EGB415

Chassis Design Assignment

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Table of Contents

Table of Figures

Aim

Formula SAE (FSAE) is an international competition that challenges students to design, build, and race a small formula-style race car. The competition allows students to apply their engineering knowledge and skills to a real-world project. Participants must comprehensively understand various engineering disciplines such as mechanical design, aerodynamics, vehicle dynamics, and manufacturing processes. The competition focuses on the vehicle's performance and the aspects of cost and business presentation, promoting a well-rounded education for the students involved. Formula SAE events allow students to showcase their technical abilities and network with industry professionals and potential employers.

The aim of this report is to design, evaluate, and optimise a formula SAE space frame chassis. The project uses computer-aided software and finite element analysis tools such as SolidWorks and ANSYS to ensure the chassis meets the FSAE rulebook, performance requirements and safety standards. Additionally, the report outlines the client's request for a 1:5 scale model of the final production prototype to be presented at the final presentation. Adhering to the rules outlined in the formula SAE international rulebook is crucial to ensure compliance and eligibility for the competition. The report is a comprehensive guide for developing the formula SAE space frame chassis, detailing the design process, evaluation criteria, and the client's requirements.

This report is divided into three sections. Section one discusses the experimental method, laboratory results, and chassis torsion test evaluation. Section two is design phase one, where the team discusses the initial approach for the initial design phase, the FSAE rules it must adhere to, and a suggested design to carry out refinements. Section three is the final phase, where FEA analysis is conducted, a final design discussion is made, and where verification will be met.

Background & Theory

Space Frame Vs. Monocoque

Upon comparing spaceframe and monocoque chassis designs, the spaceframe emerges as an ideal option for its ease of manufacturing and cost-effectiveness. Spaceframe chassis are constructed using a network of interconnected tubes, providing excellent strength and rigidity while being relatively straightforward to fabricate, as shown in Figure 1. This construction method allows for easier access to materials, making it a more practical choice for manufacturers. Additionally, the cost of producing a spaceframe chassis is generally lower than a monocoque, making it an attractive option for many applications. A spaceframe manufacturing process is less complex than a monocoque, reducing production time and cost. With simpler manufacturing techniques and easier access to materials, the spaceframe chassis presents itself as a more feasible and cost-effective solution for a student-run engineering project.

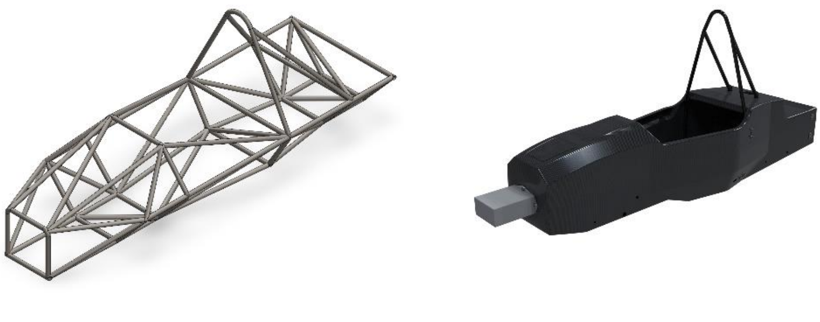


Figure 1 - Spaceframe (Left), Monocoque (Right)

Formula SAE International Rules

The FSAE rule book is a comprehensive guide that governs various aspects of automotive engineering, including stringent requirements for spaceframe chassis design. The rule book outlines specific criteria to ensure safety, performance, and competition standards. The spaceframe chassis design requirements cover structural integrity, material specifications, and dimensional constraints. For example, the rule book mandates using specific materials, such as steel alloys with defined tensile and yield strengths, to ensure the chassis can withstand anticipated loads and impacts. Additionally, this is followed by several dimensional parameters for critical structural elements such as tubing diameter, wall thickness, and joint configurations, which are stipulated to maintain structural integrity while optimising weight and performance.

Furthermore, the FSAE rule book emphasises safety considerations within spaceframe chassis design. This includes provisions for crashworthiness, roll-over protection, and driver safety. The rule book outlines requirements for impact attenuation zones, energy absorption materials, and cockpit protection measures to mitigate injury risks during collisions or rollovers. Moreover, structural integrity testing protocols, including finite element analysis and physical load testing, are required to verify compliance with safety standards. The FSAE rule book is a vital reference for formula student teams, ensuring that spaceframe chassis designs meet rigorous performance, safety, and compliance criteria.

Design Fundamentals

The design of a Formula SAE chassis is governed by several fundamental principles that influence the overall structure and performance of the vehicle. A crucial aspect is the seamless integration of the suspension system within the chassis design. This entails accommodating various suspension components such as control arms, dampers, and springs while ensuring optimal positioning for handling characteristics and vehicle dynamics. Furthermore, the chassis must provide sufficient mounting points and clearances to facilitate suspension adjustments, aligning with the team's tuning objectives. The suspension geometry significantly influences the chassis layout, with considerations for camber, caster, and toe angles affecting both handling and tyre wear. Therefore, the chassis design must balance packaging constraints, suspension kinematics, and overall vehicle performance.

In addition to the suspension system, the Formula SAE chassis must accommodate all relevant infrastructure components, including the powertrain, drivetrain, cooling systems, and driver interface elements. Serving as the backbone of the vehicle, the chassis integrates these systems while ensuring accessibility for maintenance and serviceability. Torsional stiffness is critical to maintain adequate structural integrity and vehicle dynamics performance in chassis design. The chassis can closely mitigate the impact on roll stiffness and enhance handling responsiveness by optimising material selection, cross-sectional geometry, and reinforcement strategies.

In conclusion, a holistic approach to chassis design in Formula SAE encompasses suspension integration, infrastructure accommodation, torsional stiffness optimisation, and aerodynamic refinement to achieve a well-balanced and competitive racing machine.

Experiment

Method

The following steps should be followed to prepare for the chassis torsion testing. Replace all four springs/dampers with solid dummy dampers to directly transmit loads to the chassis. Remove the wheels and attach the torsion rig adapter plates to the hubs. Ensuring the rear plates are rigid while the front plates enable rotation. Secure the plates to the torsion rig. After that, attach the three-square bars to the chassis to serve as measurement points. The bars must be perpendicular to the vehicle's longitudinal axis and situated at the front axle, front roll hoop and rear roll hoop. Secure the bars to the chassis using clamps. Position all dial gauges and angle gauges at specified locations (Figure 3), measuring the distance from each dial gauge to the chassis' centreline. The length of the torsion rig's lever arm, shown in Figure 2, must also be measured for later calculations.

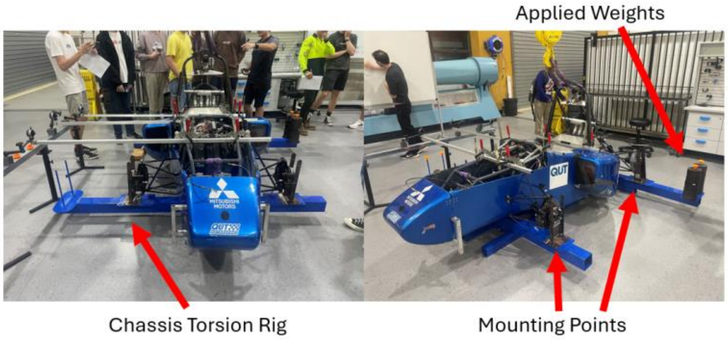


Figure 2 - Preparing for Torsion Testing

A systematic approach during testing is crucial, as the system is highly sensitive to movement and vibrations. Load the maximum applied weight as a counterbalance to minimise disturbance throughout the test. Zero the dial indicators and note any initial values should they deviate. Next, add standard mass increments to the front lever arm. Then, monitor the dial gauges and wait for the readings to stabilise. Record the outputs of the dial gauges for the applied mass. Continue adding mass at desired increments and recording resultant deflection until the maximum mass is reached. Finally, remove the applied mass in the same increments on both the front and rear applicator bars, recording each reduction in deflection until the applied mass is reduced to zero.

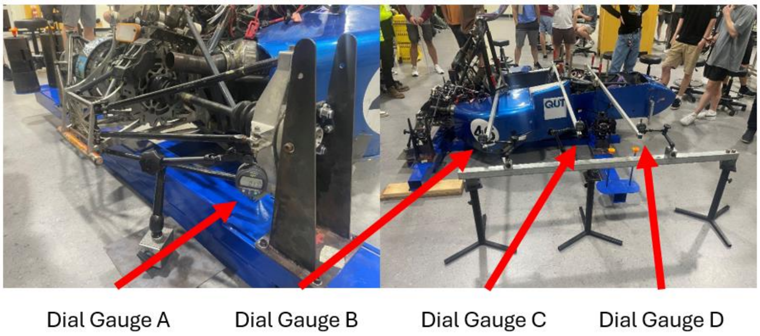


Figure 3 - Dial Gauge Locations

To obtain the relevant experimental outputs, the measured displacement must be converted to angular displacement. For each load, the applied moment is divided by its corresponding angular displacement to obtain torsional stiffness.

