigma_3 = \tau_{13} = \tau_{23} = 0$. In that case, the individual criteria can be reduced to [39]:

Tensile fibre failure for $\sigma_1 \geq 0$:

$$\left(\frac{\sigma_1}{X_t}\right)^2 + \left(\frac{\tau_{12}}{S}\right)^2 \geq 1 \quad (3.44)$$

Compressive fibre failure for $\sigma_1 < 0$:

$$\left(\frac{\sigma_1}{X_c}\right)^2 \geq 1 \quad (3.45)$$

Tensile matrix failure for $\sigma_2 > 0$:

$$\left(\frac{\sigma_2}{Y_t}\right)^2 + \left(\frac{\tau_{12}}{S}\right)^2 \geq 1 \quad (3.46)$$

Compressive matrix failure for $\sigma_2 < 0$:

$$\left(\frac{\sigma_2}{Y_c}\right)^2 + \left(\frac{\tau_{12}}{S}\right)^2 \geq 1 \quad (3.47)$$

In this thesis, the material under study is woven CFRP. In this case, the distinction between fibre and matrix failure cannot be interpreted in the same way of unidirectional composites. Consequently, the Hashin formulation is adapted so that the same form of the criterion is applied to both warp and weft directions. The modified criteria are:

Tensile failure in longitudinal fibres (warp) for $\sigma_1 \geq 0$:

$$\left(\frac{\sigma_1}{X_t}\right)^2 + \left(\frac{\tau_{12}}{S}\right)^2 \geq 1 \quad (3.48)$$

Compressive failure in longitudinal fibres (warp) for $\sigma_1 < 0$:

$$\left(\frac{\sigma_1}{X_c}\right)^2 \geq 1 \quad (3.49)$$

Tensile failure in transverse fibres (weft) for $\sigma_2 > 0$:

$$\left(\frac{\sigma_2}{Y_t}\right)^2 + \left(\frac{\tau_{12}}{S}\right)^2 \geq 1 \quad (3.50)$$

Compressive failure in transverse fibres (weft) for $\sigma_2 < 0$:

$$\left(\frac{\sigma_2}{Y_c}\right)^2 \geq 1 \quad (3.51)$$

3.6.3 Baseline Steel Structure

The steel baseline structure specified in the Formula Student rules is idealised as a simply supported circular tube (or a parallel set of n identical tubes) subjected to a central point load over a span L . The peak steel stress occurs at the outer surface beneath the punch and is evaluated using linear elastic bending. With inner diameter

$$d = D - 2t, \quad (3.52)$$

and second moment of area

$$I = \frac{\pi}{64} (D^4 - d^4), \quad (3.53)$$

the outer-surface bending stress follows from

$$\sigma = \frac{MD}{2I}, \quad M = \frac{FL}{4}. \quad (3.54)$$

Equating the bending stress to the steel yield stress, $\sigma = \sigma_y$, gives the single-tube capacity in three-point bending. For n identical baseline tubes sharing the load uniformly, the reference load for structural equivalency becomes

$$F_{\text{baseline}} = n \cdot \frac{8\sigma_y}{DL} \cdot \frac{\pi}{64} (D^4 - d^4). \quad (3.55)$$

This result is used as the benchmark against which the analytically predicted failure load of the composite sandwich panel is compared within the SES framework.

3.6.4 Implementation

The failure load prediction tool was implemented in MATLAB using an iterative load progression approach. Instead of solving for a single analytical load, the tool gradually increases the applied load on the sandwich panel section in discrete increments. At each load level, the internal stress state of the laminate is evaluated using Classical Laminate Theory (CLT). The procedure follows these steps:

1. Determine the bending moment per unit width at mid-span from the applied load and the support span specified by the Formula Student rules.
2. Using the panel's bending compliance matrix (Section 3.5), compute the mid-plane strains and curvatures corresponding to this moment.

3.7. Energy Absorption Estimation

3. Recover the through-thickness strain state at the centroid of each ply in both facesheets, transform it to the local material axes, and compute the corresponding ply stresses from the lamina stiffness properties.
4. Evaluate the ply stress state against the selected failure criteria: Maximum Stress, Tsai–Hill, Tsai–Wu, and Hashin.
5. For each criterion, identify the applied load at which the first ply reaches a failure index of 1.0; this load is taken as the predicted panel failure load for that criterion.
6. The minimum of the three predicted failure loads is adopted as the governing value for comparison with the baseline steel structure yield force.

The simulation continues until all three failure criteria reach their respective thresholds. Although each criterion offers different theoretical advantages, such as Hashin’s ability to distinguish between failure modes, all are retained to enable a direct comparison with experimental three-point bending results. This approach allows the accuracy of each failure model to be assessed in practice, supporting both the refinement of the analytical framework and the selection of the most reliable criterion for future design validation. In doing so, the tool contributes to early-stage design optimisation and compliance with SES regulations.

The MATLAB code developed for this analysis is provided in Appendices A.5 and A.6.

3.7 Energy Absorption Estimation

In addition to flexural rigidity and load-bearing capacity, the energy absorption capability of the monocoque panels is a critical parameter for ensuring occupant safety under crash conditions. Formula Student SES regulations require composite panels to demonstrate sufficient energy absorption in the event of impact. This section presents the analytical methodology developed to estimate the energy absorbed by a composite sandwich panel under loading, based on Classical Laminated Theory (CLT) and laminate failure predictions.

3.7.1 Theoretical Background

The energy absorption U of a structure under flexural loading can be approximated by integrating the area under the force–displacement curve up to the point of failure. Assuming a linear elastic response up to first-ply failure, this is given by

$$U = \frac{1}{2} F_{\text{fail}} \delta_{\text{fail}}, \quad (3.56)$$

where:

- F_{fail} is the analytically predicted failure load from the 3PB failure analysis (Section 3.6);
- δ_{fail} is the corresponding mid-span deflection at failure.

For a simply supported beam with a central point load, the mid-span deflection is

$$\delta_{\text{fail}} = \frac{F_{\text{fail}} L^3}{48 EI_{\text{panel}}}, \quad (3.57)$$

with L being the support span and EI_{panel} the flexural rigidity defined in Section 3.5.

Substituting gives:

$$U = \frac{F_{\text{fail}}^2 L^3}{96 EI_{\text{panel}}}. \quad (3.58)$$

This formulation neglects shear deformation, local indentation, and post-failure load carrying; it is therefore most accurate for stiff-core sandwich panels with failure dominated by facesheet rupture under bending.

3.7.2 Implementation

The energy absorption estimation is implemented in the MATLAB tool alongside the bending failure analysis. For each laminate configuration:

1. Predict F_{fail} from the selected failure criteria (Section 3.6);
2. Calculate δ_{fail} using beam theory;
3. Compute U from Equation 3.58;
4. Compare U to the minimum SES energy absorption requirement.

This method enables direct comparison between candidate panel designs in terms of stiffness, strength, and crash energy mitigation. Predictions will later be validated against energy absorption values obtained from experimental three-point bending tests. The relevant code is available in Appendix A.7.

3.8 Perimeter Shear Strength

In addition to flexural rigidity, Formula Student rules require that each panel of the composite monocoque exhibit adequate resistance to perimeter shear failure. This requirement ensures that composite structures can withstand localised loads and maintain structural integrity under crash, impact, or high-deformation scenarios. The perimeter shear strength is evaluated by comparing the predicted resistance of the laminate to a regulatory threshold, and verified experimentally through standardised test procedures.

3.8.1 Theoretical Background

The Structural Equivalency Spreadsheet (SES) [40] defines the perimeter shear strength of a sandwich panel according to the following expression:

$$F_{\text{per. shear}} = 2 \cdot (W + L) \cdot t \cdot \sigma_{\text{shear}} \quad (3.59)$$

where:

- W is the sandwich panel width [m];
- L is the sandwich panel length [m];
- $2 \cdot (W + L)$ is the perimeter of the sandwich panel [m];
- t is the total facesheet thickness [m];
- σ_{shear} is the out-of-plane shear strength of the sandwich panel, determined experimentally [Pa].

3.9. Compliance and Safety Factor Evaluation

The parameter σ_{shear} must be determined experimentally for each specific sandwich panel layup configuration, using perimeter shear testing. However, to facilitate preliminary design iterations without the need for extensive experimental testing, an approximate estimation method was considered. In this approach, σ_{shear} is conservatively approximated by the interlaminar shear strength (ILSS) of the facesheet material. This yields the following modified expression for the estimated perimeter shear strength:

$$F_{\text{per. shear}} \approx 2 \cdot (W + L) \cdot n \cdot t_{\text{ply}} \cdot \sigma_{\text{ILSS}} \quad (3.60)$$

where:

- n is the number of plies in the thinner facesheet of the sandwich panel;
- t_{ply} is the nominal ply thickness [m];
- σ_{ILSS} is the interlaminar shear strength of the facesheet material [Pa].

This estimation method is intended to provide conservative preliminary layup designs, which must subsequently be validated through physical testing to ensure compliance with the SES requirements.

3.8.2 Implementation

The calculation of perimeter shear strength was implemented in MATLAB to enable systematic evaluation across all monocoque panels. The procedure is as follows:

1. Identify the thinner laminate skin, particularly in the case of asymmetric layups;
2. Extract the interlaminar shear strength from material characterisation or datasheets;
3. Compute the total perimeter engaged in shear, typically as the sum of relevant panel edges depending on the loading scenario;
4. Apply Equation 3.60;
5. Compare the result with the required minimum for the given structural section (F_{required});
6. Calculate the safety factor.

This approach enables a rapid and conservative assessment of perimeter shear compliance and provides design feedback when skin thickness or shear strength must be increased to meet regulatory targets.

The corresponding MATLAB routine is detailed in Appendix A.8.

3.9 Compliance and Safety Factor Evaluation

This final section consolidates the results of all analytical evaluations carried out to assess the structural equivalency of the composite sandwich panels. The metrics calculated throughout the chapter - flexural rigidity, perimeter shear strength, failure load under three-point bending, and energy absorption - are directly compared against the minimum thresholds specified in the Formula Student rulebook and SES documentation.

Each panel is assessed on a pass/fail basis for every criterion, and a conservative safety factor is computed to quantify the degree of compliance. Only panels that exceed the required mini-

mum in all categories are considered structurally equivalent and eligible for implementation in the monocoque chassis.

To aid in this evaluation, Table 3.3 summarises the key analytical outputs obtained from the validation framework. These results form the basis for compliance checks and guide the selection of promising panel configurations for experimental testing and final SES submission.

Table 3.3: Key analytical outputs from panel validation.

Metric Category	Output Description	Symbol / Unit
Flexural Rigidity	Global flexural rigidity of the panel	EI_{panel} [N · mm ²]
Failure Load (3PB)	Predicted failure load - Maximum Stress criterion	$F_{\text{fail}}^{(\text{MS})}$ [N]
	Predicted failure load - Tsai-Hill criterion	$F_{\text{fail}}^{(\text{TH})}$ [N]
	Predicted failure load - Tsai-Wu criterion	$F_{\text{fail}}^{(\text{TW})}$ [N]
	Predicted failure load - Hashin criterion	$F_{\text{fail}}^{(\text{H})}$ [N]
Energy Absorption	Predicted absorbed energy to failure	U [J]
Perimeter Shear	Theoretical perimeter shear capacity	F_{shear} [N]

This evaluation framework ensures that all critical mechanical behaviours are verified through both theory and direct comparison against regulatory targets. By identifying compliant panels early in the design process, it supports informed decision-making, minimises development risk, and streamlines the SES approval workflow.

Chapter 4

Sizing Methods Validation

This chapter validates the analytical methods developed in the preceding chapter for the analysis of composite sandwich panels. The validation is carried out through dedicated experimental testing. It begins with the mechanical characterisation of the carbon fibre-reinforced polymer (CFRP) material used throughout the test campaign, followed by the comparison of experimental results from three-point bending and perimeter shear tests with the predictions from the analytical formulations.

4.1 Material Selection and Characterisation

The CFRP used in this study consists of a prepreg orthogonal spread tow fabric ($0^\circ/90^\circ$) with an epoxy matrix. Each ply has an areal density of approximately 160 g/m^2 and a nominal thickness of 0.160 mm . Characterisation of its mechanical properties was conducted to support the validation of the proposed sizing.

Composite materials exhibit strong anisotropic behaviour, and their mechanical response is highly dependent on fibre orientation, stacking sequence, and loading direction. To ensure that the analytical and numerical tools reflect the actual behaviour of the laminate, it is crucial to obtain accurate values of stiffness and strength in the principal material directions. Therefore, standardised tests were performed to characterise the tensile, compressive, and in-plane shear properties of the laminate in both the warp (longitudinal) and weft (transverse) directions.

These tests serve two distinct purposes. Firstly, they provide the necessary input parameters for the analytical formulations, such as Young's moduli, shear moduli, Poisson's ratio and in-plane strengths. Secondly, they offer a benchmark for evaluating the predictive capability of the models, by allowing direct comparison between simulated and experimental response.

To this end, a test matrix was established following ASTM standards: ASTM D3039 for uniaxial tension, ASTM D3518 for $\pm 45^\circ$ in-plane shear, and ASTM D6641 for uniaxial compression. Each configuration was tested using six specimens to ensure statistical consistency and minimise the influence of outliers. The layups were designed to isolate the response in each direction or loading mode, as detailed in Table 4.1.

4.1. Material Selection and Characterisation

Table 4.1: Test matrix for CFRP characterisation.

Test	Orientation	Lay-up	No. Plies	Width (mm)	Length (mm)	Gauge length (mm)
Tension	Longitudinal (warp)	[0] ₁₀	10	25	250	100
Tension	Transverse (weft)	[90] ₁₀	10	25	250	100
Shear	In-plane	[45/-45] _{3s}	12	25	250	100
Compression	Longitudinal (warp)	[0] ₂₀	20	12	140	13
Compression	Transverse (weft)	[90] ₂₀	20	12	140	13

4.1.1 Specimen Manufacturing

Three types of composite panels were manufactured to produce specimens for the tension, shear, and compression tests. The layups were defined according to the requirements of each standard: ten plies for tension tests, twelve for shear, and twenty for compression.

Curing was carried out in an autoclave under a two-stage cycle: an initial vacuum-only stage at 80° C, followed by a second stage at 130° C and 6 bar for approximately one hour. All panels were produced with square caul plates on the top side, measuring 325 × 325 mm². Once cured, the panels were cut to final specimen dimensions using a water jet machine.

The 2 mm thick aluminium tabs were bonded to the specimens using epoxy adhesive (Loctite EA 9466), cured at room temperature under ambient pressure. Prior to bonding, both the CFRP specimens and aluminium tabs were meticulously cleaned to ensure surface cleanliness and uniform adhesion. The bonding surfaces of the composite specimens were lightly sanded to enhance roughness, while the aluminium tabs had been sandblasted beforehand to improve mechanical interlocking.

The adhesive was applied carefully to avoid excessive thickness, and the specimens were placed into a dedicated alignment jig provided by INEGI, ensuring proper axial alignment. Pressure was applied by means of a heavy steel plate secured with bolts and nuts, ensuring uniform contact across all bonded surfaces. After approximately one hour, the specimens were removed from the jig to prevent unintentional bonding between the adhesive and the fixture. The epoxy was then allowed to cure fully for 24 hours at room temperature. Any excess adhesive was removed, and the bonded ends were manually sanded to ensure flatness and perpendicularity. All specimens were visually inspected, and final dimensional checks were conducted to ensure compliance with the respective test standards.

With all specimens prepared and tabbed, the next stage involved mechanical testing under controlled conditions to extract the required material properties.

4.1.2 Testing

All mechanical tests were performed using an MTS 810 Material Test System equipped with a 100 kN load cell (MTS Systems Corporation, Model 661.20F-03, Serial No. V71740) and a displacement transducer (Model LVDT318.25, Serial No. 0179699), calibrated for a ±100 mm displacement range. The testing protocol was tailored to each loading mode, following the corresponding ASTM standard, with controlled loading rates and appropriate strain measurement techniques.

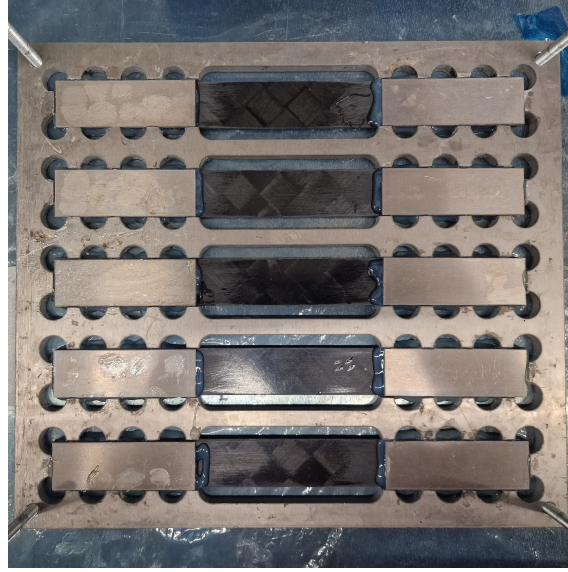


Figure 4.1: Jig used for accurate alignment of aluminium tabs during bonding.

Compression testing was carried out at a constant crosshead speed of 1 mm/min, using a Combined Loading Compression (CLC) Test Fixture in accordance with ASTM D6641. The specimen layups and dimensions were specifically chosen to ensure stable failure modes and minimise buckling. Load and displacement data were recorded at a sampling rate of 10 Hz.

Tensile characterisation was conducted following ASTM D3039, also at a crosshead speed of 1 mm/min. Digital Image Correlation (DIC) was used to measure longitudinal strain, with images acquired every 2 seconds, corresponding to a sampling frequency of 0.5 Hz. For validation purposes, one specimen per configuration (0° and 90°) was equipped with an axial strain gauge, allowing for a direct comparison between DIC-based and traditional strain measurements.

In-plane shear testing followed ASTM D3518 and was performed at a higher loading rate of 5 mm/min. Strain data was obtained using DIC, capturing both longitudinal and transverse strains with a frequency of 0.5 Hz. To validate the DIC measurements, one specimen was fitted with two strain gauges aligned with the principal strain directions. The load and displacement signals were sampled at 10 Hz.

DIC analysis was performed using the Zeiss Correlate software to extract full-field strain data from the tensile and shear tests. For each recorded image (stage), the software computed the strain fields in both the longitudinal and transverse directions. To mitigate local noise and ensure consistency, the strain values were averaged across the entire field of view, with attention paid to pattern quality metrics to exclude regions with low correlation or poor speckle contrast. The resulting strain values were then post-processed in MATLAB, where temporal interpolation was performed to align the DIC-derived strain data with the higher-frequency time stamps of the force and displacement measurements obtained from the test machine. This synchronisation enabled the construction of accurate stress-strain curves and the computation of material properties.

All tests were conducted under ambient laboratory conditions. The combined use of traditional strain gauges and full-field DIC provided robust and redundant strain measurements, enabling reliable validation of the experimental results discussed in the following section.

4.1. Material Selection and Characterisation

4.1.3 Results

Experimental results obtained from the tensile, shear, and compression tests performed on the CFRP laminate are presented here. For each test configuration, the relevant strength and stiffness parameters are reported, along with statistical descriptors to assess data consistency. Where applicable, comparisons between different strain measurement techniques are included to evaluate measurement reliability. Boxplots and stress-strain curves are used to illustrate the overall mechanical response and to highlight the laminate’s anisotropic behaviour. The final part of this subsection consolidates the derived values into a single set of properties to be used in all subsequent analyses.

4.1.3.1 Tension Tests

Uniaxial tension tests were performed on six specimens for both the 0° and 90° directions, in accordance with ASTM D3039. The primary goal was to determine the tensile strength, Young’s modulus, and Poisson’s ratio in each fibre orientation, thereby capturing the directional stiffness and anisotropic behaviour of the laminate.

From these tests, maximum stress values were extracted for all twelve specimens. Five of the six specimens in each configuration were analysed using DIC. Figures 4.2 to 4.5 present representative strain field images obtained from the analysis conducted in ZEISS Correlate. For each direction, both the raw strain field and the corresponding image overlaid on the specimen surface are shown, at two different applied strain levels (0.4% and 0.6%). The strain field in the longitudinal direction is reasonably uniform, but several points of poor correlation can be identified, which can be partially attributed to the speckle pattern quality (e.g., due to the presence of some larger black dots). It is recognised that some additional work in setting the DIC post-processing parameters would have been required to achieve more uniform correlation. Nevertheless, the average strain across the entire field of view seems to provide a reasonable strain measure to determine the elastic moduli. Similarly, the transverse strain fields show significant noise due to the fact that the transverse strains are significantly smaller, especially in orthogonal woven composites which exhibit low Poisson’s ratios. Therefore, this introduces some uncertainty into the corresponding Poisson’s ratio calculations.

Yet, to validate the accuracy of the DIC strain measurements, one specimen per configuration (LT06 for 0° and LT96 for 90°) was instrumented with a strain gauge. The time-aligned strain data from both techniques showed good agreement. For the 0° specimen, the root-mean-square (RMS) error between DIC and strain gauge measurements was an absolute strain difference of 0.0596% (i.e., percentage strain), corresponding to a relative error of 7.86% when normalised against the strain gauge values. Similarly, the 90° specimen exhibited an RMS strain error of 0.0419%, equivalent to a relative error of 7.27%. These results confirm the reliability of the DIC method for this test setup. The comparison plots are shown in Figures 4.6 and 4.7. It should be noted that in some cases, such as in Figure 4.6, strain does not appear to increase immediately at the start of the test. This delay is caused by the initial phase during which the testing machine compensates for slack or minor misalignments in the fixture, before actual loading of the specimen begins. Once contact is fully established, both strain measurement techniques show a consistent and linear response.

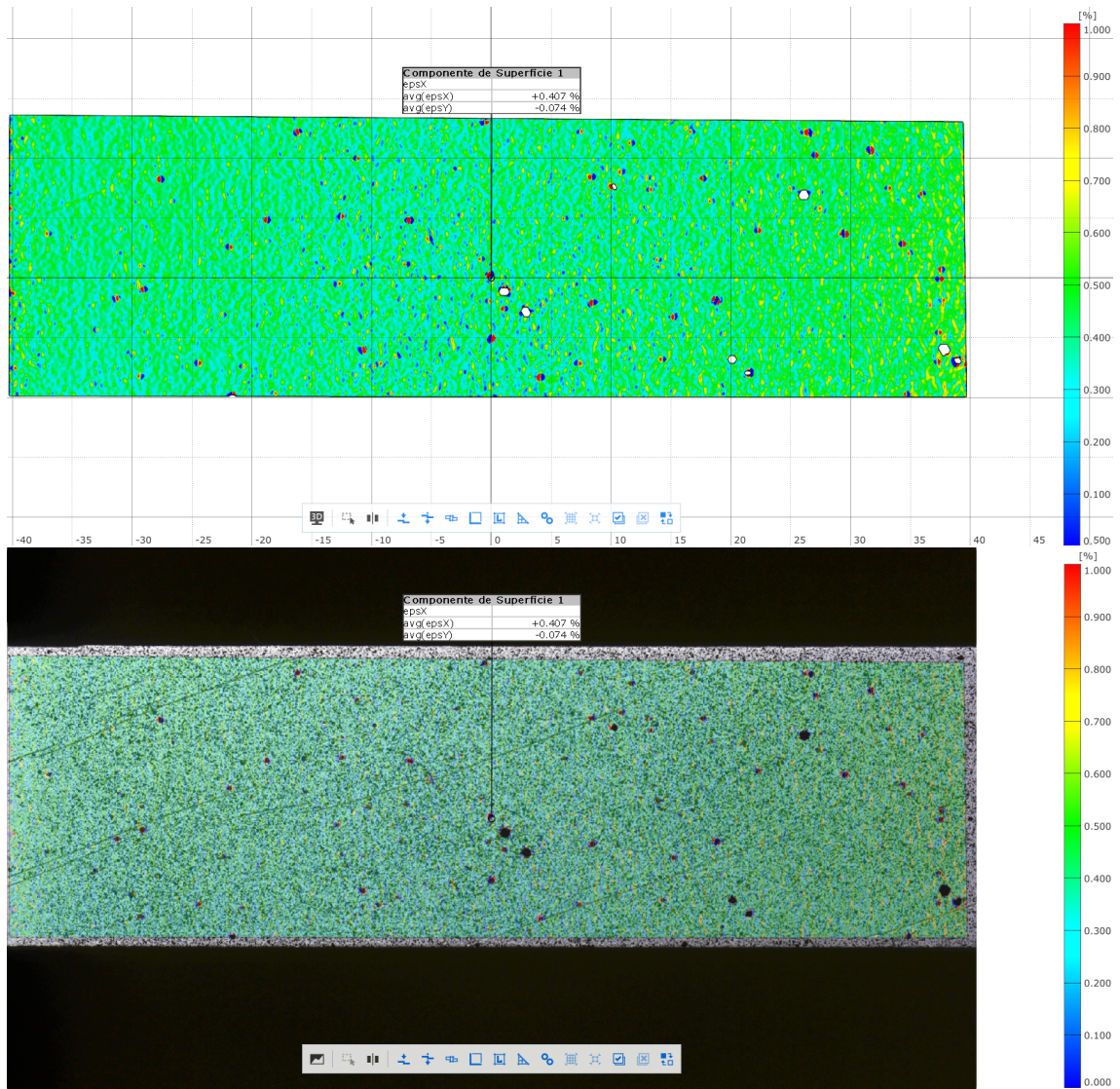


Figure 4.2: Longitudinal strain field at 0.4% strain: without background (top) and with specimen background (bottom).

4.1. Material Selection and Characterisation

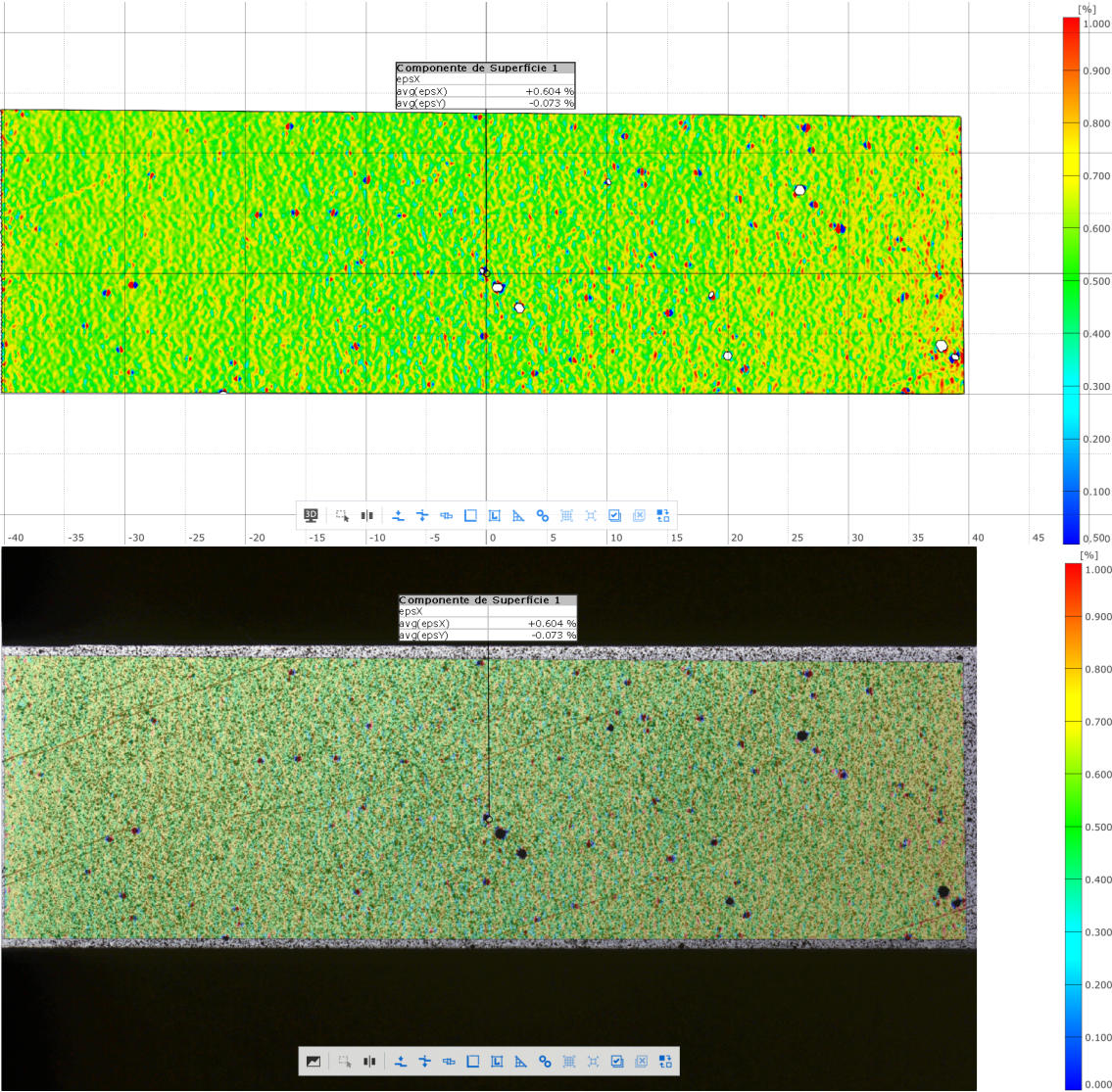


Figure 4.3: Longitudinal strain field at 0.6% strain: without background (top) and with specimen background (bottom).

