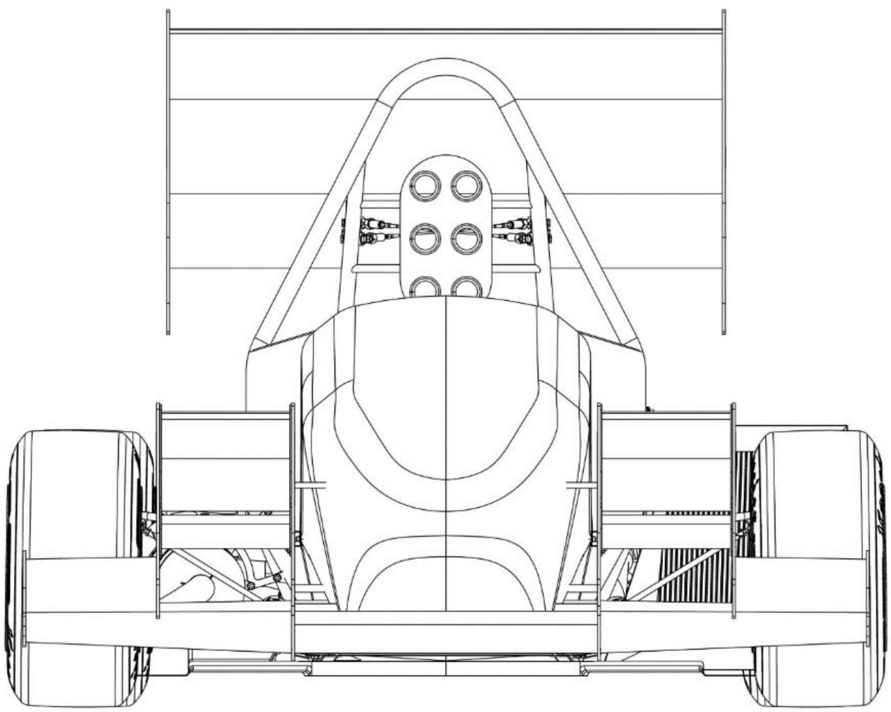
Formula SAE

Fall Final Design Report



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Acronyms and Abbreviations

SAE – Society for Automotive Engineers

FSAE – Formula SAE

ICE – Internal combustion engine

CAD – Computer-aided design

CFD – Computational fluid dynamics

FEA – Finite element analysis

CG – Center of gravity

RC – Roll center

IC – Instant center

PADS – Pre-approved design Figure 1: Economic impact of the Indianapolis effects on Speedway in 2024   8in Hoop

FH – Front hoop

FCS – Front crash support

SIS – Side-impact structure

SES – Structural equivalency spreadsheet

RCS – Rear crash support

UCA – Upper control arm

LCA – Lower control arm

ARB – Anti-roll bar

Executive Summary

The Society for Automotive Engineers (SAE) hosts an annual student design competition in which teams from universities around the world design, manufacture, and race single-seat, open cockpit race cars. Vehicles are judged on performance, cost, design, efficiency, and business applications. Students are challenged to use skills gained through their engineering curriculum, as well as skills learned outside the classroom, such as fabrication, project management, and technical automotive understanding.

The goal of this senior design project is to complete Phase One: developing a complete rolling chassis that includes the chassis frame, suspension system, braking system, and steering system. These subsystems will be designed and integrated to meet Formula SAE regulations while emphasizing performance, safety, reliability, and ease of manufacturing. The focus is to create a solid foundation that can be built upon in future years.

This rolling chassis will serve as the platform for the next senior design team, who will complete Phase Two: adding the drivetrain, aerodynamic package, and electronics. Through this phased approach, the program ensures steady progress toward delivering a fully functional Formula SAE vehicle by the beginning of the 2027 Formula SAE competition, while also providing continuity, knowledge transfer, and hands-on experience for future teams.

Team motivation

The team started the proposal process in September of 2024. Since then, much time has been spent doing research, gathering resources, and speaking to possible sponsors to ensure the success of this project. On top of that, some of the members have used their coursework in courses like Mechanics of Machinery and Computer-Aided Machine Design to focus on components for a rolling chassis. These courses have helped with concepts like kinematics, dynamics, finite element analysis (FEA), as well as project management techniques.

The main goal of the team is to establish a strong foundation for a Formula SAE team at Trine University and build a long-lasting legacy of success. By developing a competitive rolling chassis, the team aims to demonstrate that it is possible to compete with larger schools that have greater financial support and resources.

Economic Implications

The Midwest has always been a hub for motor sports in the USA. From June 2022 to May 2023, the Indiana Motor Speedway contributed $1.06 billion to Indiana's economy and created nearly 8,500 jobs. (IMS Impact, n.d.). These jobs include engineering, manufacturing, marketing, logistics and transportation, and operations, among others. Projects like FSAE help prepare students for careers in motorsports by providing hands-on experience.

Figure 1 Economic impact of the Indiana Motor Speedway in 2024

When it comes to engineering, this project provides students with exposure to iterative design, advanced manufacturing methods, project management, and business implications. In addition to this, motorsports are a proving ground for new technology and innovations. Some of these innovations eventually make their way into consumer products, meaning motorsport-derived advancements are continuously making progress in automotive markets.

Environmental Impact

Whether it's the steel used to protect the driver or the carbon fiber used to decrease mass and increase stiffness, many of the commonly used materials in motorsports come with environmental implications. The three most prevalent materials in motorsports are steel, aluminum, and carbon fiber. Each of these materials has its place in a racecar, but they all have very different levels of recyclability. The US Department of Energy estimates that production of recycled aluminum requires 90% less energy than its primary production. (Recycling by material, 2024). This means its recyclability greatly reduces the energy cost of future material production. Steel is also a highly recycled material, with close to 100% of it being reused. (American Iron and Steel institute, n.d.).

Figure 2 Mercedes AMG PETRONAS Formula 1 Team W14 sporting an exposed carbon livery

This is not the case for carbon fiber as an effective way of recycling; it has not yet been discovered. Traditional recycling methods for carbon fiber compromise the quality of the fibers and diminish their mechanical properties. This has resulted in experimentation with chemical and thermal techniques to increase the recovery rates for carbon fiber.

Even though an individual Formula SAE car has a small environmental footprint, considering the environmental impact of the material selection in this project can help shape more eco-conscious approaches in mainstream automotive and motorsports production (Sustainable manufacturing expo, n.d.).

Customer Needs

1. The chassis must be lightweight
2. Phase One of the project must be completed such that space is allocated for Phase Two subsystems (Sarikaya)
    - Phase One consists of a rolling chassis, including a suspension system, wheels, a frame (either tube steel or monocoque), steering, and brakes.
    - Phase Two consists of a powertrain, drivetrain, electrical subsystems, and aerodynamic package.

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3. Proper safety and cleanliness procedures in lab areas must be followed. (Joe/Webber)
4. The rolling chassis must be completed by 4/24/26 and ideally should be completed by 3/14/26. (Webber)
5. The chassis must be designed with sustainability in mind by ensuring a portion of the components are recyclable.
6. The design must obtain funding for Phase One of the project and should obtain at least part of the funding for Phase Two.

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Commented [CC3]: I may have misheard but it sounded like he wanted us to include this?

Requirements

Customer Requirements

The parenthesis-bracketed numbers at the end of each requirement refer to the Customer Need it helps meet:

1. Design a chassis or monocoque capable of pulling 1.5g during cornering (in steady state). (2)
2. Design a rolling chassis with a weight under 135 kg. (1)
3. Design a chassis that twists less than 20% roll compliance at maximum lateral Load (2)
4. Design must complete 4/8 of the total vehicle subsystems, including all 4 components of Phase One (Completed suspension, steering, brake, and frame subsystems), as well as determining the specifications for the Phase Two engine. (2)

5. Design must complete all Phase one subsystems (1)
    a. Phase One consists of a rolling chassis, including a suspension system, wheels, a frame (either tube steel or monocoque), steering, and brakes. Additionally, the chassis will be designed for a specific engine/transmission package.
    b. Phase Two consists of a powertrain, drivetrain, electrical subsystems, and aerodynamic package.
6. Design must accommodate drivers from any size between the 5% female and 95% male percentiles (2)
7. Design must pass 100% of the practical FSAE Tech tests applicable to Phase One subsystems, including but not limited to the following: (2)
    - Vehicle does not roll when tilted 60 degrees from the horizontal, corresponding to 1.7g

- Gaps between Floor Closeout Panels must not exceed 3mm
- Accelerator Pedal must return to 0% Pedal Travel when not pushed
- Sufficient heat insulation must be provided such that the driver will not contact any materials that may be heated to a surface temperature above 60 degrees Celsius

8. Design must be made such that a minimum of 25% of the rolling chassis weight is recyclable at the end of the vehicle's life cycle. (5)
9. Design must be completed by April 24, 2026. (4)
10. There must be 0 safety infractions during the design and manufacture of the rolling chassis. (3)
11. A minimum of $18,000 must be raised for the project's Phase One subsystems, and $7,000 should be raised for future Phase Two subsystems. (6)

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Team Requirements

A team must consist of a faculty advisor appointed by the university, as well as team members. Team members must be enrolled as degree-seeking undergraduates or graduate students in the college or university of the team they are participating with. Team members must be members of SAE International.