Practical Results

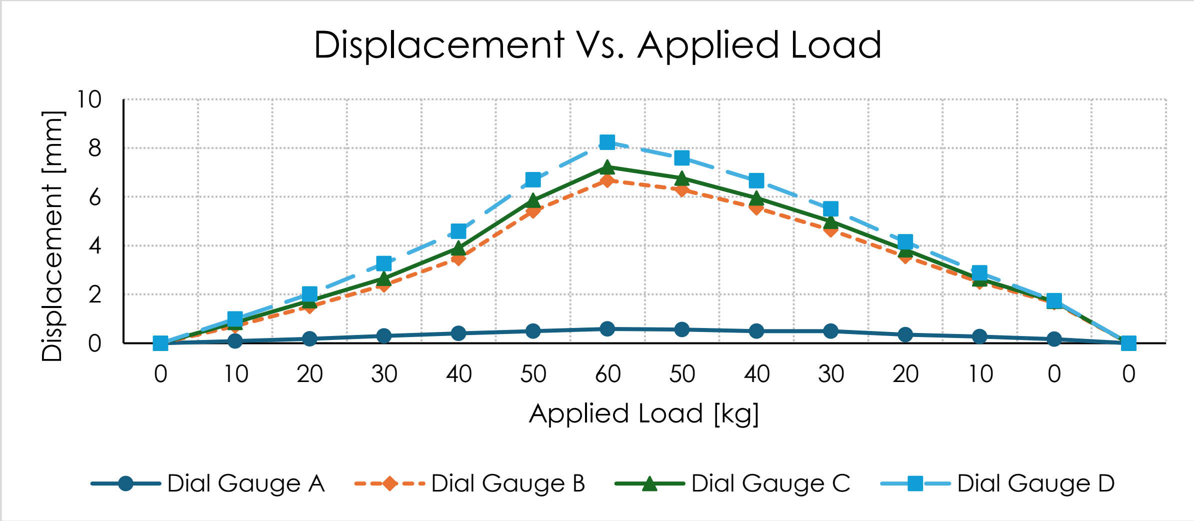


Figure 4 - Displacement Vs. Applied Load

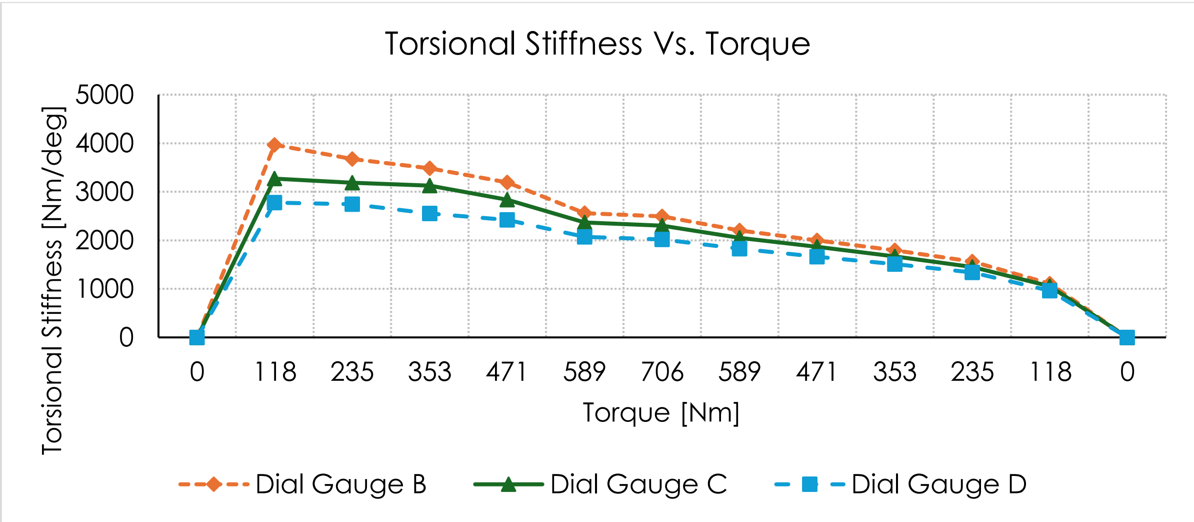


Figure 5 - Torsional Stiffness Vs. Torque

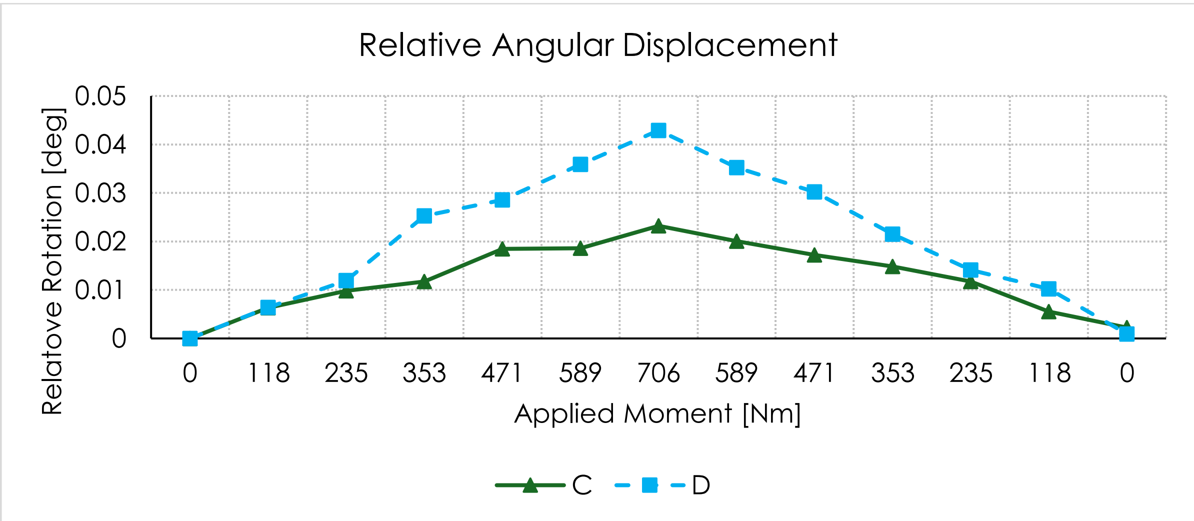


Figure 6 - Relative Angular Displacement

Experiment Discussion

Fundamentally, the test did not measure the torsional stiffness of the chassis itself but the combined stiffness of the chassis and suspension members. Applying a load through suspension members compounds the chassis deflection due to their individual compliances. This is a valuable metric, but it may be attributed to discrepancies in finite element models that do not consider such compliances. Similarly, the multitude of bolts and bearings on the vehicle and associated tolerances further contribute to the deflections observed in the experiment.

Dial gauge 'D' was situated in line with the front axle and is therefore indicative of the 'hub to hub' torsional stiffness. The maximum total vehicle stiffness was observed to be 2742 Nm/deg, when subject to an applied moment of 117.8 Nm. At all measured points, torsional stiffness was observed to decrease in accordance with increasing load. The deflection of each point was non-linear. Total stiffness decreased to 2020 Nm/deg at maximum load.

While performing the unloading sequence, it was obvious that hysteresis was happening. Hysteresis refers to the discrepancy in energy required to generate a given stress in a material or body and the elastic energy stored within that body [1]. This is a consequence of inherent material-damping properties. Energy is dissipated as heat. For an equivalent moment, deflection measured during the unloading of weights was consistently higher than that during loading. All points had a residual deflection of approximately 2mm when all weight had been removed. Consequently, the measured stiffness during unloading was less than that during loading.

Figure 6 depicts the relative angular displacements of the bars situated at the front roll hoop and front axle. Considering a cylinder of constant cross-section subject to torsion, the angle of twist is a linear function of distance from the fixed end. That is, for any random pair of points along the cylinder separated by a fixed distance, the change in angle between these points is constant. Although the bars were not positioned at regular intervals along the chassis' length, a disproportionate change in displacement was observed. It is apparent that the front roll hoop was displaced less, relative to the rear roll hoop, than the front axle was displaced relative to the front roll hoop. This indicates the front portion of the chassis is less stiff than the cockpit. This may be attributed to the reduction in frontal area as the footwell narrows and the reduction in structural members. Rear roll hoop displacement was omitted for reasons identified hereafter.

Design Implications

Understanding a chassis torsional stiffness is critical for accurately designing and tuning a vehicle's suspension system. Lateral load transfer distribution is one of the most powerful tuning tools available to vehicle dynamics, allowing adjustment of the vehicle balance during cornering. Altering this distribution induces torsional loading through the chassis. The chassis acts as an additional spring, connecting the front and rear suspension. If a chassis is sufficiently stiff relative to the suspension, this effect can typically be neglected when defining the vehicle setup. This effect must be considered in the instance of a relatively soft chassis. A soft chassis necessitates a greater change in suspension parameters to achieve an equivalent change in load transfer distribution. This requires a high degree of adjustability, which may not be realistic to implement when designing the car.

