Design and Analysis of a Monocoque Chassis for an Electric Formula SAE Vehicle

by

Oliver MacNeely

Submitted to the
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ABSTRACT

The chassis design of a racecar is fundamental to unlocking the maximum vehicle performance by supporting the applied external loads with a low-mass, high-stiffness structure. Thus far, MIT's Formula SAE team has developed and utilized welded steel tube chassis designs for their racecars due to their relative simplicity and economy.

By taking advantage of the directional stiffness of anisotropic carbon fiber materials, a monocoque design offers a stiffer structure with similar or lower total mass resulting in a high specific stiffness design that out-performs conventional welded designs. This thesis provides a detailed background on chassis design and the necessary design requirements to meet performance targets and competition regulations. Composite laminate designs are proposed and then integrated into a full-chassis structural design that is then analyzed for torsional stiffness in the ANSYS FEM package. Finally, potted insert design and avenues for future development are considered.

Thesis Supervisor: Wim M. van Rees
Title: Assistant Professor of Mechanical Engineering

Acknowledgements

I would like to thank the entire MIT Motorsports team for their support and friendship as well as for the memories I was able to share with them during my time on the team. It was truly an amazing experience that I will cherish forever.

I would also like to thank all the teachers that helped grow me into the engineer I am today, particularly Professor Wim van Rees for his support and patience during this thesis project.

Table of Contents

List of Figures

List of Tables

1 INTRODUCTION

1.1 Thesis Scope

This thesis seeks to provide a foundation for the MIT Motorsports (MIT FSAE) team to advance their race car design development by discussing the fundamentals necessary for designing a composite monocoque. To that end, this thesis will provide the rules-imposed design requirements and testing for a composite chassis. It will then discuss the different performance-based requirements for the chassis, particularly as they relate to the vehicle suspension and dynamics. Following this, this thesis will determine the necessary ply layup schedules to meet these requirements and analyze a proposed composite design in the ANSYS finite element analysis package. It will then investigate the loads experienced by the various inserts that must be attached to the chassis and develop recommended insert designs for those connections. Finally, this thesis will describe the various avenues for additional developmental work.

1.2 Competition Background

Formula SAE (FSAE) is a collegiate engineering competition run by the Society of Automotive Engineers (SAE) where engineering students form teams to design, build, test, and compete with formula-style racecars. This competition operates on a yearly cadence where one year corresponds to a single design, build, and test cycle.

The FSAE competition is primarily sub-divided into two sections – one for electric racecars and one for combustion racecars. In addition to these two sections, there is also a hybrid competition where teams use a combination of electric and conventional combustion technologies, although this competition has yet to achieve the same level of popularity as the purely combustion and purely electric competitions.

Figure 1-1: MIT Motorsports 2019
Competition Car
Photo courtesy of MIT Motorsports

During each competition, the performance of both the car as well as the engineering team is tested in what are known as "static" events and "dynamic" events. Team performance in both types of events is scored using a points system. Final placements in the competition are determined by the total number of points that a team is able to accumulate.

Static events consist of a business presentation, a cost presentation, and a design presentation that are worth 75, 100, and 150 points respectively. In each of these presentations, industry experts will judge the team's work against a stringent set of requirements and score the teams based on how well they've done [1].

Dynamic events consist of several racing-based events: the acceleration event (100 points), the skidpad event (75 points), the autocross event (125 points), and the endurance/efficiency event (375) . In the acceleration event, each team competes to finish a 75-meter straight line course as quickly as possible. This event primarily evaluates the straight-line or longitudinal acceleration of the vehicle. The skidpad event involves completing singular circular loops of a figure-eight track as quickly as possible. In this event, the car is travelling circumferentially for the entire lap, evaluating the lateral acceleration of the car. Finally, the autocross and endurance events utilize courses consisting of multiple straights, constant radius turns, hairpin turns, slaloms, and chicanes. These courses are intended to replicate the typical Grand Prix courses found in the FIA Formula One racing series. For the autocross event, the length of a single lap is regulated to be approximately 0.8km long. The autocross event consists of a single lap of this course and evaluates the entire vehicle performance under the shortest racing conditions. This allows teams to operate at the highest vehicle powers and perform at the very limits of vehicle design. By contrast, the endurance event consists of 22km of continuous racing over multiple repeated laps. This event tests the reliability and efficiency of the cars and their drivers.

1.3 Team Background

MIT Motorsports was founded in 2001 to allow MIT students to engage in this technical competition. From 2001 to 2012, the team focused on the combustion competition, but was never able to achieve a top finish. After 2012, the team decided to pivot from combustion vehicles to electric vehicles as they recognized the future importance of a strong background in electric vehicle design and manufacturing. From 2013 to 2016, the team focused on developing the necessary battery and powertrain skills to enable the design of a high-performance vehicle culminating in a car design that was able to complete all dynamic events by 2016. From 2017 to 2018, the team focused on iteration and tuning resulting in a 5^{th} place competition finish during the 2018 competition – the highest ever achieved by the team.

Figure 1-2: The 2018 design achieved a 5^{th} place finish, the highest in the team's history.
Photo courtesy of MIT Motorsports

Since then, the team has primarily focused on improving electric vehicle performance by increasing available electrical power by developing water-cooled batteries that are more difficult to overheat and by increasing vehicle acceleration performance with a four-wheel drive powertrain.

1.4 Thesis Motivation

One area that the team has yet to investigate thoroughly is the development of a higher performance chassis design. On an FSAE car, the chassis or frame provides mounting features for all of the vehicle hardware, a secure structure for the driver during crash situations, and aerodynamic surfaces to help maximize the downforce generated during racing. In addition, the chassis itself operates as a member of the suspension system since all external and inertial loads are reacted through the chassis and into the ground. As a result, the chassis must fulfill a variety of strength and stiffness requirements, both to maximize the tractive performance of the car to generate the fastest possible lap times as well as to meet the regulations imposed by SAE for all car designs entering the competition [1].

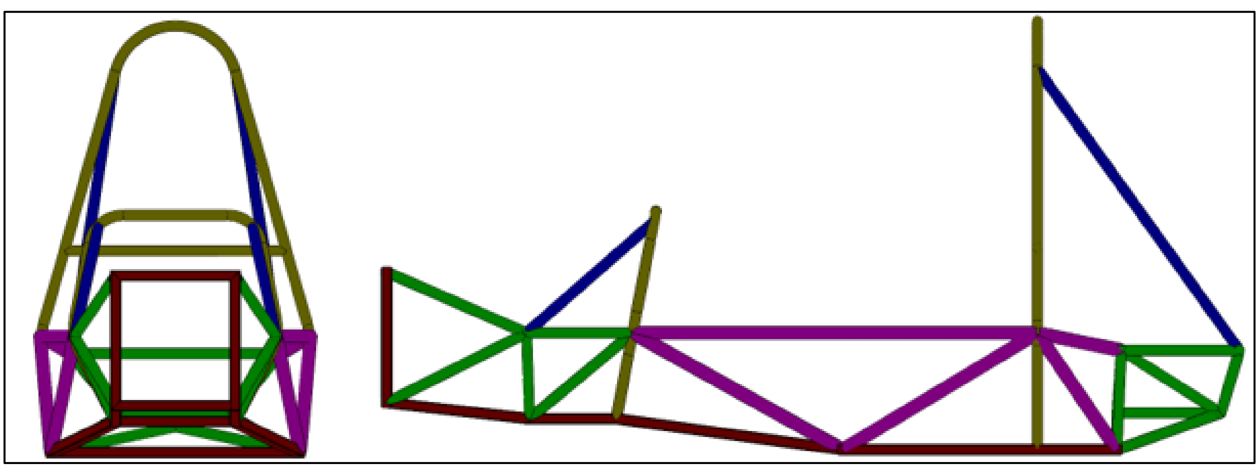


Figure 1-3. Model Year 2019 tube frame chassis design

Standard FSAE chassis designs utilize thin-walled low-alloy chromoly steel tubes that are welded together with a TIG (Tungsten Inert Gas) welding process. For new and experienced teams alike, this design approach has multiple benefits. Steel tubes are low-cost, easy to cut, and forgiving when welded.

Fundamentally, maximizing the performance of the race car is tied to maintaining the fastest average speed around the track. Ignoring aerodynamic effects, which are outside the scope of this thesis, the maximum speed of the car is constrained by the maximum accelerations that the car can achieve on any given track. Since the track design is an uncontrollable variable, maximizing acceleration is dictated by increasing the longitudinal force that the car can apply to the ground and reducing the mass of the car. The maximum longitudinal force that the car can exert on the ground is dependent on the tire design which is out of the scope of this thesis.