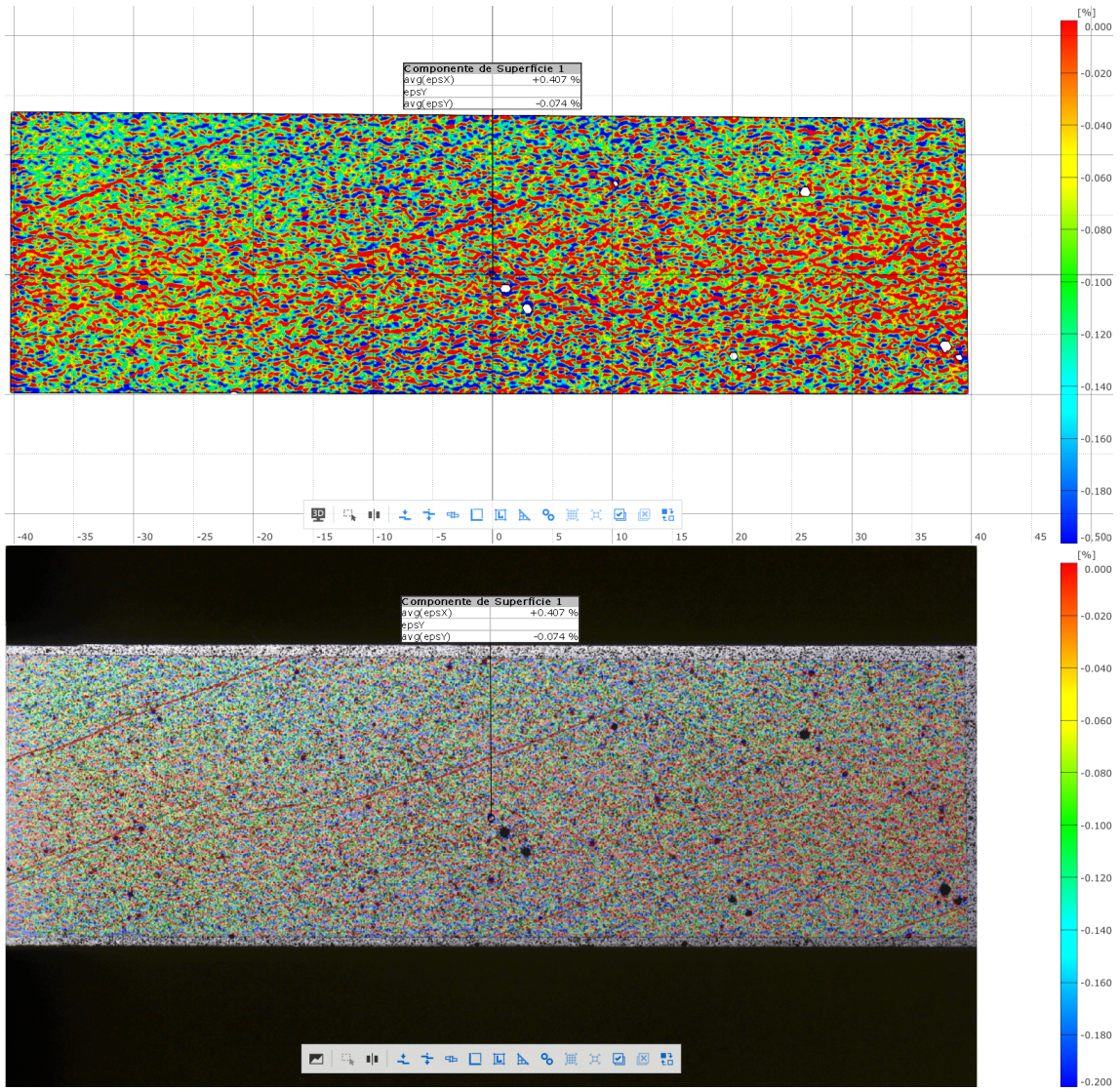


Figure 4.4: Transverse strain field at 0.4% strain: without background (top) and with specimen background (bottom).

4.1. Material Selection and Characterisation

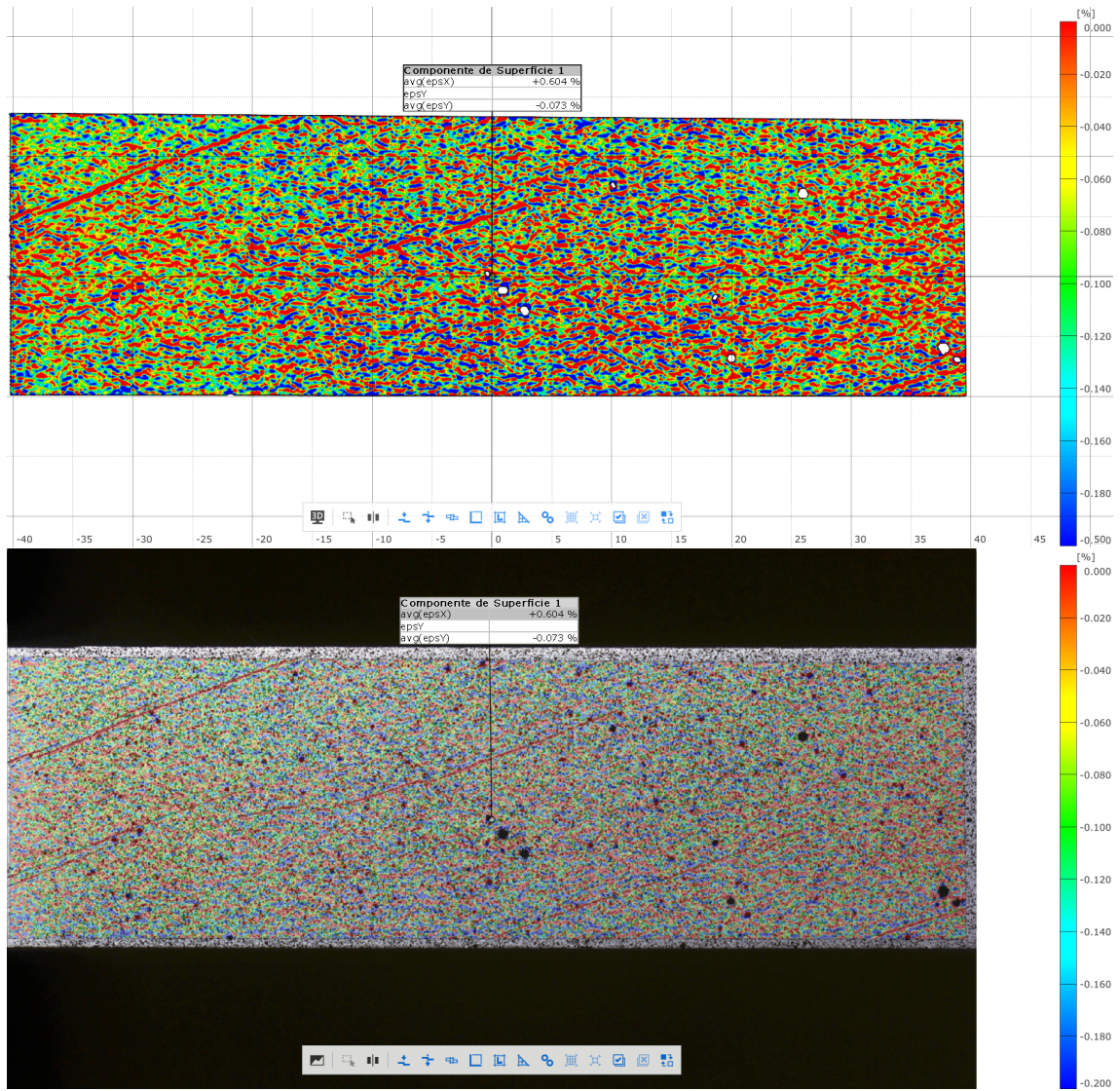


Figure 4.5: Transverse strain field at 0.6% strain: without background (top) and with specimen background (bottom).

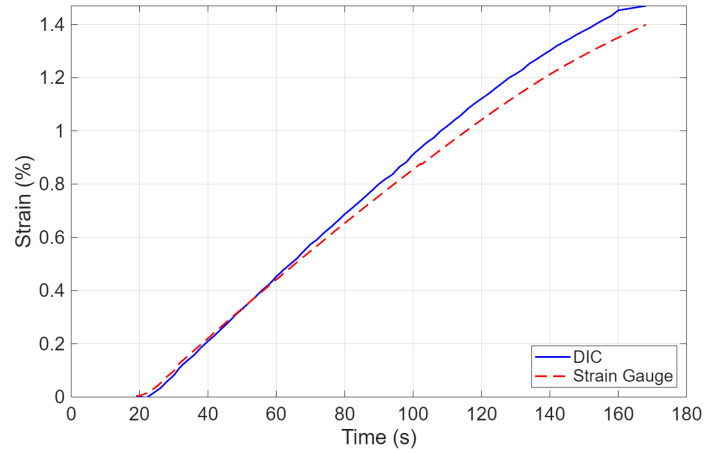


Figure 4.6: Strain-time comparison between DIC and strain gauge (LT06, 0° specimen).

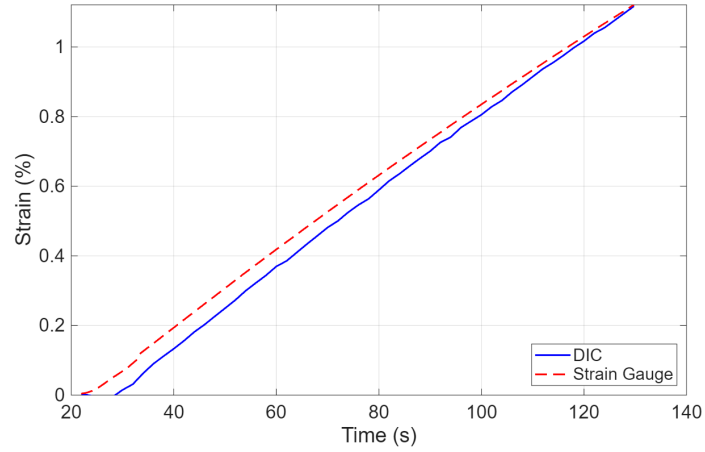


Figure 4.7: Strain-time comparison between DIC and strain gauge (LT96, 90° specimen).

The stress-strain curves using DIC strain data are presented in Figures 4.8 to 4.11. These were used to compute the tensile modulus and Poisson's ratio for each specimen via linear regression in the strain interval between 0.4% and 0.6%. Although ASTM D3039 recommends the use of the 0.1%-0.3% strain interval for modulus evaluation, it also permits alternative ranges within the initial half of the test, provided they are adequately justified. The chosen interval of 0.4%-0.6% exhibited reduced noise and more stable linearity in the measured response, leading to more consistent modulus values across specimens.

It should be noted that the initial portion of some curves displays unexpected behaviour, such as apparent preload or even negative strain, which may result from a combination of factors, including fixture slack, the strain measurement method, or alignment imperfections during specimen setup.

Furthermore, the transverse strain response, seen in Figures 4.9 and 4.11, appears particularly noisy. This issue is exacerbated by the very small absolute strain values involved, which makes their detection by DIC more challenging and substantially reduces measurement accuracy. Consequently, the Poisson's ratio values obtained must be interpreted with caution, as they are highly sensitive to noise.

4.1. Material Selection and Characterisation

Additionally, in Figure 4.10, one specimen exhibits an anomalous response, with strain seemingly accumulating before any measurable force is applied, and only later developing resistance to loading. This specimen also displays a noticeably steeper initial slope, implying a higher modulus, despite reaching a maximum load similar to the other specimens. The cause of this behaviour remains unclear but may be linked to test setup artefacts or irregularities specific to that specimen. This specimen was therefore considered an outlier and was not included in the subsequent calculations.

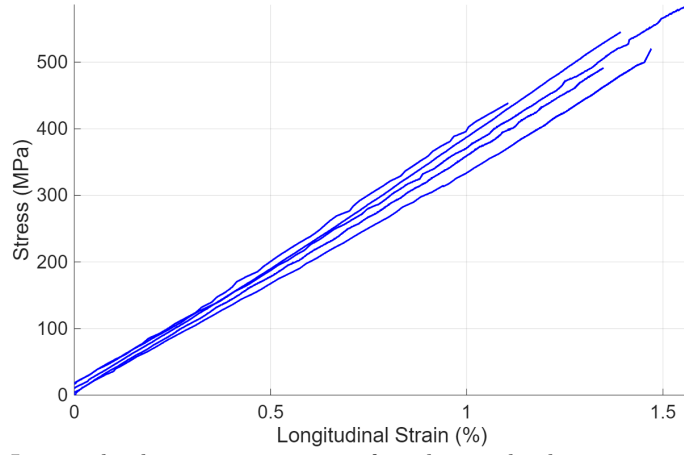


Figure 4.8: Longitudinal stress-strain curves from longitudinal tension tests (0° specimens).

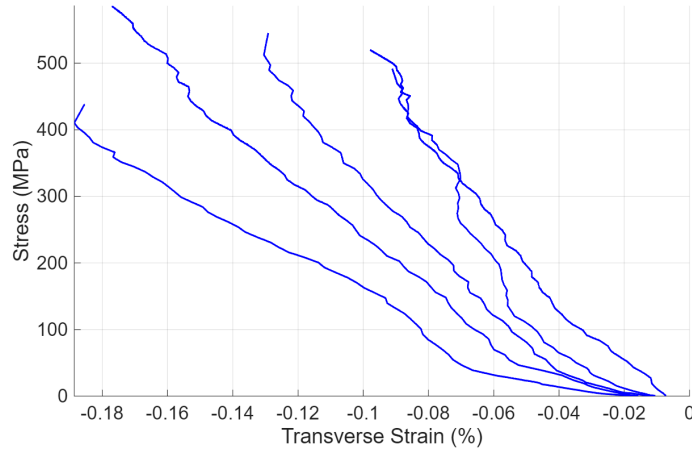


Figure 4.9: Transverse stress-strain curves from longitudinal tension tests (0° specimens).

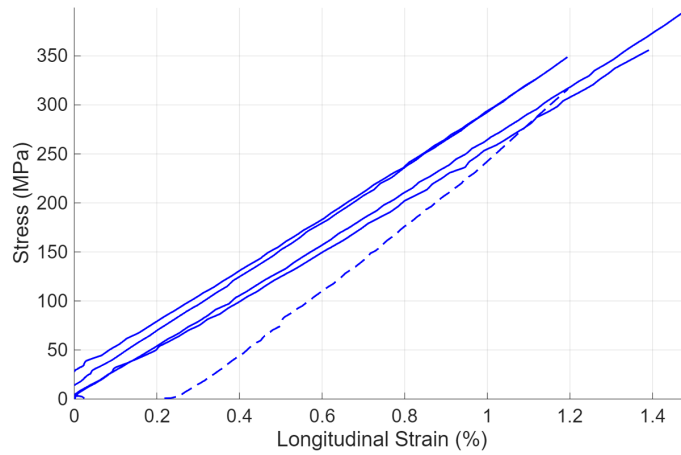


Figure 4.10: Longitudinal stress-strain curves from transverse tension tests (90° specimens).

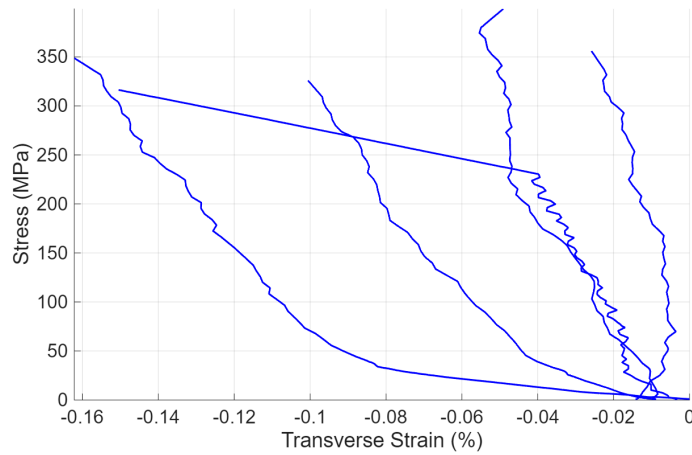


Figure 4.11: Transverse stress-strain curves from transverse tension tests (90° specimens).

Boxplots of the tensile strength, elastic modulus, and Poisson's ratio are shown in Figures 4.12, 4.13, and 4.14, respectively. In each boxplot, the red horizontal line within the box represents the median of the dataset, providing a measure of central tendency. The blue box spans from the first quartile (Q1) to the third quartile (Q3), thereby capturing the interquartile range (IQR) which contains the middle 50% of the data. The black whiskers extend from the box to the minimum and maximum values within 1.5 times the IQR from the lower and upper quartiles, respectively. Any points lying outside this range are considered statistical outliers and are represented by red crosses. This visual representation allows for a compact comparison of the dispersion and symmetry of the datasets, as well as the identification of potential outliers.

4.1. Material Selection and Characterisation

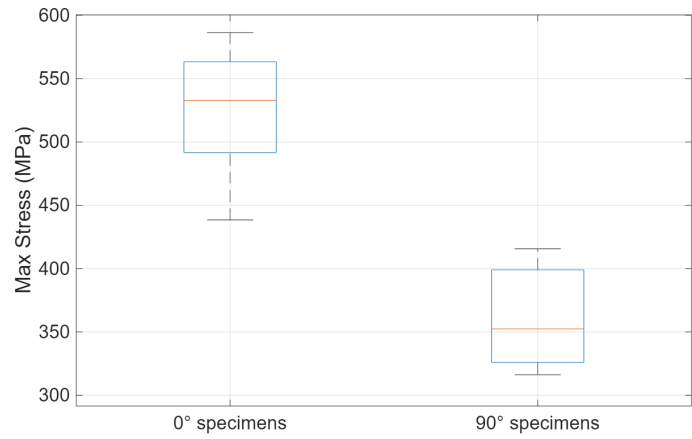


Figure 4.12: Boxplot of maximum stress for each test configuration.

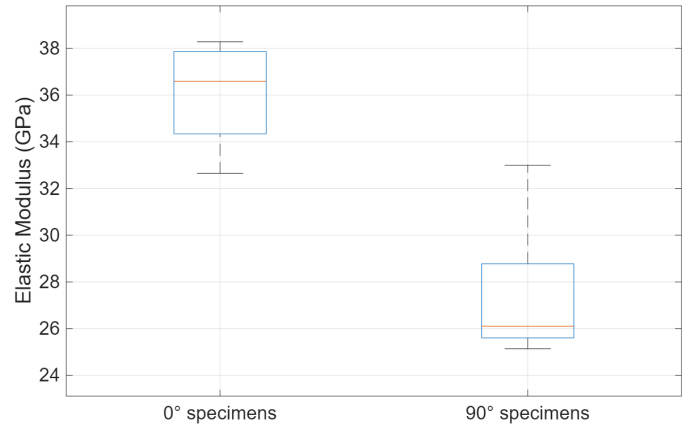


Figure 4.13: Boxplot of calculated modulus for each test configuration.

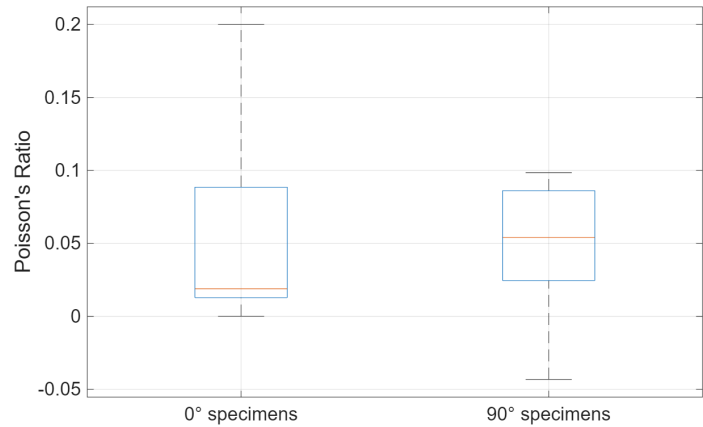


Figure 4.14: Boxplot of Poisson's ratio for each test configuration.

Table 4.2 summarises the mean, standard deviation, and coefficient of variation (CV) for each measured parameter.

Table 4.2: Summary of tensile test results.

Orientation	Parameter	Mean	Standard Deviation	CV (%)
0°	Max stress (MPa)	524.1	53.39	10.19
	Elastic Modulus (GPa)	36.03	2.291	6.358
	Poisson's ratio	0.05725	0.08212	142.3
90°	Max stress (MPa)	360.3	39.58	10.99
	Elastic Modulus (GPa)	27.47	3.195	11.63
	Poisson's ratio	0.04834	0.05541	115.1

Contrary to typical expectations for balanced woven CFRP fabrics, the laminate exhibits substantially greater tensile strength in the warp direction (0°) compared to the weft direction (90°), suggesting a possible imbalance in fibre packing, tension during weaving, or manufacturing variability. While tensile strength shows acceptable variation in both directions (CV $\approx$ 10%), the modulus in the weft direction exhibits higher variability. The Poisson's ratio measurements presented high coefficients of variation, highlighting the sensitivity of this parameter to both measurement technique and noise in strain data, particularly at small strain values. Consequently, Poisson's ratio values reported here should be interpreted with caution.

4.1.3.2 Shear Tests

In-plane shear characterisation was conducted using the $\pm 45^\circ$ tension method in accordance with ASTM D3518. Six specimens were tested, with the objective of determining the shear modulus (G) and shear strength, defined as the stress corresponding to 5% engineering shear strain, as recommended by the standard.

Five specimens were analysed using Digital Image Correlation (DIC). Figures 4.15 to 4.18 show representative strain field images for the shear tests, in both longitudinal and transverse directions. For each case, the raw strain field and the corresponding image overlaid on the specimen are displayed at two deformation levels (0.3% and 0.6% engineering shear strain). The longitudinal and transverse strain fields display a noisy distribution, which can, again, be attributed to the speckle pattern quality and partially inadequate DIC post-processing parameters.

To evaluate the reliability of DIC-based strain measurements under shear loading, specimen LTS6 was instrumented with two strain gauges: one aligned with the longitudinal axis and another with the transverse axis. The DIC strain data were compared directly to the strain gauge signals. For the longitudinal strain, the RMS error was 0.1627%, corresponding to a relative error of 14.41% when normalised against the strain gauge values. For the transverse strain, the RMS error increased to 0.4858%, or 51.83% in relative terms.

This comparison was limited to the initial 35 seconds of the test, as the strain gauge signals beyond this point exhibited abrupt discontinuities, likely due to debonding or signal saturation. As shown in Figures 4.19 and 4.20, the DIC measurements remained smooth throughout the test duration, whereas the strain gauge data showed sudden deviations. Accurately capturing in-plane shear behaviour requires at least two strain gauges, increasing the risk of mounting errors and alignment issues. Additionally, strain gauges are susceptible to failure under large deformations, as observed in this case, which can compromise signal integrity.

The purpose of this comparison was to determine whether DIC could provide sufficiently reliable strain data for the purposes of material characterisation. The agreement observed in the longitudinal strain supports its use, although the higher error in the transverse direction requires

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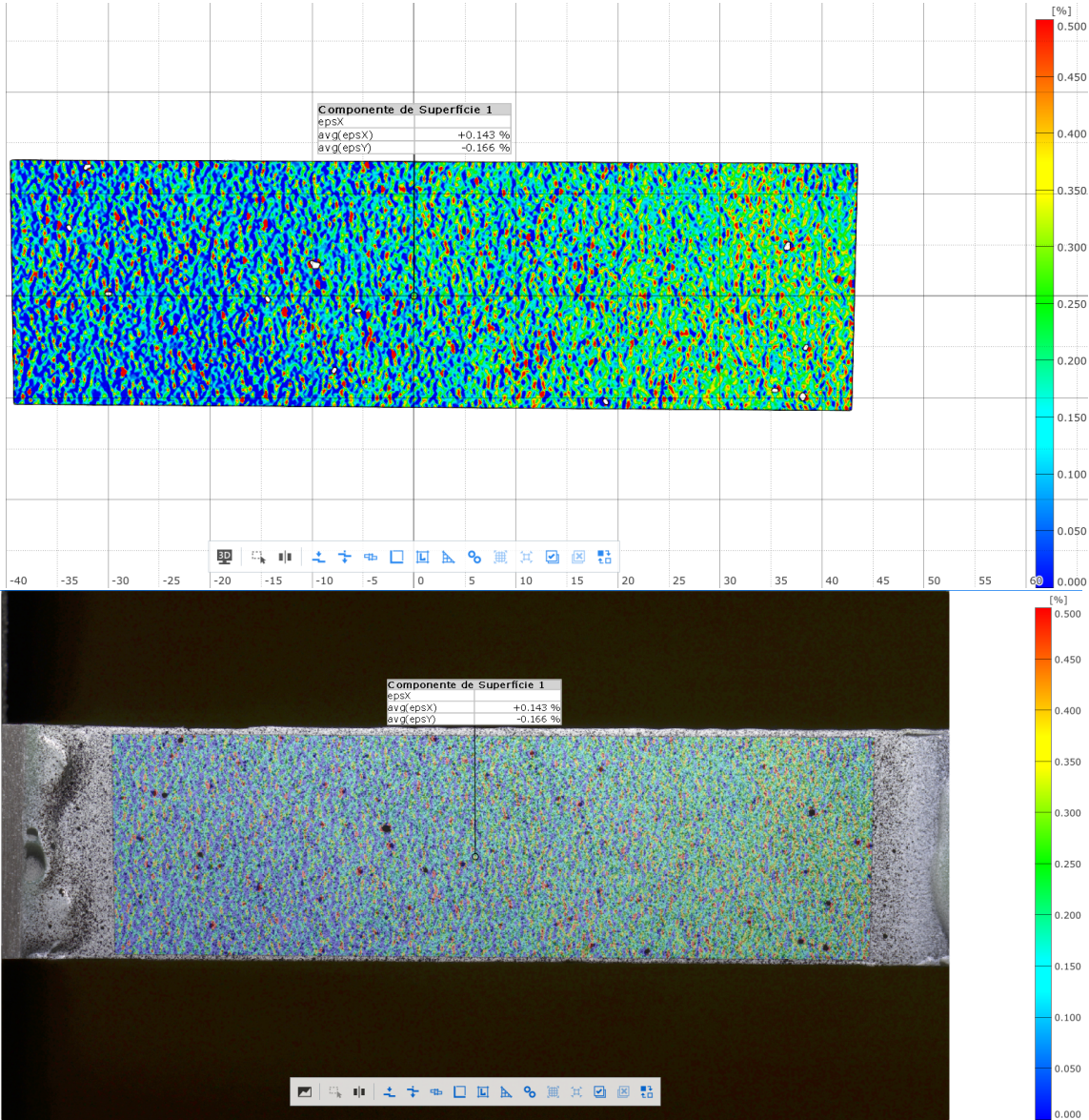


Figure 4.15: Shear test longitudinal strain field at 0.3% shear strain: without background (top) and with specimen background (bottom).