Errors & Suggested Improvements

There were numerous potential sources of error owing to the experimental setup and methodology. After conducting the experiment, dial indicators were relocated to measure the deflection of the rig itself. Upon application of an arbitrary load, deflection of the rig's foundation was observed, as shown in Figure 7. This may be the flexure of the torsion rig, a consequence of the soft floor on which it was situated, or a combination of both. Regardless, as the chassis' deflection was measured relative to the ground, any deflection of the torsion rig was included in the results. This may be mitigated by measuring the deflection of the chassis relative to itself. That is, including an additional beam at the rearmost point of the chassis, theoretically stationary as it is behind the fixture. This beam would serve as a reference point, the deflection of which could be subtracted from the remaining points.

Figure 7 - Deflection of Torsion Rig Given, 65kg Applied Mass

Due to their low stiffness, the aluminium bars on which the displacement was measured were observed to deflect under the weight of the dial indicator. Consequently, any stiction of the dial indicators would inhibit movement, impairing the accuracy of results. The method by which these bars were secured to the chassis was also susceptible to variation throughout the experiment. They were each secured only by clamps at two points. This may be mitigated by selecting a stiffer beam and creating dedicated mounting hardware.

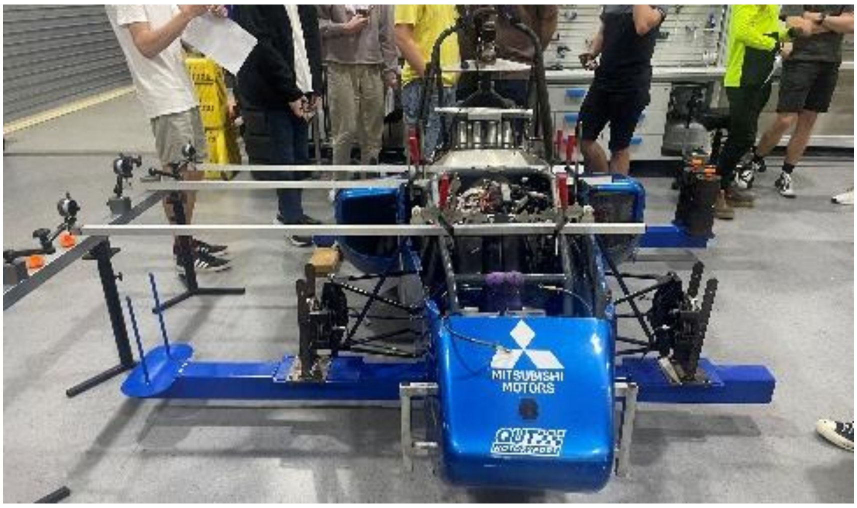


Figure 8 - Fixture of Aluminium Bars to Chassis

The moment arm and all indicated arm lengths were measured imprecisely. Reference points to measure were unclear, and all measurements were taken with a measuring tape. This introduced a degree of uncertainty in lengths used for subsequent calculations. The exact weight of each plate was not measured. There may have been some variance in applied weights as they were not pristine.

1st Design Phase

Multiple chassis designs were created as part of a solution-candidate selection process. These designs followed a standardised methodology influenced by the FSAE regulations. Each design underwent modelling in SolidWorks and finite element analysis (FEA) in Ansys. To facilitate an equitable comparison among the various chassis designs, comparable FEA and structural evaluation procedures were compared in a decision matrix.

Approach

Chassis Proposal 1

First, the pedal box and engine templates were added to the Percy sketch. The pedal box cube was located at the minimum distance in front of Percy's torso to minimise wheelbase and overall chassis length. The main hoop is then located vertically tangent to Percy's head. The engine block was then spaced as close to the main hoop while allowing for the thickness of the chassis tubes. The rear axle centreline was then positioned at its minimum distance behind the engine block. To locate the sketch, the minimum wheelbase was specified from the rear axle to the front axle centreline at the sketch origin. To reduce nose overhang, the front bulkhead plane was positioned at a minimum distance in front of the pedal box allowing for tube thicknesses. Percy was then angled at an appropriate seating angle of 130° to balance ergonomics, height, and length. To aid simplicity and minimise weight, the front hoop was made to be the rear mounting point for the front suspension.

The front bulkhead was then sized to fit the required 250mm cube pedal box with the surrounding tube thicknesses considered. The lower section of the front hoop was specified to the suspension mounting points. The top point of the hoop was located at the horizontal tangent of the bottom of Percy's head to allow enough height to incorporate a steering wheel. The main hoop starts at a regulation lower width, reaching a 95th percentile shoulder width of 600mm. This extends up to the height of Percy's shoulder. The hoop then extends up to a height resulting from them the roll hoop head position requirements. The main hoop braces were positioned at the lowest allowed position to enable a brace angle of 30° while minimising the wheelbase. At this stage, the wheelbase had to be extended by 10mm. The upper width between the braces was made 10% larger than Percy's head to maximise safety. Each radius was the minimal regulation degree. The cross members in the main hoop not only increase the rigidity of the hoop, but also help to separate the cockpit and engine compartments keeping the driver safe. The rear suspension and bulkhead planes were specified to the suspension mount positions. Finally, the front suspension mount plane was designed to the suspension mounts and to intersect the single-piece front hoop brace pair.

From here, each node was braced to maximise strength and force transfer. The required support, brace, and impact members were drawn to the regulation standards. The upper side impact member was located in line with the upper suspension mounts to retain simplicity and minimise out-of-plane forces. The main hoop cross members were drawn to the forward rear suspension mounts to minimise length. Finally, the lower cross members were added to increase stiffness in the large open cockpit and engine compartments.

Chassis Proposal 2

This space frame design differentiated itself from the other designs through its intricate reinforced members. Although this approach reduced manufacturing feasibility, it aimed to strengthen weaker structural components while directing force absorption to the thicker, stronger members. Additionally, the design featured a taller main hoop section, accommodating a more diverse range of drivers. Additional bracing was added to the main hoop, extending to the forward rear suspension mounts to decrease rear deformation.

Diagonal bracing was incorporated into the rear suspension mounts to enhance stiffness. This additional rear bracing is aimed to reduce torsional deformation throughout the chassis. To address the weakest part of the chassis, the cockpit and parallel impact, structural members were installed, connecting the front and main hoop. The front bulkhead was positioned close to the pedal box to minimise the need for additional support members. Triangulated bracing was integrated into the structural members of the front bulkhead and throughout the chassis to improve torsional rigidity.

Chassis Proposal 3

Minimum weight, owing to simplicity of design, was prioritised during the design of this chassis. Care was taken to minimise unnecessary chassis nodes. Where a node was necessitated by a structure such as suspension, it was sought to be utilised by other structures. Examples include the front roll hoop, which is integrated with the rear pickups of the front suspension. Similarly, the main hoop bracing is integrated with the front upper pickups of the rear suspension. Such location of the front roll hoop allowed the driver compartment to move forward, providing ample space in the aft portion of the chassis for the engine. The chassis assumes a reclined seating position to reduce the main hoop height without true ergonomic considerations. Cross bracing is adopted in the floor and between front hoop supports to maintain acceptable torsional stiffness. The front bulkhead structure was the minimum allowable size such that additional support members were not necessary, but footwell template regulations were satisfied.

FSAE Key Rules

Below are the key FSAE rules relevant to this chassis design report. A condensed selection of the rules mentioned can be found in Appendix B.

Table 1 - FSAE Relevant Rules

FSAE Reference Number	Description of Rule
V.1.2	**Wheelbase:** states the minimum wheelbase of the vehicle
F.3.2	**Tubing Requirements:** details the minimum size specification (per F.3.4) for each tubing application such as front bulkhead and main hoop bracing.
F.3.4	**Steel Tubing and Material:** details the minimum size and area requirements for each size specification. It also details the minimum material requirements for the tubing.
F.5.2	**Bent Tubes or Multiple Tubes:** details the requirements for bend radii and additional supports.
F.5.6	**Roll Hoops:** details the driver template and location within the roll hoop shell and triangulation.
F.5.7	**Front Hoop:** details the design and position requirements of the front hoop.
F5.8	**Main Hoop:** details the design and position requirements of the main hoop.
F5.9	**Main Hoop Bracing:** details the design and position requirements of the main hoop bracing.
F6.2	**Front Bulkhead:** supports details the design and position requirements of the bulkhead supports.
F.6.3	**Front Hoop Bracing:** details the design and position requirements of the front hoop bracing.
F.6.4	**Side Impact Structure:** details the design and position requirements of the side impact structure.
F.6.6	**Main Hoop Bracing Supports:** details the design and position requirements of the supports.

Two additional requirements were imposed: accommodation of a previously designed suspension geometry and allowances for the unobstructed insertion and removal of the pedal box and engine block templates within the chassis.

Design Process

Modelling

SolidWorks was used for solid model construction. A standard procedure was adopted for all candidate chassis. The model's origin was set to intercept the chassis floor, car centreline, and front axle centreline for consistency in reference points. The X-Z plane runs parallel to the floor, the X-Y plane aligns with the chassis centreline, and the Y-Z plane is perpendicular to the centreline. Along the front axle centreline, the front and rear suspension mounting points were situated with an upper and lower mounting width of 744 mm and 290 mm, respectively. The lower mounts were positioned 90 mm above the ground plane, while the upper member was placed 250mm above that. These measurements were consistent across all designs. A planar sketching approach was utilised. Planes were created for the front bulkhead, front suspension, front hoop, main hoop, rear suspension, and rear bulkhead. These planes were dimensioned based on distances defined by a 2D sketch of "Percy", and other relevant templates with each dimension individually specified, as shown in Figure 9.