Fundraising Requirements

The overall estimated cost for both phases of the project is estimated to be around $25,000. The team will be responsible for fundraising at least $10,500 by September 5 to ensure the feasibility and success of the project. Additionally, the team must continue to fundraise through the remainder of phase one to raise enough funds to provide phase 2 with funding.

Project Management

The team is expected to utilize a Gantt chart, PERT chart, and WBS chart, as well as any other additional resources. These project management resources ensure the project is staying on schedule, assigning people responsible for tasks, and help identify the critical path.

PERT chart

The PERT chart is used to help estimate how long a task will take and what tasks are dependent on its completion, as well as to identify the critical path. For this project, the critical path is the suspension system, outlined in red below.

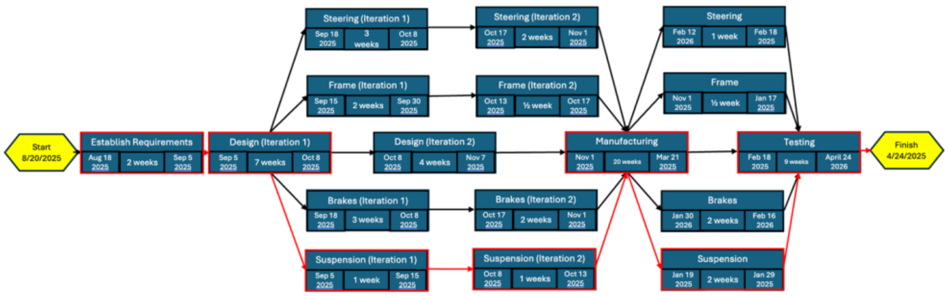


Figure 3 PERT Chart

Gantt chart

A Gantt chart is utilized to provide a visual of the project schedule and the progress of individual tasks.

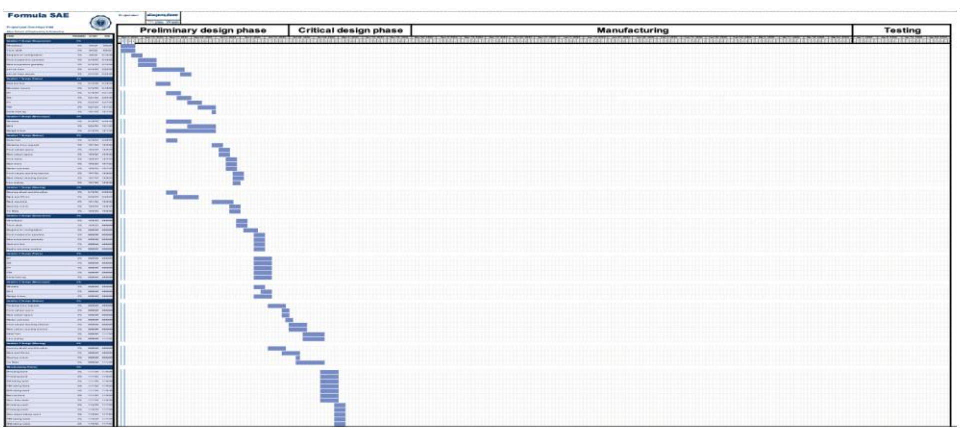


Figure 4 Portion of Gantt chart

The schedule for this project is broken down into 4 stages: Preliminary design phase, Critical design phase, Manufacturing, and Testing. The preliminary design phase will include things like Concept generation, Concept selection, and material selection, as well as the first iteration design. The critical design phase will be utilized to make any second iteration changes needed before the beginning of the manufacturing phase, which will be followed by the testing and validation phase.

WBS

The WBS is similar to a PERT chart, but more focused on the time and resource requirements of each task comprising the larger sections of the project.

WBS ID	Task	Est Hrs	Who	Resources
0	Formula SAE	2240	ALL	ALL
1	Design iteration 1	120	ALL	ALL
1.1	Suspension	32	CC	NX
1.1.1	Wheelbase	2	CC	NX
1.1.2	Track width	2	CC	NX
1.1.3	Suspension config	10	CC	NX
1.1.4	Front geometry	6	CC	NX
1.1.5	Rear Geometry	6	CC	NX
1.1.6	Anti-roll bars	4	CC	NX
1.2	Frame	32	GD	NX
1.2.1	Seat position	2	GD	NX
1.2.2	Drivetrain mounts	4	GD	NX
1.2.3	MH	6	GD	NX
1.2.4	SIS	6	GD	NX
1.2.5	FH	4	GD	NX
1.2.6	FBH	4	GD	NX
1.2.7	Frame bracing	6	GD	NX
1.3	Brakes	32	MS	NX
1.3.1	Pedal box	4	MS	NX
1.3.2	Clamping force	2	MS	NX
1.3.3	Fornt caliper	2	MS	NX
1.3.4	Rear caliper	2	MS	NX
1.3.5	Front rotors	2	MS	NX
1.3.6	Rear rotor	2	MS	NX
1.3.7	Master Cylinder	4	MS	NX
1.3.8	Front mounting bracker	4	MS	NX
1.3.9	Rear mouting bracket	4	MS	NX
1.3.10	Line routing	4	MS	NX
1.4	Steering	24	KV/AV	NX
1.4.1	Steering wheel location	6	KV	NX
1.4.2	Rack and pinion	6	AV	NX
1.4.3	Steering column	6	KV/AV	NX
1.4.4	Tie Rods	6	KV/AV	NX
2	Manufacture	50	ML	ALL

Figure 5: Work Breakdown Structure

The Work Breakdown Structure helps document each component of each subsystem needed to design the project. Shown are the hourly estimates for how long it will take to design the first iteration of the project. It is important to note that the “Who” element of the WBS determines who will take point on that component, but all members will be involved in each subsystem in all capacities.

Risk management

A project of this scope comes with many risks. To effectively navigate these risks, it is important to conduct a risk assessment matrix ranking all the possible risks in terms of severity and likelihood. On top of ranking these risks, it's important to also identify mitigations to reduce the likelihood of the risks occurring throughout the course of the project. These mitigations should be engineered into the design of the rolling chassis and must be specific actions. The major risks of this project were identified as the following:

- A: Binds in suspension links
- B: Bump steer
- C: Loose steering
- D: Unwanted dynamic suspension geometry changes
- E: Frame buckling
- F: Frame cannot withstand impacts
- G: Failure in integration of all subsystems (Future or Present)
- H: Failure to raise enough funds

High priority risks

The three risks with the highest weight for project completion were identified as: failure to design a chassis capable of withstanding an impact, designing a rolling chassis that fails to integrate all present and future subsystems, and failing to raise the funds necessary to cover the cost of the project. These risks carry such high weights that they can result in the inability to complete the project or unsafe conditions for the driver.

The risk with the highest priority in this project is designing a frame that cannot withstand an impact in case of an accident. If the chassis fails to dissipate the energy of a collision, the safety of the driver can be at risk, which could result in injury or death. Some mitigation actions that can be taken to navigate this risk are to conduct an impact test and force calculations by hand and validating them with the help of FEA, add crumple zones to dissipate the energy of the collision, and design all the impact structures with a higher factor of safety than the rest of the rolling chassis.

The next most likely and costly risk is the failure to design a rolling chassis where all future or present subsystems have issues with integration. To mitigate this risk, all CAD and design work this semester will be done in collaboration between all subsystems, and future component estimations will be added to the current CAD to estimate space constraints.

The last of the high-priority risks is the failure to raise the funds necessary to complete the project. To mitigate this risk, the team will reach out to smaller businesses for in-kind donations of materials or labor, prioritize critical components, and scale back on non-essential parts.

Medium priority risks

The remaining risks carry moderate weight as they impact the overall performance of the car and can limit the number of points that the team can score. These risks don't put the driver's safety or the completion of the project at risk, but can result in unwanted characteristics in the car.