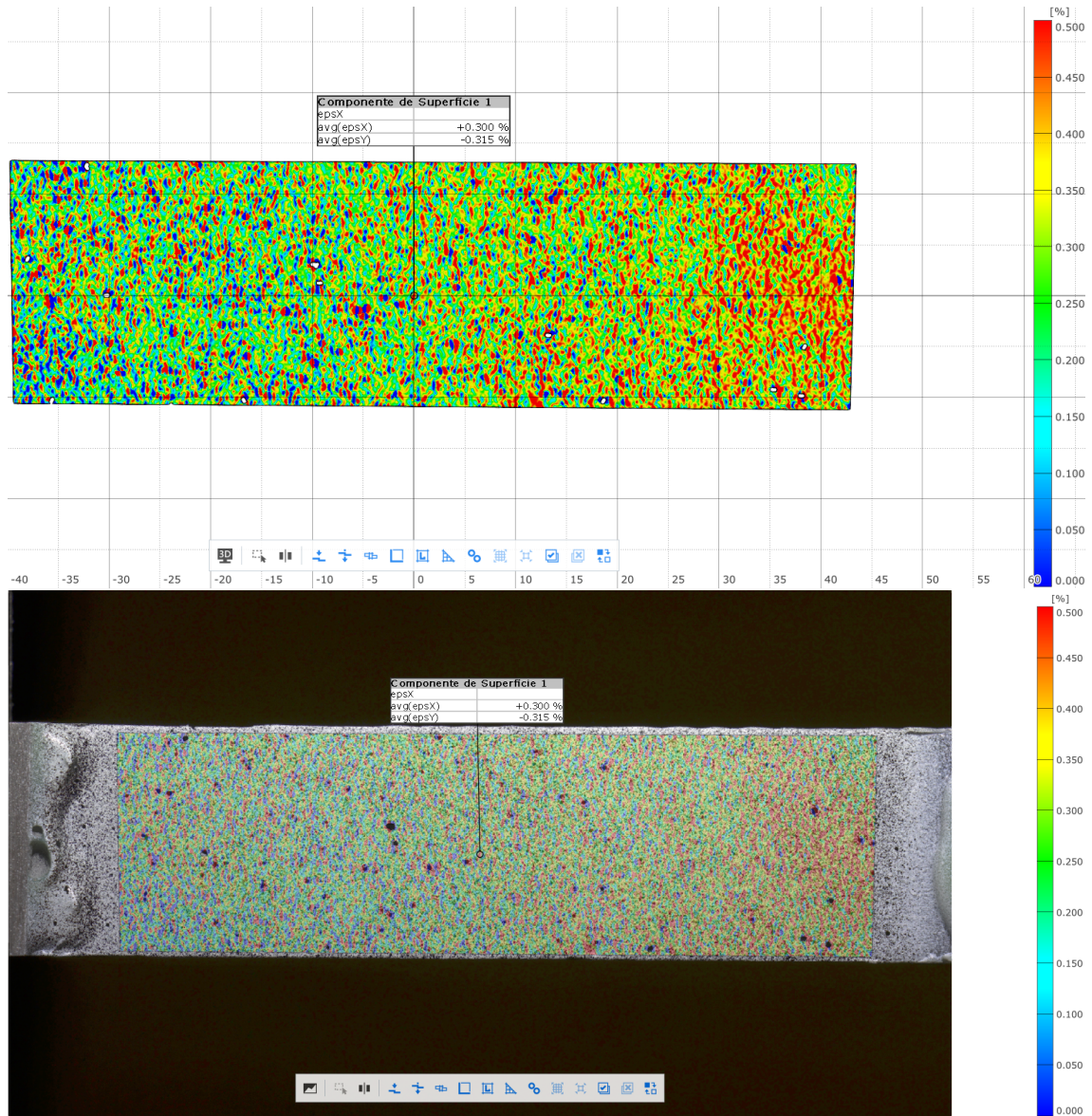


Figure 4.16: Shear test longitudinal strain field at 0.6% shear strain: without background (top) and with specimen background (bottom).

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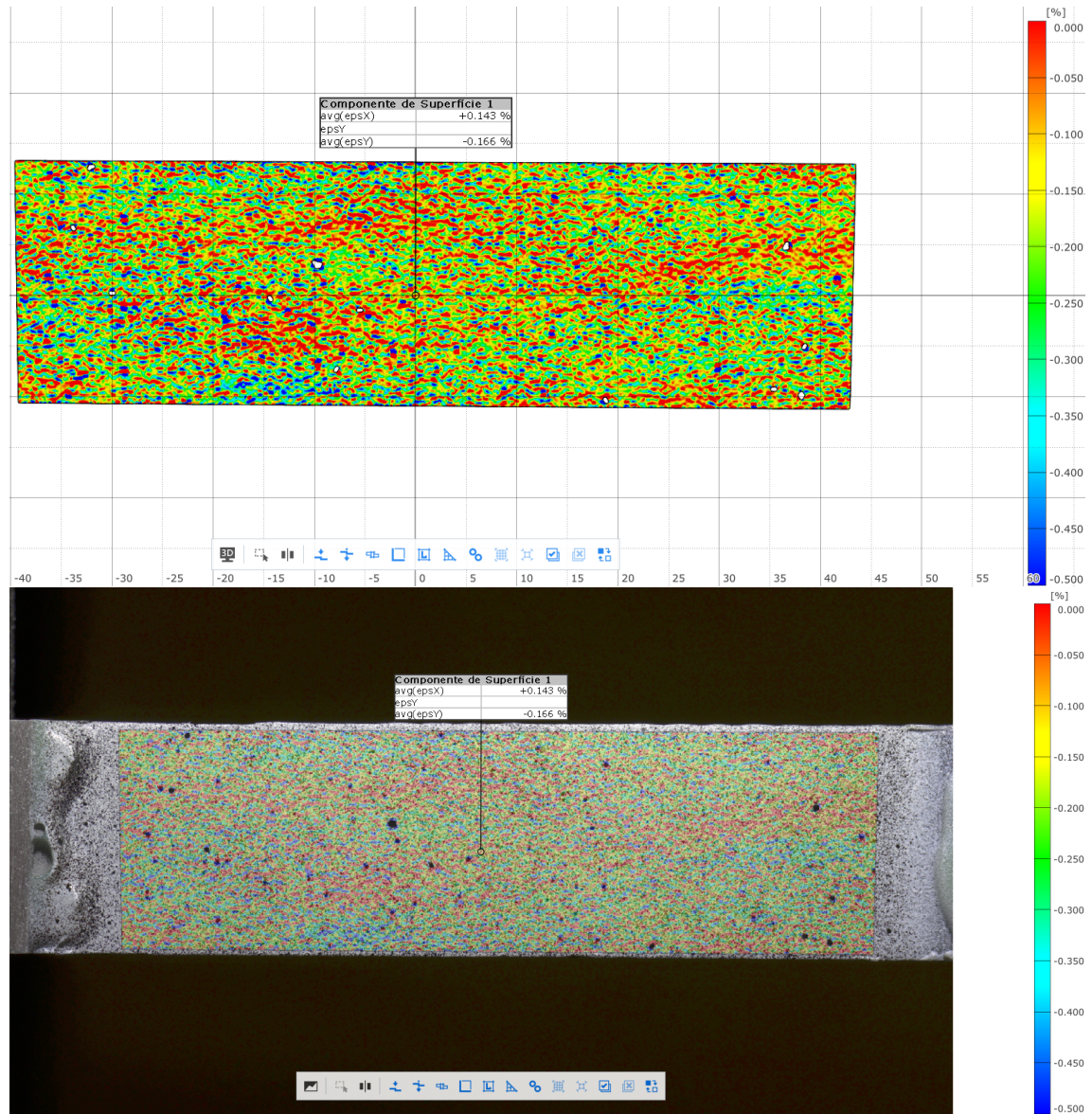


Figure 4.17: Shear test transverse strain field at 0.3% shear strain: without background (top) and with specimen background (bottom).

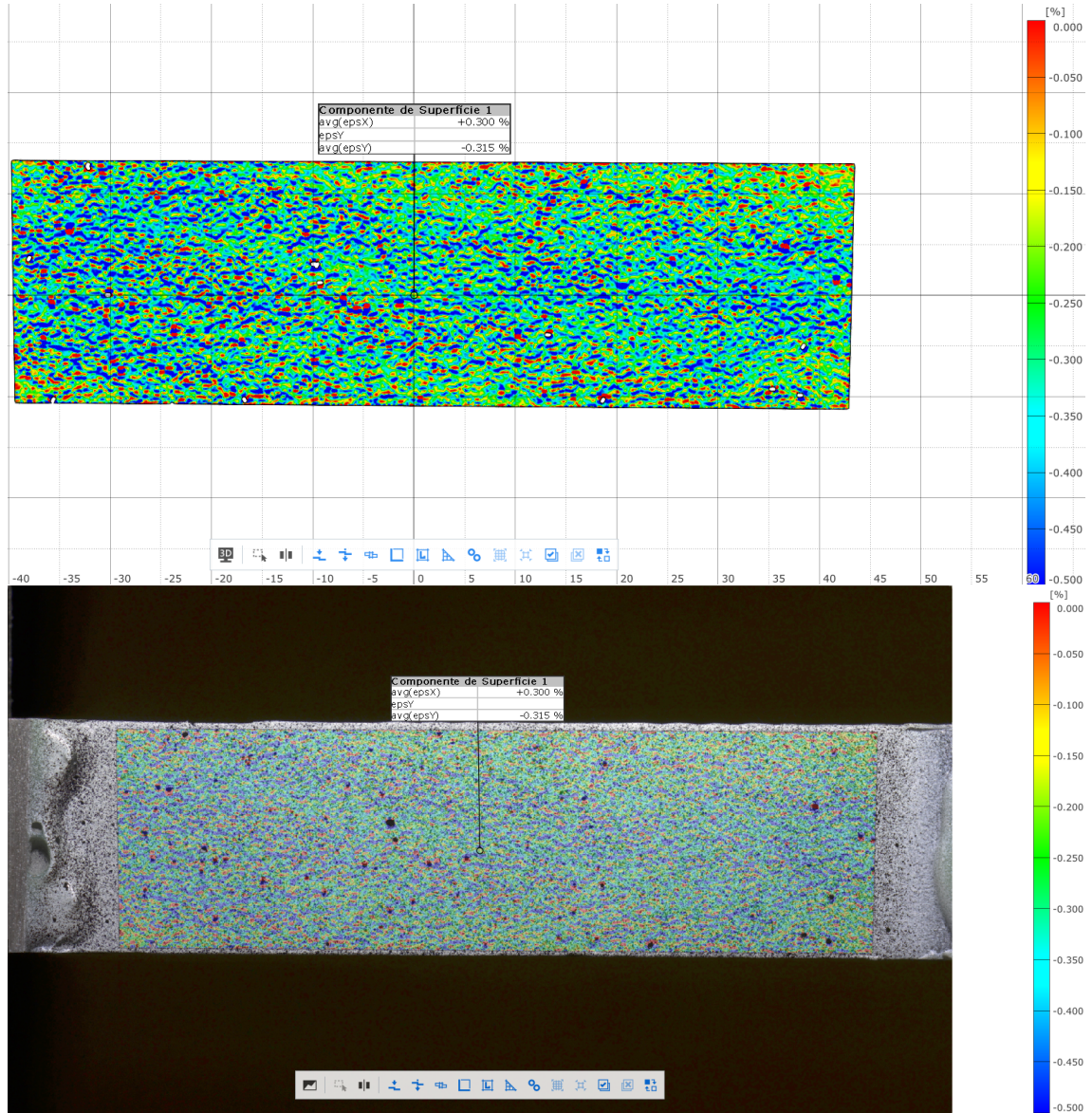


Figure 4.18: Shear test transverse strain field at 0.6% shear strain: without background (top) and with specimen background (bottom).

4.1. Material Selection and Characterisation

caution when interpreting multi-directional strain states. Ultimately, DIC was selected as the primary strain measurement technique in this work due to its consistency, full-field capability, and resilience in higher strain regimes, while recognising the uncertainty introduced.

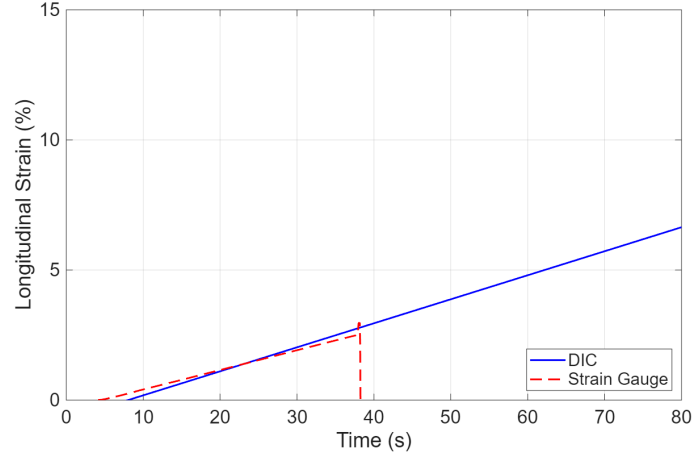


Figure 4.19: Comparison of longitudinal strain from DIC and strain gauge measurements (LTS6).

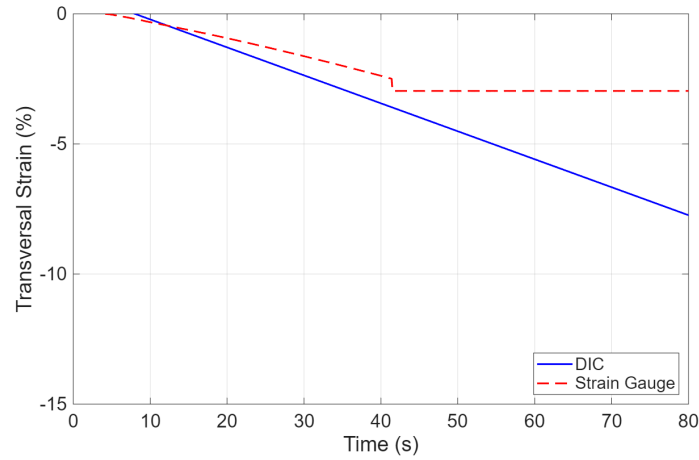


Figure 4.20: Comparison of transverse strain from DIC and strain gauge measurements (LTS6).

Shear stress-strain curves obtained from DIC data are presented in Figure 4.21. The curves display an initial linear elastic region followed by a progressive nonlinear response. This transition reflects the onset of plastic deformation, during which the fibres gradually align with the loading direction. Although this alignment enhances the load-carrying capacity of the laminate, often leading to a peak load higher than the initial yield, the associated stress is not used to define shear strength, as it does not correspond to the intrinsic shear resistance of the original laminate configuration.

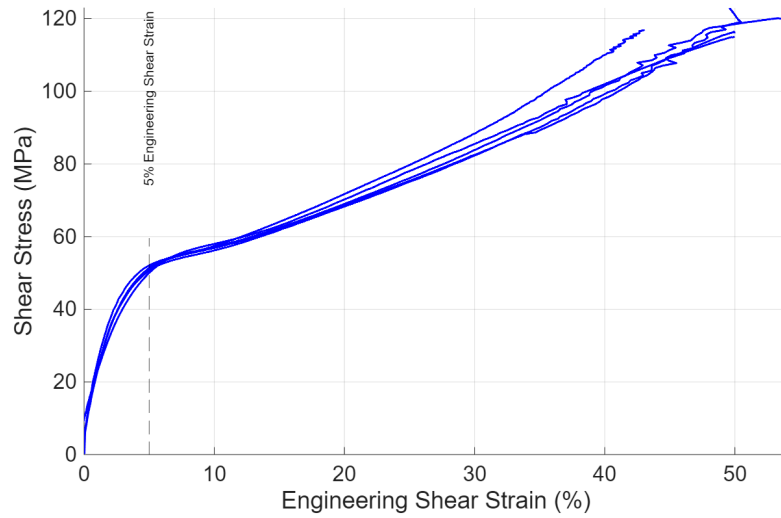


Figure 4.21: Shear stress-strain curves from $\pm 45^\circ$ tension tests.

The shear modulus was calculated using linear regression between 0.2% and 0.6% engineering shear strain, in accordance with ASTM D3518 recommendations. Shear strength was extracted as the stress value at 5% engineering shear strain. These results are displayed in boxplot form in Figures 4.22 and 4.23, and summarised in Table 4.3.

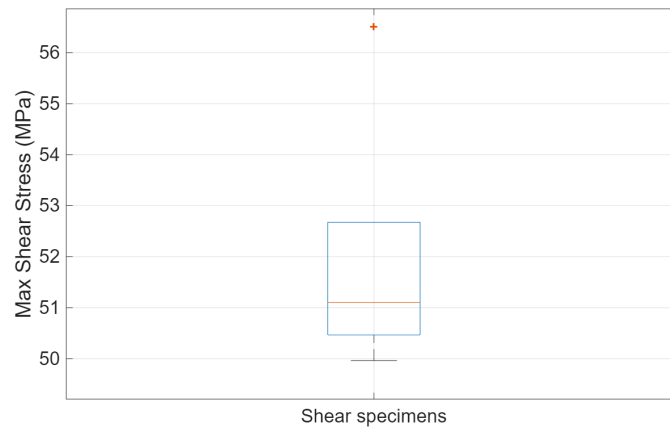


Figure 4.22: Boxplot of shear strength (shear stress at 5% engineering shear strain).

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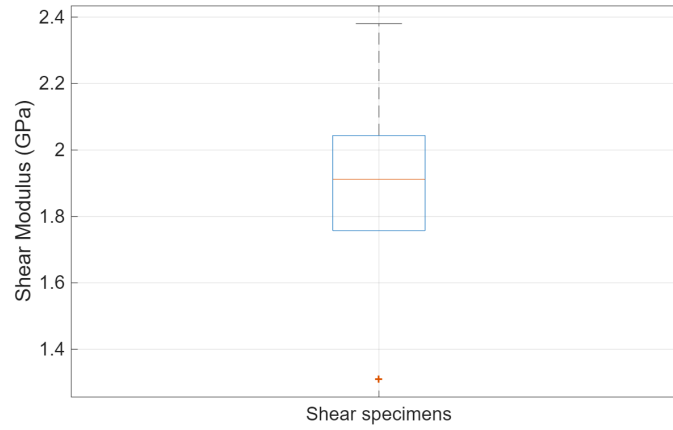


Figure 4.23: Boxplot of shear modulus calculated between 0.2% and 0.6% engineering shear strain.

Table 4.3: Summary of shear test results.

Parameter	Mean	Standard Deviation	CV (%)
Shear strength (MPa)	51.03	0.7856	1.540
Shear modulus (GPa)	1.850	0.3250	17.56

The results show highly consistent shear strength values across specimens, with a low coefficient of variation ($CV \approx 1.5\%$). In contrast, the shear modulus exhibited slightly higher variability ($CV \approx 17.6\%$), which may be attributed to specimen preparation or minor misalignments in fibre orientation. While this level of scatter is greater than that observed for tensile modulus, it remains acceptable for polymer-matrix composites, particularly in shear testing, where matrix-dominated behaviour and material non-linearity often lead to increased variability.

4.1.3.3 Compression Tests

The compressive strength of the laminate was determined for both 0° and 90° specimens using the maximum load recorded during testing. The results, summarised in Table 4.4 and illustrated in Figure 4.24, show a mean compressive strength of 352.1 MPa for the 0° specimens and 355.8 MPa for the 90° specimens, with coefficients of variation of 8.4% and 7.1%, respectively. This similarity in performance between fibre directions suggests a balanced behaviour under compressive loading, which is typical of woven reinforcements. However, this apparent symmetry under compressive loading contrasts with the tension test results, which revealed a significant directional imbalance.

Table 4.4: Summary of compression test results.

Orientation	Parameter	Mean	Standard Deviation	CV (%)
0°	Max stress (MPa)	352.1	29.46	8.366
90°	Max stress (MPa)	355.8	25.13	7.063

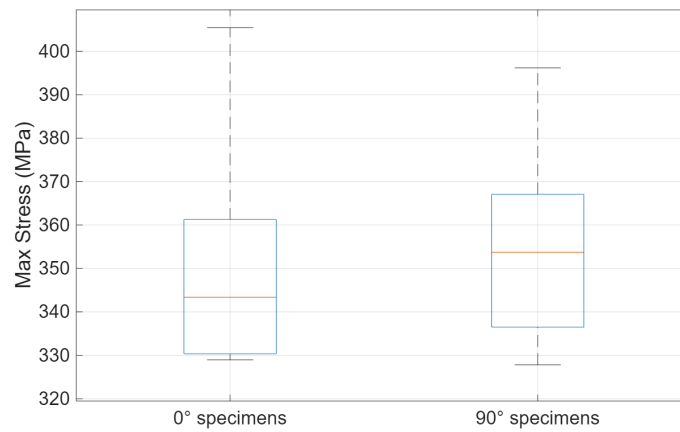


Figure 4.24: Boxplot of maximum compressive stress for each configuration.

Although the primary objective was to determine compressive strength, stress-strain curves were also extracted from machine displacement data to visualise the overall response (Figures 4.25 and 4.26). It is worth noting that the initial portion of the curves displays an irregular behaviour, which may be attributed to factors such as friction effects, slack or settling in the test fixture, or other mechanical irregularities prior to full engagement of the load.

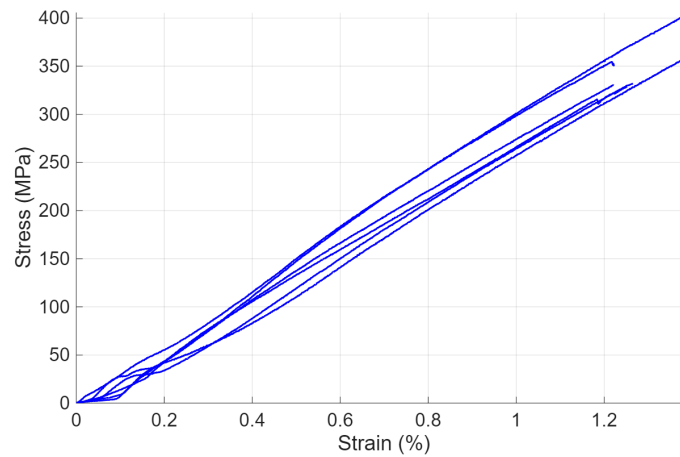


Figure 4.25: Stress-strain curves from longitudinal compression tests (0° specimens).

4.1. Material Selection and Characterisation

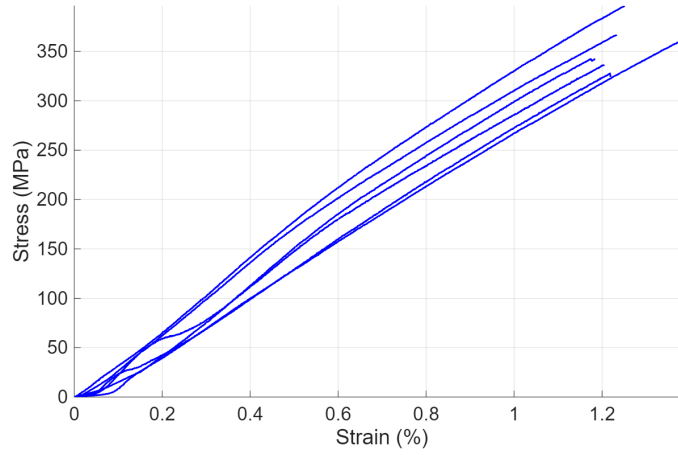


Figure 4.26: Stress-strain curves from transverse compression tests (90° specimens).

Additionally, one specimen per configuration (LC04 and LC92) was fitted with a strain gauge to assess the consistency of strain measurements. The comparison with machine displacement is shown in Figures 4.27 and 4.28, with root-mean-square (RMS) strain errors of 0.0696% for the 0° specimen (LC04) and 0.1754% for the 90° specimen (LC92). When normalised by the respective machine strain signals, these correspond to relative errors of 9.77% and 23.94%. While these values are relatively high, they may still be considered marginally acceptable given the limitations of using crosshead displacement as a reference for strain measurement. These comparisons are primarily useful for qualitative validation within the elastic regime. However, compressive modulus values were not extracted, as machine-based displacement readings are not sufficiently accurate for precise stiffness characterisation.

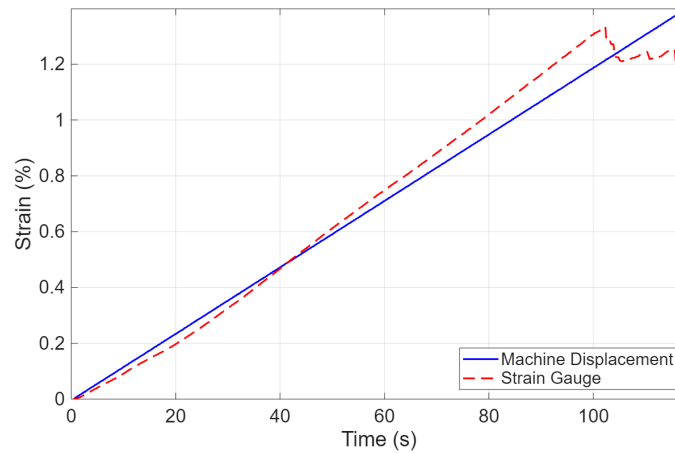


Figure 4.27: Strain-time comparison between machine displacement and strain gauge (LC04, 0° specimen).

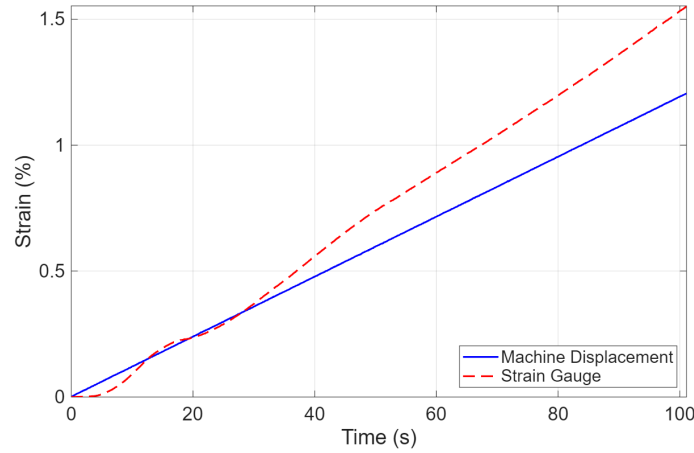


Figure 4.28: Strain-time comparison between machine displacement and strain gauge (LC92, 90° specimen).

The mechanical characterisation campaign provided a comprehensive set of stiffness and strength parameters for the CFRP laminate in all relevant loading directions. While some variability was observed, particularly in Poisson’s ratio and shear modulus, most results demonstrated good repeatability and coherence with expected material behaviour. For the purpose of subsequent analytical calculations and finite element simulations, material property values were selected based on the average results of each test configuration. These values are summarised in Table 4.5, and will serve as input data for the structural models presented in the following chapters.

Table 4.5: Selected mechanical properties of the CFRP material for analysis and simulation.

Property	Symbol	Value
Longitudinal tensile modulus	E_1	36.03 GPa
Transverse tensile modulus	E_2	27.47 GPa
Poisson’s ratio	ν_{12}	0.05725
	ν_{21}	0.04834
In-plane shear modulus	G_{12}	1.850 GPa
Longitudinal tensile strength	X_t	524.1 MPa
Transverse tensile strength	Y_t	360.3 MPa
Longitudinal compressive strength	X_c	352.1 MPa
Transverse compressive strength	Y_c	355.8 MPa
In-plane shear strength	S	51.03 MPa

4.2 Sandwich Panel Material Selection and Manufacture

The sandwich panels developed in this work consist of two carbon fibre-reinforced polymer (CFRP) facesheets bonded to an aluminium honeycomb core using a structural adhesive film. The selection of materials was guided by the need to produce lightweight yet mechanically robust structures, representative of high-performance automotive and aerospace applications.

The CFRP used in all facesheets is the same orthogonal spread tow prepreg fabric previously characterised in Section 4.1. It consists of a 0°/90° weave with an epoxy matrix, where each ply has an areal density of approximately 160 g/m² and a nominal thickness of 0.160 mm. The mechanical

4.2. Sandwich Panel Material Selection and Manufacture

characterisation of this material, performed through standardised tests, provided the stiffness and strength parameters used as input in the analytical and numerical models.

The core of the sandwich structure is a high-performance aluminium honeycomb manufactured from 5052-grade alloy. Supplied in unexpanded block form, the honeycomb was manually expanded to its final sheet size of $1250 \times 652 \text{ mm}^2$ using custom fixtures. It features a nominal cell size of $1/8''$ (3.2 mm), a thickness of 20 mm, and a nominal density of 76.9 kg/m^3 . This configuration provides an optimal balance of stiffness, weight, and compressive strength, and is commonly used in motorsport, aerospace, and structural panel applications. The aluminium foil has a thickness of $35 \mu\text{m}$ and is neither perforated nor corrosion treated. The full manufacturer's datasheet for the selected core, including its detailed mechanical properties, is provided in Appendix B.1 for reference.

The adhesive film used to bond the CFRP facesheets to the aluminium core is XPREG® XA120. This toughened epoxy-based prepreg adhesive is designed for high-performance composite sandwich construction, offering excellent bond and peel strength. XA120 is compatible with a range of core materials, including aluminium honeycomb, and can be cured in autoclave, oven, or heated platen press at temperatures ranging from 80°C to 120°C . The film's nominal resin weight is 150 gsm, and it incorporates a lightweight glass scrim for improved handling. The full manufacturer's datasheet for the selected adhesive is provided in Appendix B.2 for reference.