In addition, a rear wheel drive powertrain can provide enough power at any given instant to exceed the tractive capability of the tires. This can be surmised from the fact that drivers are constantly at risk of spinning their tires from applying a greater force to the ground than friction is able to react. As a result, increasing acceleration further relies on reducing the vehicle mass. Outside of the battery system, powertrain, and the driver themselves, a steel tube chassis is the single heaviest part on the car, frequently weighing between 30 and 40kg and contributing approximately 12% of the total vehicle mass based on a mass study conducted by Cheyenne Hua '20. Optimizing this large percentage of vehicle mass deserves further attention.

The FSAE rules allow frames to be made from multiple materials such as steel, aluminum, titanium, and magnesium tubing as well as a composite structure. Composite structures offer multiple benefits over welded tube structures.

From a design perspective, a composite structure offers benefits in both strength and stiffness. Fundamentally, the materials in question (carbon, fiberglass, and aramid fibers) have specific tensile stiffnesses and strengths that are six to ten times higher than those of aluminum and steel [2]. In addition, carbon fiber structures are resilient to fatigue failure [2] which is important because of the cyclic loading that the chassis undergoes during a lap. Most importantly, by utilizing multiple plies or layers of material with different fiber orientation, the stiffness and strength of a given composite structure can be designed to meet specific requirements in different locations. This enables each portion of the vehicle to be optimized for the load it experiences.

2 CHASSIS REQUIREMENTS

The engineering requirements for the chassis design are split between performance-based requirements and rules-based requirements. There are several pieces of nomenclature that will be used in the following sections that should be defined. These definitions come from the Version 2.1 of the 2022 Formula SAE Rules [1]. The rules defining the allowed configurations of the chassis and the structural design of the car constitute approximately 30 pages of material. As a result, the most pertinent rules for both monocoque design and for the work completed in this thesis have been distilled and included below.

2.1 Rules-Based Requirements

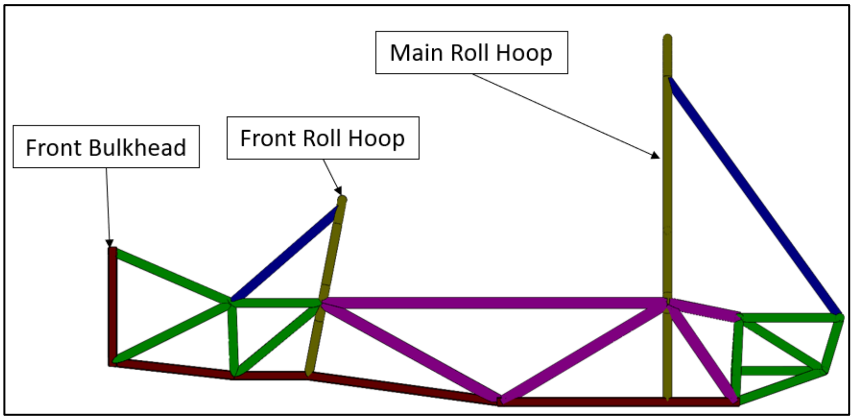


Figure 2-1: SAE nomenclature on MY19 tube chassis

From left to right in Figure 2-1, the *front bulkhead* is a planar structure that provides protection for the driver's feet, the *front (roll) hoop*, is a roll bar located above the driver's legs, in proximity to the steering wheel, and the *main (roll) hoop* is a roll bar located alongside or just behind the driver's torso.

In order to determine whether a student-designed structure follows the rules enforced by SAE, there's a Structural Equivalency Spreadsheet (SES) that each team must fill out and have approved before competing in the competition. This form documents the construction of the chassis to SAE rules and provides a consistent process for determining the equivalency of alternative structures proposed by the teams

In principle, the chassis rules are designed with a steel tube-structure in mind. This is the *baseline* configuration. The rules specify the minimum required tube sizes that the chassis must be built from. For instance, the front roll hoop must be built from a tube with the following properties [1]:

Table 2-1

Min. Area Moment of Inertia	Min. Cross Sectional Area	Min. Outside Diameter	Min. Wall Thickness
11320 mm^4	172mm^2	25.0 mm	1.0 mm

The rules also specify that even with a monocoque chassis design, the front bulkhead, front roll hoop, and main roll hoop must remain steel structures [1]. All other structural portions of the chassis can be formed from alternative materials, which in this case will be a laminate. In order to demonstrate equivalency of a potential composite laminate structure, it must be shown that the proposed composite design is equivalent to the steel tube design in terms of energy dissipation, and yield and ultimate strengths in bending, buckling, and tension.

2.2 Rules-Required Structural Testing

For each ply schedule utilized in the primary structure of the chassis design, several rounds of testing must be performed in order to meet SAE specifications. Firstly, a 3-point bending test must be performed on a representative *flat* panel. The results of this 3-point bending test will be used to derive the stiffness, yield strength, ultimate strength, and absorbed energy properties to meet equivalence with the steel tube structure that it's replacing.

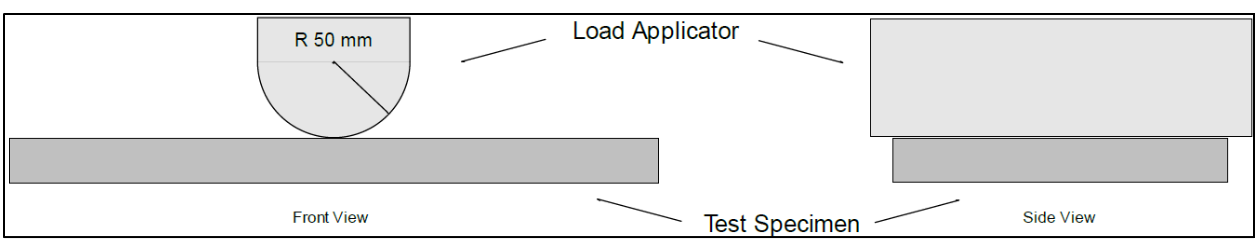


Figure 2-2: 3-point bending setup per FSAE rules [1]

It's important to note that the final cured shape of the composite is not considered in this 3-point bend test. As a result, any stiffness imparted from the geometry of the cured part will not be included in this test or impact the determination of whether the design meets rules requirements.

In addition to the 3-point bend test, a perimeter shear test must also be performed on each laminate. This requires that a 25mm diameter flat punch is pushed through a representative flat panel layup. This test is used to determine (1) the skin shear strength and (2) the maximum force required to drive a 25mm punch through the entire laminate stack up which is mandated by SAE rules (F7.4.3 and F.7.6.5) [1].

When quasi-isotropic laminate schedules, ones that have equivalent mechanical properties in the four main directions - 0°, 45°, -45°, and 90°, are *not* utilized, results from the 3-point bending test will be assigned in the 0° layup direction. Additionally, the properties of all laminates in the 90-degree layup direction must also be tested and all properties must be at least 50% of those in the strongest direction [1]. Due to the additional burden of not using a quasi-isotropic laminate design as well as the manufacturing burdens often associated with using a non-isotropic layup, quasi-isotropic layup designs will be proposed.

For any skin-to-skin or panel-to-panel lap joints, lap joint testing must be performed as well. This testing requires that two separate tensile tests be completed – one where the joint is oriented parallel to the pull direction so that the adhesive joint is tested in pure shear and one where the joint is oriented perpendicular to the pull direction so that the adhesive joint is in tension. These samples must be representative of the of the actual monocoque joints in all ways (primarily composition, dimensions, and overlap). Ultimately, the rules require that the shear strength of the bond must be greater than the ultimate tensile strength of the skin so that any joint failure results in a substrate failure rather than an adhesive failure.

2.3 Area-Specific Rules Requirements

1. The Front Bulkhead laminate, when modeled as an "L" shaped section must have an equivalent EI about both the vertical and lateral axes as the tubes specified for the front bulkhead (F.7.3.1)

2. Any Front Bulkhead that supports the impact attenuator plate must have a perimeter shear strength equivalent to a 1.5mm thick steel plate. (F.7.3.3)

3. The Front Bulkhead Support must have equivalent EI to the six steel tubes that it replaces (F.7.4.1)

4. The EI of the vertical side of the Front Bulkhead support structure must be equivalent to or greater than the EI of one steel tube that it replaces when calculated per F.4.4 (F.7.4.2)

5. The perimeter shear of the monocoque laminate in the Front Bulkhead support structure must be 4 kN or more for a section with a diameter of 25mm (F.7.4.3)

6. The Side Impact Zone must have a Buckling Modulus (EI) equal to the three steel tubes that it replaces. (F.7.6.2)

7. The portion of the Side Impact Zone that is vertically between the upper surface of the floor and 320mm above the lowest point of the upper surface of the floor must have:

8. Buckling Modulus (EI) equivalent to minimum two steel tubes (F.3.2.1.e) per F.4.4

9. Absorbed energy equivalent to minimum two Steel Tubes (F.3.2.1.e)

10. Proof of equivalent absorbed energy is determined by physical testing per F.4.3.1 and F.4.3.3

11. Horizontal floor equivalence must be calculated per F.4.4

12. The perimeter shear strength of the monocoque laminate must be 7.5kN or more for a section with a diameter of 25mm

13. This must be proven by a physical test completed per F.4.3.5 and the results included in the SES

14. Each attachment point between the monocoque and the other Primary Structure must be able to carry a minimum load of 30kN in any direction.