The following are the mitigating actions that will be taken for the remaining risks:

Binds in suspension links

- Use joint hardware and materials that reduce friction (PTFE-lined bearings or low-friction rod-ends)
- Use proper dimensions and tolerances to ensure design conditions are met in fabrication

Bump Steer

- Adjust tie rod and control arm angles to maintain parallel motion
- Perform a bump steer analysis with the use of Optimum kinematics

Loose steering

- Provide geometry for preload or interference where needed
- Design in support bearings and a clear location for the steering column so axial and radial play are removed by geometry, not just fasteners

Unwanted dynamic suspension geometry changes

- PTFE Spherical bearings and rod ends will be used on all arms to minimize deflection at all joints
- Triangulate and arrange members, so loads are carried in axial tension/compression where possible, not bending.
- Define acceptable deflection targets for control-arm endpoints and design members to meet those through simple stiffness calculations/FE iterations

Binds in suspension links

- Increase wall thickness or use higher-grade tubing
- Add triangulation or gussets in high-load areas
- Create a detailed weld procedure to ensure good penetration in high-stress areas

Literature review

Suspension research

Selecting and designing an effective suspension system is crucial for the performance of the FSAE vehicle. The suspension geometry has major effects on the handling of the car, such as dive, squat, roll, and cornering. For the FSAE competition events, due to the slaloming nature of the race, maneuverability and cornering are typically seen as more important than straight-line speed. (Esmund. F. Gaffney III, n.d.).

The first two aspects that must be determined are the wheelbase and track width of the car, essentially, the distance between the centers of the wheels (wheelbase being between the centers of the front and back wheels, and track width being between the centers of the left and right wheels, respectively. Once they have been selected, kinematic design and analysis of geometry can begin. A longer wheelbase means better straight-line speed at the expense of maneuverability. This results in many teams attempting to make short wheelbases, leading to an FSAE minimum wheelbase of 1525mm. Similarly, a wider track width helps with cornering and lateral weight transfer. (Puhn, 1976). Track width often comes down to a trade-off between being able to provide enough packaging space, keeping the weight low, and controlling cornering and weight transfer. When designing the trackwidth and wheelbase, you must keep in mind that the narrower you make the wheelbase, the lower the CG height must be, to pass the 60º tilt test. (SAE International, 2024).

The type of suspension chosen is also important. While vehicles with the size, weight, and performance requirements of an FSAE car won't typically use suspension styles such as air springs or leaf springs, there are some different types of suspension that could be utilized, such as a McPherson strut or a double wishbone design with push rods. McPherson struts tend to be cheaper and lighter, but due to the importance of vehicle handling in FSAE, the double-wishbone push-rod design is typically considered the superior option due to its better performance.

From this point, the suspension geometry can be determined by setting some desired parameters and a kinematics program such as Optimum Kinematics. After the geometry is set and analyzed in kinematic software, numerical values are found for elements such as the scrub radius, kingpin inclination, caster/camber/toe, and Ackermann percentage or "angle." This is also where components such as anti-roll bars come in to prevent issues like over-/understeer (G. Wheatley, 2022). Since optimal settings for some of these designs require more specific knowledge of the final weight distribution or other components, estimation is required.

Chassis research

Research into FSAE revealed that most teams use either a space frame tube chassis or a partial monocoque. Other considerations are full monocoque designs and space frames with composite panels. Each construction has distinct trade-offs considering stiffness, weight, and manufacturability. A past report from the University of Delaware (Delaware, 2017) Outlines the balance of fabricating a chassis with high torsional rigidity to maintain suspension geometry under load, while remaining as light as possible to aid in acceleration and handling. Additionally, Cal Poly's report (University, Formula SAE Hybrid: Carbon Fiber Monocoque and Steel Tube Frame Chassis, 2012) Proved that a carbon fiber monocoque can provide far superior stiffness-weight ratios than a space frame, but introduces high manufacturing costs, longer fabrication schedules, and complexity in the case of repairing the chassis. These findings were crucial to the team's decision to pursue a steel space frame for the first-year car.

The Design Judges articles (Kaneb, Tube Frame Analysis, 2021) (judges, 2023) Emphasize that stiffness improvements in space frames depend on more than tubing sizes alone. Deliberate focus on bracing and load path design will yield much greater returns in terms of chassis rigidity, where triangulated structures and continuous force paths are identified as the most efficient methods for improving stiffness without sacrificing final weight. These insights will guide the team's intent to use minimum tubing sizes when allowed by FSAE regulations and focus on effective triangulation around the suspension pickup points. Design Judges also discussed how excessive bracing can quickly add extra weight with limited stiffness gains, reinforcing the decision for a calculated approach to frame geometry as opposed to triangulating all profiles.

Material selection research further supported this approach. Both 4130 chromoly and mild steel were commonly cited in FSAE and motorsport applications for their favorable properties. However, multiple references also noted the decision to utilize mild steel over 4130 due to the lower cost and similar material properties. (judges, 2023). This information, and guidance from Dr. Webber, has encouraged the team to consider using mild steel over 4130, though a final decision has not yet been made.

Simulation practices outlined by Monash Motorsports (Motorsports, 2024) Demonstrated the value of an FEA-driven iterative design approach. Monash Motorsports' use of finite element analysis refined their monocoque geometry and layup plan to improve stiffness. Though a carbon monocoque is no longer an option, the team will use a similar iterative design process in conjunction with ANSYS for the space frame. This aids in ensuring the goal of 2500+ Nm/degree is met by using proven iteration methods instead of a trial-and-error-based design approach.

Overall, the research showed that a conventional, well-optimized steel space frame remains the most practical choice for a blend of cost effectiveness and performance for a first-year FSAE team. Insights from prior university teams and technical resources will guide the team's space frame design process to focus on efficient triangulation, manufacturability, and simulation-based decisions, all aimed towards assisting the Phase Two team in fielding a competitive car.

Brakes research

The braking system is responsible for slowing the vehicle and plays a critical role in both cornering performance and longitudinal weight transfer. Brakes are also a key safety component, capable of rapidly dissipating the vehicle's kinetic energy to prevent or reduce the severity of collisions. When designing a brake system, it is important to consider total vehicle weight, front and rear brake pressures, and the level of driver control. Common braking systems include rotor brakes, limited-slip differentials, drum brakes, air brakes, and brake-by-wire systems. However, brake-by-wire is prohibited under FSAE regulations due to the risk of total brake failure from a single electrical fault, and drum brakes are banned because of their weak performance and poor heat dissipation at high speeds.

According to the FSAE rules (SAE International, 2024), brake-by-wire and drum brake systems are prohibited due to their weak performance and poor heat dissipation at high speeds, making them unsuitable for racing applications. The decision between a four-rotor braking system and a limited-slip differential setup was evaluated using a selection matrix. The four-rotor configuration was ultimately chosen based on superior performance, reduced weight, and improved packaging efficiency. Factors such as adjustability, mechanical complexity, and cost were also considered but carried less weight in the decision. Research into limited-slip braking systems revealed a notable reduction in braking performance when two wheels are actuated by a single rear rotor, further reinforcing the selection of the four-rotor design.

The 4-rotor brake system will be run on a 2-circuit fluid system required by FSAE, with a master cylinder controlling the front brakes and one controlling the rear brakes. This will provide a layer of safety because if a brake line fails, the whole circuit does not go along with providing adjustability on pressure between the front and rear brakes. While calculations will need to be made on brake rotor size and how many pistons will be in the brake caliper, it is generally better to have a brake rotor that is big enough to leave just enough space for the caliper inside the wheel, as the bigger disk gives a greater moment arm on the wheel. The difference between a single piston and two or even four piston brake calipers is significant. Additionally, more pistons allow for a bigger contact patch between the brake pads and rotor, providing more friction.

Steering research

When designing the steering system, aspects like Ackerman and bump steering must be taken into account. The steering system must have a balance of steering force and steering angle. First, a steering system had to be chosen, and using the selection matrix with choices of a steering rack, steering box, and 4-wheel steering, the steering rack and pinion was selected. Through researching steering racks, one from FSAE parts was selected. The next selection was to choose eye-to-eye length. The choices were 289.56mm, 365.76mm, and 441.96mm. Another choice that needed to be made was the rack speed, being 87.9mm/rev or 101.6mm/rev.

The steering column and intermediate shafts must be designed to minimize play while meeting safety requirements. The steering wheel position is also a key component for ergonomics, which would allow the driver to maintain proper posture and visibility during rapid movements in the car. Also, the alignment between the control arms and tie rods is so important to ensure reliable vehicle response. (How does an F1 steering wheel work? - mercedes-AMG Petronas F1 Team, 2023)

Concept selection and generation

The design of the rolling chassis is split into 4 separate subsystems. These include suspension, chassis, brakes, and steering. With the use of the research each team member did, some material testing, as well as a weighted selection matrix, the team was able to select its concepts. Selection matrices are used as tools to quantify a concept and assign a score to each concept to guide the team in the correct selection that will meet the customer requirements.