This combination of high-strength, lightweight materials was selected to replicate typical high-performance structural panel configurations, as used in Formula Student monocoque structures and similar applications.

A total of eight sandwich panels were manufactured: four for three-point bending tests and four for perimeter shear tests. Both panel configurations were defined in accordance with Formula Student rules: the three-point bending panels measured $500 \times 275 \text{ mm}^2$, while the perimeter shear panels followed the required $100 \times 100 \text{ mm}^2$ specification.

The chosen laminate configuration for the CFRP facesheets was a cross-ply layup of $[0, 90]_s$. This stacking sequence was selected for both practicality and material efficiency. Using only 0° and 90° plies avoided the need to cut 45° prepreg strips, a process that would have increased material waste and manual labour. Given the objectives of this study, the cross-ply layup was deemed sufficient to assess the validity of the analytical and numerical sizing tools developed.

All required prepreg layers were cut to size and orientation, and the layups for the eight facesheets were completed manually. Each facesheet was cured individually in the autoclave using the same two-stage cure cycle adopted for the material characterisation campaign: an initial vacuum-only stage at 80°C , followed by a second stage at 130°C and 6 bar for approximately one hour. Due to unavailability of flat caul plates in the required dimensions, the curing was performed without them. This introduced minor surface waviness in the cured laminates, although visual inspection did not reveal significant defects. The autoclave layout used during the first curing stage, showing the arrangement of the individual CFRP facesheets for both three-point bending and perimeter shear panels, is shown in Figure 4.29. The corresponding vacuum bag configuration prior to autoclave processing is presented in Figure 4.30. In Figure 4.31, the difference between the smooth and wavy surfaces of the cured CFRP facesheets can be clearly observed side by side.

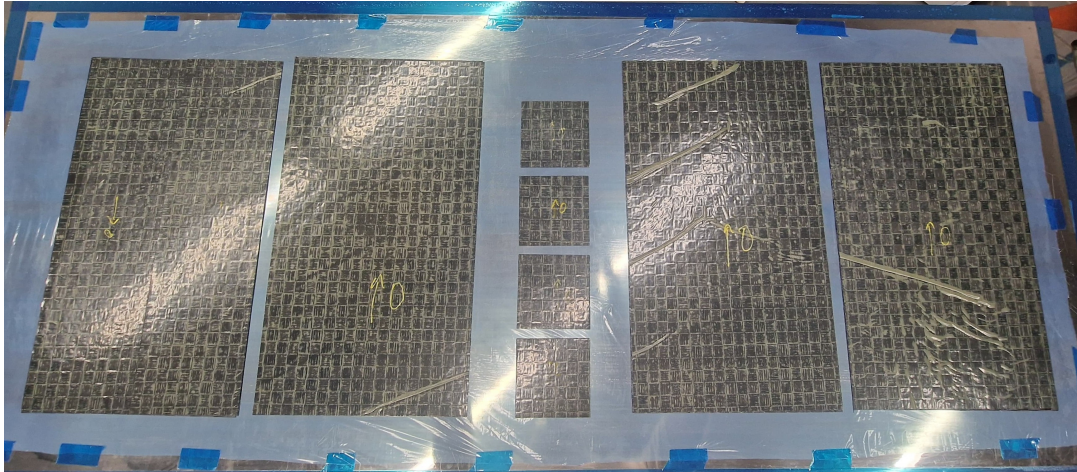


Figure 4.29: Autoclave layout of the CFRP facesheets during the first cure.

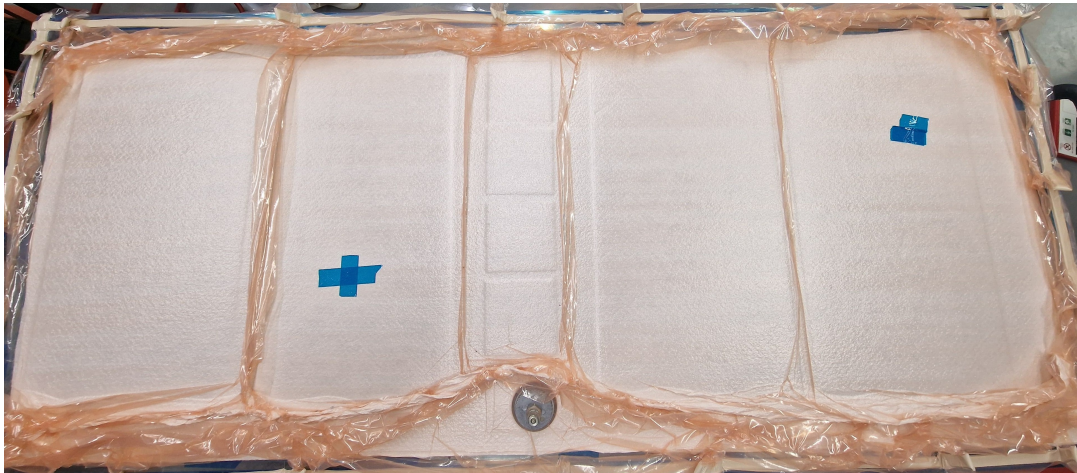
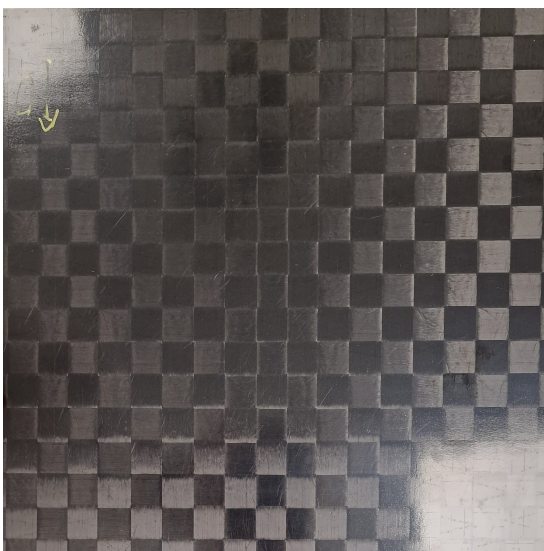
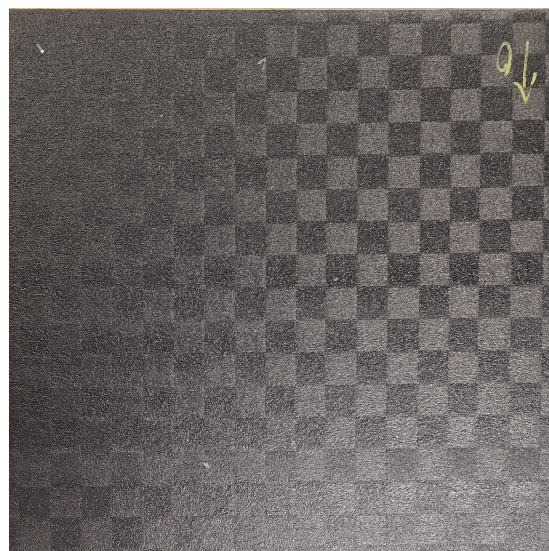


Figure 4.30: Vacuum bag configuration used for the autoclave curing of the CFRP facesheets.



(a) Smooth surface of CFRP facesheet.



(b) Wavy surface of CFRP facesheet.

Figure 4.31: Comparison between the smooth and wavy surfaces of the cured CFRP facesheets.

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Following the first cure, the aluminium honeycomb core was manually expanded and cut to the final panel dimensions. The XPREG® XA120 adhesive film was also trimmed to size. For each sandwich panel, the two cured CFRP facesheets were aligned on either side of the honeycomb core with adhesive film placed in between. Notably, the slightly wavy side of each facesheet, resulting from the absence of a caul plate during curing, was oriented towards the core to ensure better contact with the adhesive and to promote improved surface conformity with the test fixture. The assemblies were then stacked inside a single vacuum bag and sealed. This method was chosen to reduce the risk of vacuum leakage, as taping a bag directly to the autoclave table posed a higher likelihood of rupture due to the panel size and shape.

The final bonding and core cure was carried out in the autoclave using a single-stage cycle at 120°C and 1 bar for one hour. The entire stack of panels was cured simultaneously under vacuum, ensuring adequate pressure and thermal exposure throughout the bonding interface. Once removed from the autoclave, the panels were visually inspected and trimmed as required before proceeding to mechanical testing. However, a noticeable warping of the panels was observed after the final cure, likely resulting from the chosen vacuum bagging method, which did not ensure direct contact between the panels and the rectified autoclave table. For future manufacturing cycles, an alternative vacuum bagging strategy that secures panel contact with a rigid flat surface should be implemented, and a solution must be found to mitigate the associated risk of vacuum leakage or bag rupture.

4.3 Three Point Bending

The objective of these tests was to validate the analytical predictions of flexural stiffness, ultimate strength and energy absorption of the sandwich panels under three-point bending. The combination of these insights supports the validation of the developed design tools, confirming their ability to accurately predict panel performance in accordance with Formula Student requirements.

The following subsections present the predictions obtained using the analytical sizing methodology and the data extracted from mechanical testing. A final discussion integrates these results to assess the accuracy and applicability of the proposed methods.

4.3.1 Prediction Using Developed Tools

The analytical tool developed in Chapter 3 was applied to predict the flexural stiffness, strength, and energy absorption of the sandwich panels tested under three-point bending. The test specimens measured 500 × 275 mm and featured a cross-ply layup of $[0, 90]_s$ in both facesheets. The calculations were based on the mechanical properties obtained from the CFRP characterisation campaign and the geometric configuration of the test specimens. Strength predictions were carried out using four different failure criteria: Maximum Stress, Tsai-Hill, Tsai-Wu, and Hashin. Due to the simplicity of the cross-ply layup employed in the sandwich panels for experimental validation, interlaminar interactions are minimal, resulting in structural behaviour closely resembling that of a unidirectional CFRP specimen. Consequently, all failure criteria yielded identical predictions. The results are summarised in Table 4.6. These predictions provide a baseline for comparison with the experimental results presented in Section 4.3.2.

Table 4.6: Predicted properties for three-point bending specimens.

EI_{panel} (kN·mm ²)	F_{max} (kN)	U (J)
1927	17.40	104.75

4.3.2 Test Results

These tests were designed to validate the analytical stiffness and strength predictions of the sandwich panels under three-point bending loading. All tests were conducted using the same MTS 810 Material Test System employed in the previous CFRP characterisation tests, equipped with a 100 kN load cell and a displacement transducer calibrated for a ± 100 mm range.

A custom steel fixture was manufactured specifically for this work, taking into account the geometric requirements mandated by the Formula Student rules as well as the mounting constraints of the testing machine. Its design was inspired by established three-point bending fixtures, with particular attention given to robustness, straightforward specimen alignment, and compatibility with the MTS 810 system to allow easy installation and repeatable use. Basic sizing checks were performed to verify that fixture deflections would remain negligible in comparison to the expected panel deflections, ensuring that the measured responses accurately reflected the structural behaviour of the panels. In Figure 4.32, the CAD model of the fixture during development is shown alongside a photograph of the manufactured rig mounted on the testing machine. Figure 4.33 then presents the fixture in operation with a sandwich panel positioned for testing.

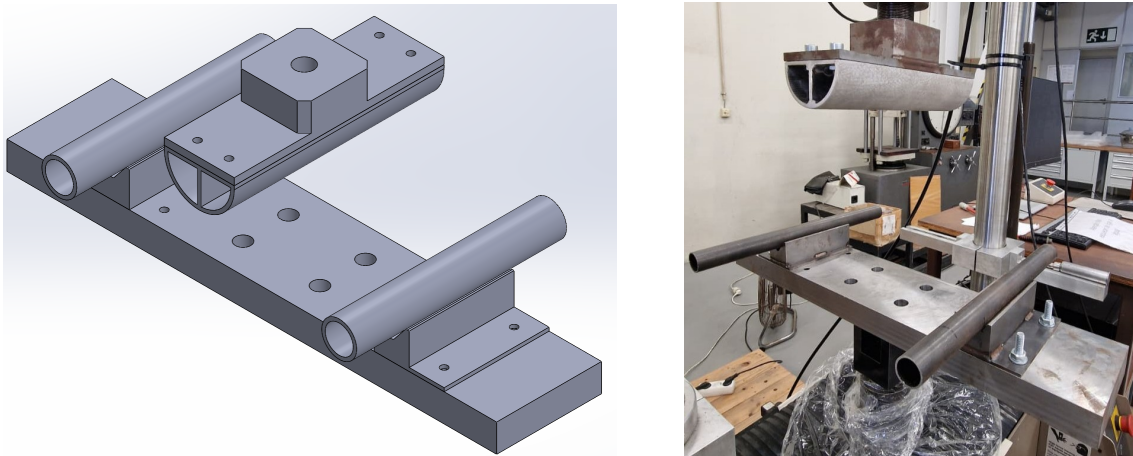


Figure 4.32: CAD model of the three-point bending fixture during development (left) and photograph of the manufactured rig mounted on the MTS 810 testing machine (right).

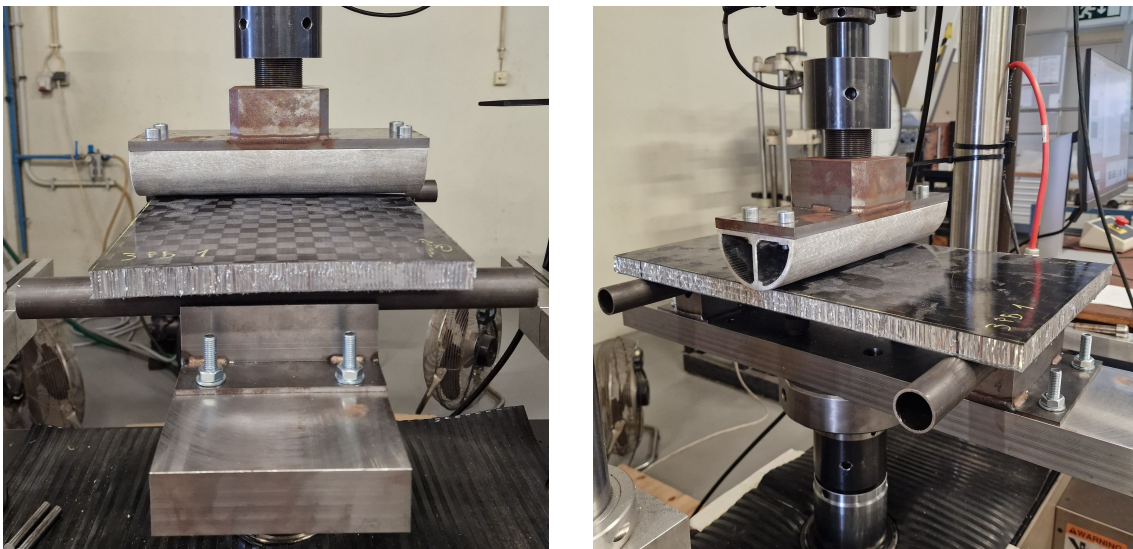


Figure 4.33: Three-point bending test fixture designed according to Formula Student specifications.

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Panel loading was performed at a crosshead displacement rate of 1 mm/min, with force and displacement signals acquired at a sampling frequency of 10 Hz. Mid-span deflection could not be measured directly because, in the testing machine employed, the punch was fixed while the supports were displaced. Consequently, the use of analogue displacement comparators at the panel ends was considered as an alternative. However, this approach was deemed unreliable due to several factors: the visible warpage of the panels, which caused misalignment of the comparators; potential local edge effects that did not accurately reflect the global deflection; and the limited resolution of the devices relative to the small displacements involved. For these reasons, comparator-based measurements were ultimately discarded. Panel warpage can be observed in Figure 4.34, as evidenced by the uneven contact between the central loading punch and the panel surface.

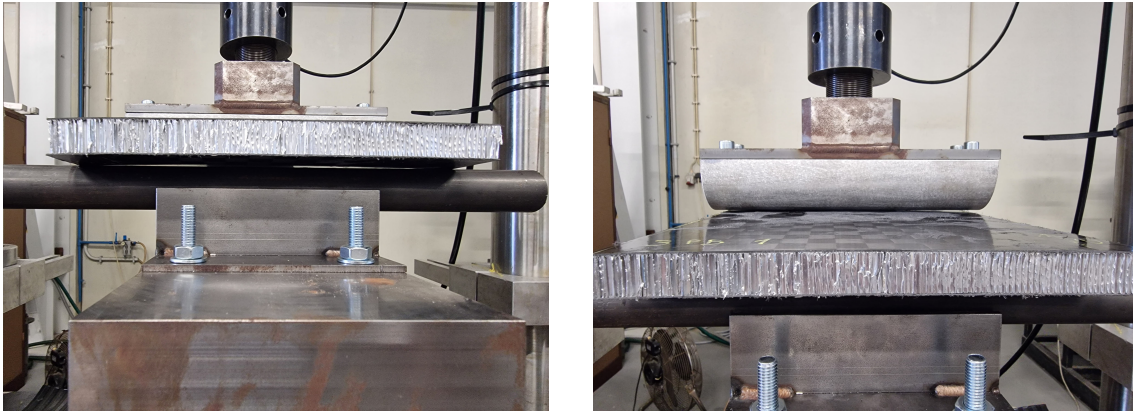


Figure 4.34: Out-of-plane warpage observed in sandwich panels after autoclave curing.

A total of four specimens were tested. Three were loaded directly using the steel punch in accordance with Formula Student rules, while one specimen was tested with a rubber layer (60 HV) inserted between the panel and the punch. This deviation, although not compliant with official regulations, was introduced to promote improved contact, more uniform load distribution, and to mitigate the effects of panel warpage. It was considered a valuable comparative case for the purposes of this work.

The load-displacement curves for all tested specimens are presented in Figure 4.35. The specimen tested with a rubber interface between the panel and the punch is plotted in red, while the remaining specimens (without rubber) are plotted in blue. All curves exhibit a linear region prior to non-linear softening and failure.

The test performed with the rubber interface displayed lower stiffness, as expected due to the local compliance introduced at the punch-panel contact. However, this setup also showed a smoother force response and a higher maximum force capacity, likely due to the better load distribution across the warped panel surface. In contrast, the specimens without rubber presented higher stiffness values and more abrupt failure, with lower maximum force capacity.

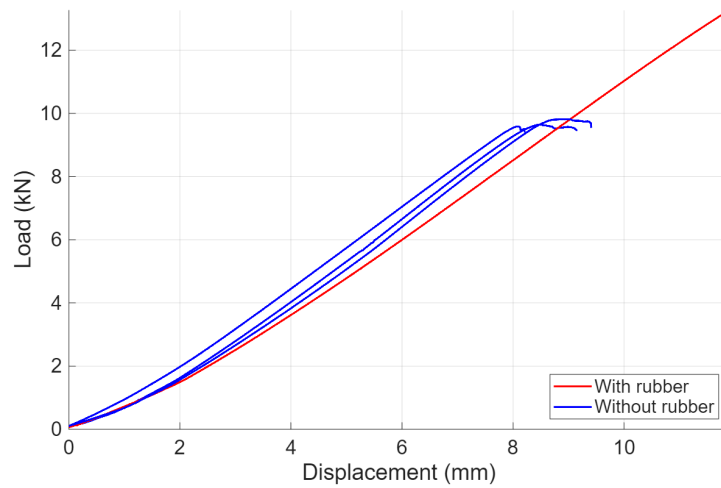


Figure 4.35: Experimental load-displacement curves for three-point bending tests.

The flexural rigidity (EI) of each panel was estimated from the slope of the force-displacement response within the identified linear region using classical beam theory. In addition to stiffness, the total energy absorption during loading was also computed by numerically integrating the area under the force-displacement curve up to the failure point. These results are summarised in Tables 4.7, 4.8 and 4.9.

Table 4.7: Maximum load statistical results.

Rubber Interface	Max Load (kN)		
	Mean	Standard Deviation	CV (%)
Without Rubber	9.685	0.121	1.253
With Rubber (single specimen)	13.259	-	-

Table 4.8: Flexural rigidity statistical results.

Rubber Interface	Flexural Rigidity ($\text{kN}\cdot\text{mm}^2$)		
	Mean	Standard Deviation	CV (%)
Without Rubber	1728	16.648	0.963
With Rubber (single specimen)	1587	-	-

Table 4.9: Energy absorption statistical results.

Rubber Interface	Energy Absorption (J)		
	Mean	Standard Deviation	CV (%)
Without Rubber	39.249	2.383	6.070
With Rubber (single specimen)	73.139	-	-

It is evident that the rubber interface had a complex effect on the test outcome: although it

4.3. Three Point Bending

provided a higher peak load and absorbed energy, the flexural rigidity estimate was lower compared to specimens tested without rubber. In contrast, the specimens without rubber showed highly consistent results across all tested metrics, reinforcing the reliability and repeatability of the test procedure.

4.3.3 Comparison and Discussion

The results obtained from experimental testing can now be compared with the analytical predictions. The comparison allows an assessment of the capability of the developed tool to predict the behaviour of sandwich panels under three-point bending.

Table 4.10: Comparison between analytical predictions and experimental results for three-point bending tests.

Property	Prediction	Experimental Mean	Deviation (%)
Flexural Rigidity EI (kN·mm ²)	1927	1728	+11.5
Maximum Load (kN)	17.40	9.69	+79.6
Energy Absorption (J)	104.75	39.25	+166.9

In terms of flexural rigidity, the analytical model predicted a value of 1927 kN · mm², while the mean experimental result was 1728 kN · mm². This corresponds to a deviation of approximately 11.5%, which can be considered a close agreement given the simplifications inherent in the modelling approach. The result demonstrates that the stiffness prediction is robust, with the small discrepancy likely attributable to panel warpage, minor imperfections in specimen geometry, and the assumption of perfectly bonded interfaces in the analytical formulation.

For maximum load, however, a larger divergence was observed. The analytical prediction of 17.40 kN considerably exceeded the experimental mean of 9.7 kN. This overestimation reflects the sensitivity of strength predictions to local imperfections and loading conditions. The analytical tool assumes idealised conditions with uniform loading and no defects, whereas in practice panel warpage, stress concentrations from the rigid punch, and material variability reduce the effective load capacity. These results highlight the limitation of using simplified failure criteria for strength estimation in sandwich panels, particularly when imperfections dominate the response. Moreover, since cross-ply specimens were employed in the validation campaign, the structural response closely resembled that of a unidirectional laminate, leading all failure criteria (Maximum Stress, Tsai-Hill, Tsai-Wu, and Hashin) to converge to identical predictions. This prevented a meaningful comparison of the criteria and represents a significant oversight in the experimental design.

The energy absorption values further illustrate this discrepancy. While the analytical approach predicted 104.8 J, the average experimental result was significantly lower at 39.2 J. This difference stems directly from the overestimated maximum load, since absorbed energy is closely linked to both peak force and the deformation capacity prior to failure.

The test performed with a rubber interface between the panel and the punch provided additional insight. This specimen exhibited a higher maximum load (13.3kN) and absorbed energy (73.1J) than the rigid-punch specimens, while presenting a lower flexural rigidity (1587 kN · mm²). The improved strength and energy absorption were achieved through a more uniform load distribution across the warped panel surface, reducing the severity of stress concentrations. However, the reduced stiffness illustrates the compliance introduced by the rubber interface. Although this setup is not compliant with Formula Student regulations, it demonstrates the sensitivity of the

results to loading conditions and validates the regulatory requirement for rigid punches, which represent a more conservative loading scenario.

Overall, the validation demonstrates that the developed analytical tool is effective in predicting flexural rigidity, providing a reliable basis for preliminary design and sizing of sandwich panels. In contrast, its predictions of strength and energy absorption consistently overestimate the experimental values, as they do not account for imperfections, contact effects, and other practical influences. However, since all parameters were found to be overestimated, the application of the tool in subsequent sections will incorporate safety factors to ensure a conservative approach to panel sizing. The specific values of the adopted safety factors are presented in Table 4.11.

Table 4.11: Safety factors applied to analytical predictions for panel sizing.

Property	Safety Factor (SF)
Flexural Rigidity	1.5
Maximum Load	2.0
Energy Absorption	3.0

4.4 Perimeter Shear

The primary goal of this section is to verify the ability of the developed analysis methods to accurately predict the shear strength of the composite sandwich panels. By comparing analytical calculations and experimental results, the robustness of the panel design and the validity of the sizing methodology can be critically evaluated.

4.4.1 Prediction Using Developed Tools

Unfortunately, the material characterisation campaign presented in Section 4.1 did not include interlaminar shear strength (ILSS) testing, which is the property required by the analytical tool to estimate perimeter shear strength. As a result, a proper validation of the prediction methodology cannot be carried out. Nonetheless, in order to provide at least a qualitative comparison between calculated and measured values, an approximate ILSS value was adopted to enable the use of the tool. This approach does not provide a rigorous validation, but it allows for an assessment of whether the tool can produce predictions of the correct order of magnitude.

The ILSS value was estimated from literature, where a similar CFRP system exhibited an interlaminar shear strength of approximately 50 MPa [41]. This reference value was therefore adopted to provide an estimate for use in the analytical calculations.

Using this input, the analytical tool predicted a perimeter shear max load of 3.927 kN for the test configuration employed in this work. It is important to note that this value corresponds to the effective load capacity along the punch perimeter of the test fixture, rather than the full perimeter of the sandwich panel. In contrast, the tool is normally applied to panel sizing calculations based on the specimen perimeter. For this reason, the predicted result should be understood as a test-specific value to be compared directly with the experimental measurements presented in Section 4.4.2, while the panel-level predictions reported elsewhere in this thesis remain based on the total specimen perimeter.

4.4. Perimeter Shear

4.4.2 Test Results

The perimeter shear tests were conducted to measure the failure load of the sandwich panels under perimeter shear loading, as defined by the Formula Student regulations. A total of four specimens were tested, each with a square geometry of $100 \times 100 \text{ mm}^2$.

All tests were carried out using an electro-mechanical Instron 5900R testing machine, equipped with a 200 kN load cell and a displacement measurement system calibrated over the required range. The crosshead speed was set to 1 mm/min, and force-displacement data were sampled at 10 Hz throughout each test.

A custom test fixture, previously developed by the team in earlier work, was employed to induce perimeter shear failure. It consisted of a rigid steel base plate with a 32 mm diameter central cut-out, on which the specimen was fully supported. A 25 mm diameter cylindrical punch was then used to apply a compressive load to the centre of the panel. This configuration created a stress concentration at the boundary of the hole, promoting shear failure around the perimeter of the loading region. Proper fixture alignment was ensured before each test.

The test setup and loading configuration are illustrated in Figure 4.36, showing the alignment of the punch and the specimen, as well as the contact between the core, facesheet, and fixture.

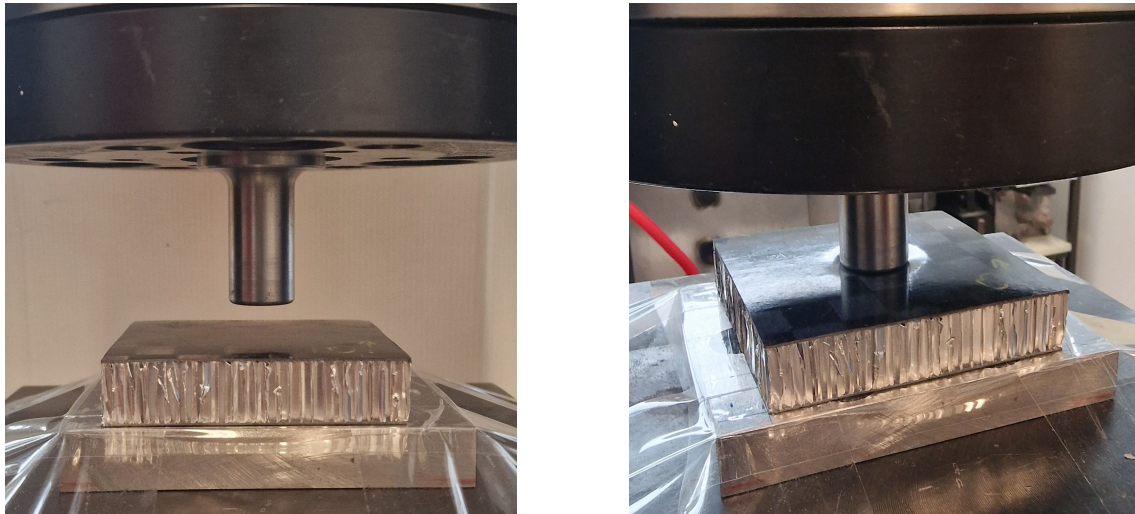


Figure 4.36: Photographs of the perimeter shear test setup and loading configuration.