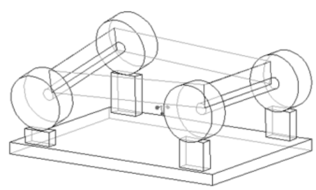
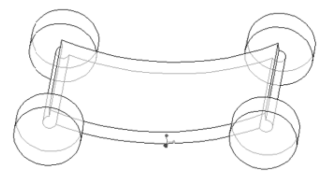
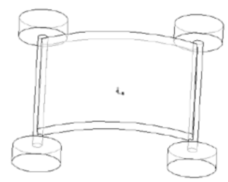
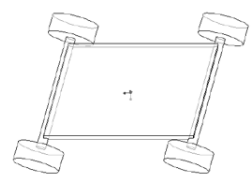
2.4 Performance-Based Requirements

The SAE rules provide significant direction for designing a carbon fiber monocoque structure. Since the manufacturing and assembly quality of FSAE teams is quite poor in comparison to the composites industry as a whole, the rules intentionally err on the side of caution and require students to design layups that are typically conservative for the stresses that they experience during nominal use. By following the rules-required design criteria, students will develop a ply layup that is over-designed for strength, however there are unique stiffness requirements that deserve attention.

While racing, a variety of loads are applied to the vehicle and in turn, the chassis. These loads are primarily split between external loads applied at the contact patch of each tire and body loads from the acceleration of the mass of the vehicle. Since these loads are transmitted through the frame, which is compliant member, the frame will deform.

Based on the layout of the vehicle chassis and the loading conditions of the vehicle, Riley and George [3] described the four primary modes of chassis deformation as longitudinal torsion, vertical bending, lateral bending, and horizontal lozenging:

Table 2-2

Longitudinal Torsion [4]	Vertical Bending [4]
	
Lateral Bending [4]	**Horizontal Lozenging** [4]
	

Longitudinal torsion occurs while the vehicle is turning around a corner or going over a disturbance in the track. During a corner, a lateral acceleration is applied to the center of gravity of the vehicle. Due to the moment arm between the center of gravity and the roll axis/center of stiffness, the lateral load due to body acceleration is resolved into a force and moment term. The moment term excites the longitudinal torsion mode and the load term excites the lateral bending mode.

Vertical bending occurs statically since the mass of the chassis, the driver, and all the components are suspended between the front and rear wheels of the car. This mode can also occur dynamically due to the influence of aerodynamic forces on the car. FSAE cars frequently use airfoils and other aerodynamic devices to increase the amount of downforce on the car to take advantage of increased frictional force between the tires and the road, ultimately enabling the engine to apply more load to the tires and more acceleration to the vehicle before slipping. Due to the variation in speed of the car over the race track, the vertical load applied to the chassis will vary, exciting this deformation mode.

Lastly, the horizontal lozenging mode is exited when opposing loads are applied to the inner wheels compared to the outer wheels. This can occur during a turn as well. While turning around a corner, weight transfer will occur, shifting load off of the outer tires and onto the inner tires. Since the frictional force exerted by the tires is a function of the normal force on the tire, this weight transfer will result in load variation between the inner and outer sets of tires, thereby exciting this deformation mode.

While all four of these deformation modes play a roll in the dynamic performance of the chassis, the longitudinal torsional mode is typically considered the most important of all four because of the impact that this deformation has on cornering dynamics [3].

While completing a steady-state, constant-radius turn around a corner ("cornering"), the car experiences a lateral centrifugal acceleration. This acceleration is applied to the center of gravity of the car, however the suspension design of the car dictates exactly how the car responds to this applied acceleration.

Milliken and Milliken show a typical suspension setup in their work, *Race Car Vehicle Dynamics* [4]*:*

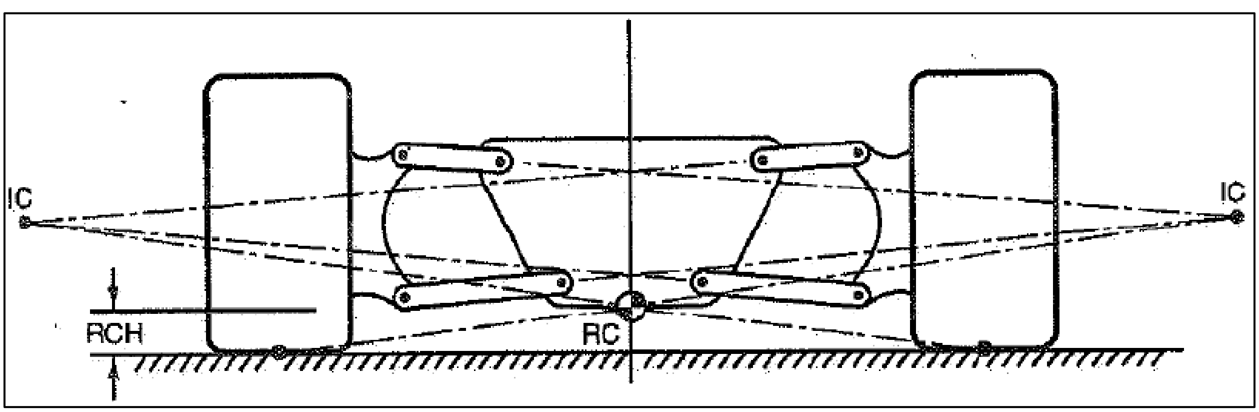


Figure 2-3: Diagram of roll center and suspension setup per Milliken and Milliken [4]

The center of stiffness of each suspension linkage is denoted above as the IC or instant center. The roll center is defined as the intersection of lines drawn from each instant center to the contact patch on the opposite wheel. This is the center of stiffness of the vehicle, at which, if a point load is applied, no rotation will be induced. However, the center of gravity or CG of the vehicle is typically located above the roll center since a large portion of the vehicle's mass is also located above the roll center. This means that there is a moment arm of some magnitude between the roll center and the center of gravity. As described in the previous section, when a lateral acceleration is applied to the center of gravity of the vehicle during a turn, a load determined by Newton's 2nd law is reacted at the CG. However, this results in both a load and moment being reacted at the roll center. This load generates lateral bending of the chassis and the moment induces torsion of the chassis.

This is where chassis stiffness plays a crucial role. If the chassis were very compliant, these loads would significantly distort the geometric shape of the chassis, leading to unpredictable suspension characteristics since the roll center and instant centers of either side of the suspension would be changing dynamically as load is applied. For drivers, this means that the car responds unpredictably when turning into a corner. This lack of predictability impacts the drivers in two ways.

Firstly, it makes driving the car fundamentally less safe since the driver won't be able to accurately predict the outcome of different driving techniques, which, for instance, could result in the driver taking too aggressive of a turn, losing traction, and sliding off the course into a barrier at very high speeds

Secondly, this lack of predictability means a performance loss for the car. Ideally, the driver constantly operates the car at the edge of the traction limit of the vehicle. In the case of turning the car around a corner, this means turning the steering wheel exactly enough to generate the slip angle in the tires that maximizes the lateral force generated by the tires. Since lateral force depends so strongly on slip angle, any error in the driver's judgement results in dramatically less lateral force and acceleration, and therefore a slower lap time. This sensitivity is clear in Figure 6 from *Race Car Vehicle Dynamics* where the lateral load as a function of normal force and slip angle is shown for an example tire compound [4]:

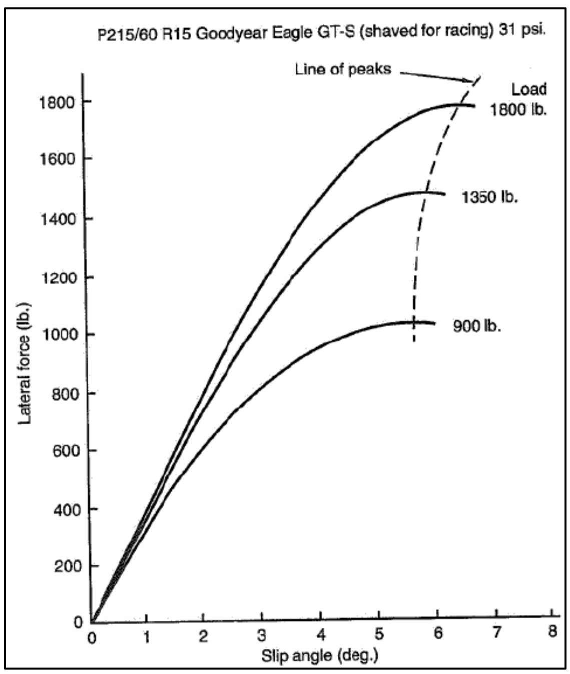


Figure 2-4: Lateral load sensitivity to slip angle and normal force [4]

In addition to dynamic stability and predictability, the chassis stiffness also helps with suspension performance characterization and optimization. Although the car is designed to operate optimally with a certain suspension setup, the final assembly frequently needs additional tuning to compensate for manufacturing and assembly defects as well as individual driver preference.