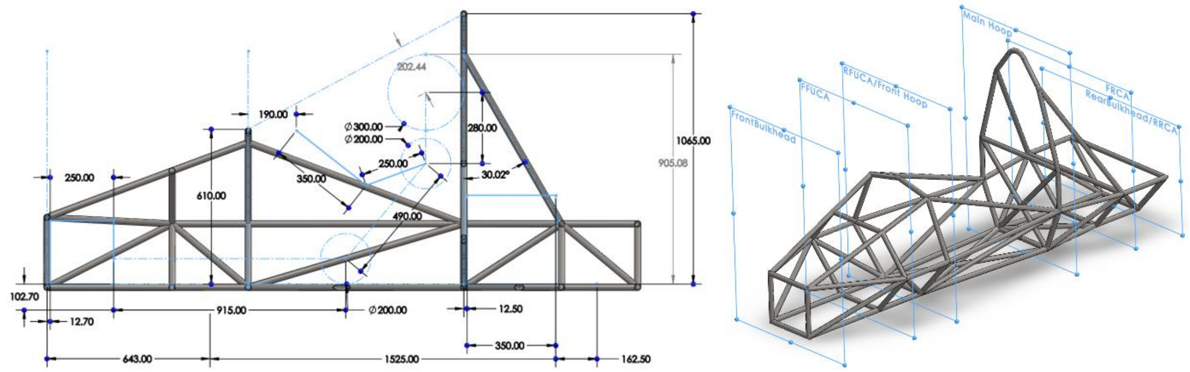


Figure 9 - Initial Modelling Example

Having defined sketches for each planar structure, a 3D sketch was used to connect each structural node. The appropriate cross sections were assigned to each sketch segment using SolidWorks weldment functions. Custom sections were necessary. A simple 2D sketch was created on the front plane in a separate SolidWorks part, where two circles were drawn to the proper dimensions and saved as a library feature part in the weldment profiles folder within the program files directory. After assigning structural members to each chassis sketch in the weldment profile, the trimming option was used to ensure correct intersections at each node. Finally, a material was assigned to the chassis to determine the weight for comparison, as shown in Figure 10. Evaluating a model's mass properties is crucial in understanding how adding more weight or beam members to a build will affect it. This step allows for an in-depth examination of how the structure reacts to changes in weight distribution and additional components.

Mass properties of FSAE space chassis2
Configuration: Default<As Machined>
Coordinate system: -- default --

Density = 0.007900 grams per cubic millimeter

Mass = 29385.031004 grams

Volume = 3719624.177763 cubic millimeters

Surface area = 5478447.212896 square millimeters

Center of mass: (millimeters)
X = -0.477303
Y = 245.901303
Z = -473.887279

Principal axes of inertia and principal moments of inertia: (grams * square millimeters)
Taken at the center of mass.
Ix = (0.000335, -0.014458, 0.999895) Px = 3227506868.946432
Iy = (0.864807, 0.502055, 0.006969) Py = 13076508131.571707
Iz = (-0.502104, 0.864715, 0.012671) Pz = 13089544496.318550

Figure 10 - Mass Properties Model Example

Preliminary Designs

Below are three tables illustrating the three preliminary chassis designs, their rule compliance and the decision matrix used to select a chassis.

Table 2 - Preliminary Chassis Designs

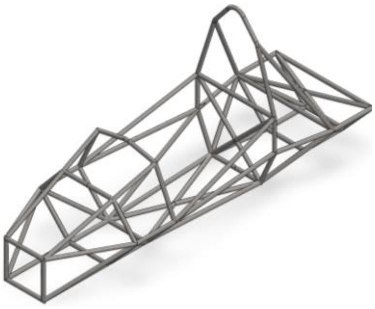
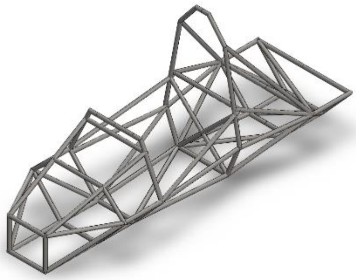
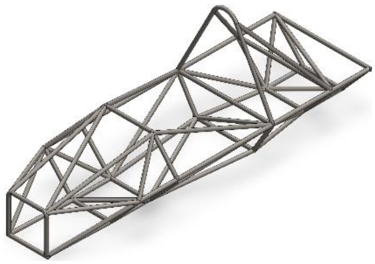
Chassis Proposal 1	Chassis Proposal 2	Chassis Proposal 3
		

Table 3 - FSAE Rule Compliance for Each Chassis Proposal

FSAE Rule	Chassis 1	Chassis 2	Chassis 3
Tubing Requirements and Material (F.3.2, F.3.4)			
Bent Tubes: Radius, angle (F.5.2)			Fail
Roll Hoops: Driver position (F.5.6)			
Front Hoop: Steering position, inclination (F.5.7)			
Main Hoop: Width, bends, inclination (F.5.8)		Fail	Fail
Main Hoop Bracing: Location, angle (F.5.9)		Fail	
Front Bulkhead: Supports (F.6.2)			
Front Hoop Bracing: Connections, straight (F.6.3)			
Side Impact Structure: Connections, locations (F.6.4)			
Main Hoop Bracing Supports: Connections (F.6.6)			
Minimum Wheelbase (V.1.2)			
Uses Existing Suspension Geometry (Addition Rule)			
Pedal Box and Engine Clearance (Addition Rule)			

Table 4 - Decision Matrix

Criteria	Weighting	Chassis 1	Chassis 2	Chassis 2
FSAE Rule Compliance	Required	Pass	Fail	Fail
Total Weight	3	2 – 30.2 kg	3 – 29.4 kg	1 – 30.3 kg
Axial Force	2	2 – 342.64 N	1 – 408 N	3 – 280 N
Complexity	1	3 – Simple	2 – Intricate	3 – Simple
Torsional Stiffness	3	3 – 3763 m/deg	1 – 2487 Nm/deg	2 – 3666 Nm/deg
Total		**22**	16	18

*Criteria ranking is a weighted scoring system from 1 (lowest) to 3 (highest), assigning weights to criteria, scoring each item for each criterion, multiplying scores by weight, summing the results and ranking items by total scores. Axial Force is relative and for comparison only.

2nd Design Phase

From the decision matrix above, it was clear that chassis candidate 1, with the highest score of 22, would be chosen to proceed with; this design was then analysed further using FEA methods.

Numerical Solution

After the chassis was designed in SolidWorks, each node location was recorded in a .txt file. This file was imported into Ansys 2024 Design Modeller, where each member was replicated using the ‘lines from points’ tool. Each member was created under three different ‘lines’ with respect to each tube size. Before the lines were generated, the operation was set to frozen. The three tube cross sections were then created using the ‘cross-section’ tool and applied to the corresponding ‘line body’. Each ‘line body’ in the tree was highlighted, and then ‘form new part’ was selected to create 1 part with 6 bodies.

In Engineering Data, a new ‘isotropic elasticity’ material was created to emulate AISI 4130 with $E = 205\ GPa$ and $v = 0.29$.

In Mechanical, a good-quality mesh was generated using default settings. As shown in Figure 11 below, two ‘remote displacement’ supports were added and applied to the appropriate rear a-arm connection nodes. Two ‘remote force’ loads of 100 N were added and applied to the appropriate front a-arm connection nodes. The left load was applied in the positive y-direction, and the right load in the negative y-direction. ‘Total deformation’, ‘axial force’, and ‘total bending’ solutions were added along with three flexible rotation probes. The probes were set to ‘Z-Axis’ and applied to the forward-front a-arm nodes, front hoop and main hoop.

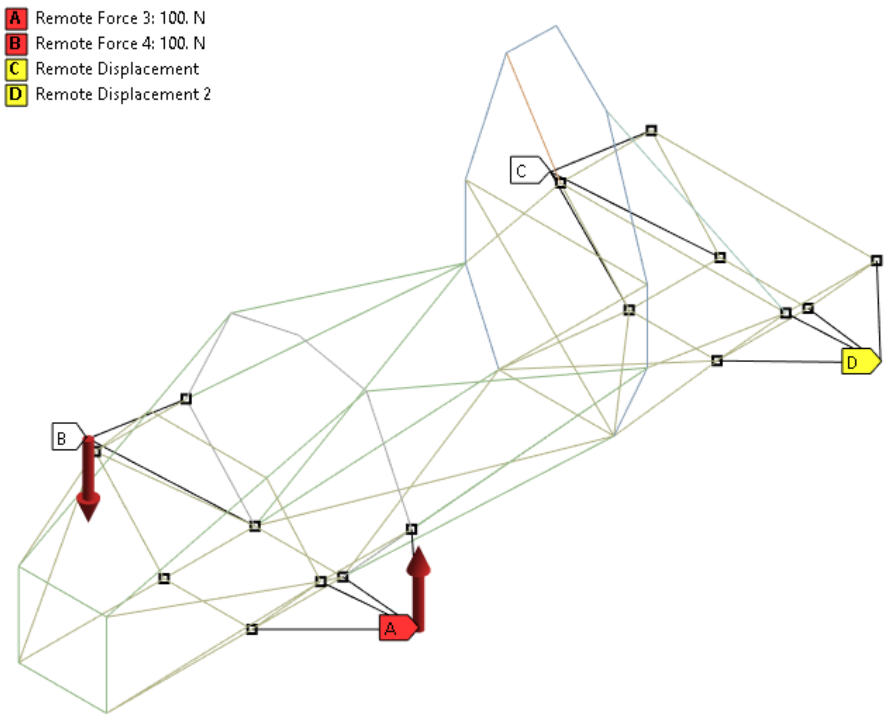


Figure 11 - Ansys Mechanical Support Set-Up

Design Iteration

Further FEA investigations identified two areas for improvement. As shown in Figure 12 below, the middle member of the side impact structure (circled in blue) was carrying large axial forces due to the chassis' rotation. This brought focus to the upmost side member (circled in red), which was found to be ineffectively resisting the twisting load.