A selection matrix consists of design criteria, concepts, weights, individual scores, and a final score. Each criterion for the matrix is given a weight percentage. Once these weights are assigned, a score can be given for each subsystem and criterion on a scale from 1-5, with 5 being the best score and 1 being the worst score. As mentioned previously, the selection matrices are used merely as tools and are not the deciding factor for the subsystems. As future engineers, engineering judgment will always have a strong effect on every decision. In this case, every selection matrix aligns with our initial predictions.

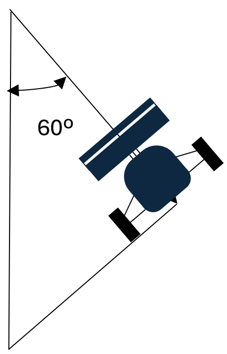
Suspension

Suspension Type:

The first step in designing a suspension is to select a wheelbase and trackwidth. These two parameters determine the foundation for the geometry of the suspension mechanism. The trackwidth is the lateral distance between the centers of the wheels and influences the way the vehicle transfers its weight laterally. The wheelbase is the distance between the centers of the front and rear axles, which affects the weight transfer longitudinally.

A wheelbase of 1550 mm was selected; this wheelbase is 25mm larger than the minimum allowed wheelbase. (SAE International, 2024) . This wheelbase was chosen to give the car good maneuverability while keeping the mass down. The two main sacrifices that are made with a shorter wheelbase are packaging and aerodynamic advantages. A longer wheelbase has an intrinsic ability to make more downforce due to its larger surface area for more complex aerodynamic components to be implemented, like long venturi tunnels or diffusers.

Trackwidth is unrestricted in the FSAE rules; however, it plays a huge role in the rollover stability and its ability to navigate the slalom-style autocross event in the competition. A track width of 1200 mm was selected and changed to 1250mm. To ensure that this track width passes the 60º tilt test. The equation used the 60º angle that is congruent with the angle with respect to the horizontal. A maximum CG height needs to be calculated to ensure and set as a design restriction for the remainder of the project. With the use of some basic Geometry, it was established that the maximum CG height was 360 mm.



$$h_{CG,max} = \frac{t}{2 * \tan{(60º)}}$$

$$h_{CG,max} = \frac{1250\ mm}{2*\tan{(60º)}} = 360 \text{ mm}$$

Requirements	Weight	Double Wishbone	Double Wishbone Pushrod	Mcphearson Strut
Weight	10%	5	4	3
Mechanical Complexity	5%	4	3	5
Handling Characteristics	25%	4	5	2
Drag	15%	3	5	2
Adjustability	15%	3	5	2
Packaging	8%	4	3	5
Unspring Weight	23%	3	5	2
Sum	100%	**3.575**	**4.65**	**2.475**

Weight: The first criterion in the matrix considers the weight of each system. This will help in meeting the weight requirement to keep our rolling chassis under 135kg. It is important to note that this criterion is based on OVERALL weight, not unsprung weight. The system scales between 5 and 1, with 5 corresponding to the lightest weight and 1 corresponding to the heaviest weight. The weights were based on the amount of material used for each design, as all systems would be made of the same materials. The weight for the entire vehicle should ideally be low, but as the difference in weight between the suspension systems would not be a major factor in the total weight, a multiplier of only 10% was applied. The Double Wishbone suspension system was determined to have the lowest weight.

Mechanical Complexity: The second matrix criterion considers the mechanical complexity of the components. More complex designs can be more costly and time-intensive to repair, so the least complex design was given the highest rank of 5, with more complex designs receiving lower ranks. The designs were based on the number and complexity of components in the system. As all suspension systems will require high cost and time to repair if damaged, and the most important criterion should be performance, the criterion was granted the lowest multiplier of 5%. The McPherson strut was determined to have the least mechanical complexity.

Handling Characteristics: The third matrix criterion considers the handling characteristics of each suspension system. This directly connects to our performance requirements, such as passing the relevant FSAE tech tests. Better handling systems were given higher ranks, with the best handling system receiving a rank of 5. This was determined by examining reports and statements by FSAE members as well as the systems themselves. This is the primary objective of the suspension system as it regards FSAE, and as such, was given the highest multiplier of 25%. The Double Wishbone suspension system with push rods was determined to have the best handling.

Drag: The fourth criterion in the matrix considers the drag caused by each suspension system. This was determined by examining the shape of each system. Systems with less drag were given higher ranks, with the system causing the least drag being given the highest rank of 5. As drag is important to vehicle performance, and saddling the Phase 2 team with high drag would hamper their progress, this criterion was given a higher multiplier of 15%. The Double Wishbone with Push Rod suspension system was determined to have the least drag.

Adjustability: The fifth criterion is with respect to the ability to adjust the suspension system once it's designed. This helps meet the requirement relating to keeping the project in a good state for the phase 2 team to modify and complete. More adjustable systems are given higher weights, with the most adjustable given the highest rank of 5. The adjustability of each matrix was determined by examining the number of adjustable points on each system and the extent to which the camber/toe/etc. Could be altered. Adjustability is very important, as optimized suspension requires weighting information that won't be known until phase 2 of the project, so it was given a high multiplier of 15%. The Double Wishbone with Pushrod suspension system was determined to have the most adjustability.

Packaging: The sixth criterion refers to the packaging, or the amount of space the suspension system will take up. Systems taking up less space were granted higher ranks, with the least space-intensive system being given the highest weight of 5. The packaging was determined by looking at the volume of each system. Ensuring proper packaging space is important for the Phase 2 readiness requirement, but as the three suspension system options occupy very similar spatial requirements, it was given a lower multiplier of 8%. The McPherson Strut was determined to take up the least space.

Unsprung Weight: The final criterion considers the unsprung weight of each system, helpful for both total weight (which is required to be kept under 135kg) but also performance. Systems with a lower unsprung weight were granted higher ranks, with the system with the least unsprung weight being given the highest rank of 5. The Unsprung weight was determined by examining the number and weight of the unsprung components of the system. Unsprung weight has much more impact on performance than sprung weight, and, as a result of this, and the importance of handling and performance to the project, this criterion received a high multiplier of 23%. The Double Wishbone with Pushrod suspension system was determined to have the lowest unsprung weight.

The total weighted score for the Double Wishbone with Pushrod suspension system received a score of 4.65, the highest of the three systems, due to its determination as the suspension system that will result in the best performance and handling.

Anti-Roll:

Requirements	Weight	Anti-Roll Bar (Blade)	Sway Bar (Solid)	Sway Bar (Hollow)
Weight	15%	5	3	4
Mechanical Complexity	10%	3	5	4
Performance	20%	5	3	4
Cost	10%	3	5	4
Adjustability	20%	5	4	4
Packaging	25%	3	2	2
Sum	100%	**4.1**	**3.35**	**3.5**

Weight: The first matrix criterion regards the weight of each anti-roll concept. This is necessary to aid in meeting the requirement that the weight must be kept under 135 kg. The lighter systems were given higher rankings, with the lightest receiving the highest rank of 5 and the heavier systems receiving lower rankings. This criterion was judged by looking at the volume and material used in each system. A multiplier of 15% was applied to this criterion as there is significant variability in the weight of the different options. The Anti-Roll blade design was determined to have the lowest weight.

Mechanical Complexity: The second criterion refers to the mechanical complexity of the systems designed. Systems with higher mechanical designs were granted higher ranks, with the least complex system being given the highest weight of 5. To determine the mechanical complexity of each system, the number of components and the complexity of each element were examined. All three systems would take roughly the same time and cost to repair, so they were granted a lower multiplier of only 10%. The Solid Sway Bar was determined to be the least complex.

Performance: The third criterion in the matrix considers the performance of each system, which is important for meeting performance-based requirements such as passing relevant FSAE tech tests. Systems with higher performance were given higher ranks, with the system judged to have the best performance being given the highest rank of 5. This was determined by questioning Andretti Motorsports staff about their anti-roll design and examining system performance judgments online. As performance and handling are extremely important to suspension design, this criterion was given a higher multiplier of 20%. The Anti-Roll blade was determined to have the highest performance.

Cost: The fourth matrix criterion considers the cost of the components. The project has a limited budget, so the system with the cheapest design was given the highest rank of 5, with more expensive designs receiving lower ranks. This ties into our fundraising requirement, as spending less on the first phase will help leave more money for the next. The determination of the cost was reached by looking at the prices of the constituent components and materials. As the difference in system cost was comparatively small compared to the budget, the cost was given a lower multiplier of 10%. The Solid Sway Bar was determined to have the lowest cost.