This tuning occurs by modifying the stiffness of various suspension components to achieve the desired dynamic response. One example of this is tuning the total lateral load transfer distribution or TLLTD of the car. The TLLTD is the ratio of weight transfer between the front portion of the car and the rear portion of the car. As-developed previously, during a turn, the car will transfer load from the outer wheels to the inner wheels due to the moment generated from the offset of the vehicle center of gravity from the vehicle roll center. According to Gaither [5], the optimal rear TLLTD for any given track, assuming a rear wheel drive powertrain, occurs around 0.55 as seen in Figure 2-5:

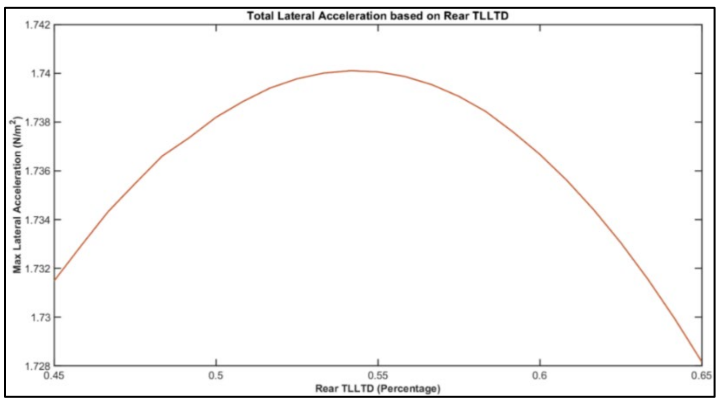


Figure 2-5: Lateral acceleration as a function of rear TLLTD [5].

As the rear TLLTD is increased above 0.55 or reduced below 0.55, oversteering starts to occur due to increased rear tire capability or adverse load transfer decreasing front steering capability, respectively [5]. However, different drivers frequently request modifications to the vehicle TLLTD settings. In order to accomplish this and correct for manufacturing defects, adjustability is built into the vehicle suspension system by varying the spring stiffness of the suspension coil springs and the anti-roll bar (ARB) stiffness which is a tuned torsion spring. Varying the stiffness of both of these springs changes the overall vertical and torsional stiffness of the car to obtain a suspension setup optimized to a particular driver.

However, if the chassis is too compliant, varying the coil spring stiffness and ARB stiffness will not significantly affect the overall torsional and vertical stiffness. This is because the total stiffness of a race car can be approximated by several springs in series: the chassis stiffness, suspension stiffness (coil spring stiffness, ARB stiffness, and control rod stiffness), and tire stiffness [3]. Since these springs are in series, the total equivalent spring stiffness, the vehicle stiffness is dominated by the most compliant spring. Similarly, adjusting this compliant spring will result in the maximum amount of overall vehicle stiffness adjustability, which directly results in the most tuning capability for a given vehicle design. It is therefore necessary that the chassis be substantially stiffer in both vertical bending as well as torsion compared to the suspension system.

Choosing a performance target for the chassis stiffness is a function of the particular vehicle design and expected suspension performance, however MIT FSAE has found that a chassis torsional stiffness of approximately 1530 ft-lbs/deg provided a stiff enough chassis that the car was extremely tunable in addition to deforming no more than 1mm for 90% of an endurance lap. Furthermore, this amount of deformation was found to produce a negligible degradation in lap time due to the movement of suspension points and the resulting change in suspension characteristics and vehicle dynamics [6]. This result agrees well with Riley and George’s conclusion that a torsional stiffness of approximately 1600 ft-lbs/deg has provided Cornell FSAE with a high performing chassis [4]. Moving forward into the laminate design process, a chassis design that meets this 1500-1600 ft-lb/deg torsional stiffness target will be designed.

3 STRUCTURAL DESIGN

In theory, a monocoque could be designed using only plies of resin-impregnated fiber. However, this would result in an extremely heavy and structurally inefficient design since individual plies are extremely compliant when subjected to any out-of-plane loads. In addition, due to their low 2^{nd} area moment of inertia, they're unstable when carrying in-plane compressive loads and buckle easily. Developing a continuous thickness layup capable of meeting the bending stiffness requirements of the chassis would result in an incredibly heavy chassis, negating the advantages of composite materials. To compensate for these characteristics of thin surfaces, plies of material are frequently bonded to either side of a core structure such as aluminum, PVC, or aramid fiber honeycomb. This core structure supports the thin laminates at the surfaces and creates a structure with a significantly larger 2^{nd} area moment of inertia. In turn, this improves the bending stiffness, in-plane buckling resistance, and torsional stiffness of the panel.

3.1 Core Design

In order to determine an optimal layup schedule for the monocoque, we must first determine the core material and thickness. Typical sandwich panel designs often make use of PVC foam, aluminum honeycomb for their core materials [2]. PVC foam is advantageous for its low density and continuous support of the face sheets, but lacks significant in-plane and out-of-plane shear and compressive strength when compared to other materials like aluminum and aramid fibers like Kevlar or Nomex. This makes it a poor choice for the monocoque, especially in the areas where concentrated compressive loading occurs such as where attachments are made for the suspension and for mounting various components. Nomex honeycomb and aluminum honeycomb are both compelling options due to their higher specific strength and stiffness. To decide between these two options, the mechanical properties of two commercially-available Nomex and Aluminum honeycomb manufactured by PLASCORE [7] were compared:

Table 3-1

Mechanical Property	¼" Cell Aluminum Honeycomb (PCGA-XR2-5.2-1/4-30-N-3000)	¼" Cell Nomex Honeycomb (PN1-1/4-3.0)
Density	5.2 lb/ft^3	3.0 lb/ft^3
Compressive Strength	680 psi	229.5 psi
Shear Modulus in the "L" Direction	67000 psi	5700 psi
Shear Modulus in the "W" Direction	37000 psi	3150 psi

Based on these properties, it appears that aluminum honeycomb has a higher specific stiffness in both the L and W directions as well as a higher specific compressive strength. Based on these results, aluminum honeycomb will be considered as the core material for this design. As a baseline, aluminum honeycomb with a cell size of ¼” and a thickness of 1” will be considered. This choice will be expanded-upon the following sections.

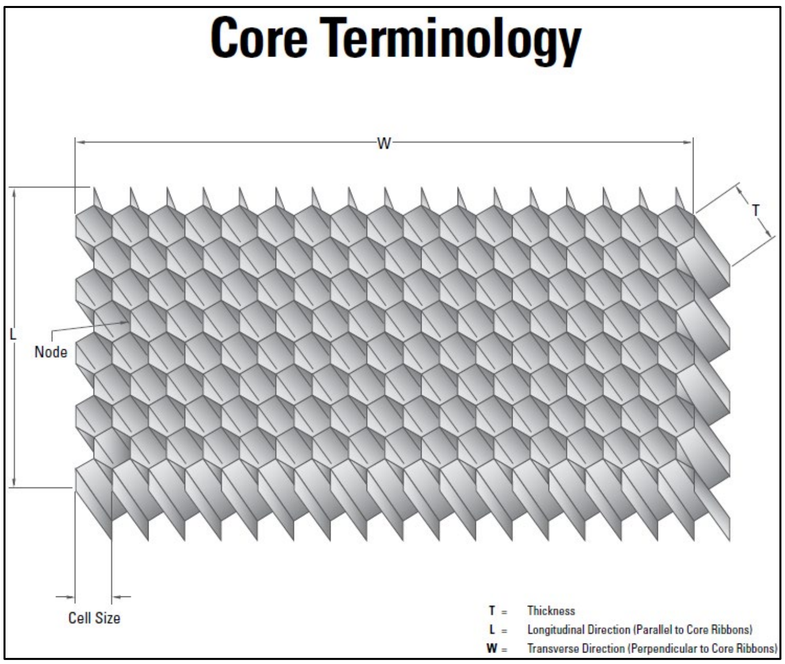


Figure 3-1: directional terminology used when referring to honeycomb core materials. [7]

For the individual laminates, the 2510 prepreg system manufactured by Toray Composite Materials America was considered. While MIT Motorsports hasn’t investigated potential sponsors for composite materials in the quantities necessary for building a monocoque, Toray sponsors multiple FSAE teams by providing them with materials for manufacturing and assembly. By using their materials for this design, the hope is that MIT Motorsports could develop a relationship with Toray for the sponsorship of MIT’s first monocoque design.