The load-displacement curves from all four tests are presented in Figure 4.37. All specimens displayed a comparable response, with an initial linear region up to the first peak load, followed by a partial load drop and a subsequent second peak at higher displacement. This behaviour suggests the occurrence of at least two stages of failure, which may correspond to an initial localised damage mechanism followed by a more global failure process. No abrupt load drops or excessive scatter were observed, indicating good repeatability, proper specimen alignment, and stable test conditions.

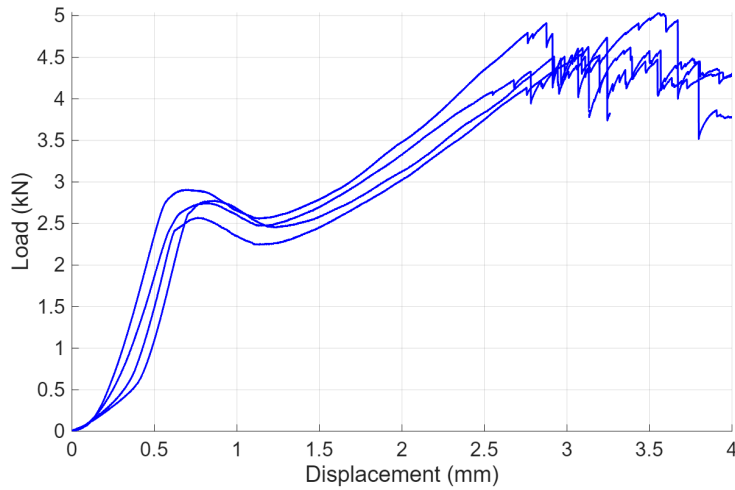


Figure 4.37: Experimental load-displacement curves for perimeter shear tests.

Table 4.12 summarises the statistical indicators for the two characteristic load peaks observed in the perimeter shear tests. For each peak, the mean value, standard deviation, and coefficient of variation (CV) are reported. The results demonstrate good consistency, with CV values below 5% in both cases, which supports the repeatability and reliability of the test method.

Table 4.12: Statistical indicators for the two load peaks recorded in perimeter shear testing.

Measured Load	Mean (kN)	Standard Deviation (kN)	CV (%)
First Peak	2.746	0.137	5.000
Max Load	4.733	0.221	4.659

4.4.3 Comparison and Discussion

The analytical tool, when supplied with the estimated ILSS value, predicted a perimeter shear strength of 3.927 kN for the punch-perimeter configuration of the test fixture. The experimental results, summarised in Table 4.13, revealed two characteristic peaks with mean values of 2.746 kN and 4.733 kN. Although it is not possible to state with certainty which of these peaks the analytical tool is estimating, the predicted value lies closer to the second (maximum) peak, deviating by approximately -17%. While this does not constitute a formal validation due to the absence of directly measured ILSS data, the results demonstrate that the tool provides predictions of the correct order of magnitude. This suggests that the methodology may serve as a useful guideline for preliminary design, albeit with caution.

Given these limitations, the validation of perimeter shear strength cannot be considered rigorous within the present work. To mitigate this uncertainty, the analytical predictions employed in subsequent panel sizing will incorporate a safety factor of 2 for perimeter shear strength. This adjustment ensures a conservative design approach that compensates for the current lack of experimentally determined ILSS data and the associated limitations in predictive accuracy.

Table 4.13: Comparison between analytical prediction and experimental values for perimeter shear testing.

Measured Load	Prediction (kN)	Experimental Mean (kN)	Deviation (%)
First Peak	3.927	2.746	+43.008
Max Load		4.733	-17.029

Chapter 5

Monocoque Design

This chapter presents the development of a monocoque chassis conceived as a proof of concept to guide future full-scale implementation within the Formula Student framework. While the final chassis will be subject to distinct and more stringent constraints, the objective here is to establish a coherent design approach that can serve as a foundation for subsequent iterations. The chapter begins by outlining the design constraints considered for this conceptual study, including simplified assumptions that balance feasibility with educational value. It then describes the overall shape and geometry of the chassis, followed by the material selection process aimed at achieving an optimal balance between strength, stiffness, and weight. The dimensioning of the composite panels in the different sections of the monocoque is supported by the engineering tools and methodologies developed in the preceding chapters of this thesis. Lastly, a finite element method (FEM) structural analysis is performed to assess the global performance of the chassis, with a particular focus on torsional rigidity.

5.1 Design Constraints

The design of a monocoque chassis for a Formula Student car is governed by two main types of constraints: those imposed by the competition rules and those defined internally by the team. The official rules primarily aim to ensure driver safety and compliance with standardised structural requirements. In contrast, the team-specific constraints focus on optimising performance, manufacturability, and integration with other vehicle subsystems. Both types of restrictions are outlined in this section, as they form the basis for the design choices made throughout the development of the chassis.

5.1.1 Team Constraints

To ensure a realistic and relevant design, the constraints of the FSFEUP 02 vehicle were adopted as the reference framework for this study. This decision enables a direct comparison between the proposed monocoque chassis concept and the team's existing tubular steel frame, allowing the advantages and limitations of each structural approach to be more clearly assessed. By grounding the design in a known and validated configuration, this study also provides the team with a structured methodology and practical insights that can inform future monocoque development. The key team-defined geometric constraints considered in this study include:

- **Suspension Pickup Points:** The location of the suspension pickup points was fixed early

5.1. Design Constraints

in the design process based on the suspension geometry optimisation conducted for FSFEUP 02. These points dictate the positioning of the hardpoints on the monocoque and must be maintained within strict tolerances to ensure proper suspension function and vehicle dynamics performance.

- **Main and Front Hoop Positioning and Shape:** The positioning and shape of the main and front roll hoops were maintained from the FSFEUP 02 layout to provide a consistent reference for comparison between the existing tubular steel frame and the proposed monocoque chassis. This allowed the study to isolate the impact of switching to a composite monocoque without introducing additional variables related to cockpit geometry or rollover protection design.
- **High Voltage System:** The monocoque design had to accommodate the internal packaging of both the FSFEUP 02 accumulator and the inboard transmission system. Together, these components defined the volume and geometry requirements of the central and rear sections of the chassis. In typical monocoque configurations, the accumulator is inserted through the rear bulkhead; however, this was not feasible due to interference with the inboard transmission layout. An alternative approach, used by some teams, involves insertion from below and closing the opening with a CFRP panel, but this can compromise stiffness and complicate structural optimisation. To address these challenges, insertion through the cockpit opening was selected. This decision influenced the overall dimensions of the cockpit and the design of the surrounding structure to ensure accessibility and integration while preserving structural integrity.
- **Half Shaft Clearance:** The geometry of the side and rear monocoque regions was designed to allow adequate clearance for the rotating half shafts. This included ensuring sufficient space to accommodate full suspension travel and articulation without interference, while also preventing excessive cut-outs that could weaken structural continuity.

Together, these team-defined geometric constraints formed the foundation upon which the overall shape of the monocoque was developed. The key components that define these constraints are highlighted in Figure 5.1.

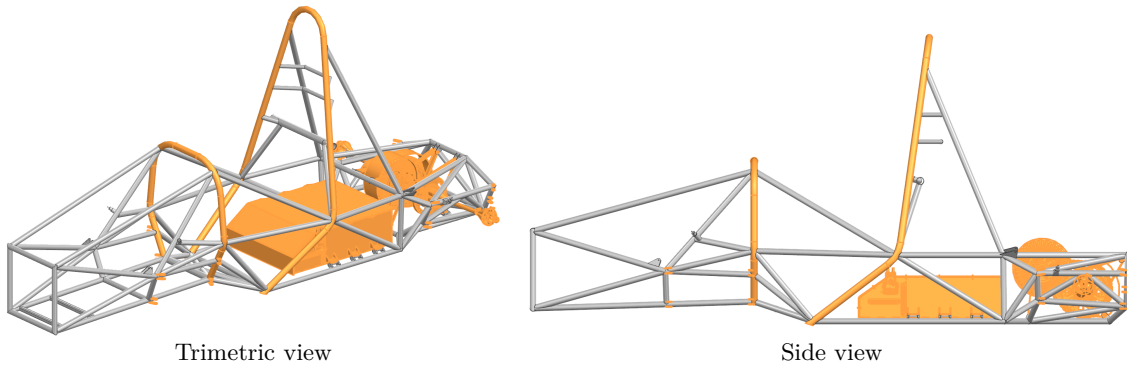


Figure 5.1: FSFEUP 02 chassis geometry including accumulator and transmission subsystems. Key components defining geometric constraints are highlighted.

While the FSFEUP 02 geometry provides a valuable reference for defining key hardpoints and packaging requirements, it was originally conceived for a tubular spaceframe chassis. As such, its integration into a monocoque concept introduces certain challenges that would not typically arise in a ground-up design. Notably, the overall shape features abrupt changes in cross-sectional area, particularly in the regions near the suspension hardpoints and cockpit. These transitions

complicate the development of a clean and continuous monocoque surface, requiring adaptations to ensure compatibility with the inherited geometry while maintaining adequate manufacturability. As a result, future iterations of the design would benefit from a revised baseline geometry, optimised specifically for composite construction.

5.1.2 Rules

The development of the monocoque chassis was carried out in alignment with the Formula Student Rules 2025 – Formula Student Germany (FSG) [40]. This ruleset establishes comprehensive safety and technical requirements that govern all vehicle systems, with particular attention to the structural design of the chassis and driver cell. Among the various sections, those related to chassis configuration, composite materials, crash safety, and driver ergonomics were especially relevant for this conceptual study.

Although this project does not include the full structural validation and testing required in a competition-ready vehicle, the monocoque was designed with the intent to satisfy key regulatory constraints. This includes compliance with cockpit template requirements, roll hoop positioning and sizing, and accumulator integration. The design also respects the Structural Equivalency Spreadsheet (SES) applicable to carbon fibre monocoques.

The following rules specifically constrain the external geometry of the monocoque and cockpit:

- T 2.1.2 The vehicle must be open-wheeled, single seat and open cockpit (a formula style body) with four wheels that are not in a straight line.
- T 2.3.1 There must be no openings through the bodywork into the cockpit other than that required for the cockpit opening. Minimal openings around the front suspension and steering system components are allowed.
- T 2.3.2 In any side view in front of the cockpit opening and outside the area defined in T8.2 all parts of the bodywork must have no external concave radii of curvatures. Any gaps between bodywork and other parts must be reduced to a minimum.
- T 2.3.3 Enclosed chassis structures and structures between the chassis and the ground must have two venting holes of at least 25 mm diameter in the lowest part of the structure to prevent accumulation of liquids. Additional holes are required when multiple local lowest parts exist in the structure.
- T 2.3.4 The bodywork in front of the front wheels with the tangent of surface above 45° relative to the forward direction must have a radius of at least 38 mm along the top, sides and bottom of all affected edges.
- T 2.4.1 For all edges that could come into contact with any standing pedestrian without reaching to the vehicle, the minimum radius of the bodywork and aerodynamic devices is 3 mm for all forward facing edges and 1 mm for all other edges.
- T 4.1.1 The size of the cockpit opening needs to be sufficient for the template shown on the left of Figure 5.2 to pass vertically from the cockpit opening to below the upper side impact member when held horizontally. The template may be moved fore and aft.
- T 4.1.2 If the side impact structure is not made of tubes, the template must pass until it is 320 mm above the lowest inside chassis point between the front and main hoop.

5.2. Modelling Strategy

- T 4.1.3 The steering wheel, seat and all padding may be removed for the template to fit. Any other parts may only be removed if they are integrated with the steering wheel.
- T 4.2.1 The cockpit must provide a free internal cross section sufficient for the template shown on the right of Figure 5.2 to pass from the cockpit opening to a point 100 mm rearwards of the face of the rearmost pedal in an inoperative position. The template may be moved up and down. Adjustable pedals must be in their most forward position.
- T 4.2.2 The steering wheel and any padding that can be removed without the use of tools while the driver is seated may be removed for the template to fit.
- T 4.2.3 The driver's feet and legs must be completely contained within the primary structure when the driver is seated normally and the driver's feet are touching the pedals. In side and front views, any part of the driver's feet or legs must not extend above or outside of this structure.

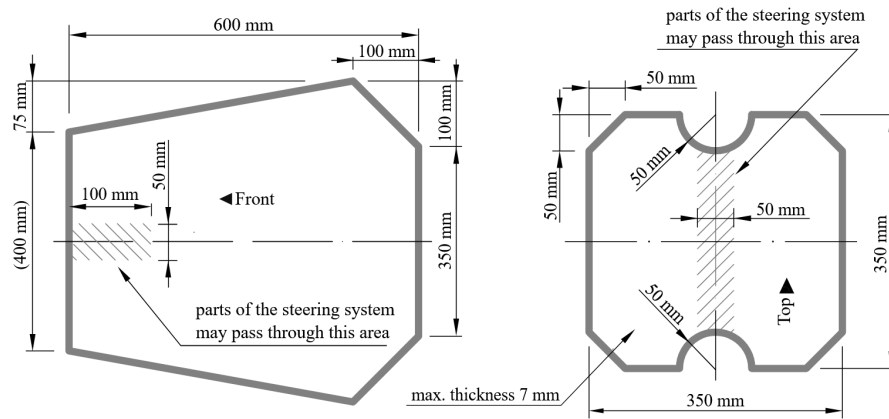


Figure 5.2: Cockpit opening template (left) and cockpit internal cross section template (right) [40].

T 3.1.1 Among other requirements, the vehicle's structure must include:

- Two roll hoops that are braced
- A front bulkhead with support system and IA
- Side impact structures

The regulatory constraints listed above serve as the foundation for key aspects of the design process. The geometric rules guided the definition of the monocoque's external shape and dimensions and were taken into account throughout the modelling stage.

5.2 Modelling Strategy

The monocoque chassis design was developed using Siemens NX, as part of the team's ongoing transition to Siemens tools, driven by the higher level of integration between key engineering functions such as CAD, CAM, FEM, and CFD, among others. The goal was to create a realistic shell model that respects both regulatory constraints and team-specific requirements, while also reflecting good composite manufacturing practices. Mechanical integration features such as inserts, brackets, or mounting reinforcements were not considered at this stage, as the focus remained on the overall shape and feasibility of the monocoque concept.

The process began by importing the FSFEUP 02 CAD model and isolating only the essential elements relevant to the current concept study: the front and main roll hoops, suspension pickup

points, accumulator, and transmission system. These elements served as fixed references that defined the key boundary conditions for the design.

With those references in place, a series of strategic sketches were created at critical sections of the car: near the suspension hardpoints, around the main and front hoops and the accumulator volume, at the front bulkhead, and at the rear end of the chassis. To streamline modelling and reduce potential asymmetries, only half of the monocoque geometry was created and later mirrored. These sketches defined the cross-sections of the monocoque along its length, to guide the overall shape. Smooth curves were then drawn to connect the corresponding sections, and the *Through Curve Mesh* tool was used to generate a continuous surface connecting all the profiles. This ensured a smooth and coherent shape with minimal geometric discontinuities.

After defining the outer shell, several cut-outs were made to accommodate functional requirements. The cockpit opening was dimensioned to comply with the Formula Student cockpit template, while also allowing for accumulator insertion, which is typically done through the top opening in monocoque designs. Additional access cut-outs were added at the front to enable serviceability in the pedal box area, and at the rear to facilitate integration of the transmission system.

Throughout the modelling process, manufacturing feasibility was kept in mind, especially considering the limitations of carbon fibre reinforced polymer (CFRP) layup. Sharp edges, small radii, and abrupt geometry changes were avoided to improve drapeability and reduce the risk of defects during lamination. The final result is a realistic shell element that balances simplicity, performance potential, and manufacturability, serving as a solid base for further design development and structural analysis.

The resulting geometry is illustrated in Figure 5.3.

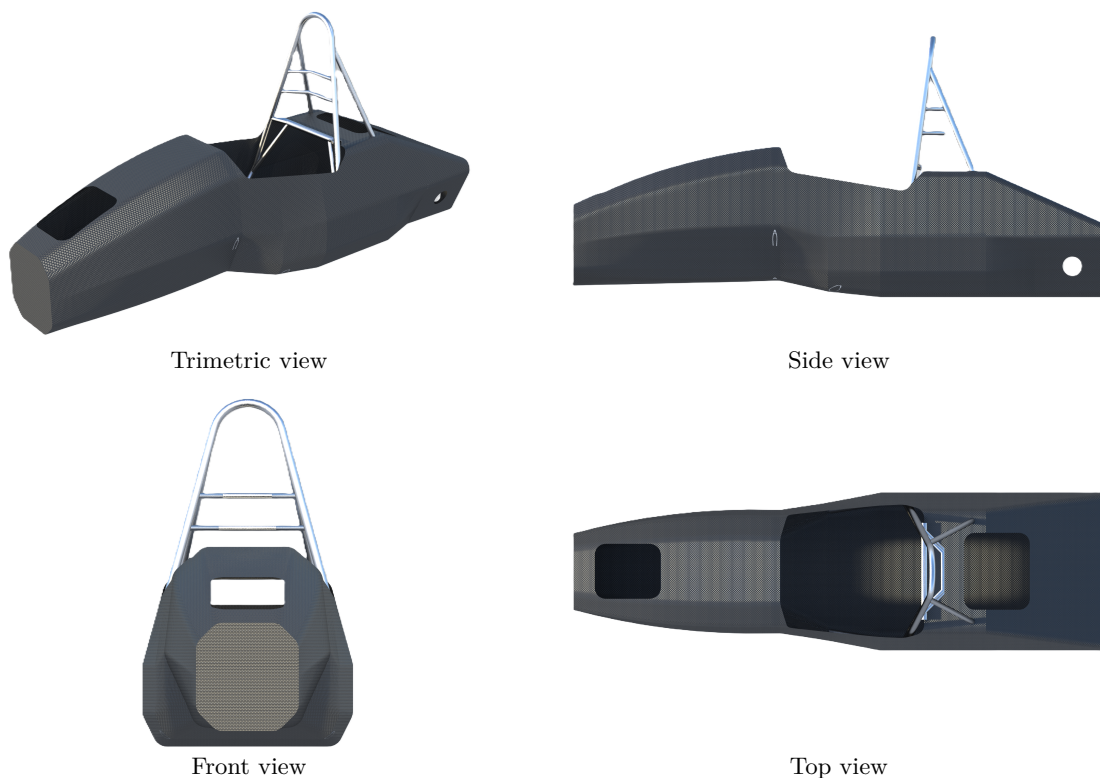


Figure 5.3: Monocoque chassis CAD model render shown in four standard views.

5.3. Material Selection

This initial design did not prioritise iteration-friendliness, as the positions of key hardpoints, such as suspension pickups, roll hoops, and packaging elements, were already fixed. As a result, the model was developed with a focus on delivering a representative concept rather than enabling design flexibility. However, future monocoque development efforts should aim to implement a more parametric modelling approach. Such an approach would facilitate faster and more efficient iterations, especially when accommodating changes driven by other subsystems, ultimately improving the overall integration and adaptability of the vehicle design.

5.3 Material Selection

This section discusses the material selection for the proof of concept monocoque being designed, from which the relevant material properties will be extracted for calculation purposes. Unlike the experimental validation phase, this design stage is not constrained by the limitations of a thesis budget, material availability or tight production timelines. Consequently, the material system does not replicate the CFRP used in the experimental work. Instead, a higher performance system, with properties taken directly from manufacturer datasheets, is selected to yield more representative results for the intended design scenario.

The carbon fibre reinforcement selected for this project is the Toray TC346 IM0221, a 200 gsm 2×2 twill fabric utilising T800 high-strength intermediate modulus fibres with a 38% resin weight. The exclusive consideration of Toray materials was dictated by the availability of complete mechanical property datasheets to the team at the time of selection, as well as the strategic intention to establish a partnership with Toray.

Only a single material type was adopted for the purposes of this study to avoid extensive finite element analysis and iterative feedback loops that would be required for layup optimisation using multiple materials. While the analytical tool developed in this work could readily be adapted to support multi-material laminates, such optimisation falls outside the scope of this work. Instead, a quasi-isotropic philosophy is maintained throughout, providing balanced stiffness and strength in multiple in-plane directions without the need for fibre-specific tailoring. It is recognised that this approach does not achieve maximum performance or optimisation, and therefore does not fully exploit the potential benefits of transitioning from a spaceframe to a monocoque. Nonetheless, the simplicity of this method enables a more straightforward preliminary analysis and sizing process, while still allowing for a sufficiently robust comparison between the two chassis types.

The TC346 IM0221 offers a combination of properties well suited to the structural demands of a Formula Student monocoque chassis. The intermediate modulus fibre grade strikes a balance between stiffness and fracture toughness, avoiding the brittleness typically associated with high modulus systems while delivering superior compressive performance. The 2×2 twill weave facilitates drapability, enabling efficient layup over the curved and intersecting geometries of a monocoque chassis, while also providing aesthetically uniform outer plies. The TC346 epoxy matrix system complements the fibre properties by providing enhanced compressive strength, fracture toughness, and property retention at elevated temperatures. These attributes, combined with the availability of detailed technical data and the anticipated benefits of supplier collaboration, render the TC346 IM0221 an optimal choice for the present design. The full manufacturer's datasheet for the selected material is provided in Appendix B.3 for reference.

The core material selected for the monocoque sandwich panels is an aluminium honeycomb, supplied by Easy Composites Ltd. Compared to polymeric foam cores, aluminium honeycomb exhibits a superior strength-to-weight ratio, combining high compressive strength with excellent

in-plane shear properties while maintaining low density. These characteristics enable the creation of exceptionally stiff and lightweight panels, which is critical in the structural design of a Formula Student monocoque.

The selected configuration features a 1/8" (3.2 mm) cell size, chosen for its combination of higher shear strength and compressive strength compared to larger-cell alternatives. The smaller cell size also improves the surface area available for bonding, enhancing the quality and reliability of the core to facesheet interface.

The specific grade used in this study is identical to that employed in the experimental validation panels. The decision to use this exact material was influenced by its ready availability to the team, thereby reducing procurement delays.

Although its metallic nature can complicate cutting and shaping operations, these challenges are outweighed by the stiffness and strength benefits provided. The full manufacturer's datasheet for the selected aluminium honeycomb core is provided in Appendix B.1 for reference.

5.4 Panel Sizing

With the selected material system described in Section 5.3, the structural requirements outlined in Chapter 3, and the analytical tools developed for SES compliance, the panel layouts for each region of the monocoque were defined through an iterative sizing approach. Starting from quasi-isotropic layouts, chosen for their balanced stiffness and strength across multiple directions, successive refinements were performed until all relevant safety factors met the regulatory thresholds. This process ensured that each panel configuration satisfies the minimum requirements while maintaining manufacturing feasibility and structural efficiency.

The method consisted of evaluating each monocoque section individually, calculating the equivalent properties and safety margins using the developed MATLAB tools, and refining the layout accordingly. Throughout this process, quasi-isotropic layouts were maintained for all panels. While this approach yielded solutions that comply with the SES minimum requirements, further optimisation was not pursued within the scope of this dissertation. Future work could explore more tailored layouts through finite element simulation of the complete chassis, enabling local adjustments based on load conditions. These optimised configurations need not remain quasi-isotropic, provided they are re-validated using the analytical tools to confirm regulatory compliance.

For all sections, the sandwich panels were defined with identical layouts on both the inner and outer skins. This choice was primarily motivated by the need to satisfy the perimeter shear requirement for certain sections, as using a thinner skin on either side would reduce overall shear capacity. Maintaining equal skins also simplifies the design process. Furthermore, the core thickness was standardised at 20 mm across all panels, as variations would provide only marginal weight savings relative to the potential loss in stiffness. From a manufacturing perspective, transitioning between different core thicknesses is challenging, often requiring core splicing and leading to abrupt geometric changes. Wherever feasible, the skins were kept symmetric about their mid-plane. Layouts such as [0/45/−45/90] were favoured to achieve near-isotropic in-plane stiffness, as even balanced woven fabrics display slight differences between the warp and weft directions.

In order to evaluate whether each section complies with the structural requirements, the analytical predictions were compared against the target safety factors established in Chapter 4. These values, summarised in Table 5.1, were defined based on the outcomes of the experimental validation campaign and represent the minimum acceptable margins for SES approval.

5.4. Panel Sizing

Table 5.1: Safety factors applied to analytical predictions for panel sizing.

Property	Safety Factor (SF)
Flexural Rigidity	1.5
3 Point Bending Maximum Load	2.0
3 Point Bending Energy Absorption	3.0
Perimeter Shear Strength	2.0

The final panel layups are summarised in Table 5.2, and the corresponding coefficients of safety are presented in Table 5.3. The results in Table 5.3 show that several sections greatly exceed the minimum safety margins. This outcome is a direct consequence of the decision to employ quasi-isotropic symmetric layups across all panels and to forego optimisation within the scope of this work. In order to ensure that every requirement was satisfied, certain layups were necessarily over-conservative in some properties, leading to disproportionately high safety factors. In some cases, values even exceed 160, which represents clear overdesign. These findings indicate that while the current solutions are compliant, there is substantial room for further optimisation in future work, where locally tailored layups could reduce weight without compromising regulatory approval.

Table 5.2: Selected laminate configurations and core thicknesses for each monocoque section.

Car Section	Outer Skin Layup	Inner Skin Layup	Core Thickness (mm)
Front Hoop Bracing	[0/45]	[45/0]	20
Front Bulkhead Support	[0/45/-45/90] _s	[0/45/-45/90] _s	20
Front Bulkhead	[0/45/-45/90] _{6s}	[0/45/-45/90] _{6s}	20
Side Impact (Vertical)	[0/45/-45/90] ₃	[90/-45/45/0] ₃	20
Side Impact (Horizontal)	[0/45/-45/90]	[90/-45/45/0]	20
TS Side Impact (Vertical)	[0/45/-45/90] ₇	[90/-45/45/0] ₇	20
TS Side Impact (Horizontal)	[0/45]	[45/0]	20
TS Rear Impact Structure	[0/45/-45/90] ₃	[90/-45/45/0] ₃	20
Main Hoop Bracing Support	[0/45/-45/90] _s	[0/45/-45/90] _s	20

Table 5.3: Safety factors achieved for each monocoque section with the selected layups.

Car Section	Safety Factors			
	Flexural Rigidity	3 Point Bending	Energy Absorption	Perimeter Shear
Front Hoop Bracing	2.344	5.774	-	-
Front Bulkhead Support	2.180	4.800	-	71.616
Front Bulkhead	19.958	77.638	-	2.180
Side Impact (Vertical)	4.805	17.103	3.065	40.105
Side Impact (Horizontal)	2.753	11.304	-	-
TS Side Impact (Vertical)	27.070	41.841	3.255	141.480
TS Side Impact (Horizontal)	2.475	5.775	-	-
TS Rear Impact Structure	3.881	17.103	3.794	35.331
Main Hoop Bracing Support	2.034	14.398	-	-

To facilitate interpretation of the results, Figure 5.4 and Table 5.4 provide a schematic of the monocoque chassis with the different structural regions highlighted. These sections correspond to the nomenclature adopted in Tables 5.2 and 5.3, enabling a clear visual reference between the analytical sizing outcomes and their physical location on the vehicle. The regions shown in grey are not defined by the regulations and therefore do not need to comply with the SES rules.

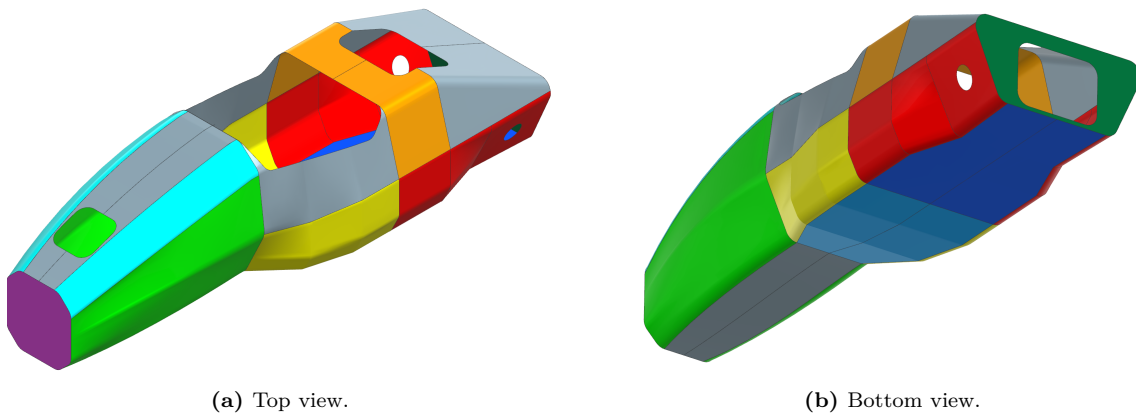


Figure 5.4: Schematics of the monocoque chassis sections corresponding to Tables 5.2 and 5.3.