The parallel nature of the two circled members resulted in both members inefficiently sharing the load of chassis rotation. Mounting the member between nodes B and D solves this issue in several ways.

First, flipping the member's direction results in a more effective moment and a wider spread. This is illustrated in Figure 13, where the resulting triangles form one large triangle with one diagonal member in tension and the other in compression.

Secondly, the force flow between the front and main hoops was not optimal due to the members' mounting node at C being horizontally unsupported in the side view. Both nodes B and D are properly supported, allowing for efficient axial force transfer through the chassis.

Finally, due to the chassis' rotation around the neutral axis, the top of the front roll hoop (point B) experiences higher rotational deformation than point C. Therefore, with correct triangulation, the topmost member would resist greater chassis twist if mounted at point B. The new upmost member was then upgraded from size C to size B tubing to help reduce deflection and increase strength. See Appendix B: FSAE Relevant Rules for exact tube sizes.

Additionally, the lower members circled in red (Figure 12) were highly stressed, experiencing high deflection and required reinforcement. A pair of cross members were added to effectively triangulate the bending forces into axial forces, as shown in Figure 13.

These changes helped reduce the maximum von mises stress to 60 MPa at 1090 Nm. This load was applied after looking into actual FSAE load data. With a yield stress of 435 Mpa for 4130 steel, this stress falls into a safe fatigue region.

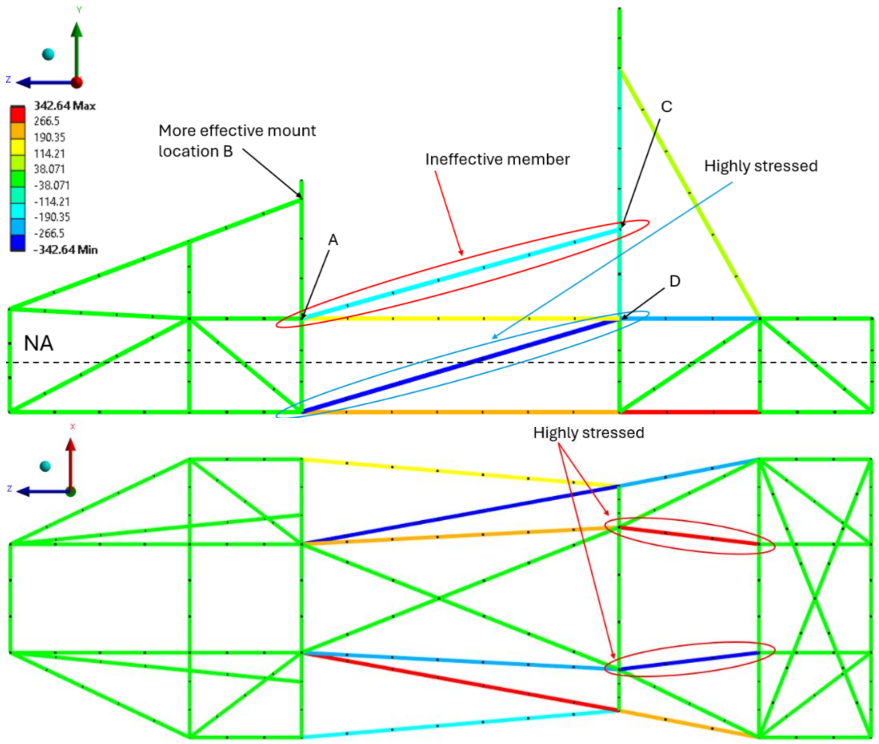


Figure 12 - Axial Force Illustration of Initial Chassis

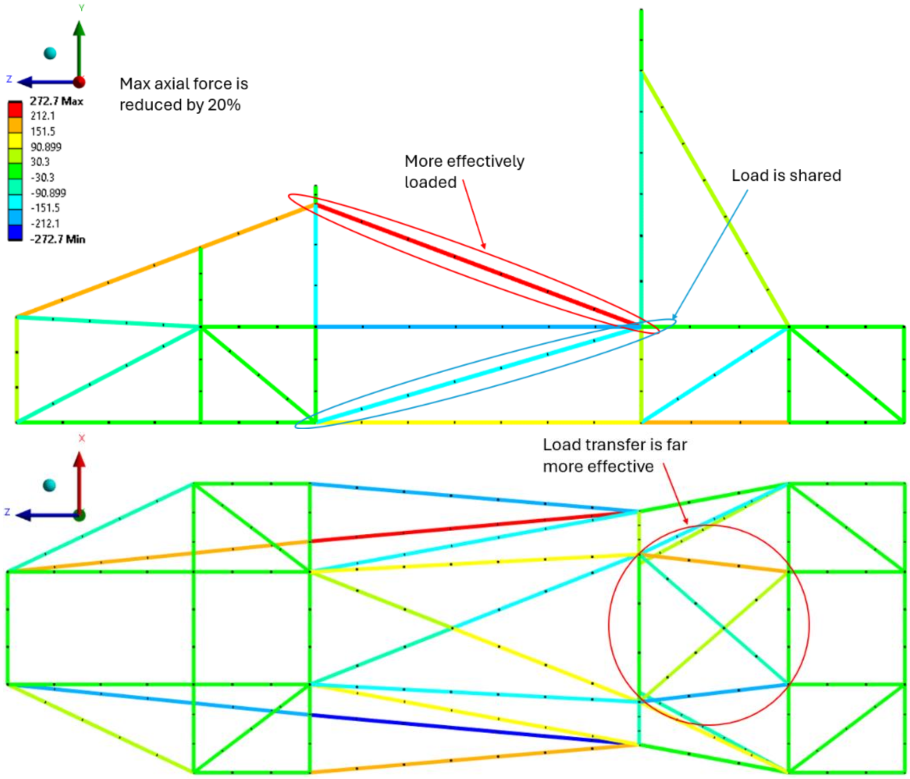


Figure 13 - Final Design with Iteration Improvements

Final Design

The final design is shown in Figure 14 below. Attached are the chassis and FEA files.

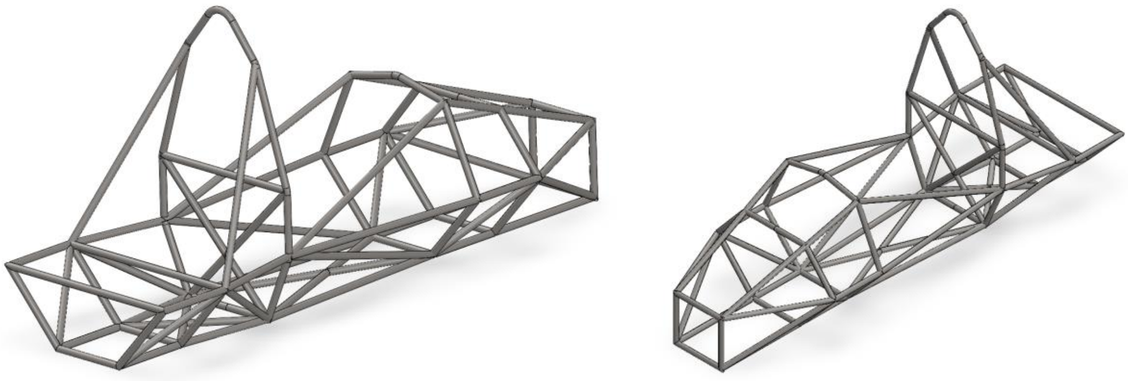


Figure 14 - Final Chassis Design

After the design iterations detailed above, the final design was again analysed via FEA methods, and an average chassis stiffness was produced. The final results are compared in Table 5 below.

Table 5 - Final Chassis Results Vs. Proposed Design

Applied Load – 100 N	Proposed Chassis	Final Chassis
Weight (Kg)	30.19	30.49
Max axial force (N)	342.64	277.75
Max deformation (mm)	0.832	0.708
Stiffness (Nm/deg)	3763	4389

Final Discussion

The iterative improvements resulted in a significant 17% increase in torsional stiffness while only experiencing a 1% increase in weight. This resulted in a chassis stiffness of 4389 Nm/deg and a weight of 30.49 Kg. 4130 chromoly steel was used throughout the chassis, with every member being equivalent to the smallest permittable-sized tube. While this material is widely used in chassis for its great strength-to-weight ratio, future iterations could reduce weight by using a lighter, equivalent-strength material.