Adjustability: The fifth matrix criterion considers the adjustability of each suspension system. This helps fulfill the requirement relating to keeping the project in a good state for the phase 2 team. More adjustable systems were given higher ranks, with the most adjustable system receiving a rank of 5. This was determined by examining the range and ease of adjusting each system post-design and implementation. As the higher adjustability would help fulfill anti-roll requirements, the system was given a higher weight of 20%. The Anti-Roll Blade was determined to have the most adjustability.

Packaging: The final criterion in the matrix considers the packaging size required by each system. This was determined by examining the volume of each system's components. Systems with a lower packaging size were given higher ranks, with the system containing the smallest packaging size being given the highest rank of 5. As there is a large difference in packaging size between the options, and it would fulfill Phase 2 readiness requirements to have a smaller packaging size used, and as such, more available for the second phase, this criterion was given a higher multiplier of 25%. The Anti-Roll Blade system was determined to have the lowest packaging size.

The total weighted score for the Anti-Roll Blade system received a score of 4.1, the highest of the three systems, due to its determination as the system that will result in the best performance, and taking up the lowest weight and packaging size.

Chassis

Requirements	Weight	Steel Tubing	Full Monocoque	Front/Rear Tube Carbon Cockpit	Carbon Monocoque w/ Tube Rear
Mass	18%	2	5	4	4
Stiffness	16%	2	5	3	4
Cost	12%	4	1	2	3
Manufacturability	18%	5	1	3	3
Recylability	3%	5	1	2	2
Safety	10%	5	4	4	5
Repairability	6%	5	2	4	3
Preparation for Next Year's Team	10%	4	2	3	5
Subsystem Integration	7%	5	4	4	3
Sum	100%	**3.76**	**3.03**	**3.26**	**3.71**

Cost**:** The cost criterion was heavily influenced by the project's limited budget and available manufacturing resources. As a student-built prototype, minimizing expenses was essential to ensure feasibility within a single academic year. Designs requiring advanced tooling, molds, or specialized fabrication techniques received lower rankings, while those that could be produced with basic welding and tubing equipment scored higher. Consequently, the steel space frame achieved the highest score in this category due to its low material cost and the accessibility of required tools, while the full carbon monocoque received the lowest score due to the significant expense of carbon fiber materials and accompanying processes.

Manufacturability**:** Manufacturability was weighted highly because the team must complete the chassis within a single academic year using university resources, as well as some limited outside resources from sponsors. The space frame ranked highest, as it can be produced using standard welding and fixturing methods without specialized training or equipment. The partial monocoque and full monocoque designs scored lower due to their complex layup processes, requirements for molds, and potentially curing ovens. Simplicity and reliability during fabrication were key factors, leading to a high weighting of 18%.

Mass and Stiffness: Mass and stiffness were both considered critical for performance. While the monocoque concepts offered potential for greater stiffness-to-weight ratios, research indicated that an optimized space frame could still achieve adequate torsional stiffness while remaining within the overall weight goal.

Safety and Repairability: Safety and repairability are not among the major considerations for chassis configuration; however, the team still viewed them as important. There is the possibility that the Phase Two team breaks something during testing, and it is important that the frame can be salvaged as much as possible. The space frame, within reason, can be cut and have new tubing welded in, so long as other stress areas are not compromised. Composite monocoques, however, cannot be easily repaired. Monocoques also risk total failure from even light incidents, as a crack or delamination can propagate throughout the entire chassis.

Recyclability and Next Year's Team: Although these criteria carried smaller weightings, they did contribute to the final decision. Steel's recyclability and the modular design of a space frame make it advantageous when presenting a general recommendation of parts needed to finish the car to next year's team. If the Phase Two team is presented with a monocoque, they cannot easily make any modifications to what the Phase One team has completed or recommended. The recyclability criterion also considers the ability to reuse components after Phase Two of the project.

Subsystem Integration**:** Integration was included to evaluate how easily other systems (suspension, steering, and drivetrain) could be mounted. The space frame scored highest because its tube-based structure allows for flexible placement of mounting tabs and brackets, while monocoque designs require additional inserts or bonded connections, increasing both complexity and weight.

The total weighted scores showed that the steel space frame design achieved the highest overall ranking with a score of 3.76. While the monocoque and hybrid designs offered theoretical performance benefits, the complexity, cost, and manufacturing challenges outweighed the advantages. The space frame was therefore selected as the optimal choice for a first-year FSAE project, providing an achievable balance of stiffness, manufacturability, cost-effectiveness, and Phase Two adaptability.

Brakes

Requirements	Weight	4 Rotors	Limited Slip Differential + 2 Front Rotors
Weight	20%	4	5
Mechanical Complexity	15%	5	3
Performance	25%	5	2
Cost	10%	3	5
Adjustability	20%	5	3
Packaging	10%	5	4
Sum	100%	**4.6**	**3.45**

Weight: This was given a 20% weight as keeping the car as light as possible is one of our main criteria. The limited-slip differential was given a higher score than the 4-rotor, as it would have fewer components. However, the 4-rotor was rated right below, as weighing each wheel would improve the stability of the car and increase control.

Complexity: This was given a 15% weighting as it does not play as big a factor affecting the performance of the vehicle. The 4-rotor uses a simple design of a hydraulic piston clamping onto the brake rotor, whereas the limited-slip incorporates a brake rotor attached to the rear drive shaft, braking with a singular caliper. This would require extra mounting points for the caliper, use of a limited-slip differential, and taking up space in the engine bay.

Performance: This was given a 25% weight because if the brakes don't effectively slow the car down quickly while still maintaining control in corners, then they should not be used. The 4-rotors provide stability at all 4 wheels, even braking, and better heat dissipation from the rotors. The limited-slip would have a more centered mass, but there would be less mass directly on the rear wheels, possibly leading to the rear sliding out on corners.

Cost: This was given a 10% weighting as it is important to save money, but it is also a very critical system, so corners must not be cut. The limited-slip was generally the cheaper option, as you would have to buy one less caliper and rotor, along with only having to run fewer brake lines. Limited-slip differentials can be bought for a little more than performance calipers.

Adjustability: This was given a 20% weighting as being able to adjust and tune the individual wheel braking is an important part of improving performance. The 4-rotor would be able to adjust each caliper at the wheel, whereas the limited-slip would have the same setting for both rear wheels since they run off one caliper.

Packaging: This was given a 10% weighting as the space the component takes up is important for both aerodynamics and component arrangements, which could affect weight. The limited-slip uses a more compact system, freeing up space around the rear wheels, but provides complexity in the axle, while the 4-rotor uses an already space in the wheel well to mount the caliper and rotor.

Steering

Requirements	Weight	Rack and Pinion	Steering Box	4 Wheel
Complexity	15%	5	2	1
Responsiveness	30%	4	3	5
Mass	40%	4	3	2
Cost	15%	3	5	2
Sum	100%	**4.00**	**3.15**	**2.75**

Complexity: The first matrix criterion has the least weight with cost, and this criterion considers how many components, the durability, and the repairability of the system. The least complex systems were given a score of 5, with the most complex a 1. The rack and pinion was given the highest score due to the low number of components when compared to the steering box and 4-wheel setup. The rack and pinion is also the simplest steering setup out of the 3. The rack and pinion was determined to be the least complex since it requires fewer moving parts, which leads to fewer possible areas of failure and less repair time. This makes the rack and pinion the clear choice for the lowest complexity.

Responsiveness: The second matrix criterion has the second most weight, and considers the force required to steer and the driver's feedback of the system. The most responsive systems were given a 5, with the least responsive given a 1. The rack and pinion was determined to have the most responsiveness. The rack and pinion systems reduce the amount of steering input required compared to other setups. (W. F. Milliken, 1995). The improved feedback allows the grip limits of the tires to be felt while driving the car, leading to more precise control and faster reaction times on track. For these reasons, the rack and pinion had the highest responsiveness rating.

Mass: The third matrix criterion has the most weight out of all the systems, the most important for a car. Systems with the most mass were given a 1, and systems with the least mass were given a 5. The rack and pinion were determined to have the least mass.

Cost: The final matrix criterion has the least weight. While being important to the scope of the project, the mass and the responsiveness were more important to succeeding in the FSAE races and meeting the customer's needs and requirements. Systems with the highest cost were given a 1, while systems with the least cost were given a 5. The steering box was determined to have the least cost.

Critical Design phase

Fundraising requirements

The team was tasked with fundraising $10,500 by September 5, a feat the team met and surpassed by funding $15,500 by the week. The team has also received sponsorship from Industrial Contracting and Engineering in Angola. They have committed to supporting the team with unlimited access to tooling, machinery, engineering support, as well as any raw materials they have in stock. This is a huge contribution, as it is likely the team will be able to utilize this to cut costs on many subsystems.