5.5. Structural Analysis and Performance

Table 5.4: Colour legend for monocoque chassis sections.

Car Section	Colour
Front Hoop Bracing	Cyan
Front Bulkhead Support	Green
Front Bulkhead	Purple
Side Impact (Vertical)	Yellow
Side Impact (Horizontal)	Blue
TS Side Impact (Vertical)	Red
TS Side Impact (Horizontal)	Dark Blue
TS Rear Impact Structure	Dark Green
Main Hoop Bracing Support	Orange

5.5 Structural Analysis and Performance

The performance of a racing chassis is strongly influenced by its structural behaviour, with torsional stiffness generally regarded as the most critical parameter. High torsional stiffness ensures that suspension forces are effectively transmitted without excessive chassis deformation, thereby improving handling response, driver confidence, and overall vehicle performance.

In this section, the torsional stiffness of the developed carbon fibre monocoque is evaluated using finite element analysis (FEA). The results are compared directly with the steel spaceframe chassis of FSFEUP 02, which serves as the reference configuration. This comparison allows for a quantitative assessment of the improvements achievable through the transition from a spaceframe to a monocoque structure. In addition to stiffness, the mass of each chassis is considered, enabling the calculation of the stiffness-to-weight ratio, which is a more meaningful indicator of structural efficiency.

5.5.1 Problem Definition

The primary objective of this analysis was to evaluate the torsional stiffness of the developed CFRP monocoque chassis proof-of-concept and to compare it with the baseline FSFEUP 02 steel spaceframe. Torsional stiffness was defined as the resistance of the chassis to twisting deformation under a wheel-to-wheel torque input. The adopted procedure consisted of restraining the rear axle of the car while applying equal and opposite vertical forces at the front wheels, thereby inducing a torque about the longitudinal axis. The resulting angular displacement between the front and rear axles provides a direct measure of chassis torsional stiffness.

5.5.2 Geometry Import

The chassis geometry was initially created in Siemens NX and imported into Abaqus as a Parasolid file. To focus exclusively on the composite monocoque behaviour, only the CFRP sandwich panels were considered in the finite element model. Auxiliary steel structures, such as the main hoop and the front hoop, were deliberately excluded from the analysis. Alongside the monocoque surfaces, the suspension pick-up points and the wheel centres were also imported, as they were required for the application of loads and boundary conditions.

5.5.3 Loads and Boundary Conditions

The transmission of forces between the wheel centres and the chassis was modelled using structural distributing couplings connected to a partition that contained the suspension wishbones pick-up points. This choice was made to avoid over-constraining the pick-up regions and to approximate the way forces are introduced through wheel hubs in conventional torsional stiffness test rigs. Although a more realistic assessment would require introducing loads through the suspension rockers, the proof-of-concept model developed here only included the wishbone pick-up points. Consequently, these were employed as the load transfer interfaces. As the same methodology is consistently applied to both the monocoque and the FSFEUP 02 spaceframe chassis, the simplification does not affect the validity of the comparative analysis.

At the rear wheels, an encastre boundary condition was imposed, constraining all degrees of freedom. For the front wheels, degrees of freedom outside the transverse plane were additionally restrained, ensuring that only torsional deformation of the chassis was permitted. Finally, a pair of vertical forces of ± 1 kN were applied to the front wheel centres in opposite directions, generating a torsional moment about the longitudinal axis of the chassis.

5.5.4 Material Properties

Both the CFRP facesheets and the aluminium honeycomb core were defined in Abaqus as orthotropic laminae. This allows their integration into a single composite layup. Each facesheet was represented by its individual ply orientations, while the honeycomb core was modelled as an additional thick layer of equivalent homogeneous material inserted between the two facesheet layups. Such an approach captures the anisotropic behaviour of both constituents while ensuring that the through-thickness stiffness contribution of the core is properly represented. Furthermore, this modelling strategy enables the use of shell elements together with the composite layup definition tool in Abaqus, which provides a straightforward and efficient representation of layered sandwich structures. In contrast, a fully solid model would complicate the layup definition and significantly increase computational cost.

The elastic constants were obtained from manufacturer datasheets in Appendix B. Where values were not available, reasonable engineering estimates were introduced, as indicated in Table 5.5. In Abaqus, no inherent unit system is imposed, meaning that all inputs and outputs are treated as dimensionless quantities. As a result, strict consistency of units is essential throughout the model definition. In this work, the following unit system was adopted: millimetres (mm) for length, newtons (N) for force, megapascals (MPa) for stress and elastic constants, kilograms (kg) for mass and kilograms per cubic millimetre (kg/mm^3) for density, ensuring internal consistency across all material definitions, loads, and boundary conditions.

5.5. Structural Analysis and Performance

Table 5.5: Orthotropic lamina properties used in Abaqus for CFRP facesheets and aluminium honeycomb core.

Property	Symbol	Composite system	Aluminium Honeycomb
Longitudinal modulus	E_1 [MPa]	74400	0.1*
Transverse modulus	E_2 [MPa]	71100	0.1*
Poisson's ratio	ν_{12}	0.04	0.33*
In-plane shear modulus	G_{12} [MPa]	4570	0.1*
Transverse shear modulus	G_{13} [MPa]	2970*	483
Transverse shear modulus	G_{23} [MPa]	2970*	214
Thickness	t_{ply} [mm]	0.2	20
Density	ρ [kg/mm ³]	1.57×10^{-6} *	7.21×10^{-8}

* Values not explicitly provided in manufacturer datasheets. These were estimated as described below.

For the 2×2 twill CFRP laminates, the out-of-plane shear response is governed primarily by matrix deformation and fibre waviness through the thickness. Direct measurement of the through-thickness shear moduli is rarely performed, and their influence is typically estimated by pragmatic engineering approximations. Following the guidelines of the Composite Materials Handbook, a common approach is to relate the transverse shear moduli to the measured in-plane shear modulus through a reduction factor α , which accounts for the reduced efficiency of fibre reinforcement in the 1–3 and 2–3 directions [42]. Accordingly, the following relation is adopted:

$$G_{13} = G_{23} = \alpha G_{12}, \quad \alpha = 0.65 \quad (5.1)$$

The choice $\alpha = 0.65$ reflects typical ranges reported for balanced woven CFRP (0.60–0.75), ensuring conservative stiffness without artificially softening the laminate in torsion. Substituting the measured G_{12} from the datasheet yields the values marked with an asterisk in Table 5.5.

The density of the CFRP system was not directly provided in the datasheets and was therefore calculated from the constituent fibre and resin densities together with the cured fibre volume fraction. The composite density ρ_c can be expressed as a rule of mixtures between fibre and matrix contributions,

$$\rho_c = V_f \rho_f + V_m \rho_m \quad (5.2)$$

where V_f and V_m are the fibre and matrix volume fractions, respectively, and ρ_f and ρ_m are the corresponding densities.

The fibre volume fraction was determined from the resin weight content reported in the prepreg datasheet (38% RW) and the known fibre and resin densities. The relation between resin weight fraction w_r and resin volume fraction V_m is given by

$$V_m = \frac{w_r / \rho_m}{w_r / \rho_m + (1 - w_r) / \rho_f} \quad (5.3)$$

with $V_f = 1 - V_m$. Substituting $w_r = 0.38$, $\rho_f \approx 1.79$ g/cm³ for T800 carbon fibre, and $\rho_m \approx 1.30$ g/cm³ for the TC346 epoxy resin, yields a fibre volume fraction of approximately $V_f = 0.55$.

Using this value in Equation 5.2 gives a final composite density of $\rho_c = 1.57 \text{ g/cm}^3 = 1.57 \times 10^{-6} \text{ kg/mm}^3$, consistent with typical values for aerospace-grade CFRP laminates.

The in-plane (membrane) stiffness of a hexagonal honeycomb is several orders of magnitude lower than its through-thickness compression and shear stiffness. When tensile moduli are not reported by the manufacturer, they can be estimated using the Gibson–Ashby scaling laws for cellular solids [43]. In practice, the values obtained for E_1 and E_2 are extremely small, typically in the kPa–MPa range, and are of little structural relevance, since the primary role of the honeycomb core is to provide out-of-plane stiffness. For the present FEM model, small but non-zero values were introduced in the material definition to ensure numerical stability, while having negligible influence on the overall torsional stiffness of the chassis. The values listed in Table 5.5 are therefore indicated with an asterisk.

Poisson’s ratios for metallic honeycombs are rarely reported and, in practice, this parameter has limited influence on the global stiffness response of sandwich panels. In the present work, the Poisson’s ratio of aluminium is therefore adopted as a reasonable and numerically stable approximation.

$$\nu_{12} = 0.33 \quad (5.4)$$

This value is flagged with an asterisk in Table 5.5 as an engineering assumption.

These estimates are specifically intended to reproduce the global torsional rigidity of the sandwich structure using composite shell sections. They are not designed to capture localised phenomena such as core failure or facesheet–core interactions, and for this reason all strength parameters were deliberately omitted from the material definitions.

5.5.5 Layup Definition

The layups applied in the finite element model were defined according to the sizing methodology described in Section 5.4. Each of the chassis sections was assigned a laminate configuration that satisfies the minimum requirements established in the rules. The resulting layup schedule is shown in Figure 5.5, where the colour coding highlights the distribution of plies across the chassis.

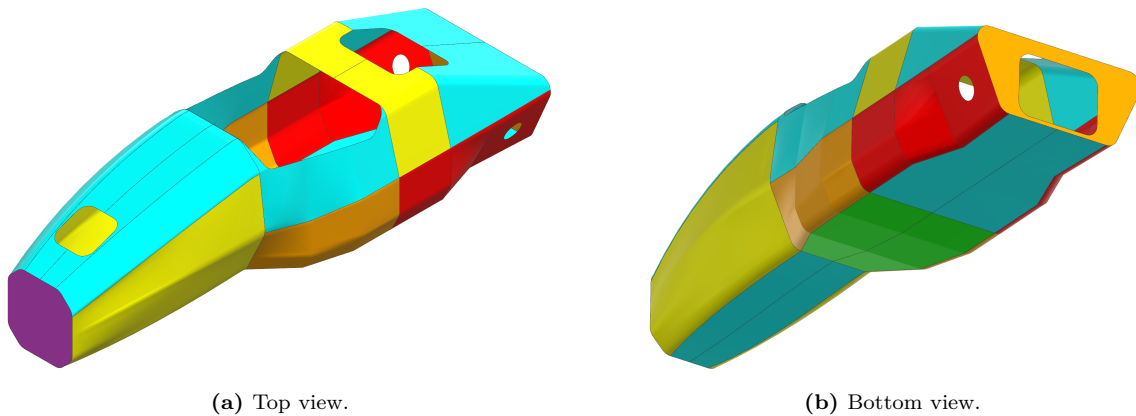
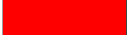


Figure 5.5: Layup schedule assigned to the different chassis regions.

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Table 5.6: Colour legend for sandwich panel layup schedules.

Colour	Outer Skin Layup	Inner Skin Layup	Core Thickness (mm)
	[0/45]	[45/0]	20
	[0/45/-45/90]	[90/-45/45/0]	20
	[0/45/-45/90] _s	[0/45/-45/90] _s	20
	[0/45/-45/90] ₃	[90/-45/45/0] ₃	20
	[0/45/-45/90] ₇	[90/-45/45/0] ₇	20
	[0/45/-45/90] _{6s}	[0/45/-45/90] _{6s}	20

In regions where no specific regulatory constraints apply, the minimum layup thickness was employed in order to reduce the overall mass of the structure. This strategy ensures compliance with the mandatory rule-defined areas, while optimising weight in less critical regions, thereby achieving a balance between structural performance and mass efficiency.

5.5.6 Simulation

The torsional stiffness analysis was carried out using the *Static, General* step in Abaqus/Standard. This procedure is appropriate since the objective is to evaluate the elastic response of the chassis under quasi-static loading, without accounting for inertial or damping effects. The incrementation parameters were defined to ensure both accuracy and robustness of convergence. The maximum number of increments was set to 1000, with an initial increment size of 0.01. To prevent premature termination of the step in case of local non-linearities, the minimum increment size was limited to 1×10^{-5} , while the maximum increment size was set to 1. These settings allowed the solver to capture the structural response within the elastic range without encountering convergence difficulties.

5.5.7 Element Selection and Mesh

The monocoque structure was modelled using shell elements, which provide an efficient and accurate representation of thin-walled sandwich components. The composite layup was defined at the element level through the *Composite Layup* module in Abaqus, enabling each ply and the core to be represented within a single shell section. This approach avoids the need for solid elements through the thickness, significantly reducing computational cost while maintaining fidelity in predicting global torsional stiffness.

The mesh was generated using quadrilateral S4R elements, which are four-node, doubly curved, general-purpose shell elements with reduced integration and hourglass control. These elements are widely adopted for composite thin-walled structures, offering a good compromise between accuracy and computational efficiency. The element size adopted in the model is discussed in Section 5.5.8, where a dedicated mesh convergence study is presented.

5.5.8 Mesh Convergence Study

To ensure that the torsional stiffness predictions were not influenced by mesh density, a convergence study was performed. The model was discretised using different global seed sizes, ranging from 50 mm to 3 mm. For each mesh, the total number of elements and nodes was recorded, together with the resulting rotation of the front shaft relative to the rear shaft about the longitudinal axis, denoted here as θ . This relative rotation was extracted from Abaqus as the displacement degree of freedom *UR2* at the reference points coupled to the front suspension pick-up

locations.

The torsional stiffness K_t was then calculated from the applied loading and the measured rotation using

$$K_t = \frac{M}{\theta} = \frac{F \cdot l}{\theta} \quad (5.5)$$

where F [N] is the vertical load applied at each front wheel, l [m] is the track width (distance between the two front wheels), and θ is the relative rotation between the front and rear shafts. Abaqus provides θ in radians, which was converted to degrees for stiffness evaluation in units of Nm/°.

Table 5.7: Summary of mesh convergence study results.

Seed size [mm]	Elements	Nodes	θ [rad]	K_t [Nm/°]
50	2840	2806	0.00264	7318
40	3871	3856	0.00263	7346
30	6203	6187	0.00278	6950
20	14443	14396	0.00284	6803
10	57882	57532	0.00289	6685
5	225604	224047	0.00291	6639
3	625025	620489	0.00292	6617

Figure 5.6 shows the variation of θ with the number of nodes, while Figure 5.7 presents the corresponding torsional stiffness. Both curves demonstrate asymptotic behaviour as the mesh is refined, with negligible changes in θ and K_t beyond a seed size of 10 mm. Based on these results, this mesh density was adopted in all subsequent simulations, as it provided a good compromise between accuracy and computational cost.

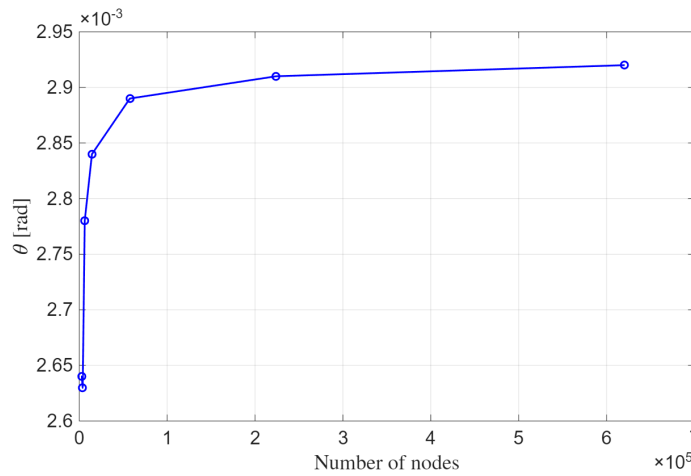


Figure 5.6: Relative rotation θ between front and rear shafts as a function of mesh density (number of nodes).

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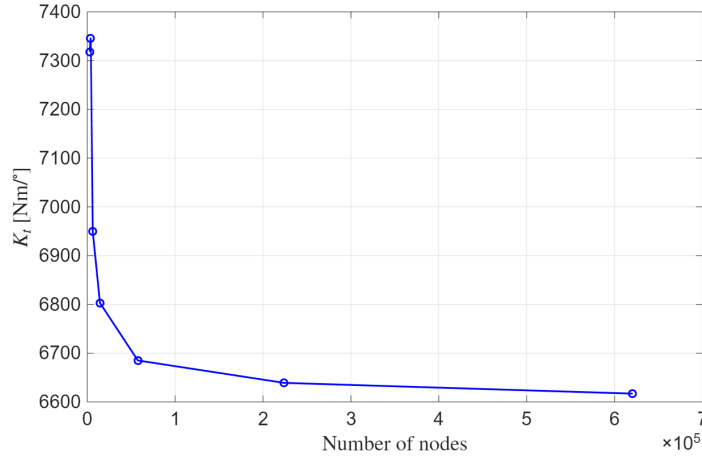


Figure 5.7: Torsional stiffness K_t as a function of mesh density (number of nodes).

5.5.9 Results

Before extracting the final torsional stiffness value, a preliminary study was carried out to confirm the linearity of the model response. Several load magnitudes were applied at the front wheels, and the corresponding torsional stiffness was calculated using Equation 5.5. In a perfectly linear elastic analysis, the stiffness should remain constant regardless of the applied load, with variations attributable only to numerical round-off or solver tolerances.

Table 5.8: Verification of model linearity: torsional stiffness calculated for different applied wheel loads.

Load per wheel F [N]	Rotation θ [rad]	K_t [Nm/°]
500	0.00145	6685
1000	0.00289	6685
2000	0.00578	6685
3000	0.00867	6685

The results presented in Table 5.8 show that the torsional stiffness remained constant across all applied load levels, with a value of 6685 Nm/° obtained in every case. The proportionality between applied force and measured rotation demonstrates that the chassis behaves in a linear elastic manner under the investigated loading conditions. The absence of variation in the calculated stiffness confirms that the model response is independent of load magnitude. This validates the use of a single reference load case in subsequent analyses, as it is sufficient to characterise the torsional stiffness of the chassis. Figure 5.8 illustrates the global deformation pattern under torsional loading, showing the expected twist between the front and rear axles.

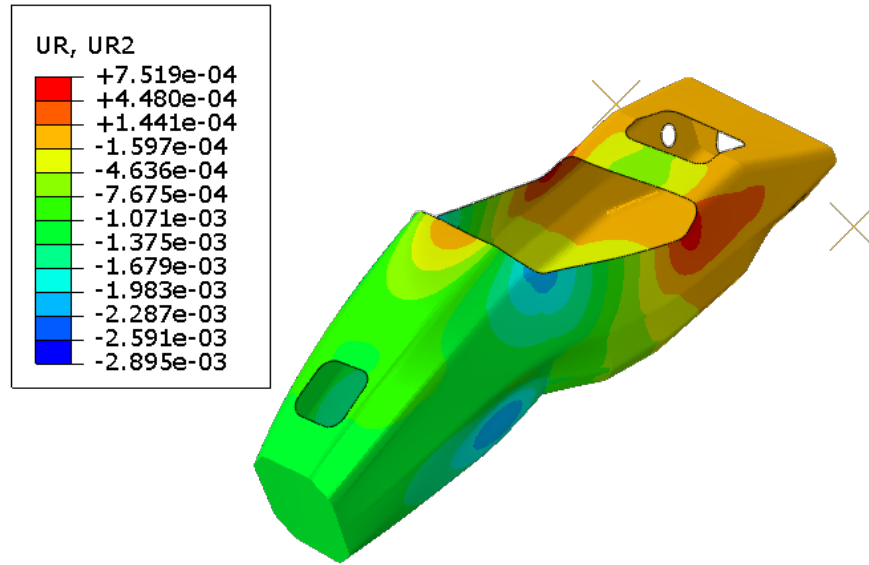


Figure 5.8: Deformation contour of the chassis under torsional loading (exaggerated scale).

The numerical model also provides an estimation of the chassis mass. For the monocoque, the raw Abaqus calculation corresponds only to the composite sandwich panels, yielding a mass of 28.07 kg. To comply with Formula Student regulations, additional structural elements must be included: the main hoop, the main hoop bracings, and the front hoop. The mass of the main hoop and its bracings was obtained from the FSFEUP 02 spaceframe, with values of 4.03 kg for the main hoop and 0.77 kg for each bracing. For the front hoop, the FSFEUP 02 spaceframe reference corresponds to a steel structure weighing 2.58 kg. However, in a monocoque chassis this component can realistically be manufactured in aluminium, reducing the mass to 0.89 kg. The mass of the metallic inserts required for attachment points was not considered, as their contribution is negligible relative to the overall chassis mass. Adding the mass of the mandatory components to the sandwich panels results in a total monocoque mass of approximately 34.62 kg. The absolute accuracy of this estimate is not critical, as it is intended solely for comparison purposes. Furthermore, layup optimisation, not undertaken in this study, would likely reduce the mass of the monocoque even further without significantly compromising torsional stiffness, meaning that the present comparison can be regarded as conservative.

For reference, the FSFEUP 02 spaceframe chassis exhibited a torsional stiffness of 2201 Nm/° with a total structural mass of 36.98 kg. A direct comparison of stiffness, mass, and stiffness-to-weight ratio is presented in Table 5.9.

Table 5.9: Comparison of torsional stiffness and stiffness-to-weight ratio between spaceframe and monocoque chassis.

Chassis type	Mass [kg]	K_t [Nm/°]	K_t/m [Nm/° · kg ⁻¹]
Spaceframe (FSFEUP 02)	36.98	2201	59.52
Monocoque (this work)	34.62	6685	193.10

The results clearly demonstrate the advantages of the monocoque concept over the traditional spaceframe. Although the overall mass is only marginally lower, primarily due to replacing the steel front hoop with an aluminium equivalent, the monocoque attains nearly double the torsional stiffness. This translates into an improvement of approximately 224% in stiffness-to-weight ratio,

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underscoring the substantial structural efficiency achievable with composite sandwich construction. Moreover, since the adoption of a monocoque inherently reduces the need for separate bodywork components, the final vehicle mass reduction will be even greater, as fewer auxiliary structures are required. It should also be noted that the present estimate is conservative, as no optimisation of the layup schedule was performed. Consequently, the proposed design not only fulfils the defined objectives but also provides a compelling justification for FSFEUP to pursue the transition from a spaceframe to a monocoque chassis architecture.

Chapter 6

Conclusions and Future Work

The primary objective of this work was to develop and validate an analytical tool to support the structural sizing of sandwich composite panels for application in a Formula Student monocoque chassis. The framework was conceived to incorporate regulatory requirements from the Structural Equivalency Spreadsheet (SES) while addressing the specific design constraints of the FSFEUP project. The overarching aim was to assess whether a transition from a steel spaceframe to a composite monocoque is justified in terms of performance, and to provide the team with the tools and methodology needed to enable this transition. In particular, the work adhered to the packaging and geometric boundaries defined by the FSFEUP 02 prototype, while seeking to achieve the dual performance targets of limiting chassis mass to below 35 kg and attaining a torsional stiffness of at least 2000 Nm/°. To evaluate these objectives, the proposed monocoque was benchmarked against the FSFEUP 02 steel spaceframe to demonstrate potential performance gains.

The methodology adopted in this work combined analytical, experimental, and numerical approaches to ensure both accuracy and applicability of the proposed sizing tool. At its core, a MATLAB-based framework was developed to implement Classical Laminate Theory and a range of failure criteria, enabling the evaluation of stiffness, strength, and energy absorption of candidate sandwich panels. This framework provided the basis for panel sizing against the requirements imposed by the SES.

To supply the necessary input data, an extensive programme of experimental characterisation of carbon fibre reinforced polymer laminates was undertaken. Tensile, compressive, and shear tests were performed to determine the elastic constants and strength parameters required for analytical modelling, with strains recorded using both digital image correlation and strain gauges. Interlaminar shear strength (ILSS) testing was not performed, which limited the validation of the perimeter shear approximation and reduced confidence in capturing the out-of-plane shear response of the sandwich panels.

Beyond laminate characterisation, the structural performance of sandwich panels was assessed through dedicated mechanical testing aimed at validating the analytical framework. Three-point bending experiments were used to compare predicted and measured flexural stiffness and failure loads, while perimeter shear tests provided data on the out-of-plane shear capacity of the sandwich construction, a metric directly relevant to impact scenarios and general out-of-plane loading.

The validation campaign was limited to cross-ply laminates, which proved insufficient to capture the full range of failure mechanisms or to enable a reliable comparison between different failure criteria. In addition, several of the test panels exhibited noticeable warpage, which likely affected the

accuracy of the results and should be addressed in future campaigns through improved manufacturing control. Broader testing with alternative layups and material systems will also be required to improve accuracy, particularly with respect to stress concentrations and defect sensitivity.

Even within these constraints, the tests confirmed that the analytical tool provided acceptable approximations of sandwich panel behaviour, though with notable limitations. Stiffness predictions were shown to be reasonably accurate, confirming the robustness of Classical Laminate Theory for elastic response. By contrast, strength predictions were consistently overestimated, which necessitated the use of relatively high safety factors in the sizing process. This tendency towards overprediction suggests that the tool is better suited as a conservative guide rather than a precise predictor of failure loads.

Building on these results, the framework was then applied to the complete sizing of a Formula Student monocoque chassis. Each region of the structure was evaluated individually, beginning from quasi-isotropic baseline layups and iteratively refined until compliance with SES requirements was achieved.

With this framework implemented and applied to the chassis design, the study proceeded to assess its performance and extract the main findings regarding structural efficiency. When compared against the FSFEUP 02 steel spaceframe baseline, the monocoque achieved a significant increase in torsional stiffness, albeit with a lower than expected mass reduction. The resulting design was characterised by stiffness values far exceeding the target minimum, leaving substantial margin for optimisation. This outcome demonstrates the inherent efficiency of the monocoque architecture and supports the team's transition from spaceframe to composite construction.

The excessive stiffness and relatively high mass were largely attributed to the reliance on quasi-isotropic symmetric layups. This design philosophy, while simplifying the SES approval process, often resulted in overbuilding of properties not critical for a given region of the chassis. Future iterations can reduce mass and balance stiffness more effectively through the adoption of optimised and multi-material layups, tailored to the loading conditions of each section. In particular, it is advisable to place $\pm 45^\circ$ plies on the outer surfaces rather than 0° plies, as implemented in this work. Such a configuration improves torsional rigidity, enhances resistance to in-plane shear stresses, and provides better damage tolerance, all of which are critical for the performance and durability of a composite monocoque. In this context, finite element modelling offers a valuable complement to the analytical tool, enabling targeted optimisation of layup schedules and variation of core thickness to reduce mass while maintaining the required torsional stiffness. Equally important, future developments must also address cost and manufacturability, ensuring that proposed layups and configurations are not only structurally efficient but also feasible within the resource and time constraints of a Formula Student team. Incorporating such considerations would enhance competition readiness and strengthen the practical impact of the tool.

These findings, and the refinement opportunities they suggest, highlight both the current limitations and the considerable potential of the approach. In summary, while the developed tool is limited in accuracy for strength prediction, it has proven useful as a practical means of preliminary design. Beyond enabling the successful sizing of an SES-compliant chassis, the tool now provides FSFEUP with a structured methodology that can support future iterations. More broadly, the integration of analytical modelling, experimental characterisation, and numerical simulation constitutes a transferable workflow that may be adopted by other Formula Student teams undertaking similar transitions to composite monocoques. The work has thus established both the feasibility and the potential of a composite monocoque chassis within FSFEUP, providing the groundwork for optimisation strategies that can unlock the full performance benefits of this architecture while

reinforcing the team's capability to compete at a higher technical level.

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References

Appendix A

MATLAB Code for Analytical Tools

This appendix presents the MATLAB[®] code developed during the course of this thesis to support the analytical evaluation of the carbon fibre monocoque. These scripts were used to implement classical laminate theory calculations, estimate flexural stiffness, evaluate three-point bending loads and respective failure criteria, estimate energy absorption under bending, estimate perimeter shear loads, and support other numerical checks performed during the design and validation phases in Chapters 3, 4 and 5.