The 2510 prepreg system was chosen because it is specifically formulated for processing without an autoclave and can be cured at relatively low temperatures (121C – 132C) [8]. Both of these qualities are advantageous since MIT Motorsports does not possess an autoclave, nor does it have access to one from a sponsor. In addition, the 2510 prepreg system is recommended as a good structural laminate. Since it doesn't feature any specialized high-modulus fibers, it's likely an economical choice as well. For this design, the unidirectional tape in the 2510 prepreg system, P707AG-15, was considered.

3.2 Ply Schedule Requirements

Several guiding principles will be applied for determining the ply schedule of each portion of the chassis. Firstly, the laminates are designed to be quasi-isotropic. Isotropic refers to the quality of having the same properties in all directions. Homogenous materials such as aluminum or steel are considered to be largely isotropic. Since composite laminates are typically manufactured as thin sheets, they will rarely achieve isotropic properties, but they are able to achieve *quasi-isotropic* qualities whereby the stiffness and strength in the primary in-plane directions are equivalent. This requires an equivalent number of fibers (or areal weight) oriented in the two primary in-plane directions. From the SAE rules perspective, a quasi-isotropic layup requires less additional testing which relieves an additional developmental burden from the team.

In addition to requiring that the laminates be quasi-isotropic, the laminates must also be both symmetric and balanced. A symmetric laminate has the same ply orientations above and below the midplane of the laminate. Non-symmetric laminates experience warpage during curing due to differences in coefficients of thermal expansion as well as coupling of the extension and bending deformation modes. A balanced laminate is one in which there is an equal number of plies in both $\pm\theta°$ directions. This eliminates the coupling of the shear and extension modes [2]. By requiring that the proposed laminates are symmetric and balanced we've eliminated unwanted coupling of deformation modes and improved the reliability of our manufacturing process by reducing the change of warpage and deflection during cure.

Lastly, a number of design considerations from Niu [2] shall be incorporated in order to generate a laminate design that follows the best practices of the industry. These are summarized below:

1. Adjacent plies should be oriented with no more than 60° between them
2. Avoid grouping 90° plies to minimize interlaminar shear and normal stresses
3. Maintain a homogenous stacking sequence and avoid grouping of similar plies. If plies must be grouped, avoid grouping more than 5 plies of the same orientation together to minimize edge splitting.
4. Exterior surface plies should be continuous and 45° or -45° to the primary load direction
5. Provide at least 10% of each of the four ply orientations to prevent direct loading of the matrix in any direction.
6. Use at least one group of 45°/90°/-45° lies at the surface of the laminate
7. Use at least 40% ±45° plies to maximize bearing strength in the location of mechanically fastened joints
8. Avoid grouping more than six plies of the same orientation
9. Avoid grouping of 90° plies

3.3 Individual Ply Schedules

A complete chassis design requires the integration of the front and main roll hoops into the composite structure of the chassis. Integrating steel tubing into the monocoque chassis and the joints necessary are out of scope of this thesis, so this analysis will focus exclusively on the continuous composite structure. There are seven sections of the chassis that need ply schedules that are equivalent to the steel tube structure that they replace. These sections are: front bulkhead (FBH), front bulkhead supports (FBHS), side impact structure (SIS), main hoop brace support (MHBS), accumulator side protection (electric vehicle only), tractive side protection (electric vehicle only), and rear impact protection (electric vehicle only). The remaining portions of the chassis will be tuned to meet the stiffness requirements defined previously.

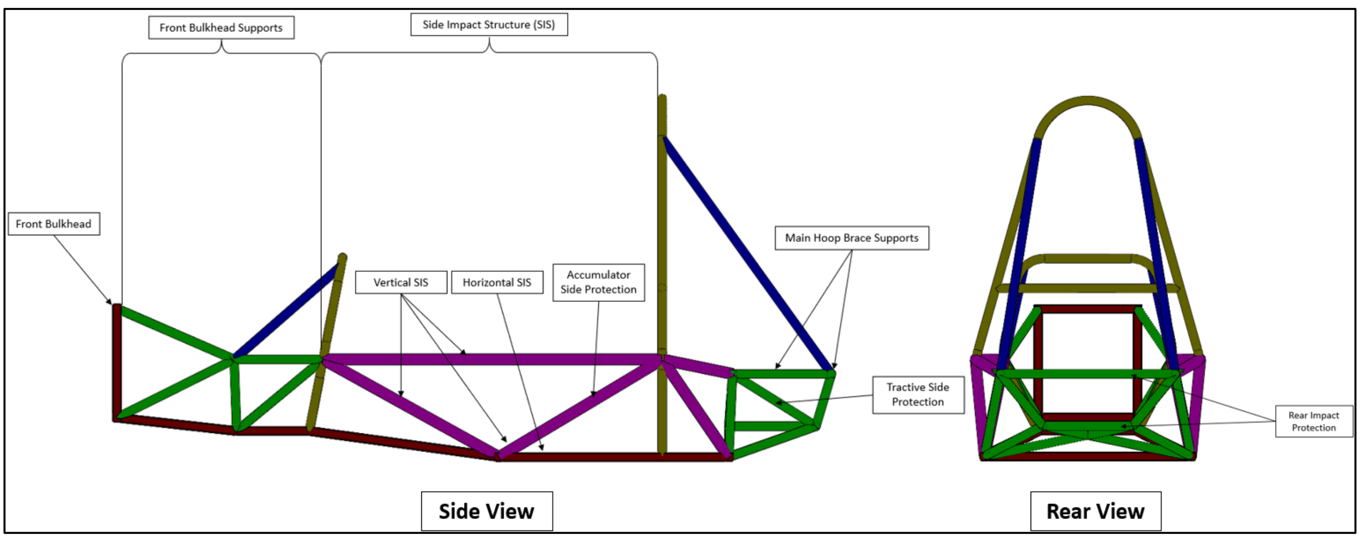


Figure 3-2: Terminology for chassis sections

For each area of the chassis, the standard steel tube structure is provided, along with the calculated flexural rigidity. Since this thesis is focused on a stiffness analysis of the chassis, rather than a strength-based one, ply schedules are designed to match the flexural rigidity of the steel structure as required by the SAE rules. It was found that a single ply schedule is actually capable of providing the necessary stiffness to exceed the equivalent steel structure in flexural rigidity. This is accomplished by judicious usage of the chassis geometry to maximize the 2nd area moment of inertia and increase the flexural rigidity without requiring an increased Young's modulus. This enables the same ply schedule to be used in all parts of the chassis, significantly simplifying manufacturing and assembly. This ply schedule consists of 8 unidirectional plies on either side of 25.4mm thick 3000 series aluminum honeycomb core: [45°/90°/-45°/0°]s. Below, a table shows the required flexural rigidity in each portion of the chassis and the corresponding flexural rigidity of the proposed ply schedule.

Table 3-2

Chassis Section	Required Flexural Rigidity	Calculated Flexural Rigidity (proposed layup)
Front Bulkhead	3400 Nm^2	3478 Nm^2
Front Bulkhead Supports	4020 Nm^2	7547 Nm^2
Horizontal Side Impact Structure	1700 Nm^2	4340 Nm^2
Vertical Side Impact Structure	2680 Nm^2	5232 Nm^2
Main Hoop Brace Supports	4020 Nm^2	6084 Nm^2
Battery Side Protection	1340 Nm^2	6084 Nm^2
Tractive Side Protection	4020 Nm^2	4903 Nm^2
Rear Impact Protection	4020 Nm^2	6680 Nm^2

In Table 3-2, the required flexural rigidity which is the product of Young's Modulus, *E*, and 2nd area moment of inertia, *I*, was calculated by taking the product of the values of *E* and *I* for the necessary number of steel tubes required by the competition regulations. The required values for *E* and *I* are supplied in the regulations in a similar form to Table 2-1.

The calculated flexural rigidity of the proposed composite design was determined with Classical Laminate Theory by first finding the gross laminate elastic constant, E_{11},using the inverse of the [A] matrix as described by Niu [2]. In this case, E_{xx}= E_{yy} since the laminate is quasi-isotropic. After this, the 2nd area moment of inertia of each chassis section was calculated by using section cuts of the chassis surface model in SolidWorks. To be conservative, the section cut was swept along the length of each chassis section and the minimum 2nd area moment of inertia was found and used in this calculation. The product of these two values was used for the calculated flexural rigidity.