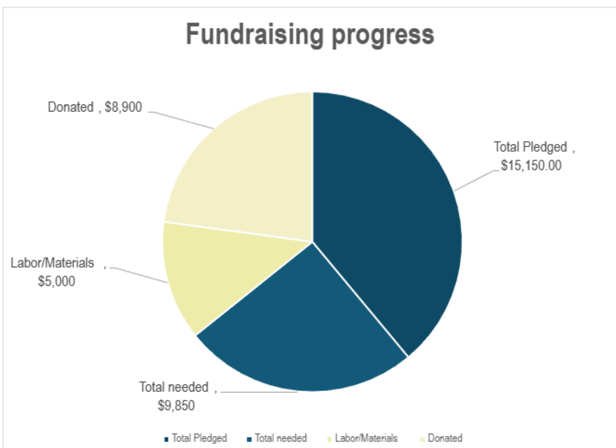


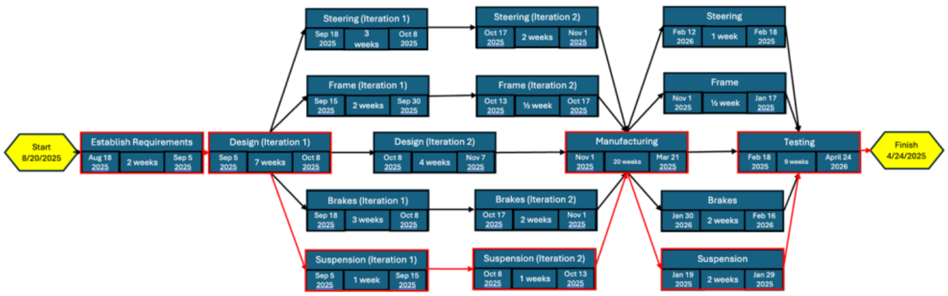
Figure 6 Pie Chart for fundraising progress as of (9/9/2025)

Project Management

Gantt chart:

A detailed Gantt chart with updated manufacturing and testing plans was created to ensure the project remained on schedule.

PERT chart



Start 8/20/2025
Establish Requirements
Aug 18 2025
2 weeks
Sep 5 2025
Design (Iteration 1)
Sep 5 2025
7 weeks
Oct 8 2025
Steering (Iteration 1)
Sep 18 2025
3 weeks
Oct 8 2025
Frame (Iteration 1)
Sep 15 2025
2 weeks
Sep 30 2025
Brakes (Iteration 1)
Sep 18 2025
3 weeks
Oct 8 2025
Suspension (Iteration 1)
Sep 5 2025
1 week
Sep 15 2025
Steering (Iteration 2)
Oct 17 2025
2 weeks
Nov 1 2025
Frame (Iteration 2)
Oct 13 2025
½ week
Oct 17 2025
Design (Iteration 2)
Oct 8 2025
4 weeks
Nov 7 2025
Brakes (Iteration 2)
Oct 17 2025
2 weeks
Nov 1 2025
Suspension (Iteration 2)
Oct 8 2025
1 weeks
Oct 13 2025
Manufacturing
Nov 1 2025
20 weeks
Mar 21 2025
Steering
Feb 12 2026
1 week
Feb 18 2025
Frame
Nov 1 2025
½ week
Jan 17 2025
Brakes
Jan 30 2026
2 weeks
Feb 16 2026
Suspension
Jan 19 2025
2 weeks
Jan 29 2025
Testing
Feb 18 2025
9 weeks
April 24 2026
Finish 4/24/2025

BOM

The BOM currently accounts for most major components in all subsystems, as well as most of the raw materials necessary to manufacture the car. The main things missing from the BOM are the fasteners and miscellaneous items necessary for manufacturing and assembly. The suspension tabs and mounts are still being designed, and as such, they are not included in the BOM.

Item	Brand	Model/Dimensions	Quantity	List price	FSAE Discoun	Full Pric	Shipping/fee	FSAE Pric	Subsyste	Link
Calipers (LH)	Wilwood	N/120-5453 - Caliper-PS 1 LH	2	$ 116.55	26.67%	$ 233.10		$ 170.93	Brakes	wilwood.com
Calipers (RH)	Wilwood	N/120-5453 - Caliper-PS 1 RH	2	$ 116.55	26.67%	$ 233.10		$ 170.93	Brakes	wilwood.com
Pads	Wilwood	4108	4	$ 65.97	26.67%	$ 263.88		$ 193.50	Brakes	wilwood.com
Master Cyinder	Tilton	78 Series 5/8"bore	3	$ 295.00	27.50%	$ 885.00		$ 641.63	Brakes	tiltonracing.com
Balance bar	Tilton	600 Series	1	$ 74.00	27.50%	$ 74.00		$ 53.65	Brakes	tiltonracing.com
Front wheel hub kit	RCV performance	FW220HA	2	$ 1,089.00	0.00%	$ 2,178.00		$ 2,178.00	Brakes	rcvperformance.com
Rear wheel hub kit	RCV performance	D5580-HA	2	$ 770.55	0.00%	$ 1,541.10		$ 1,541.10	Brakes	rcvperformance.com
1026 tubing	N/A	1.000" OD 0.065" wall	1	$ 2,500.00	0.00%	$ 2,500.00	$ -	$ 2,500.00	Chassis	N/A
1026 tubing	N/A	1.000" OD 0.095 wall	1	$ 1,000.00	0.00%	$ 1,000.00	$ -	$ 1,000.00	Chassis	N/A
1026 tubing	N/A	1.750" OD 0.065 wall	1	$ 500.00	0.00%	$ 500.00	$ -	$ 500.00	Chassis	N/A
1026 plate	N/A					$ -	$ -	$ -	Chassis	N/A
Impact attenuator	Plascore	FSAE Impact attenuator	2	$ 75.00	0.00%	$ 150.00	$ -	$ 150.00	Chassis	
Steering rack	NARRco	V2 (11.4" 13T)	1	$ 650.00	0.00%	$ 650.00	$ 25.96	$ 675.96	Steering	fsaeparts.com
Steering rack bolts	Mcmaster Carr	AN4-10A(10 count)	1	$ 8.62	0.00%	$ 8.62		$ 8.62	Steering	Mcmastercarr.com
Rack mounts	NARRco	N/A	1	$ 75.00	0.00%	$ 75.00		$ 75.00	Steering	fsaeparts.com
U-Joint	Sweet Manufacturing	3/4" Smooth	1	$ 132.90	0.00%	$ 132.90		$ 132.90	Steering	sweetmanufacturing.com
Column Tube	Speedy Metals	3/4" OD .065 Wall (2ft)	1	$ 26.32	0.00%	$ 26.32		$ 26.32	Steering	speedymetals.com
Bearing	Grainger	3/4" Bore 2" OD	1	$ 9.88	0.00%	$ 9.88		$ 9.88	Steering	grainger.com
Quick Disconnect	Race Parts	5/8" OD (slug)	1	$ 183.15	0.00%	$ 183.15		$ 183.15	Steering	race-parts.com
Spring pin	Bolt Depot	1/8" x 1"	2	$ 0.14	0.00%	$ 0.28		$ 0.28	Steering	boltdepot.com
Bearing to Support Tube	Speedy Metals	2-1/4" OD 2.010" ID (1-1/2")	1	$ 2.91	0.00%	$ 2.91		$ 2.91	Steering	speedymetals.com
Support to Frame Tube		1" OD (1 ft)			0.00%	$ -		$ -	Steering	
Rack Mount Bolts	NARRco	1/4" D 1" L (min)			0.00%	$ -		$ -	Steering	
Raack Mount Plate	Online metals	3"x16"x0.25"	1	$ 49.21	0.00%	$ 49.21		$ 49.21	Steering	onlinemetals.com
Rack Mount Extension	Online metals	2.25"x0.5"x0.5"	2	$ 35.50	0.00%	$ 71.00		$ 71.00	Steering	onlinemetals.com
Tires	Hoosier	20.5X 7.0-13 R20A	4	$ 338.00	15.00%	$ 1,352.00	$ 86.54	$ 1,235.74	Suspension	hoosiertirewest.com
Shocks	Ohlins	TTX25 mkii	1	$ 2,600.00	0.00%	$ 2,600.00	$ -	$ 2,600.00	Suspension	performanceshocks.com
Wheels	Keizer	Slepen	4	$ 375.00	0.00%	$ 1,500.00	$ 110.00	$ 1,610.00	Suspension	Keizerwheels.com
6061 T6 Billet	Online metals	6"x12"x2"	2	$ 247.18	0.00%	$ 494.36	$ -	$ 494.36	Suspension	Onlinemetals.com
6061 T6 Billet	Online metals	8"x12"x2"	2	$ 330.49	0.00%	$ 660.98	$ -	$ 660.98	Suspension	Onlinemetals.com
4130 tubing	N/A	1/2" OD x .049"	224	$ 0.55	0.00%	$ 123.20		$ 123.20	Suspension	stockcarsteel.com
Spehrical bearings	FK rod ends	HIN4T	16	$ 14.06	35.49%	$ 224.96		$ 145.12	Suspension	fkrodends.com
Rod ends 1/4" (RH)	Fk rod ends	JMX4T	8	$ 17.99	38.02%	$ 143.92		$ 89.20	Suspension	fkrodends.com
Rod ends 1/4" (LH)	Fk rod ends	JMXL4T	8	$ 17.99	38.02%	$ 143.92		$ 89.20	Suspension	fkrodends.com
Rod ends .19" (RH)	Fk rod ends	JMX3T	4	$ 17.99	38.35%	$ 71.96		$ 44.36	Suspension	fkrodends.com
Rod ends .19" (LH)	Fk rod ends	JMXL3T	12	$ 17.99	38.35%	$ 215.88		$ 133.09	Suspension	fkrodends.com
Springs	Hyperco	18FS	4	$ 54.00	0.00%	$ 216.00		$ 216.00	Suspension	hypercoils.com
Rod End spacer	Midwest Control	DLS-04	25	$ 1.86	0.00%	$ 46.50		$ 46.50	Suspension	midwestcontrol.com
Suspension bolts			2	$ 14.16	0.00%	$ 28.32		$ 28.32	Suspension	Mcmastercarr.com
Suspension nuts			1	$ 8.00	0.00%	$ 8.00		$ 8.00	Suspension	Mcmastercarr.com
1026 plate			1			$ -		$ -	Suspension	Mcmastercarr.com
CF tubes	Clearwater composites	1/2" ID x .744" OD 72"	1	$ 164.00	15.00%	$ 164.00	$ 27.13	$ 166.53	Suspension	Clearwatercomposites.com
4130 round stock	Metal supermarket	1" 10" length	1	$ 43.03	0.00%	$ 43.03		$ 43.03	Suspension	Metalsupermarket.com
Snap ring	N/A	98455A124	1	$ 4.92	0.00%	$ 4.92		$ 4.92	Suspension	Mcmastercarr.com
Ball bearing	SKF	6000-2Z	4	$ 7.35	0.00%	$ 29.40		$ 29.40	Suspension	amazon.com
Carbon fiber inserts	Dragonplate	.75" short threaded insert	10	$ 15.00		$ 150.00		$ 150.00	Suspension	Dragonplate.com
Anti-Roll Bolts	Mcmaster Carr	92240A542	1	$ 8.92	0.00%	$ 8.92		$ 8.92	Suspension	Mcmastercarr.com
Anti-Roll nuts	Mcmaster Carr	94846A505	1	$ 7.69	0.00%	$ 7.69		$ 7.69	Suspension	Mcmastercarr.com
Anti-Roll Rod End Nuts	Mcmaster Carr	90760A150	1	$ 6.73	0.00%	$ 6.73		$ 6.73	Suspension	Mcmastercarr.com
Anti-Roll Link Tubing	Mcmaster Carr	91125A501	4	$ 3.06	0.00%	$ 12.24		$ 12.24	Suspension	Mcmastercarr.com
ARB Lever Arm sheets	McMaster Carr	9146T32 (1ft length)	2	$ 4.17	0.00%	$ 8.34		$ 8.34	Suspension	Mcmastercarr.com
ARB Tube (Placeholder)	McMaster Carr	9965K357 (6ft length)	1	$ 40.75	0.00%	$ 40.75		$ 40.75	Suspension	Mcmastercarr.com
Pedal Pivot Pin	McMaster Carr	98306A679	2	$ 8.41	0.00%	$ 16.82		$ 16.82	Brakes	Mcmastercarr.com
Pedal Box Rails Sheet Metal	McMaster Carr	2' x 3" x 1/8"	1	$ 22.27	0.00%	$ 22.27		$ 22.27	Brakes	Mcmastercarr.com
Pedal box mount plate	McMaster Carr	2' x 10" x 1/8"	1	$ 9.69	0.00%	$ 9.69		$ 9.69	Brakes	Mcmastercarr.com
Pedal mount plate	McMaster Carr	12" x 4 1/2" x 1/4"	1	$ 16.71	0.00%	$ 16.71		$ 16.71	Brakes	Mcmastercarr.com
Pedal face plate	McMaster Carr	6" x 4 1/2" x 1/8"	1	$ 5.74	0.00%	$ 5.74		$ 5.74	Brakes	Mcmastercarr.com
Pedals	McMaster Carr	12" x 4 1/2" x 1"	1	$ 4.81	0.00%	$ 4.81		$ 4.81	Brakes	Mcmastercarr.com