A.1 Panel Property Definition and Input Parameters

```
1 % Code to test the validity of all sandwich panel for each section
2
3 clc          % Clears command window
4 clear        % Clears all variables from workspace
5 close all    % Closes all open figure windows
6
7 % --- Carbon Fibre properties struct ---
8
9 CFRP.t_ply = 0.20e-3;          % m, ply thickness (estimado com base no
   areal weight)
10 CFRP.E1 = 74.4e9;              % Pa, Young's modulus (warp direction)
11 CFRP.E2 = 71.1e9;              % Pa, Young's modulus (weft direction)
12 CFRP.G12 = 4.57e9;            % Pa, in-plane shear modulus
13 CFRP.v12 = 0.04;              % Poisson's ratio
14 CFRP.sigma_xt = 985e6;         % Pa, tensile strength (warp direction)
15 CFRP.sigma_xc = 841e6;         % Pa, compressive strength (warp direction)
16 CFRP.sigma_yt = 928e6;         % Pa, tensile strength (weft direction)
17 CFRP.sigma_yc = 795e6;         % Pa, compressive strength (weft direction)
18 CFRP.tau_shear = 144e6;        % Pa, maximum in-plane shear strength
19 CFRP.ilss = 74.6e6;            % Pa, interlaminar shear strength
20
21
22
23 % --- Sandwich Panel Layup Schedule ---
```

References

```
24
25 layup1 = [0 45 -45 90];
26 layup2 = [90 -45 45 0];
27
28 % Front Hoop Bracing
29 Panel(1).l = 0.040;           % m, panel length
30 Panel(1).w = 0.900;           % m, panel width
31 Panel(1).t_core = 20e-3;       % m, core thickness
32 Panel(1).layup_out = [0 45];  % degree, ply orientations
33 Panel(1).layup_in = [45 0];   % degree, ply orientations
34
35 % Front Bulkhead Support
36 Panel(2).l = 0.300;
37 Panel(2).w = 0.900;
38 Panel(2).t_core = 20e-3;
39 Panel(2).layup_out = [layup1 layup2];
40 Panel(2).layup_in = [layup1 layup2];
41
42 % Front Bulkhead
43 Panel(3).l = 0.280;
44 Panel(3).w = 0.300;
45 Panel(3).t_core = 20e-3;
46 Panel(3).layup_out = [repmat(layup1,1,6) repmat(layup2,1,6)];
47 Panel(3).layup_in = [repmat(layup1,1,6) repmat(layup2,1,6)];
48
49 % Side Impact Structure Vertical
50 Panel(4).l = 0.320;
51 Panel(4).w = 0.520;
52 Panel(4).t_core = 20e-3;
53 Panel(4).layup_out = [repmat(layup1,1,3)];
54 Panel(4).layup_in = [repmat(layup2,1,3)];
55
56 % Side Impact Structure Horizontal
57 Panel(5).l = 0.140;
58 Panel(5).w = 0.520;
59 Panel(5).t_core = 20e-3;
60 Panel(5).layup_out = layup1;
61 Panel(5).layup_in = layup2;
62
63 % Tractive system Side Impact Structure Vertical
64 Panel(6).l = 0.320;
65 Panel(6).w = 0.950;
66 Panel(6).t_core = 20e-3;
67 Panel(6).layup_out = [repmat(layup1,1,7)];
68 Panel(6).layup_in = [repmat(layup2,1,7)];
69
70 % Tractive system Side Impact Structure Horizontal
71 Panel(7).l = 0.210;
72 Panel(7).w = 0.950;
73 Panel(7).t_core = 20e-3;
74 Panel(7).layup_out = [0 45];
75 Panel(7).layup_in = [45 0];
76
```

```

77 % Tractive system Rear Impact Structure
78 Panel(8).l = 0.320;
79 Panel(8).w = 0.420;
80 Panel(8).t_core = 20e-3;
81 Panel(8).layup_out = [repmat(layup1,1,3)];
82 Panel(8).layup_in = [repmat(layup2,1,3)];
83
84 % Main Hoop Bracing Support
85 Panel(9).l = 0.280;
86 Panel(9).w = 0.280;
87 Panel(9).t_core = 20e-3;
88 Panel(9).layup_out = [layup1 layup2];
89 Panel(9).layup_in = [layup1 layup2];
90
91
92
93 % --- Baseline Steel Tube properties ---
94
95 % Front Hoop Bracing
96 Tube(1).E = 200e9; % Pa, Young's modulus
97 Tube(1).sigma_y = 305e6; % Pa, yield strength
98 Tube(1).D = 25.4e-3; % m, outer diameter
99 Tube(1).t = 1.6e-3; % m, wall thickness
100 Tube(1).n = 1; % number of tubes
101 Tube(1).F_shear = 0e3; % N, perimeter shear load required
102 Tube(1).energy = 0; % J, energy absorption required
103
104 % Front Bulkhead Support
105 Tube(2).E = 200e9;
106 Tube(2).sigma_y = 305e6;
107 Tube(2).D = 25.4e-3;
108 Tube(2).t = 1.2e-3;
109 Tube(2).n = 6;
110 Tube(2).F_shear = 4e3;
111 Tube(2).energy = 0;
112
113 % Front Bulkhead
114 Tube(3).E = 200e9;
115 Tube(3).sigma_y = 305e6;
116 Tube(3).D = 25.4e-3;
117 Tube(3).t = 1.6e-3;
118 Tube(3).n = 2;
119 Tube(3).F_shear = 2 * (Panel(3).l + Panel(3).w) * 0.0015 * 219e6; %
    equivalent to 1.5 mm thick steel plate
120 Tube(3).energy = 0;
121
122 % Side Impact Structure Vertical
123 Tube(4).E = 200e9;
124 Tube(4).sigma_y = 305e6;
125 Tube(4).D = 25.4e-3;
126 Tube(4).t = 1.6e-3;
127 Tube(4).n = 2;
128 Tube(4).F_shear = 7.5e3;

```

References

```
129 Tube(4).energy = 65;
130
131 % Side Impact Structure Horizontal
132 Tube(5).E = 200e9;
133 Tube(5).sigma_y = 305e6;
134 Tube(5).D = 25.4e-3;
135 Tube(5).t = 1.6e-3;
136 Tube(5).n = 1;
137 Tube(5).F_shear = 0e3;
138 Tube(5).energy = 0;
139
140 % Tractive system Side Impact Structure Vertical
141 Tube(6).E = 200e9;
142 Tube(6).sigma_y = 305e6;
143 Tube(6).D = 25.4e-3;
144 Tube(6).t = 1.6e-3;
145 Tube(6).n = 2;
146 Tube(6).F_shear = 7.5e3;
147 Tube(6).energy = 65;
148
149 % Tractive system Side Impact Structure Horizontal
150 Tube(7).E = 200e9;
151 Tube(7).sigma_y = 305e6;
152 Tube(7).D = 25.4e-3;
153 Tube(7).t = 1.6e-3;
154 Tube(7).n = 1;
155 Tube(7).F_shear = 0e3;
156 Tube(7).energy = 0;
157
158 % Tractive system Rear Impact Structure
159 Tube(8).E = 200e9;
160 Tube(8).sigma_y = 305e6;
161 Tube(8).D = 25.4e-3;
162 Tube(8).t = 1.6e-3;
163 Tube(8).n = 2;
164 Tube(8).F_shear = 7.5e3;
165 Tube(8).energy = 65;
166
167 % Main Hoop Bracing Support
168
169 Tube(9).E = 200e9;
170 Tube(9).sigma_y = 305e6;
171 Tube(9).D = 25.4e-3;
172 Tube(9).t = 1.2e-3;
173 Tube(9).n = 2;
174 Tube(9).F_shear = 0e3;
175 Tube(9).energy = 0;
176
177
178
179 % --- Calculations ---
180
181 Results(9) = struct('EI_panel', [], 'SF_EI', [], ...
```

```

182         'F_MaxStress', [], 'F_TsaiHill', [], 'F_TsaiWu', [],
183         'F_Hashin', [], 'SF_3pb', [], ...
184         'U_MaxStress', [], 'U_TsaiHill', [], 'U_TsaiWu', [],
185         'U_Hashin', [], 'SF_U', [], ...
186         'F_per_shear', [], 'SF_ps', []]);
187
188 Matrix.Q = Q_matrix(CFRP); % inherent property of the CFRP
189
190 for i = 1:9
191     % CLT
192     [Matrix.A, Matrix.B, Matrix.D] = clt(Panel(i), CFRP, Matrix);
193
194     % EI, Flexural Rigidity
195     [EI_panel, SF.EI] = flexural_rigidity(Matrix, Panel(i), Tube(i));
196
197     % 3PB Failure Load
198     [F.MaxStress, F.TsaiHill, F.TsaiWu, F.Hashin, SF.threepb] =
199         ThreePB(Matrix, Panel(i), CFRP, Tube(i));
200
201     % U, Absorbed Energy
202     [U.MaxStress, U.TsaiHill, U.TsaiWu, U.Hashin, SF.U] = energy(F,
203         EI_panel, Tube(i));
204
205     % Perimeter Shear Failure Load
206     [F_per_shear, SF.ps] = perimeter_shear(CFRP, Panel(i), Tube(i));
207
208     % Safety Factor Evaluation
209     Results(i).EI_panel = EI_panel;
210     Results(i).SF_EI = SF.EI;
211
212     Results(i).F_MaxStress = F.MaxStress;
213     Results(i).F_TsaiHill = F.TsaiHill;
214     Results(i).F_TsaiWu = F.TsaiWu;
215     Results(i).F_Hashin = F.Hashin;
216     Results(i).SF_3pb = SF.threepb;
217
218     Results(i).U_MaxStress = U.MaxStress;
219     Results(i).U_TsaiHill = U.TsaiHill;
220     Results(i).U_TsaiWu = U.TsaiWu;
221     Results(i).U_Hashin = U.Hashin;
222     Results(i).SF_U = SF.U;
223
224     Results(i).F_per_shear = F_per_shear;
225     Results(i).SF_ps = SF.ps;
226
227 end
228
229 % --- Build summary table ---
230 sectionNames = { ...
231     'Front Hoop Bracing', ...
232     'Front Bulkhead Support', ...

```

References

```
231     'Front Bulkhead', ...
232     'SIS Vertical', ...
233     'SIS Horizontal', ...
234     'TS SIS Vertical', ...
235     'TS SIS Horizontal', ...
236     'TS Rear Impact Structure', ...
237     'Main Hoop Bracing Support'}];
238
239 n = numel(Panel);
240
241 dims    = strings(n,1);
242 layOut  = strings(n,1);
243 layIn   = strings(n,1);
244 tCore   = strings(n,1);
245
246 for i = 1:n
247     % dimensions in mm, shown as "L * W"
248     dims(i)    = sprintf('%0f * %0f', Panel(i).l*1e3, Panel(i).w*1e3);
249
250     % layups as "[0 / 90 / +45/-45 ...]"
251     layOut(i) = "[" + strjoin(string(Panel(i).layup_out), " / ") + "]";
252     layIn(i)  = "[" + strjoin(string(Panel(i).layup_in), " / ") + "]";
253
254     % core thickness in mm
255     tCore(i)  = sprintf('%0.1f', Panel(i).t_core*1e3);
256 end
257
258 T = table( ...
259     sectionNames(:), ...
260     dims, ...
261     layOut, ...
262     layIn, ...
263     tCore, ...
264     [Results.SF_EI].', ...
265     [Results.SF_3pb].', ...
266     [Results.SF_U].', ...
267     [Results.SF_ps].', ...
268     'VariableNames', {'chassis_section','dimensions_lxw_mm', ...
269     'outer_layup','inner_layup','core_thickness_mm', ...
270     'SF_EI','SF_3pb','SF_U','SF_ps'});
271
272 disp(T) % show in Command Window
273 writetable(T,'panel_summary.csv') % CSV for quick sharing
```

Listing A.1: Matlab file input.m

A.2 Plane Stress Reduced Stiffness Matrix

```
1 function Q = Q_matrix(CFRP)
2
3     % Material properties
4     E1 = CFRP.E1;
5     E2 = CFRP.E2;
6     G12 = CFRP.G12;
7     v12 = CFRP.v12;
8
9     v21 = (E2 / E1) * v12;
10
11    % Q matrix in material coordinates
12    Q11 = E1 / (1 - v12 * v21);
13    Q22 = E2 / (1 - v12 * v21);
14    Q12 = v12 * E2 / (1 - v12 * v21);
15    Q66 = G12;
16
17    Q = [Q11, Q12, 0; Q12, Q22, 0; 0, 0, Q66];
18
19 end
```

Listing A.2: Matlab file Q_matrix.m

A.3 Laminate Stiffness Matrices

```

1 function [A, B, D] = clt(Panel, CFRP, Matrix)
2
3     N_out = numel(Panel.layup_out);
4     N_in  = numel(Panel.layup_in);
5     N_total = N_out + 1 + N_in; % +1 for the core
6
7     % Full layup including core placeholder (angle irrelevant for core)
8     layup_full = [Panel.layup_out  0  Panel.layup_in];
9
10    % Total thickness and z-interfaces (mid-plane at z=0)
11    total_thickness = (N_out + N_in)*CFRP.t_ply + Panel.t_core;
12    z = zeros(1, N_total + 1);
13    z(1) = -total_thickness/2;
14
15    % Top facesheet
16    for i = 2:(N_out+1)
17        z(i) = z(i-1) + CFRP.t_ply;
18    end
19    % Core
20    z(N_out+2) = z(N_out+1) + Panel.t_core;
21    % Bottom facesheet
22    for i = (N_out+3):(N_total+1)
23        z(i) = z(i-1) + CFRP.t_ply;
24    end
25
26    % Initialise A, B and D
27    A = zeros(3,3);
28    B = zeros(3,3);
29    D = zeros(3,3);
30
31    % Pre-extract CFRP Q terms
32    Q11 = Matrix.Q(1,1);
33    Q22 = Matrix.Q(2,2);
34    Q12 = Matrix.Q(1,2);
35    Q66 = Matrix.Q(3,3);
36
37    for k = 1:N_total
38
39        if k == N_out + 1
40            Q_bar = zeros(3,3); % "empty" core for in-plane stiffness
41        else
42            m = cosd(layup_full(k));
43            n = sind(layup_full(k));
44
45            % Transform Q -> Q_bar
46            Q_bar = zeros(3,3);
47            Q_bar(1,1) = Q11*m^4 + 2*(Q12+2*Q66)*m^2*n^2 + Q22*n^4;
48            Q_bar(1,2) = (Q11+Q22-4*Q66)*m^2*n^2 + Q12*(m^4+n^4);
49            Q_bar(2,2) = Q11*n^4 + 2*(Q12+2*Q66)*m^2*n^2 + Q22*m^4;
50            Q_bar(1,3) = (Q11-Q12-2*Q66)*m^3*n - (Q22-Q12-2*Q66)*m*n^3;
51            Q_bar(2,3) = (Q11-Q12-2*Q66)*m*n^3 - (Q22-Q12-2*Q66)*m^3*n;

```

```

52         Q_bar(3,3) = (Q11+Q22-2*Q12-2*Q66)*m^2*n^2 + Q66*(m^4+n^4);
53         Q_bar(2,1) = Q_bar(1,2); Q_bar(3,1) = Q_bar(1,3); Q_bar(3,2) =
           Q_bar(2,3);
54     end
55
56     % Integrate over layer thickness
57     z_bot = z(k);
58     z_top = z(k+1);
59     dz = z_top - z_bot;
60     dz2 = z_top^2 - z_bot^2;
61     dz3 = z_top^3 - z_bot^3;
62
63     A = A + Q_bar * dz;
64     B = B + 0.5 * Q_bar * dz2;
65     D = D + (1/3) * Q_bar * dz3;
66 end
67 end

```

Listing A.3: Matlab file clt.m

A.4 Flexural Rigidity Evaluation

```
1 function [EI_panel, EI_tube] = flexural_rigidity(Matrix, Panel, Tube)
2
3     A = Matrix.A;
4     B = Matrix.B;
5     D = Matrix.D;
6     d = inv(D - B*(A\B));
7
8     % EI for composite sandwich panel
9     EI_panel = Panel.width/d(1,1);
10
11    % EI for baseline steel tube
12    d = Tube.D - 2*Tube.t; % m, inner diameter
13    I = pi/64 * (Tube.D^4 - d^4); % m^4, second moment of area
14    EI_tube = Tube.E * I * Tube.n; % N*m^2, flexural rigidity of n tubes
15 end
```

Listing A.4: Matlab file flexural_rigidity.m

A.5 Failure Load Prediction under Three-point Bending

```

1 function [F_MaxStress, F_TsaiHill, F_TsaiWu, F_Hashin, SF_3pb] =
   ThreePB(Matrix, Panel, CFRP, Tube)
2
3 % Test fixture setup
4 L = 0.400; % m, span length
5 W = 0.275; % m, panel width
6
7 % --- Face failure limits Calculation ---
8 F_guess = 0;
9 F_step = 100;
10 max_iter = 10000;
11 iter = 0;
12
13 F_MaxStress = NaN;
14 F_TsaiHill = NaN;
15 F_TsaiWu = NaN;
16 F_Hashin = NaN;
17
18 A = Matrix.A;
19 B = Matrix.B;
20 D = Matrix.D;
21
22 stiffness = [A, B; B, D];
23 compliance = inv(stiffness);
24
25 n_out = length(Panel.layup_out);
26 t_out = n_out*CFRP.t_ply;
27 n_in = length(Panel.layup_in);
28 t_in = n_in*CFRP.t_ply;
29 h = t_out + t_in + Panel.t_core;
30 z_out = (-h/2) + ((1:n_out) - 0.5) * CFRP.t_ply; % compression
31 z_in = (+h/2) - ((1:n_in) - 0.5) * CFRP.t_ply; % tension
32
33 while (isnan(F_MaxStress) || isnan(F_TsaiHill) || isnan(F_TsaiWu) ||
   isnan(F_Hashin)) && iter < max_iter
34
35     iter = iter + 1;
36     F_guess = F_guess + F_step;
37     M = F_guess * L / (4*W);
38     force = [0; 0; 0; M; 0; 0];
39     strain_0 = compliance*force;
40
41     FI_MaxStress = zeros(n_out+n_in,1);
42     FI_TsaiHill = zeros(n_out+n_in,1);
43     FI_TsaiWu = zeros(n_out+n_in,1);
44     FI_Hashin = zeros(n_out+n_in,1);
45
46     % Calculate failure load on outer skin (compression)
47     for k = 1:n_out
48         m = cosd(Panel.layup_out(k));
49         n = sind(Panel.layup_out(k));

```

References

```

50     T = [m^2, n^2, m*n; n^2, m^2, -m*n; -2*m*n, 2*m*n, m^2-n^2];
51
52     % Global strain transformation into local material coordinate
        system
53     strain_z = strain_0(1:3) + z_out(k)*strain_0(4:6);
54     epsilon = T*strain_z;
55
56     % Stress calculation
57     sigma = Matrix.Q * epsilon;
58     sigma1 = sigma(1);
59     sigma2 = sigma(2);
60     tau12 = sigma(3);
61
62     % Failure index calculation
63
64     [FI_MaxStress(k), FI_TsaiHill(k), FI_TsaiWu(k), FI_Hashin(k)]
        = failure_criteria(sigma1, sigma2, tau12, CFRP);
65 end
66
67 % Calculate failure load on inner skin (compression)
68 for k = 1:n_in
69     m = cosd(Panel.layup_in(k));
70     n = sind(Panel.layup_in(k));
71     T = [m^2, n^2, m*n; n^2, m^2, -m*n; -2*m*n, 2*m*n, m^2-n^2];
72
73     % Global strain transformation into local material coordinate
        system
74     strain_z = strain_0(1:3) + z_in(k)*strain_0(4:6);
75     epsilon = T*strain_z;
76
77     % Stress calculation
78     sigma = Matrix.Q * epsilon;
79     sigma1 = sigma(1);
80     sigma2 = sigma(2);
81     tau12 = sigma(3);
82
83     % Failure index calculation
84
85     [FI_MaxStress(n_out+k), FI_TsaiHill(n_out+k),
        FI_TsaiWu(n_out+k), FI_Hashin(n_out+k)] =
        failure_criteria(sigma1, sigma2, tau12, CFRP);
86 end
87
88 FI_MaxStress_max = max(FI_MaxStress);
89 FI_TsaiHill_max = max(FI_TsaiHill);
90 FI_TsaiWu_max = max(FI_TsaiWu);
91 FI_Hashin_max = max(FI_Hashin);
92
93 if isnan(F_MaxStress) && FI_MaxStress_max >= 1
94     F_MaxStress = F_guess;
95 end
96
97 if isnan(F_TsaiHill) && FI_TsaiHill_max >= 1

```

```

98         F_TsaiHill = F_guess;
99     end
100
101     if isnan(F_TsaiWu) && FI_TsaiWu_max >= 1
102         F_TsaiWu = F_guess;
103     end
104
105     if isnan(F_Hashin) && FI_Hashin_max >= 1
106         F_Hashin = F_guess;
107     end
108 end
109 n = Tube.n;
110 D = Tube.D;
111 t = Tube.t;
112 F_tube = n * (8*Tube.sigma_y)/(D*L)*(pi/64)*(D^4 - (D-2*t)^4);
113
114 SF_3pb = min(min(min(F_MaxStress, F_TsaiHill), F_TsaiWu), F_Hashin) /
115         F_tube;
116 end

```

Listing A.5: Matlab file ThreePB.m

A.6 Failure Criteria

```

1 function [FI_MaxStress, FI_TsaiHill, FI_TsaiWu, FI_Hashin] =
   failure_criteria(sigma1, sigma2, tau12, CFRP)
2
3 % Maximum Stress failure index calculation
4 FI = zeros(3,1);
5
6 if sigma1 >= 0
7     FI(1) = sigma1 / CFRP.sigma_xt;
8 else
9     FI(1) = -sigma1 / CFRP.sigma_xc;
10 end
11
12 if sigma2 >= 0
13     FI(2) = sigma2 / CFRP.sigma_yt;
14 else
15     FI(2) = -sigma2 / CFRP.sigma_yc;
16 end
17
18 FI(3) = abs(tau12) / CFRP.tau_shear;
19
20 FI_MaxStress = max(FI);
21
22 % Tsai-Hill failure index calculation
23 if sigma1 >= 0
24     sigma_x = CFRP.sigma_xt;
25 else
26     sigma_x = CFRP.sigma_xc;
27 end
28
29 if sigma2 >= 0
30     sigma_y = CFRP.sigma_yt;
31 else
32     sigma_y = CFRP.sigma_yc;
33 end
34
35 FI_TsaiHill = (sigma1 / sigma_x)^2 - (sigma1 * sigma2) / sigma_x^2 +
   (sigma2 / sigma_y)^2 + (tau12 / CFRP.tau_shear)^2;
36
37 % Tsai-Wu failure index calculation
38 F1 = 1/CFRP.sigma_xt - 1/CFRP.sigma_xc;
39 F2 = 1/CFRP.sigma_yt - 1/CFRP.sigma_yc;
40 F11 = 1/(CFRP.sigma_xt * CFRP.sigma_xc);
41 F22 = 1/(CFRP.sigma_yt * CFRP.sigma_yc);
42 F66 = 1/CFRP.tau_shear^2;
43 F12 = -0.5 * sqrt(F11 * F22);
44
45 FI_TsaiWu = F1*sigma1 + F2*sigma2 + F11*sigma1^2 + F22*sigma2^2 +
   F66*tau12^2 + 2*F12*sigma1*sigma2;
46
47 % Hashin failure index calculation
48 if sigma1 >= 0

```



```
49     FI_long = (sigma1 / CFRP.sigma_xt)^2 + (tau12 / CFRP.tau_shear)^2;
50 else
51     FI_long = (sigma1 / CFRP.sigma_xc)^2;
52 end
53
54 if sigma2 >= 0
55     FI_trans = (sigma2 / CFRP.sigma_yt)^2 + (tau12 / CFRP.tau_shear)^2;
56 else
57     FI_trans = (sigma2 / CFRP.sigma_yc)^2;
58 end
59
60 FI_Hashin = max(FI_long, FI_trans);
61
62 end
```

Listing A.6: Matlab file failure_criteria.m

A.7 Energy Absorption Estimation

```
1 function [U_MaxStress, U_TsaiHill, U_TsaiWu, U_Hashin, SF_U] = energy(F,  
    EI_panel, Tube)  
2  
3 L = 0.400; %m, span length  
4 U_MaxStress = (F.MaxStress^2 * L^3) / (96 * EI_panel);  
5 U_TsaiHill = (F.TsaiHill^2 * L^3) / (96 * EI_panel);  
6 U_TsaiWu = (F.TsaiWu^2 * L^3) / (96 * EI_panel);  
7 U_Hashin = (F.Hashin^2 * L^3) / (96 * EI_panel);  
8  
9 if Tube.energy == 0  
10     SF_U = NaN;  
11 else  
12     SF_U = min(min(min(U_MaxStress, U_TsaiHill), U_TsaiWu), U_Hashin) /  
        Tube.energy;  
13 end  
14  
15 end
```

Listing A.7: Matlab file energy.m

A.8 Perimeter Shear Strength

```
1 function [F_per_shear, SF_ps] = perimeter_shear(CFRP, Panel, Tube)
2
3 n_out = length(Panel.layup_out);
4 t_out = n_out*CFRP.t_ply;
5 n_in = length(Panel.layup_in);
6 t_in = n_in*CFRP.t_ply;
7 t = min(t_out, t_in);
8 F_per_shear = 2 * (Panel.l + Panel.w) * t * CFRP.ilss;
9
10 if Tube.F_shear == 0
11     SF_ps = NaN;
12 else
13     SF_ps = F_per_shear/Tube.F_shear;
14 end
15
16 end
```

Listing A.8: Matlab file perimeter_shear.m

References

Appendix B

Material Datasheets

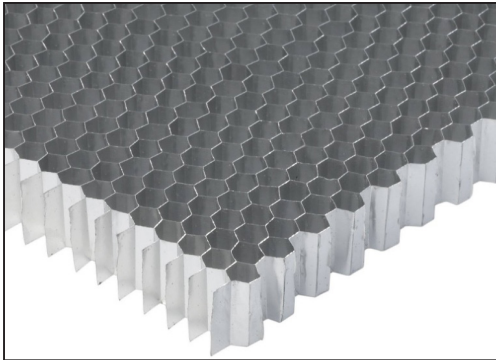
This appendix compiles the technical datasheets of the materials used in the development and manufacture of the sandwich panels and composite structures analysed in this work. The datasheets provide the baseline mechanical and physical properties directly from the suppliers, forming the reference values for both the analytical sizing tools and the finite element simulations presented in the main body of the work.

B.1 Aluminium Honeycomb Core Datasheet

The aluminium honeycomb was used as the lightweight core material in the sandwich panels. This material was employed both in the experimental testing presented in Chapter 4 and in the theoretical panel sizing described in Chapter 5. Its datasheet provides the mechanical properties adopted in this study.



CORE MATERIAL ALUMINIUM HONEYCOMB



Key Features

- High Shear Strength
- High Compressive Strength
- Low Density
- Corrosion & Temperature Resistant
- Accurate Thickness Tolerance

Specification & Properties

The following table lists the specification and properties for the different cell sizes of honeycomb we supply.

Property	1/8" (3.2mm)	1/4" (6.4mm)	3/4" (19.1mm)
Nominal Density	72.1 kg/m ³	83.3kg/m ³	28.8kg/m ³
Aluminium Series	5052	3003	3003
Foil Thickness (micron)	35	70	50
Perforated?	No	Yes	Yes
Corrosion Treated	No	Yes	Yes
Comp Strength - Bare	539 psi	625 psi	115 psi
Comp Strength - Stabilised	559 psi	655 psi	125 psi
Crush Strength	255 psi	235 psi	40 psi
Plate Shear (L dir.) Strength	340 psi	360 psi	95 psi
Plate Shear (L dir.) Modulus	70 ksi	65 ksi	22 ksi
Plate Shear (W dir.) Strength	220 psi	210 psi	60 psi
Plate Shear (W dir.) Modulus	31 ksi	35 ksi	10 ksi
Max Service Temperature	120 °C	120 °C	120 °C

Product Description

High performance aluminium honeycomb core exactly as supplied to the world's top composites engineering, aerospace and motorsport manufacturers. Aluminium honeycomb core is one of the most widely used high performance honeycomb materials; chosen for its excellent strength to weight ratio (the best of all core materials) and bonding characteristics.

Aluminium honeycomb is used extensively in the motorsport, marine and aerospace industries and increasing now in construction industry where reducing weight whilst maintaining or improving the strength of a composite is of key importance.

In its un-expanded (block) form aluminium honeycomb is also very compact and easy to ship cost effectively and without risk of damage. Un-expanded blocks can be easily expanded into their full sheet form without the need for sophisticated equipment. Sheets are available in blocks which will expand to a full sheet size of 2500mm x 1250mm or to a smaller sheet size of 1250mm x 652mm.

Like all core materials aluminium honeycomb is designed to be sandwiched between two skins of material to create a sandwich panel. By creating distance between the two skins and ensuring that these skins cannot move relative to each other great mechanical advantage is gained making for very stiff panels with only minimal increase in weight.