Since the calculated flexural rigidities exceed the requirements significantly in some cases, this layup is will be overbuilt to meet the stiffness requirements of the regulations, however exceeding the local stiffness of the equivalent steel structure does not necessarily indicate whether this design meets the global torsional stiffness requirements previously set forth previously. To evaluate this, a 3D model of the monocoque was developed in SolidWorks 2021 and then imported into ANSYS ACP for analysis.

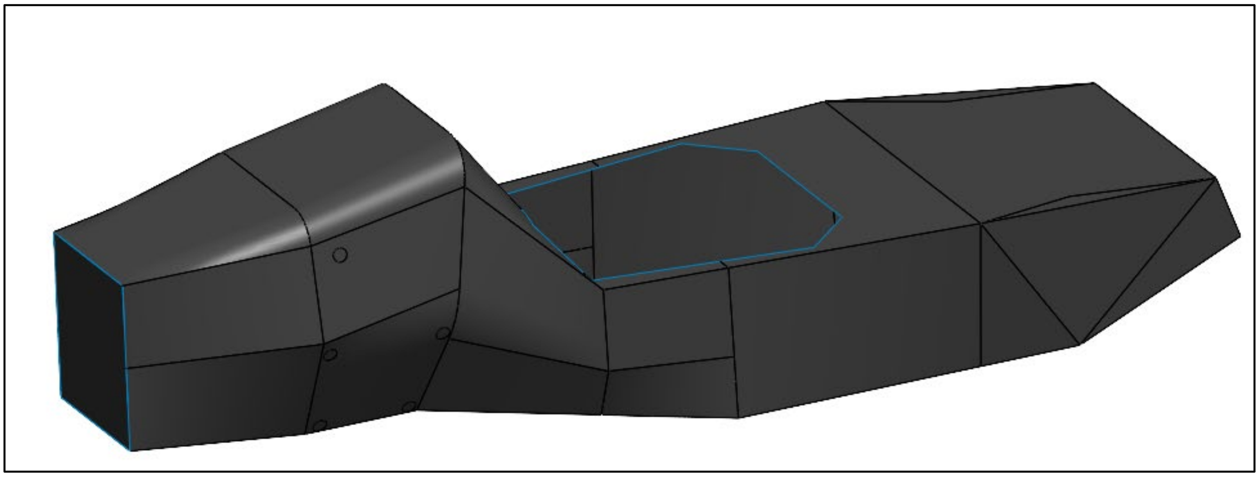


Figure 3-3: Surface model of monocoque structure

3.4 Finite Element Analysis

The surface model was first imported into ANSYS ACP Pre for mesh generation, ply setup, ply orientation, draping, and connecting. An example of one part of the preparation process is shown below in Figure 3-3. Here we can see the laminate thickness as a function of location on the chassis. With a 25.4mm thick core, and approximately 2mm of plies on either side of the core, the total thickness is about 27mm. This varies across the chassis due to the draping algorithm implemented by ANSYS ACP to compensate for the sharp corners and irregular geometry. Although there seems to be a large discrepancy due to the color scale, the thickness variation shown below corresponds to less than 1% of the thickness. Checking the laminate thickness across the model serves to ensure that the ply stack-up is being modeled correctly in ANSYS and matches the 3D design being considered.

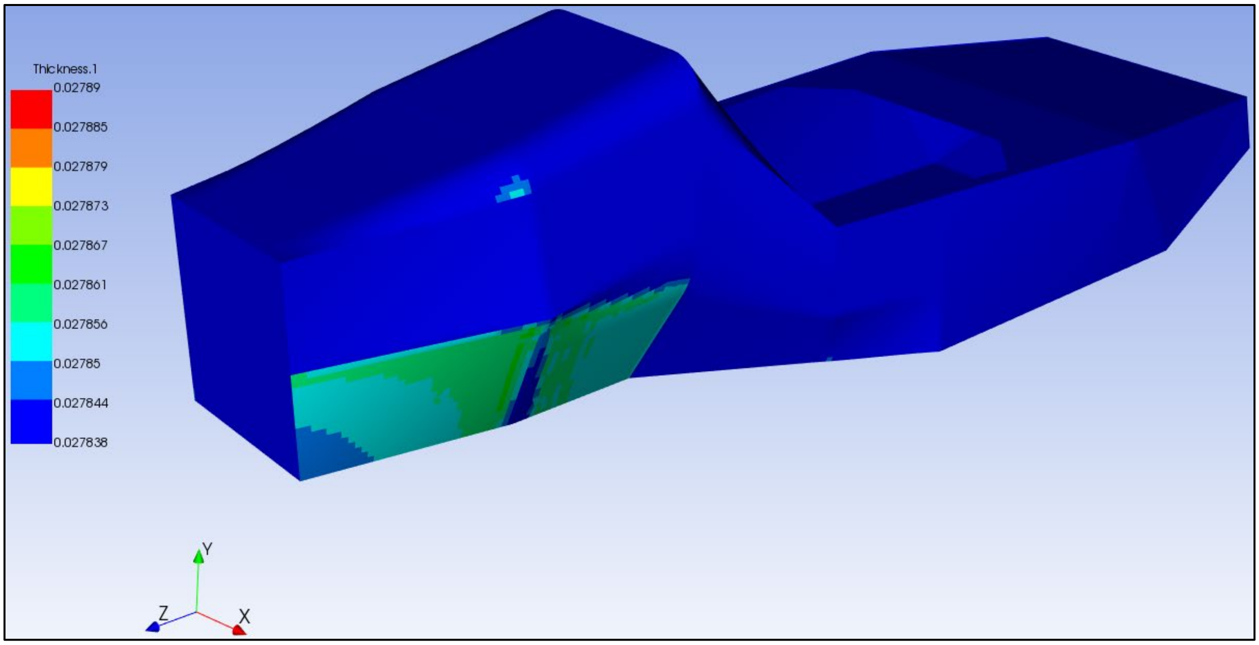


Figure 3-4: Thickness variation in ANSYS ACP Pre

Due to the large size of the model and the nodal limits imposed by an ANSYS student license, a mesh size of 10mm was chosen. Since the goal of this model is to evaluate the global stiffness of the model by applying a load and measuring the displacement, a fine mesh was not needed. Break-out models may be used to achieve high mesh density at areas of stress concentration.

In addition to modeling the curvature of the composite panels that form the monocoque, surfaces were also included to simulate the attachment points where the suspension connects to the chassis. These were only included in the front portion of the chassis since the suspension attachment points in the rear of the car correspond to the intersection points of the composite panels and no additional features were needed to scope loads or boundary conditions in the software.