The graph illustrated below (Figure 15) indicates a linear stiffness as load increases. This is to be expected within a simulation environment and highlights one of the largest differences between simulation and a real-world chassis.

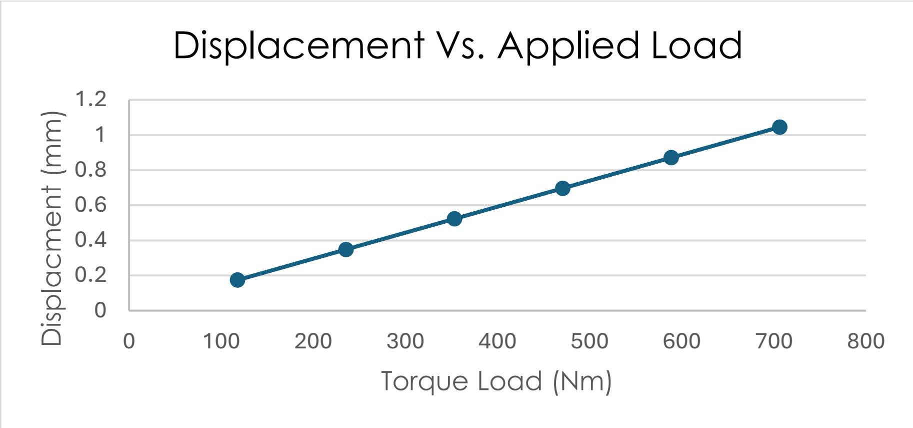


Figure 15 - Displacement Vs. Torque

As illustrated in Figure 16 below, the major source of torsional compliance is between the main and front hoops or the cockpit. This result is expected as it is the largest compartment of the chassis and, by definition, must be on an open top. This means all the forces must travel through the floor and side members, severely limiting the potential stiffness.

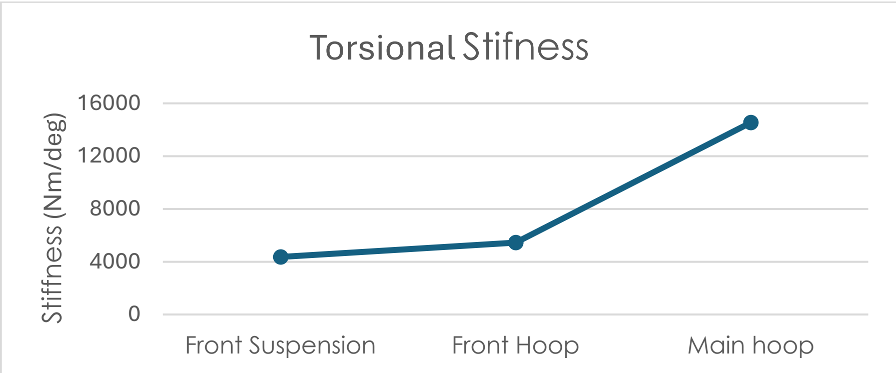


Figure 16 - Torsional Stiffness at the Three Testing Locations

As shown in Figure 17 below, only two members experience relatively high axial loads. These members are designed specifically to handle these loads while taking the stress off key safety members. This aims to free up and maximise the structural capacity of these safety members in the case of an incident. Most of the force within the chassis is transmitted axially, with relatively minor bending moments appearing in the front hoop and front bulkhead. A maximum von mises stress of 60 MPa was experienced by the front hoop at an applied torque of 1090 Nm. If the chassis were pushed beyond these limits, the connection labelled A would likely be the first to fail.

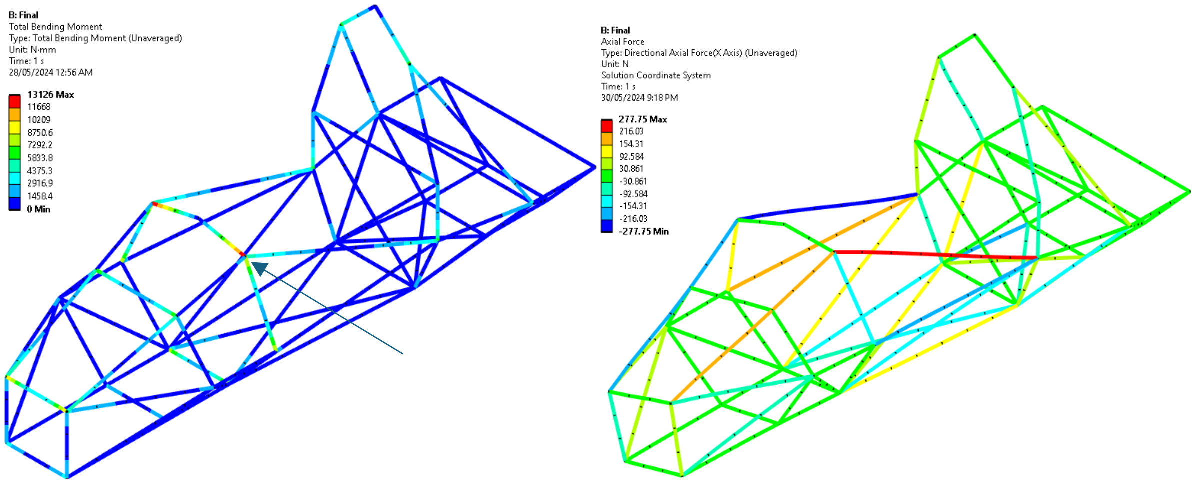


Figure 17 - Bending Moment (Left) and Axial Force (Right) of the Final Design

Limitations

The limitations and potential errors in simulating a chassis must be carefully considered to ensure the accuracy and reliability of the analysis. One significant limitation is the assumption of linear material behaviour and homogeneity. This assumes the structure is a single part with no consideration of the negative effects of welding, leading to inaccuracies in predicting the structural response, especially in scenarios involving large deformations or material yielding. Similarly, simplifications in modelling the connections between each tube member will introduce errors, as these connections are vital in transmitting forces and moments throughout the chassis structure. Neglecting the complexity of these connections can result in an inaccurate representation of the chassis's behaviour under various loading conditions, compromising the reliability of the simulation results. Additionally, the accuracy of the simulation heavily relies on the quality of input data, including material properties and boundary conditions. Uncertainties in these inputs can introduce errors in the simulation results, emphasising the need to validate with experimental and analytical methods.

Verification

A preliminary 1/5-scale model was constructed using balsa wood to validate the design principles and fundamental specifications. In week 13, this model will undergo torsion testing, be weighed, and undergo clearance checks. This evaluation will occur after this proposal is submitted.

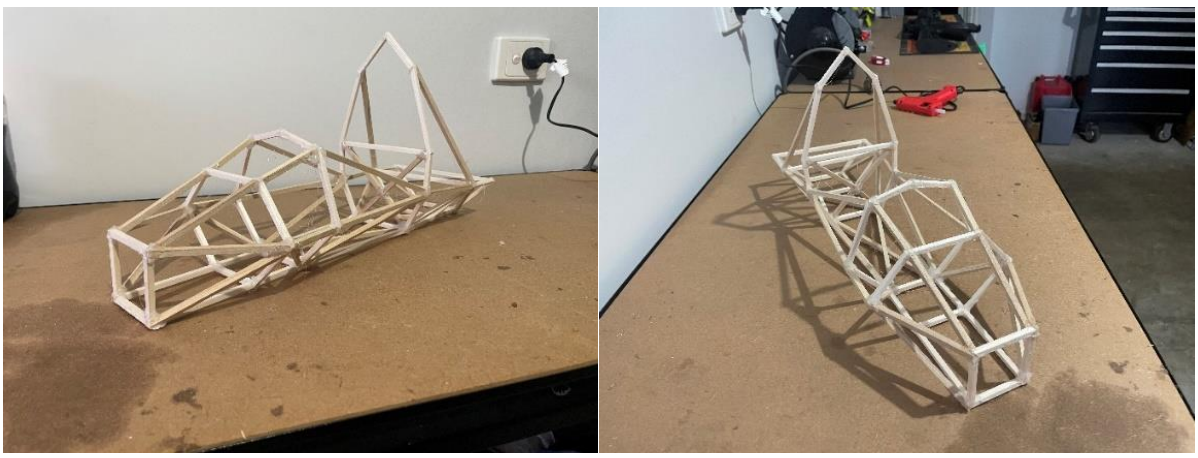


Figure 18 - Final Design Model

Conclusion

Three preliminary chassis designs were created and assessed based on FSAE rules and FEA simulations. They were compared using a decision matrix, with the first design emerging as the optimal choice, fully complying with FSAE regulations. Further refinement of this design included reorienting structural members and improving triangulation, resulting in lower axial force and torsional deformation in the topmost member. Additionally, cross members were added to reinforce the lower-stressed members, significantly improving the chassis' resistance to twisting and deflection. These iterative improvements led to a significant increase in torsional stiffness, with only a minimal increase in weight. The open-top configuration of the cockpit area made the most significant contribution to torsional stiffness. As a result of optimising key structural members, the overall model saw a 17% increase in torsional stiffness, resulting in a chassis stiffness of 4389 Nm/deg, weighing 30.49 Kg.