Recommended Uses

Aluminium honeycomb is used in a wide range of applications, including:

- Motorsport (monocoques, splitters, floors, diffusers, crash cells)
- Aviation (bulkheads, dashboards, floors)
- Marine (decks, bulkheads, dashboards)
- Renewable energy (turbine housings, blade construction)
- Architectural (panels, doors, floors)
- Mass transit
- Crash attenuation (impact cells for crash tests, crumple zones)
- Air straighteners (wind tunnels)
- Laser cutting beds

Expanding the Honeycomb

IMPORTANT - wear gloves before handling the block.

Commonly Aluminium Honeycomb is supplied as an unexpanded block. This is for two reasons, firstly because expanded honeycomb would be expensive to ship, and secondly because in its expanded un-stabilised form, the honeycomb is very susceptible to damage and would unlikely survive transport intact.

In large scale manufacturing, a specialist expanding machine is used to expand the honeycomb. However, for smaller applications and one off jobs, such machinery is prohibitively expensive and large thus an alternative method is needed.

Equipment needed

You will need 2 wooden battens, both as wide as the sheet you are expanding and stiff enough not to flex too much. 2" by 4" wooden battens are usually ideal and easily obtained. You will also need a bag of wood nails. They need to be long enough so that they can go through both the wooden batten and still protrude enough to go through the honeycomb.

For cutting honeycomb you need a steel ruler and a sharp Stanley type knife with a blade long enough to cut through the honeycomb to its full thickness in one go.

Aluminium Honeycomb

Making tools to aid expansion

Take one of the battens and mark out a series of dots on the wood along the battens length spaced at approximately 50mm intervals.

Once marked, hammer the nails through the dots marked on the batten (taking care nothing is underneath!). The final result should be a batten with regular nails poking through.

Repeat for the second batten so you have two identical battens with nails sticking out.

You now just need a flat surface to work on and an assistant to help you expand the honeycomb.

Expanding the honeycomb

The first step is to lay the honeycomb onto a flat surface and by hand tease out gently the first few cells. Take care as you work across the sheet to ensure you do not pull too much out in one go as otherwise the honeycomb will kink making it harder to work.

The reason for expanding the first few cells by hand is to give the nails enough material to pull out the sheet without just ripping through the cells at the edge. The sheets are deliberately cut oversize to allow a little waste at each end due to expansion.

Repeat on the other side of the honeycomb.

At this stage you will need an assistant. Insert the batten tools into each side of the honeycomb. With one person holding each batten, slowly apply tension to expand the honeycomb.

If you find it is just ripping through the cells you have teased out by hand, then stop, pull out a few more cells by hand then start the process again. Expect the cells at the edge to distort and stretch as the sheet expands.

As you build up tension the honeycomb will begin to expand. Be careful you don't put on too much force and distort excessively the honeycomb as it expands. Gentle but firm is the method to use.

Continue to apply the tension to the battens to expand the whole sheet, taking care as it expands to keep the sheet as straight as possible. Once you are happy the honeycomb has been expanded sufficiently, then you can remove the battens and then use the honeycomb for your project.

Cutting the honeycomb

You will need a sharp Stanley type knife and a metal ruler. The blade needs to be long enough to cut the cell foils in one action for the chosen thickness of honeycomb.

The method is relatively simple. Measure the sheet out and use the metal rule as the cutting edge. Working from one side of the planned cut, individually cut through each cell foil from top to bottom using the knife. You need to ensure each foil is fully cut before moving onto the next one, so that when you try to separate the honeycomb, it comes apart neatly and doesn't catch or distort the cells.

Do not attempt to cut across the honeycomb in one go as it is easy to tear and distort the cells as you drag the knife blade across the honeycomb.

Disclaimer

This data is not to be used for specifications. Values listed are for typical properties and should not be considered minimum or maximum.

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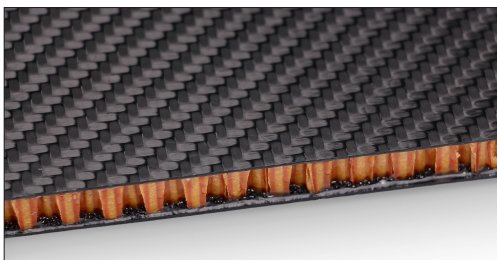
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B.2 Adhesive Film Datasheet

The adhesive film was used to bond the CFRP skins to the aluminium honeycomb core. This material was employed exclusively in the experimental testing presented in Chapter 4. The datasheet presents the properties relevant for this work.



Nomex[®] honeycomb carbon fibre panel made using XA120 prepreg and XA120 adhesive film.

- Industry-leading bond strength
- Service temperature up to 120°C
- Easy handling (4 week out-life)
- Minimal shrinkage, very low CTE
- Compatible with most prepregs

XA120 | PREPREG ADHESIVE FILM

XPREG[®] XA120 is a structural epoxy adhesive film used to provide a high strength bond between composite laminate and core materials such as aluminium honeycomb, Nomex[®] or foam core (such as Rohacell[®]). This adhesive film can also be used to provide additional resin around bonding fasteners and inserts or underneath peel-ply when using peel-ply to provide a secondary bonding surface.

XA120's advanced, toughened epoxy formulation results in outstanding bond and peel-strength at the very forefront of adhesive film performance.

This adhesive film prepreg is suitable for autoclave, oven or heated platen press cure at temperatures from 80°C to 120°C. XA120 cannot be cured at room temperature.

STANDARD SPECIFICATION

XPREG[®] XA120 is held in stock in the following specification:

SKU	Resin Weight (gsm)	Scrim	Width (mm)
XA120-GPL-150(1250)	150	13g Glass	1250

CURING

Recommended initial cure is 8hrs at 90°C (ramp rate 2°C/min). Where the maximum T_g is required, a post cure of 120°C for 1hr should be followed (ramp rate 0.3°C/min).

CURE CYCLES AND RESULTS

Cure	Comments	Duration	T _g
80°C	(minimum)	16 hours	90°C
90°C		8 hours	100°C
100°C		4 hours	110°C
120°C	(maximum)	1 hour	120°C
120°C	Post cure	1 hour	120°C

Cure cycles listed are a guide only and are subject to a number of factors. Ramp rates should not exceed 3°C per minute during initial cure and 0.3°C per minute during post-cure.

TECHNICAL SPECIFICATION

GENERAL PROPERTIES

Cure temperature range	80°C to 120°C
Maximum service temperature	120°C (after post cure)
Out-life (at 20°C)	30 days
Freezer-life (at -18 °C)	12 months
VOC content	Very low (solvent free)

CURED MECHANICAL PROPERTIES

Tests performed on XA120-GPL-150(1250) cured sample.

Test	Standard	Results	Units	Result
Climbing Drum Peel	ASTM D3165	Peel Strength (L)	N/mm ²	437
	ASTM D1781	Peel Strength (L)	N/mm ²	538
DMA	AITM 1-003	T _g Storage Modulus Onset	°C	141
		Tan Peak	°C	150

STORAGE & HANDLING

When not in use, XPREG[®] prepregs should be kept frozen at -18°C (0°F) in sealed plastic packaging. When ready to use, the material should be removed from the freezer and allowed to thaw fully to room temperature before being removed from the packaging.

Remaining material should be re-sealed before returning to the freezer to avoid the risk of moisture uptake.

SAFETY INFORMATION

This material contains uncured epoxy resin which can cause allergic reactions with skin contact. Repeated and prolonged skin contact must be avoided.

Please refer to the product safety data sheet before working with this material.



PROCESSING GUIDELINES

WORKING WITH PEEL-PLY

Most construction methods using a honeycomb core one or both skins of the laminate to be cured first before a second cure to bond the honeycomb to the skins. When following this procedure, peel-ply can be layered onto the laminate where the adhesive film will be used to aid with bonding and eliminate the need for surface preparation. If a peel-ply is not used to provide a bonding surface for the adhesive film then the surface of the laminate should be prepared by keying the surface using a coarse abrasive paper once the laminate has been cured.

The use of peel-ply will absorb some of the resin from the laminate. Testing has shown that peel-ply applied to the reverse (inside) of a laminate made up of three or more plies will not affect the surface finish of the component. Peel-ply applied to the reverse of a laminate of only one or two plies will absorb some of the resin needed for the laminate itself and therefore could locally affect the surface finish (causing pin holing). In this situation, an additional layer of XPREG® XC120 Prepreg Adhesive Film can be positioned under the peel ply at the time of laminating the inner or outer skin in order to provide additional resin content.

Nylon 66 peel-ply such as AeroFilm® PP230 Aero-Grade Nylon 66 Peel-Ply due to the increased ease with which this grade of peel-ply will release from the prepreg.

HONEYCOMB CORE CONSTRUCTION METHODS

2-Shot Cure Process

The simplest construction method for a honeycomb-cored panel or component is a '2-shot' cure. In a 2-shot cure the outer (visible) skin of the panel is laminated and cured first before the adhesive film, core material and inside skin laminate are added to the cured outer skin, vacuum bagged and then oven (or autoclave) cured for a second time.

The advantage to this process is that the component requires only two vacuum bagging and cure cycles and the process does not require a separate mould for the inside skin. The disadvantage however is that the laminate of the inside skin will be pressed into the honeycomb structure as it cures resulting in less flat fibre alignment, known as crimping. Crimped fibres in a laminate can straighten under tension, reducing the tensile modulus of the laminate and therefore the overall stiffness of the completed panel.

To prepare the laminate for bonding to the adhesive film, a peel-ply is usually layered onto the inside of the laminate before vacuum bagging (see notes about peel-ply in previous section).

Once the laminate is cured, the peel ply is removed leaving the bonding surface. The XA120 adhesive film is positioned onto the textured bonding surface left by the peel ply. The honeycomb core (Nomex®, aluminium etc.) is then placed into the adhesive film. To create the inside skin, XA120 adhesive film is positioned over the honeycomb and the remaining plies of prepreg are laid directly onto the uncured adhesive film. The structure is then vacuum bagged and oven (or autoclave) cured to cure the adhesive film and the inside skin prepreg laminate.

3-Shot Cure Process

More sophisticated honeycomb core constructions require a 3-shot cure process whereby the inside and outside skins are laminated and cured separately before then being combined together with the honeycomb core using adhesive film before being vacuum bagged and oven (or autoclave) cured to flow and cure the adhesive.

The advantage to a 3-shot cure is that the inside and outside skins of the panel or component are cured with full consolidating pressure evenly distributed across them. Because the laminate is cured against a flat mould-surface, the fibres remain straight, maximises their mechanical performance. The disadvantage to this method is the requirement for a separate mould for the inside skin and the need for a total of 3 vacuum bagging and cure cycles (although both inside and outside skin components could be cured at the same time).

The bonding surfaces of the cured skins should be prepared in advance using peel-ply or otherwise will require keying with abrasive paper to allow the adhesive film to bond properly.

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B.3 Facesheet CFRP Datasheet

The CFRP fabric was used as the reinforcement for the composite facesheets of the sandwich structure. This material was employed exclusively in the theoretical panel sizing described in Chapter 5. The datasheet provides the reference fibre properties applied in the study.

References



Toray Advanced Composites

Resin System	TC346 High temperature epoxy component prepreg for structural applications within F1 and high performance automotive				
TC346 IM0221 200gsm 2x2T T800 HB 40B 38% RW					
Property	Test Temp	Cure Cycle	Test Orientation	Actual (52.85% Vf)	Normalized (55% Vf)
Tensile Strength (MPa) BS EN ISO 527-4	22	2 hrs @ 180°C @ 2°C/min 90psi AC	Warp - 0°	985.00	1025.07
Tensile Modulus (GPa) BS EN ISO 527-4	22	2 hrs @ 180°C @ 2°C/min 90psi AC	Warp - 0°	74.40	77.43
Poissons Ratio BS EN ISO 527-4	22	2 hrs @ 180°C @ 2°C/min 90psi AC	Warp - 0°	0.04	
Tensile Strength (MPa) BS EN ISO 527-4	22	2 hrs @ 180°C @ 2°C/min 90psi AC	Weft - 90°	928.00	965.75
Tensile Modulus (GPa) BS EN ISO 527-4	22	2 hrs @ 180°C @ 2°C/min 90psi AC	Weft - 90°	71.10	73.99
Poissons Ratio BS EN ISO 527-4	22	2 hrs @ 180°C @ 2°C/min 90psi AC	Weft - 90°	0.04	
Compression Strength (MPa) pr EN 2850 (Type B)	22	2 hrs @ 180°C @ 2°C/min 90psi AC	Warp - 0°	841.00	875.21
Compression Modulus (GPa) pr EN 2850 (Type B)	22	2 hrs @ 180°C @ 2°C/min 90psi AC	Warp - 0°	51.30	53.39
Compression Strength (MPa) pr EN 2850 (Type B)	22	2 hrs @ 180°C @ 2°C/min 90psi AC	Weft - 90°	795.00	827.34
Compression Modulus (GPa) pr EN 2850 (Type B)	22	2 hrs @ 180°C @ 2°C/min 90psi AC	Weft - 90°	50.50	52.55
Maximum In Plane Shear Strength (MPa) ASTM D3518M	22	2 hrs @ 180°C @ 2°C/min 90psi AC	±45°	144.00	
In Plane Shear Modulus (GPa) ASTM D3518M	22	2 hrs @ 180°C @ 2°C/min 90psi AC	±45°	4.57	
Poissons Ratio ASTM D3518M	22	2 hrs @ 180°C @ 2°C/min 90psi AC	±45°	0.80	
ILSS(MPa) ASTM D2344M	22	2 hrs @ 180°C @ 2°C/min 90psi AC	Warp - 0°	74.60	
ILSS(MPa) ASTM D2344M	22	2 hrs @ 180°C @ 2°C/min 90psi AC	Weft - 90°	75.10	
GICJ / m2 pr EN 6033	22	2 hrs @ 180°C @ 2°C/min 90psi AC	Warp - 0°	1014.00	
DMA Tg Onset (°C) DMA	22	2 hrs @ 180°C @ 2°C/min 90psi AC	Warp - 0°	206.00	
DMA Peak Tan Delta (°C) DMA	22	2 hrs @ 180°C @ 2°C/min 90psi AC	Warp - 0°	225.00	

B.4 Facesheet Resin Datasheet

The epoxy resin system was used as the matrix material for the composite facesheets. This material was employed exclusively in the theoretical panel sizing described in Chapter 5. Its datasheet contains the properties considered in the analysis and design.

Toray TC346

TORAY
Toray Advanced Composites

PRODUCT DATA SHEET

DESCRIPTION

Toray TC346 is a high temperature epoxy component prepreg developed for structural applications within Formula 1 and high performance automotive. Toray TC346 is designed to deliver enhanced compression properties to manufactured parts while maintaining a good fracture toughness. When exposed to elevated temperature environments, TC346 composites demonstrate excellent retention of mechanical performance.

FEATURES

- ▶ **T_g 208°C (406°F) after 180°C (356°F) cure cycle**
- ▶ **Structural system—mechanical performance optimized for compression, short-beam shear, fracture toughness**
- ▶ **Excellent retention of properties at elevated temperatures**
- ▶ **Toughened system—good interlaminar fracture toughness energy achieved without significant compromise of compression properties**
- ▶ **Controlled tack level to aid laminating**
- ▶ **Controlled resin flow during recommended cure cycle**
- ▶ **21-day out life at 20°C (68°F)**
- ▶ **12-month storage life at -18°C (0°F)**

PRODUCT TYPE

135°C (275°F) to 180°C (356°F) Cure High Temperature Epoxy
Component Prepreg

TYPICAL APPLICATIONS

- ▶ Structural applications within Formula 1 and high performance automotive

SHELF LIFE

Out Life:	21 days at 20°C (68°F)
Storage Life:	12 months at -18°C (0°F)

Out life is the maximum time allowed at room temperature before cure.

To avoid moisture condensation

Following removal from cold storage, allow the prepreg to reach room temperature before opening the polythene bag. Typically, the thaw time for a full roll of material will be 4 to 6 hours.

TYPICAL NEAT RESIN PROPERTIES

Density (ASTM D792)	1.3 g/cm ³ (81.2 lbs/ft ³)
T _g (DMA) after 2 hrs at 180°C (356°F) (ASTM D7028)	Onset: 208°C (406°F); Peak tan δ: 228°C (442°F)
T _g (DMA) after 2 hrs at 200°C (392°F) post cure (ASTM D7028)	Onset: 221°C (430°F); Peak tan δ: 236°C (457°F)
T _g (DMA) after 3 hrs at 135°C (275°F) (ASTM D7028)	Onset: 156°C (313°F); Peak tan δ: 178°C (352°F)



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Toray TC346

TORAY
Toray Advanced Composites

PRODUCT DATA SHEET

TYPICAL LAMINATE PROPERTIES

Intermediate Modulus 6K Carbon 200gsm 2x2 Twill 42% RC			
Property	Method	Test Temperature	
		RTD (Vf 52.6%)*	RTD Normalized to 55% Vf
Tensile Strength 0°	BS EN ISO 527-4	1034 MPa	1081 MPa
Tensile Modulus 0°	BS EN ISO 527-4	72.1 GPa	72.3 GPa
Poisson's Ratio 0°	BS EN ISO 527-4	0.04	-
Tensile Strength 90°	BS EN ISO 527-4	956 MPa	1000 MPa
Tensile Modulus 90°	BS EN ISO 527-4	72.7 GPa	76.0 GPa
Poisson's Ratio 90°	BS EN ISO 527-4	0.04	-
Compression Strength 0°	prEN 2580	910 MPa	952 MPa
Compression Modulus 0°	prEN 2580	65.4 GPa	68.3 GPa
Compression Strength 90°	prEN 2580	866 MPa	906 MPa
Compression Modulus 90°	prEN 2580	62.0 GPa	64.8 GPa
In-Plane Shear Strength	ASTM D 3518 M	148 MPa	-
In-Plane Shear Modulus	ASTM D 3518 M	4.70 GPa	-
Poisson's Ratio	ASTM D 3518 M	0.78	-
Interlaminar Shear Strength 0°	ASTM D 3518 M	105.6 MPa	-
Interlaminar Shear Strength 90°	ASTM D 3518 M	105.7 MPa	-
Mode I Interlaminar Fracture Toughness (G _{IC} Strain Energy Release Rate)	prEN 6033	935 J/m ²	-
IM7 fiber Cured 2 hrs at 180°C (356°F) at 2°C/min 90 psi autoclave cure cycle *Data is average of 2 batches			

Toray TC346



PRODUCT DATA SHEET

TYPICAL LAMINATE PROPERTIES

High Modulus M46JB 12K Carbon 140gsm UD 34% RC			
Property	Method	Test Temperature	
		RTD (VF 62.1%)	RTD Normalized to 60% Vf
Tensile Strength 0°	BS EN ISO 527-4	1906 MPa	1842 MPa
Tensile Modulus 0°	BS EN ISO 527-4	258.9 GPa	250.1 GPa
Poisson's Ratio 0°	BS EN ISO 527-4	0.30	-
Tensile Strength 90°	BS EN ISO 527-4	35 MPa	-
Tensile Modulus 90°	BS EN ISO 527-4	7.4 GPa	-
Poisson's Ratio 90°	BS EN ISO 527-4	0.00	-
Compression Strength 0°	prEN 2580 Type B	1057 MPa	1021 MPa
Compression Modulus 0°	prEN 2580 Type B	229.9 GPa	222.1 GPa
Compression Strength 90°	prEN 2580 Type B	256 MPa	-
Compression Modulus 90°	prEN 2580 Type B	7.7 GPa	-
In-Plane Shear Strength	ASTM D 3518 M	66 MPa	-
In-Plane Shear Modulus	ASTM D 3518 M	4.8 GPa	-
Poisson's Ratio	ASTM D 3518 M	0.80	-
Interlaminar Shear Strength 0°	ASTM D 2344 M	95.4 MPa	-
Flexural Strength 0°	BS EN ISO 14125	1650 MPa	1594 MPa
Flexural Modulus 0°	BS EN ISO 14125	234 GPa	226.1 GPa
Mode I Interlaminar Fracture Toughness (G _{IC} Strain Energy Release Rate)	prEN 6033	193 J/m²	

Cured 2 hrs at 180°C (356°F) at 2°C/min 90 psi autoclave cure cycle

Toray TC346



PRODUCT DATA SHEET

High Modulus M46JB 6K Carbon 200gsm 2x2 Twill 38% RC			
Property	Method	Test Temperature	
		RTD (VF 55.65%)	RTD Normalized to 55% Vf
Tensile Strength 0°	BS EN ISO 527-4	596 MPa	589 MPa
Tensile Modulus 0°	BS EN ISO 527-4	109.6 GPa	108.3 GPa
Poisson's Ratio 0°	BS EN ISO 527-4	0.01	-
Tensile Strength 90°	BS EN ISO 527-4	529 MPa	522.8 MPa
Tensile Modulus 90°	BS EN ISO 527-4	103.7 GPa	102.5 GPa
Poisson's Ratio 90°	BS EN ISO 527-4	0.02	-
Compression Strength 0°	prEN 2580 Type B	526 MPa	519.9 MPa
Compression Modulus 0°	prEN 2580 Type B	101.1 GPa	99.9 GPa
Compression Strength 90°	prEN 2580 Type B	493 MPa	487.2 MPa
Compression Modulus 90°	prEN 2580 Type B	97.1 GPa	96.0 GPa
In-Plane Shear Strength	ASTM D 3518 M	102 MPa	-
In-Plane Shear Modulus	ASTM D 3518 M	4.47 GPa	-
Poisson's Ratio	ASTM D 3518 M	0.90	-
Interlaminar Shear Strength 0°	ASTM D 2344 M	74.3 MPa	-
Interlaminar Shear Strength 90°	ASTM D 2344 M	75.6 GPa	-
Mode I Interlaminar Fracture Toughness (G _{IC} Strain Energy Release Rate)	ASTM D 2344 M	503 J/m ²	-

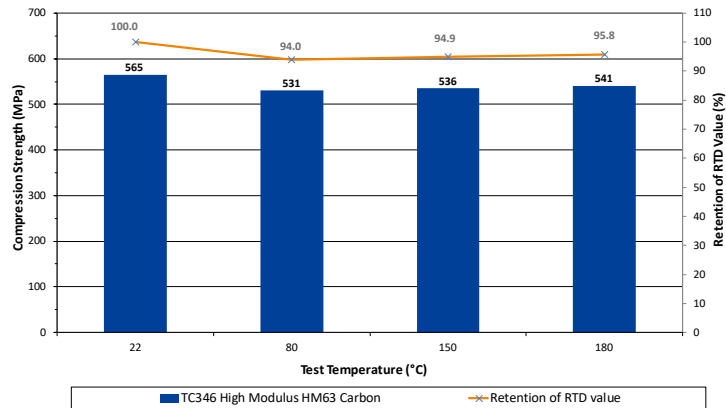
Cured 2 hrs at 180°C (356°F) at 2°C/min 90 psi autoclave cure cycle

RETENTION OF MECHANICAL PROPERTIES AT ELEVATED TEMPERATURES

Elevated Temperature Dry (ETD) Cured 2 Hours at 180°C (356°F) at 2°C/min at 90 psi

ETD Compression Strength vs. Test Temperature

High Modulus 12K Carbon 285gsm SHS 42% RC TC346 Laminate

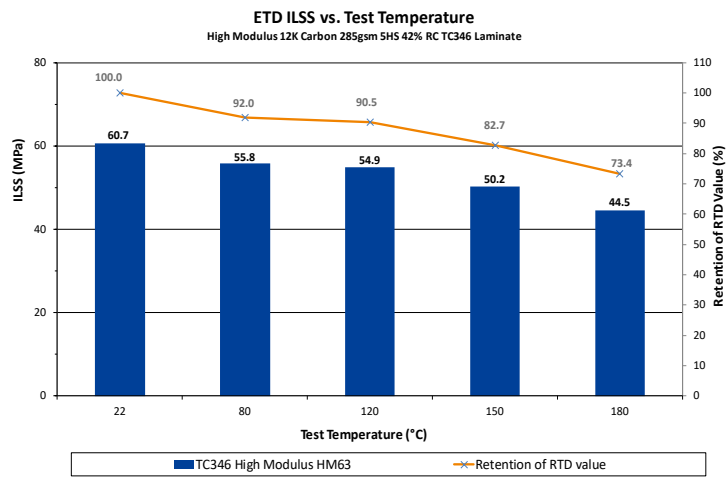


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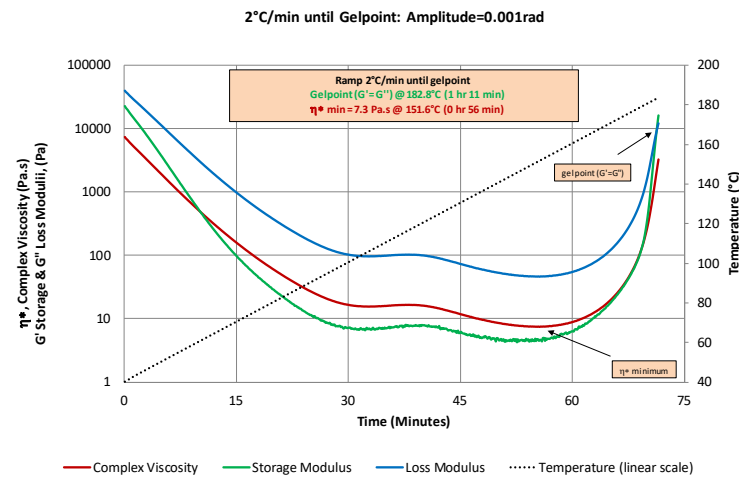
TORAY

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PRODUCT DATA SHEET



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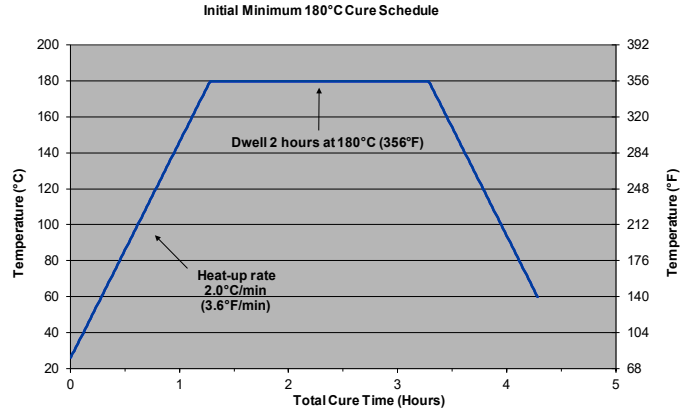


PRODUCT DATA SHEET

TYPICAL CURE PROFILE

180°C (356°F) Cure Temperature with 90 psi Pressure		
Ramp	2.0°C (3.6°F)/min to 180°C (356°F)	Dwell for 2 hours
Cool	2.0°C (3.6°F)/min to 60°C (140°F)	Followed by demold
Total time: 4 hours 30 minutes		

CURE SCHEDULE



A 60-minute dwell at 107°C (225°F) may be also be used.
An alternative cure cycle at 135°C (275°F) may be used followed by a 180°C (356°F) freestanding post cure.

135°C (275°F) Cure Temperature with 90 psi Pressure		
Ramp	2.0°C (3.6°F)/min to 135°C (275°F)	Dwell for 3 hours
Cool	2.0°C (3.6°F)/min to 60°C (140°F)	Followed by demold
Followed by freestanding post cure for 2 hours at 180°C (356°F)		
Total time: 4 hours 40 minutes		

Alternative cure cycles and molding techniques may be employed. Please consult our technical support engineers for advice on your specific cure requirements.

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PRODUCT DATA SHEET

EXOTHERM

In certain circumstances, such as the production of thick section laminates rapid heat-up rates or highly insulating masters, Toray TC346 can undergo exothermic heating leading to rapid temperature rise and component degradation in extreme cases. Where this is likely, a cure incorporating an intermediate dwell is recommended in order to minimize the risk.

HANDLING SAFETY

Observe established precautions for handling epoxy resins and fibrous materials—wear gloves. For further information, refer to the Safety Data Sheet.

PROCESSING

Following removal from refrigerated storage, allow the prepreg to reach room temperature before opening the polythene bag, to avoid moisture condensation. Typically, the thaw time for a full roll of material will be 4 to 6 hours. Cut patterns to size and lay-up the laminate in line with design instructions taking care not to distort the prepreg. If necessary, the tack of the prepreg may be increased by gentle warming with hot air. The lay-up should be vacuum debulked at regular intervals using a P3 (pin pricked) release film on the prepreg surface; a vacuum of 948 mbar (28 in Hg) is applied for 10 minutes.

For autoclave cures, use of a nonperforated release film on the prepreg surface trimmed to within 25–30 mm of the prepreg edge is recommended for the cure cycle and a vacuum bag should be installed using standard techniques.

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