Subtotal:	$ 18,408.83
Suspension	$ 8,002.63
Chassis	$ 4,150.00
Brakes	$ 4,949.74
Steering	$ 1,235.23

Figure 7: BOM with subsystem breakdown

The current BOM includes the cost of hub kits, which is nearly $5,000. A proposal for additional funding has been submitted to the Dean of Engineering, which would help cover this cost. In case the dean does not approve of the proposal, some additional design work will be needed to manufacture proprietary hubs.

Phase 2 integration:

For smooth integration with phase 2, careful steps have been taken when designing the rolling chassis. Many of the large components are included in the CAD model, including the engine, differential, battery, and fuel cell.

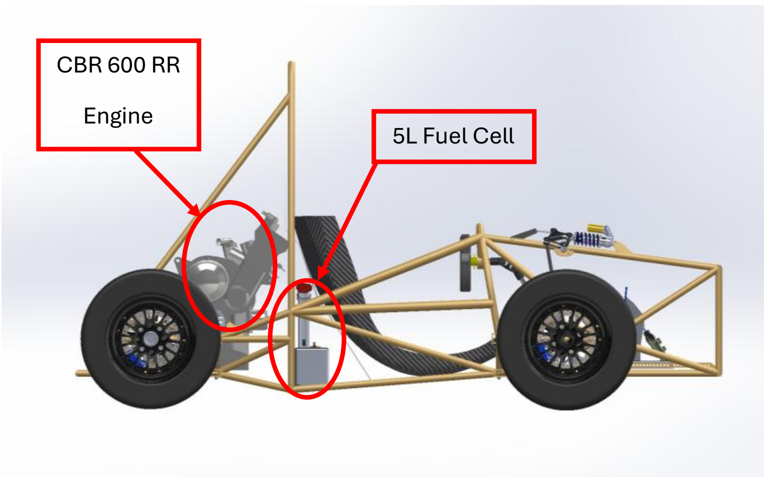


Figure 8: Right Side View of the full assembly

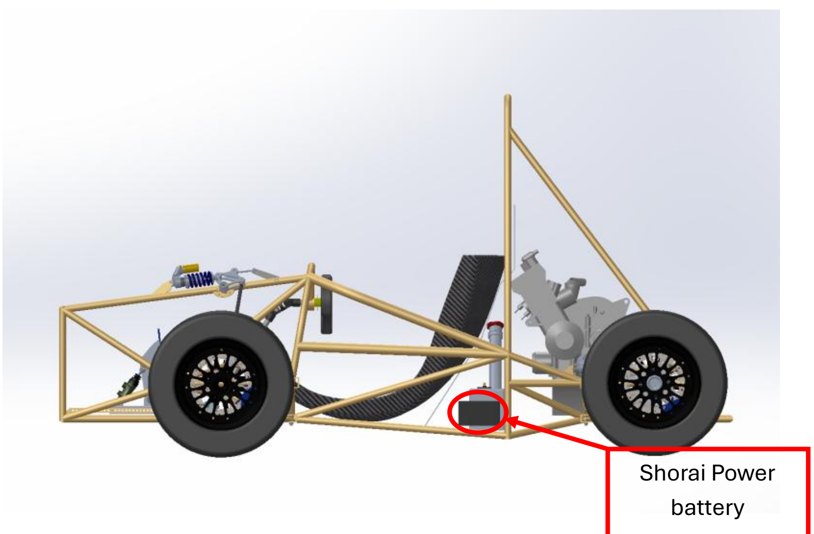


Figure 9: Left Side view of the full assembly

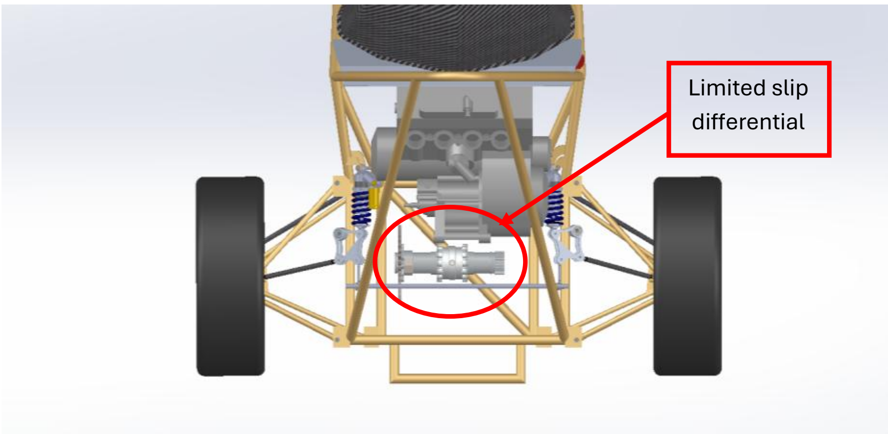


Figure 10: Top View of the full assembly

The differential has been placed in the ideal location to ensure there are no issues with clearances between the driveshaft and the rear pushrod, as well as maximum misalignment angles between the differential and the wheel hubs.