References

[1] Instron, "Elastic Hysteresis," 2024. [Online]. Available: https://www.instron.com/en/resources/glossary/elastic-hysteresis.

Appendix

Appendix A: Experimental Results Data

Table 6 - Displacement Vs. Applied Load

	Displacement [mm]			
Load [kg]	Dial Gauge A	Dial Gauge B	Dial Gauge C	Dial Gauge D
0	0	0	0	0
10.00	0.09	0.70	0.85	1.00
20.00	0.18	1.51	1.74	2.02
30.00	0.29	2.39	2.66	3.26
40.00	0.40	3.47	3.91	4.58
50.00	0.50	5.42	5.86	6.70
60.00	0.59	6.68	7.23	8.24
50.00	0.56	6.29	6.76	7.59
40.00	0.49	5.55	5.95	6.67
30.00	0.49	4.65	5.00	5.50
20.00	0.36	3.55	3.82	4.16
10.00	0.27	2.51	2.64	2.88
0.00	0.17	1.67	1.72	1.74

Table 7 - Torsional Stiffness Vs. Torque

	Stiffness [Nm/deg]			
Torque [Nm]	Dial Gauge A	Dial Gauge B	Dial Gauge C	Dial Gauge D
0	0	0	0	0
117.72	12469.31	3968.11	3270.89	2776.49
235.44	11987.07	3671.36	3184.51	2742.18
353.16	11069.69	3486.02	3123.55	2552.49
470.88	10848.30	3193.68	2838.28	2421.39
588.60	10870.03	2559.24	2367.45	2068.69
706.32	11126.46	2492.09	2303.42	2020.66
588.60	9668.71	2204.50	2050.34	1826.24
470.88	8801.86	1999.43	1863.42	1664.39
353.16	6614.81	1790.64	1665.22	1512.10
235.44	6111.72	1564.42	1451.44	1335.12
117.72	3988.34	1105.95	1051.44	963.43

Appendix B: FSAE Relevant Rules

Tubing

F.3.2 Tubing Requirements

F.3.2.1 Requirements by Application

	Application	Steel Tube Must Meet Size per **F.3.4**:	Alternative Tubing Material Permitted per **F.3.5** ?
a.	Front Bulkhead	Size B	Yes
b.	Front Bulkhead Support	Size C	Yes
c.	Front Hoop	Size A	Yes
d.	Front Hoop Bracing	Size B	Yes
e.	Side Impact Structure	Size B	Yes
f.	Bent / Multi Upper Side Impact Member	Size D	Yes
g.	Main Hoop	Size A	NO
h.	Main Hoop Bracing	Size B	NO
i.	Main Hoop Bracing Supports	Size C	Yes
j.	Driver Restraint Harness Attachment	Size B	Yes
k.	Shoulder Harness Mounting Bar	Size A	NO
l.	Shoulder Harness Mounting Bar Bracing	Size C	Yes
m.	Accumulator Protection Structure	Size B	Yes
n.	Component Protection	Size C	Yes
o.	Structural Tubing	Size C	Yes

F.3.3 Non Structural Tubing

F.3.3.1 Definition

Any tubing which does NOT meet **F.3.2.1.o Structural Tubing**

F.3.3.2 Applicability

Non Structural Tubing is ignored when assessing compliance to any rule

F.3.4 Steel Tubing and Material

F.3.4.1 Minimum Requirements for Steel Tubing

A tube must meet all four minimum requirements for each Size specified:

	Tube	Minimum Area Moment of Inertia	Minimum Cross Sectional Area	Minimum Outside Diameter or Square Width	Minimum Wall Thickness	*Example Sizes of Round Tube*
a.	Size A	11320 mm^4	173 mm^2	25.0 mm	2.0 mm	*1.0" x 0.095"* *25 x 2.5 mm*
b.	Size B	8509 mm^4	114 mm^2	25.0 mm	1.2 mm	*1.0" x 0.065"* *25.4 x 1.6 mm*
c.	Size C	6695 mm^4	91 mm^2	25.0 mm	1.2 mm	*1.0" x 0.049"* *25.4 x 1.2 mm*
d.	Size D	18015 mm^4	126 mm^2	35.0 mm	1.2 mm	*1.375" x 0.049"* *35 x 1.2 mm*

F.3.4.2 Properties for ANY steel material for calculations submitted in an SES must be:

a. Non Welded Properties for continuous material calculations:

Young's Modulus (E) = 200 GPa (29,000 ksi)

Yield Strength (Sy) = 305 MPa (44.2 ksi)

Ultimate Strength (Su) = 365 MPa (52.9 ksi)

b. Welded Properties for discontinuous material such as joint calculations:

Yield Strength (Sy) = 180 MPa (26 ksi)

Ultimate Strength (Su) = 300 MPa (43.5 ksi)

F.3.4.3 Where Welded tubing reinforcements are required (such as inserts for bolt holes or material to support suspension cutouts), Equivalence of the Welded tube and reinforcement must be shown to the original Non Welded tube in the SES

F.5.2 Bent Tubes or Multiple Tubes

F.5.2.1 The minimum radius of any bend, measured at the tube centerline, must be three or more times the tube outside diameter (3 x OD).

F.5.2.2 Bends must be smooth and continuous with no evidence of crimping or wall failure.

F.5.2.3 If a bent tube (or member consisting of multiple tubes that are not in a line) is used anywhere in the Primary Structure other than the Roll Hoops (see **F.5.6.2**), an additional tube must be attached to support it.

a. The support tube attachment point must be at the position along the bent tube where it deviates farthest from a straight line connecting both ends

b. The support tube must terminate at a node of the chassis

c. The support tube for any bent tube (other than the Upper Side Impact Member or Shoulder Harness Mounting Bar) must be:

- The same diameter and thickness as the bent tube
- Angled no more than 30° from the plane of the bent tube

Hoops

F.5.6 Roll Hoops

F.5.6.1 The Chassis must include both a Main Hoop and a Front Hoop.

F.5.6.2 The Main Hoop and Front Hoop must be Triangulated into the Primary Structure with Structural Tubing

The Triangulation must be at a node in side view for:

a. Bends in side view

b. Bends in front view below the Upper Side Impact Structure **F.6.4**, **F.7.5**

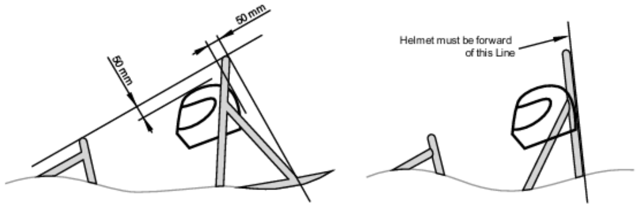
F.5.6.3 Roll Hoop and Driver Position

When seated normally and restrained by the Driver Restraint System, the helmet of a 95th percentile male (see **V.2.1.1**) and all of the team's drivers must:

a. Be a minimum of 50 mm from the straight line drawn from the top of the Main Hoop to the top of the Front Hoop.

b. Be a minimum of 50 mm from the straight line drawn from the top of the Main Hoop to the lower end of the Main Hoop Bracing if the bracing extends rearwards.

c. Be no further rearwards than the rear surface of the Main Hoop if the Main Hoop Bracing extends forwards.



F.5.6.4 Driver Template

A two dimensional template used to represent the 95th percentile male is made to the following dimensions (see figure below):

- A circle of diameter 200 mm will represent the hips and buttocks.
- A circle of diameter 200 mm will represent the shoulder/cervical region.
- A circle of diameter 300 mm will represent the head (with helmet).
- A straight line measuring 490 mm will connect the centers of the two 200 mm circles.
- A straight line measuring 280 mm will connect the centers of the upper 200 mm circle and the 300 mm head circle.

F.5.9.7 The Main Hoop Braces must be:

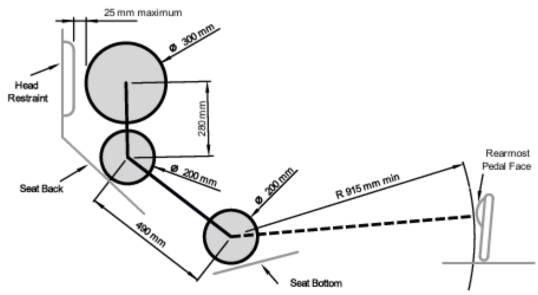
a. Securely integrated into the Frame

b. Capable of transmitting all loads from the Main Hoop into the Major Structure of the Chassis without failing

F.5.6.5 Driver Template Position

The Driver Template will be positioned as follows:

- The seat will be adjusted to the rearmost position
- The pedals will be placed in the most forward position
- The bottom 200 mm circle will be placed on the seat bottom where the distance between the center of this circle and the rearmost face of the pedals is no less than 915 mm
- The middle 200 mm circle, representing the shoulders, will be positioned on the seat back
- The upper 300 mm circle will be positioned no more than 25 mm away from the head restraint (where the driver's helmet would normally be located while driving)



F.5.7 Front Hoop

F.5.7.1 The Front Hoop must be constructed of closed section metal tubing meeting **F.3.2.1.c**

F.5.7.2 With proper Triangulation, the Front Hoop may be fabricated from more than one piece of tubing

F.5.7.3 The Front Hoop must extend from the lowest Frame Member on one side of the Frame, up, over and down to the lowest Frame Member on the other side of the Frame.