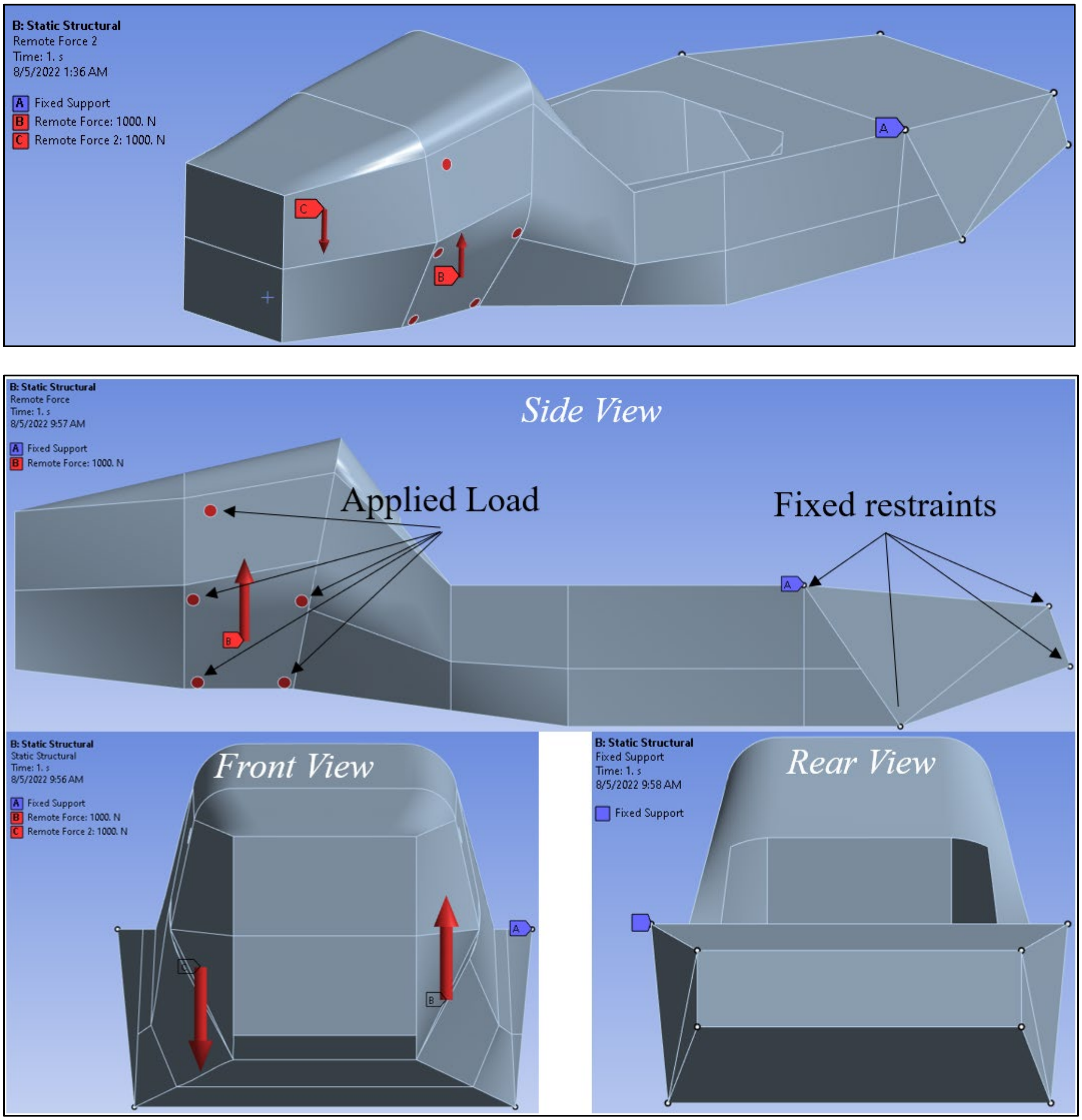


Figure 3-5: Exported shell model with symmetrical boundary conditions in ANSYS Mechanical

Seen above in Figure 3-5, two vertical loads in alternate directions were applied to the front suspension attachment points in red. These loads are symmetric around the vertical mid-plane of the chassis so as to generate a pure moment. 1000 newtons were applied on each side of the chassis to generate a moment of approximately 480 newton-meters. In the rear of the chassis, the suspension attachment points at the corner of the panels were restrained with a fixed support to react the shear and moment being applied to the front. Nodal rotational displacement of the front bulkhead was measured after load was applied. In this way, the torsional stiffness of the chassis can be calculated since the loads were low enough to remain well within the linear regime of structural deformations. This was verified by examining the maximum stress generated in the plies by this loading which was found to be well under 1 MPa, which is significantly lower than the tensile strength of the unidirectional prepreg material which is 2.1 GPa [8]. Since it's known that carbon fiber laminates are largely elastic until failure [2], it's reasonable to assume that this loading kept the structure within the elastic regime.

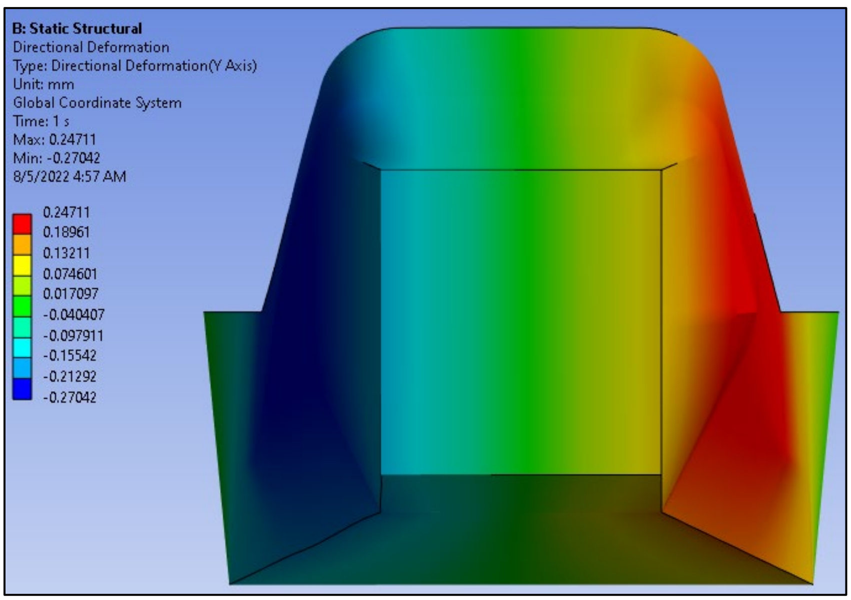


Figure 3-6: Deformed chassis under 480Nm load. Vertical deformation shown in color bar.

When 480Nm was applied to the front of the chassis, the chassis was found to rotate by approximately 0.001 radians. This corresponds to a torsional stiffness of 480,000 $\frac{Nm}{rad}$ or 6075 $\frac{ft-lbs}{deg}$. This is almost a 4x increase over the requirement of 1530 $\frac{ft-lbs}{deg}$. Additionally, this design has a total mass of 18.4kg or 40.5lbs. This is dramatically lighter than recent chassis designs that typically weight 30-40kg. Ultimately this means that the specific stiffness associated with this design is approximately an order of magnitude higher than MIT FSAE's conventional spaceframe designs.

3.5 Potted Inserts and Attachments

While composite structures are remarkably efficient at carrying loads distributed across the structure, they struggle at reacting concentrated loads because of low local stiffness, which then leads to high strains and failure. This is particularly a concern when the face sheets are supported by a honeycomb core which offers no compressive support between cell walls. In order to solve this problem, stiffened hardpoints are inserted into the monocoque in addition to a build-up of plies local to where the concentrated loads are applied. These hardpoints are composed of lightweight metallic materials like titanium and aluminum or stiff engineering plastics like Torlon and PEEK and commonly come in the form of cylindrical inserts with various features (ribs, knurls, flanges, etc) for reacting tensile, compressive, shear, and moment loads.

In order to engage both the shear strength of the core as well as the face sheets, these inserts are adhered into the thickness of a sandwich panel with a potting adhesive. First, the insert is fixtured into place with a jig. This ensures that the orientation and placement of the insert is correct. Orientation is critical for insert installation due to the risk of induced moments from off-axis tensile loading. When the insert is installed at an angle to an applied tensile load, a moment load is induced, which results in peeling action leading to failure earlier than expected due to the high peek stresses generated from peeling. Similarly, if an insert is installed sub-flush to the skin surface and an in-plane shear load is applied, the insert acts as a wedge, delaminating the skin from the core [10].

For an FSAE chassis, the primary sources of concentrated load are the steel tube suspension elements that attach the wheels to the chassis. To capture all six degrees of freedom, six tubes are used to attach each wheel to the chassis. Each of these tubes has a rod-end at either end. This prevents the tubes from carrying bending loads and ensures that they efficiently act as two-force members reacting the loads from the wheels into the chassis entirely as tension and compression. This significantly simplifies the sizing of inserts.

MIT FSAE uses a student-develop MATLAB script in order to determine the maximum loads experienced by each of the individual suspension tubes as a function of steering angle under the worst loading situation. For an FSAE car, the highest suspension loads are typically experienced while braking and turning into a corner [4]. Figure 3-4 displays an example of this most-conservative loading.

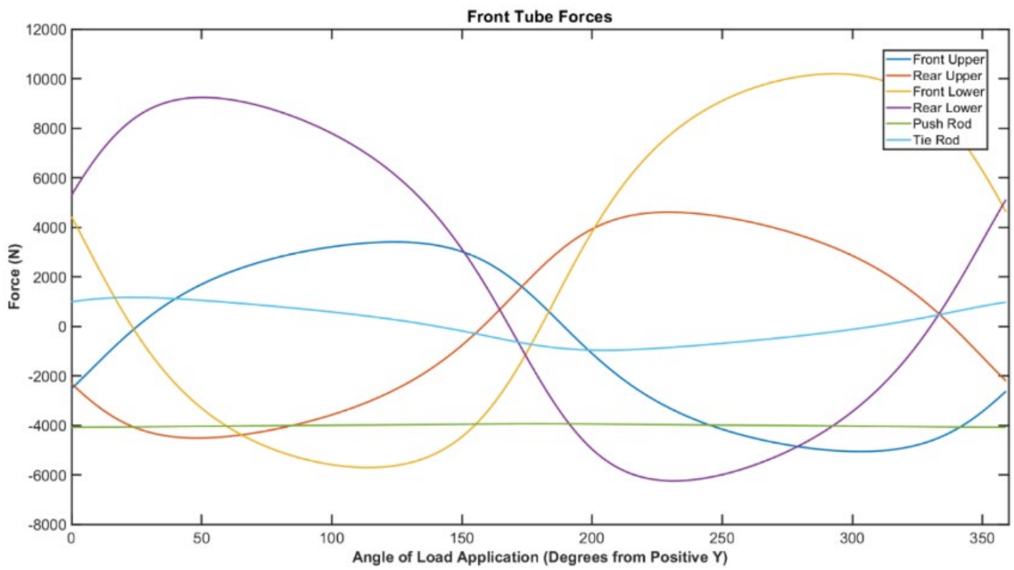


Figure 3-7: Front tube force model [4]

From these results, we can see that the maximum load expected to be reacted through an insert attachment is approximately 10kN. The Insert Design Handbook [10] details the multiple failure modes that should be expected for potted inserts. For the suspension attachments, the tensile and compressive loads will be reacted entirely as shear between the surface of the insert, through the potting, and into the core of the sandwich panel. For this application, 3M's DP460NS was chosen. DP460NS is a high viscosity, high strength epoxy that has a strong heritage on the MIT FSAE team for bonding composites for the aerodynamic components. After a six-hour cure at 49 degrees Celsius, the maximum lap shear strength of this epoxy is expected to be approximately 38 MPa. Testing conducted by the author in 2017 demonstrated that a strength reduction of approximately 12% is expected when comparing bonded joints created by MIT FSAE to the data provided by 3M.