F.5.7.4 The top-most surface of the Front Hoop must be no lower than the top of the steering wheel in any angular position. See figure following **F.5.9.6 below**

F.5.7.5 The Front Hoop must be no more than 250 mm forward of the steering wheel.

This distance is measured horizontally, on the vehicle centerline, from the rear surface of the Front Hoop to the forward most surface of the steering wheel rim with the steering in the straight ahead position.

F.5.7.6 In side view, any part of the Front Hoop above the Upper Side Impact Structure must be inclined less than 20° from the vertical.

F.5.7.7 A Front Hoop that is not steel must have a 4 mm hole drilled in a location to access during Technical Inspection

F.5.8 Main Hoop

F.5.8.1 The Main Hoop must be a single piece of uncut, continuous, closed section steel tubing meeting **F.3.2.1.g**

F.5.8.2 The Main Hoop must extend from the lowest Frame Member / bottom of Monocoque on one side of the Frame, up, over and down to the lowest Frame Member / bottom of Monocoque on the other side of the Frame.

F.5.8.3 In the side view of the vehicle,

a. The part of the Main Hoop that lies above its attachment point to the upper Side Impact Tube must be less than 10° from vertical.

b. Any bends in the Main Hoop above its attachment point to the Major Structure of the Chassis must be braced to a node or Attachment point **F.7.8** with tubing meeting **F.3.2.1.h** and **F.5.9.5**

c. The part of the Main Hoop below the Upper Side Impact Member attachment:

- May be forward at any angle
- Must not be rearward more than 10° from vertical

F.5.8.4 In the front view of the vehicle, the vertical members of the Main Hoop must be minimum 380 mm apart (inside dimension) at the location where the Main Hoop is attached to the bottom tubes of the Major Structure of the Chassis.

F.5.9 Main Hoop Braces

F.5.9.1 Main Hoop Braces must be constructed of closed section steel tubing meeting **F.3.2.1.h**

F.5.9.2 The Main Hoop must be supported by two Braces extending in the forward or rearward direction, one on each of the left and right sides of the Main Hoop.

F.5.9.3 In the side view of the Frame, the Main Hoop and the Main Hoop Braces must not lie on the same side of the vertical line through the top of the Main Hoop.

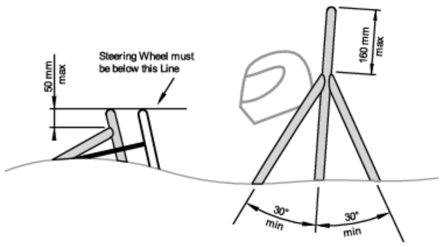
(If the Main Hoop leans forward, the Braces must be forward of the Main Hoop, and if the Main Hoop leans rearward, the Braces must be rearward of the Main Hoop)

F.5.9.4 The Main Hoop Braces must be attached 160 mm or less below the top most surface of the Main Hoop.

The Main Hoop Braces should be attached as near as possible to the top of the Main Hoop

F.5.9.5 The included angle formed by the Main Hoop and the Main Hoop Braces must be 30° or more.

F.5.9.6 The Main Hoop Braces must be straight, without any bends.



F.6 TUBE FRAMES

F.6.1 Front Bulkhead

The Front Bulkhead must be constructed of closed section tubing meeting **F.3.2.1.a**

F.6.2 Front Bulkhead Support

F.6.2.1 Frame Members of the Front Bulkhead Support system must be constructed of closed section tubing meeting **F.3.2.1.b**

F.6.2.2 The Front Bulkhead must be securely integrated into the Frame.

F.6.2.3 The Front Bulkhead must be supported back to the Front Hoop by a minimum of three Frame Members on each side of the vehicle; an upper member; lower member and diagonal brace to provide Triangulation.

a. The upper support member must be attached 50 mm or less from the top surface of the Front Bulkhead, and attach to the Front Hoop inside a zone extending 100 mm above and 50 mm below the Upper Side Impact member.

b. If the upper support member is further than 100 mm above the Upper Side Impact member, then properly Triangulated bracing is required to transfer load to the Main Hoop by one of:

- the Upper Side Impact member
- an additional member transmitting load from the junction of the Upper Support Member with the Front Hoop

c. The lower support member must be attached to the base of the Front Bulkhead and the base of the Front Hoop.

d. The diagonal brace must properly Triangulate the upper and lower support members

F.6.2.4 Each of the above members may be multiple or bent tubes provided the requirements of **F.5.2** are met.

F.6.2.5 Examples of acceptable configurations of members may be found in the SES

F.6.3 Front Hoop Bracing

F.6.3.1 Front Hoop Braces must be constructed of material meeting **F.3.2.1.d**

F.6.3.2 The Front Hoop must be supported by two Braces extending in the forward direction, one on each of the left and right sides of the Front Hoop.

F.6.3.3 The Front Hoop Braces must be constructed to protect the driver's legs and should extend to the structure in front of the driver's feet.

F.6.3.4 The Front Hoop Braces must be attached as near as possible to the top of the Front Hoop but not more than 50 mm below the top-most surface of the Front Hoop. See figure following **F.5.9.6 above**

F.6.3.5 If the Front Hoop above the Upper Side Impact Structure leans rearwards by more than 10° from the vertical, it must be supported by additional rearward Front Hoop Braces to a fully Triangulated structural node.

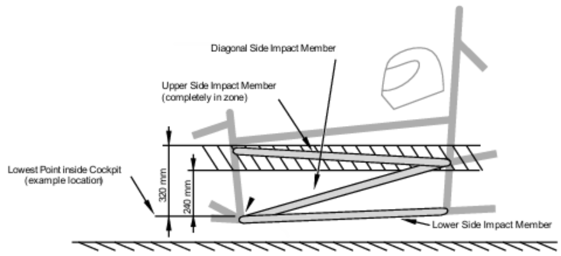
F.6.3.6 The Front Hoop Braces must be straight, without any bends

F.6.4 Side Impact Structure

F.6.4.1 Frame Members of the Side Impact Structure must be constructed of closed section tubing meeting **F.3.2.1.e** or **F.3.2.1.f**, as applicable

F.6.4.2 With proper Triangulation, Side Impact Structure members may be fabricated from more than one piece of tubing.

F.6.4.3 The Side Impact Structure must be comprised of three or more tubular members located on each side of the driver while seated in the normal driving position



F.6.4.4 The Upper Side Impact Member must:

a. Connect the Main Hoop and the Front Hoop.

b. Be entirely in a zone that is parallel to the ground between 240 mm and 320 mm above the lowest point of the top surface of the Lower Side Impact Member

F.6.4.5 The Lower Side Impact Structure member must connect the bottom of the Main Hoop and the bottom of the Front Hoop.

F.6.4.6 The Diagonal Side Impact Member must:

a. Connect the Upper Side Impact Member and Lower Side Impact Member forward of the Main Hoop and rearward of the Front Hoop

b. Completely Triangulate the bays created by the Upper and Lower Side Impact Members.

F.6.6 Main Hoop Bracing Supports

F.6.6.1 Frame Members of the Main Hoop Bracing Support system must be constructed of closed section tubing meeting **F.3.2.1.i**

F.6.6.2 The lower end of the Main Hoop Braces must be supported back to the Main Hoop by a minimum of two Frame Members on each side of the vehicle: an upper member and a lower member in a properly Triangulated configuration.

a. The upper support member must attach to the node where the upper Side Impact Member attaches to the Main Hoop.

b. The lower support member must attach to the node where the lower Side Impact Member attaches to the Main Hoop.

c. Each of the above members may be multiple or bent tubes provided the requirements of **F.5.2** are met.

d. Examples of acceptable configurations of members may be found in the SES.

Appendix C: Team Member Contribution Statement

Table 8 - Team Member Contribution Statement

Name (Printed)	Student Number	% Contribution to Current Assessment	Signature
Aaron Brading	N10211705	25 %	A. Brading
Ryan Brown	N11031662	25 %	R. Brown
Mitchell Eickenloff	N11161701	25 %	M. Eickenloff
Jayden Landroth	N10269819	25 %	J. Landroth

*Please indicate the percent of the total work that may be attributed to you for this portion of the assessment. These percentages should total 100%. For instance, if both partners contributed equally put 50% for each.

Aaron Brading:
Designed a chassis proposal and wrote the approach for chassis proposal 1. Listed the key FSAE rules and rules checklist. Performed the final design iterations in Ansys and wrote the 2nd design phase. Helped design the proposed model.

Jayden Landroth:

I have completed a design and chassis proposal, along with the approach for chassis proposal 2. The First Design Phase, Design process, Preliminary design, and a conclusion were completed as well. Helped design the proposed model.

Mitchell Eickenloff:

I completed the background and theory section, Experiment Method, Practical Results, Errors and suggested improvements and Second Design Phase Limitation. I also formatted the report and helped build the proposed model.

Ryan Brown:

Completed a chassis design and proposal, produced a standardised methodology for finite element analysis of chassis, processed experimental data and wrote the discussion of results and experimental methods.