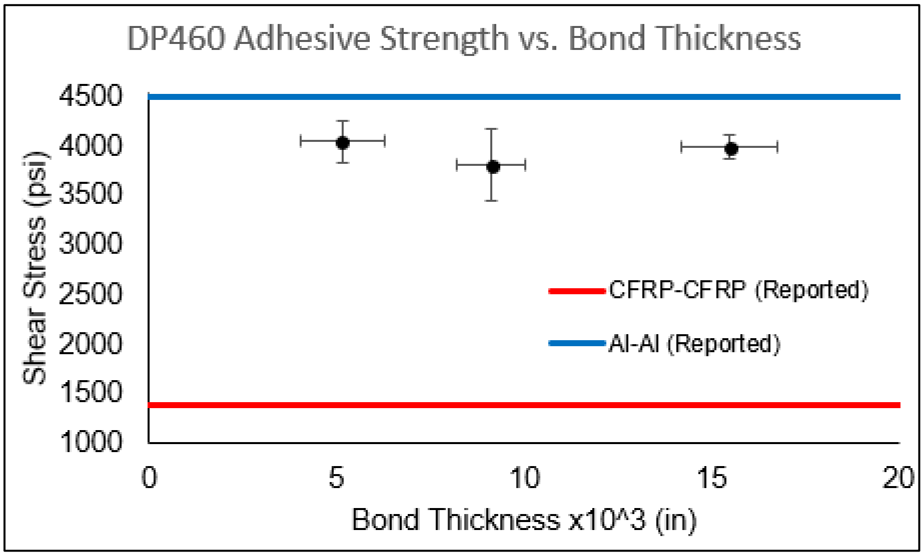


Figure 3-8: DP460NS lap shear strength testing

This results in an allowable shear strength of approximately 29 MPa. For an insert potted into a 25.4mm thick piece of aluminum honeycomb core, a diameter of 12.7mm would yield an adequate surface area to prevent adhesive failure. In this case, the epoxy would experience at most 9.9 MPa, which is well under the limit of 29 MPa. To meet this requirement, the NAS1834 series of inserts, specifically NAS1834-4N-1000 shown below from Clarendon Specialty Fasteners was chosen:


Figure 3-9: NAS1834 Insert, courtesy of Clarendon Specialty Fasteners

For an insert potted entirely through a sandwich panel, shear rupture of the metallic honeycomb foils is an additional concerning failure mode. The Insert Design Handbook, defines the effective shear strength of the metallic core material as

$$\tau_{crit} = 1.36\tau_{w,crit} \text{ [10]}$$

For the ¼" cell size, 1" thick aluminum honeycomb being considered for this monocoque design using the properties provided by PLASCORE in Table 3-1, the resulting effective shear strength of the core is equal to approximately 3.6 MPa. However, the shear stress resulting from a 10 kN tensile load would exceed this strength significantly at 9.9 MPa. As a result, it's expected that reacting these tensile and compressive loads in pure shear would quickly result in the rupture of the core and failure of the bolted joint. To compensate for this low shear strength, it is recommended that backing plates be used on either side of the sandwich panel. These backing plates will spread the load applied by the suspension members across a larger area, reducing the stress at the attachment point. The compressive strength of the core material is 4.7 MPa [7], therefore an aluminum square backing plate with an edge length of 60mm would spread the compressive/tensile load across a large enough area to prevent yielding of the core material. This design would result in a compressive stress of 2.78 MPa, well under the limit of the core in compression. An example of such a joint design is shown below with the insert denoted in orange:

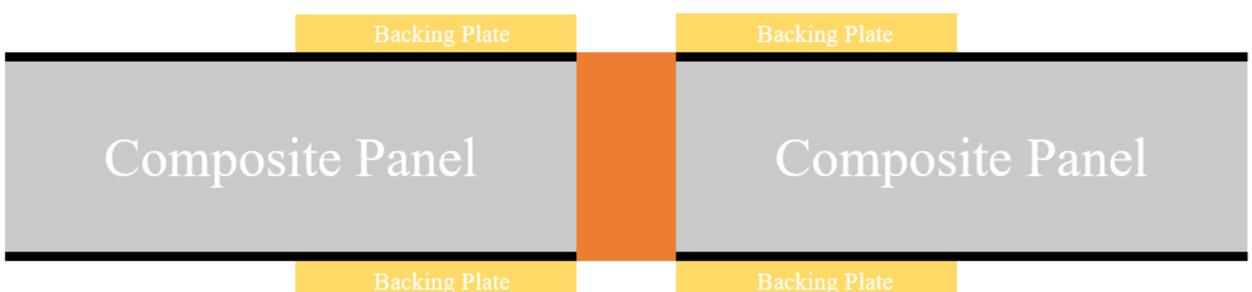


Figure 3-10: Backing plate bolted joint design recommendation

4 CONCLUSION AND FUTURE DEVELOPMENT

4.1 Conclusion

This thesis focused primarily on designing a chassis that meets or exceeds the specific stiffness of the current steel tube chassis architecture. MIT FSAE's team history, motivations, and background were discussed in addition to an overview of the FSAE competition itself and the regulations mandated therein. The specific rules constraining potential chassis designs were enumerated and followed by a detailed discussion about performance-based requirements, particularly those that stem from vehicle dynamics and suspension characteristics.

Following this, a carbon fiber sandwich panel monocoque design was proposed and demonstrated to exceed the area-specific flexural rigidity regulations. This design was then pre-processed in ANSYS ACP Pre and then evaluated for torsional stiffness in ANSYS Mechanical using the static structural package. It was found that a torsional stiffness of 6075 $\frac{ft-lbs}{deg}$ was achieved, exceeding the performance target of 1500-1600 $\frac{ft-lbs}{deg}$ by approximately a factor of six. It was also determined that the proposed design would weight approximately 40lbs, achieving a specific stiffness almost an order of magnitude higher than that of the current steel tube design. Finally, a brief section on potted insets, attachments, and externally applied suspension loads was included for completeness.

However, there is still a significant amount of work required to achieve the goal of building MIT FSAE's first composite monocoque chassis. This work is largely grouped into the following: strength considerations, testing considerations, and manufacturing considerations.

4.2 Future Development

First and foremost are strength considerations. The strength requirements of the chassis vary considerably from year to year because of the variation in suspension design. Even slight changes to the angle of the suspension control arms can result in dramatically more induced load due to the suspension control arm geometry [4]. The finite element model developed by the author in this work can be further expanded upon and evaluated for laminate and ply failure criterion to gauge the strength of the design.

In addition, this thesis has treated the composite structure and design purely analytically. Aerospace and automotive industry, which both use a large amount of composite materials in their production designs, utilize a significant amount of mechanical testing to inform their material allowable and guide their designs. Due to the safety-critical nature of primary structure composites, it's necessary to obtain a large amount of testing data to ensure that the design is manufactured and assembled as intended. This work is likely the most limited by the fact that it was purely analytical and largely ignores failure modes expected in real life due to issues such as poor cure, foreign object debris (FOD), ply and core damage, and inexact resin distribution between plies.

Similarly, the torsional stiffness of the structure was calculated based on a finite element model. MIT FSAE has designed and assembled a torsion jig that enables torsional stiffness measurements of steel tube frames and has been used in the past to validate designs and correlate finite element models [9]. The author highly recommends that the team pursue a similar validation campaign with any composite monocoque design to safeguard the team against unexpected poor chassis performance.

5 REFERENCES

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