Regarding the steering subsystem, a steering wheel has been designed with a cutout in the middle to accept a small LCD with switches for improved driver controls and visibility of vital engine and drivetrain parameters.

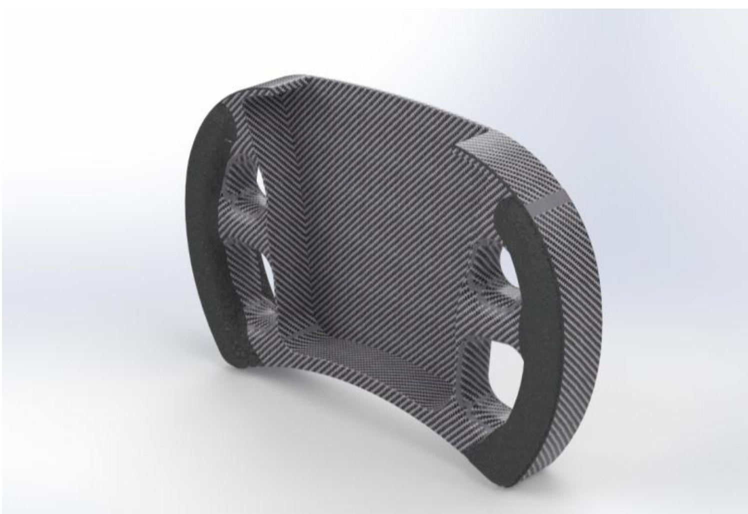


Figure 11: Steering wheel model

Additionally, the team has worked in collaboration with Ben Mendenhall, who will be the driver for the competition. Things like steering wheel location, ideal sitting position, as well as seat shape have been optimized to his preferences.

Design

Suspension

Suspension Geometry/OptimumK:

The suspension geometry setup in OptimumK went through 13 major published iterations over the course of the past two months, changing almost every aspect of the design to accommodate requests from other subsystems as the design of those subsystems progressed, as well as meeting desired values for caster, camber, toe, scrub radius, roll centers, and the like. The final suspension geometry, as of 12/4, is shown in the figures below:

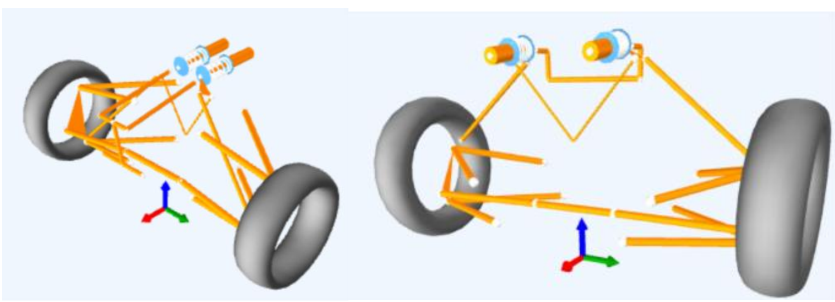


Figure 12: Front Suspension Geometry, with the concept design (left) and final design as of 12/4 (right)

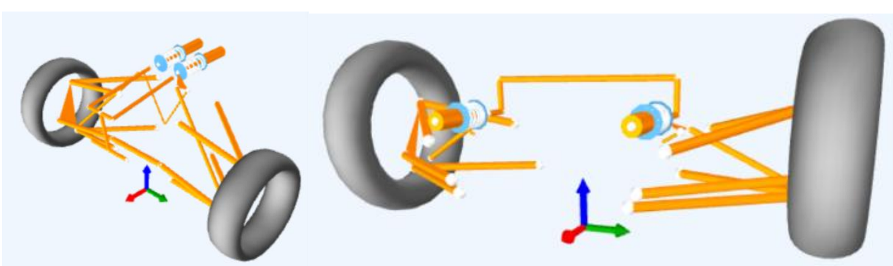


Figure 13: Rear Suspension Geometry, with the concept design (left) and final design as of 12/4 (right)

The Suspension now has different setups for the pushrods and rockers, as well as the anti-roll flipped in a different direction between the front and rear setups, although the geometry of the wishbones remains the same between each side. This geometry results in the following values:

Criterion	Kingpin Inclination	Caster	Scrub Radius	Center of Gravity	Front Roll Center	Rear Roll Center
Value	7.17	8.62	23.2	203.8	96.9	108
Unit	degrees	degrees	mm	mm	mm	mm

Figure 14: Table of Suspension Values generated by Optimum Kinematics

All values are within acceptable parameters to pass applicable FSAE tech tests, meeting our seventh requirement relating to passing all applicable FSAE tech tests. The suspension geometry was also designed to keep weight low in accordance with our second requirement, and to aid in the execution of our first requirement relating to performance. The Roll Center is slightly higher than optimal, but a compromise that needed to be made to accommodate other subsystems.

The final suspension pickup points were made to fit along specific tubes on the frame to allow for easier mounting, as well as upright points to accommodate the required steering ratio via the toe rod position. Other considerations included preventing binding, ensuring proper suspension travel, and confirming camber angle gain, all of which were checked manually in CAD.

Suspension Calculations:

Suspension Force and spring rate calculations were done via Excel. By setting up matrices based on the suspension geometry and information such as the car mass and force at each tire contact patch, the force at each chassis mounting point was located. From that, another table was created with a variety of force inputs depending on the motion the car is expected to undergo. The figures below serve to provide a rough overview of the calculation process (moving from pickup points to necessary forces and moments to the calculation of the operating forces and are not meant to be legible.

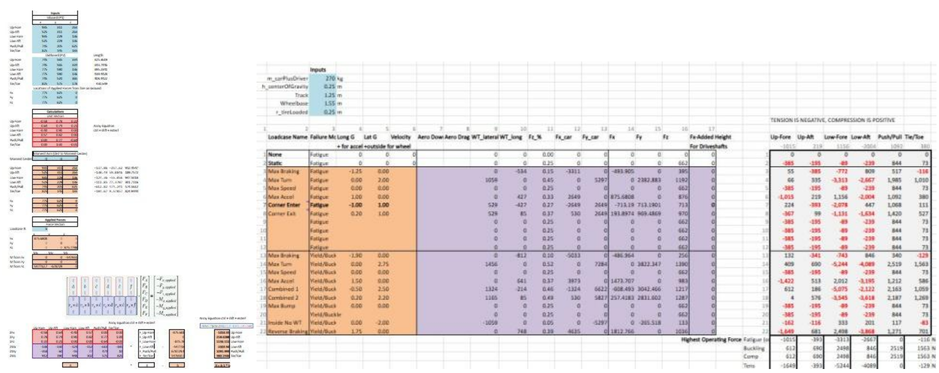


Figure 15: Suspension Calculation Overview

The forces were calculated in the X, Y, and Z directions at different load cases according to the following equations:

$$WT_{lateral} := \frac{m_{carPlusDriver} \cdot accel_{lat} \cdot h_{centerOfGravity}}{Track} = 588\ N$$

$$WT_{longitudinal} := \frac{m_{carPlusDriver} \cdot accel_{long} \cdot h_{centerOfGravity}}{Wheelbase} = 441\ N$$

$$F_{z\_applied} := \frac{m_{carPlusDriver} \cdot g}{4} + \frac{\left(WT_{lateral} + WT_{longitudinal}\right)}{2} = 1250\ N$$

$$F_{z\_percent} := \frac{F_{z\_applied}}{m_{carPlusDriver} \cdot g} = 43\%$$

This share of the vehicles weight on our wheel of interest.

$$F_{y\_car} := m_{carPlusDriver} \cdot accel_{lat} = 2942\ N$$

Total lateral force on the car (sum of the 4 tires)

$$F_{y\_applied} := F_{y\_car} \cdot F_{z\_percent} = 1250\ N$$

Assume that the lateral force of each tire is directly proportional to it's share of the entire vehicle's lateral force. This is a major oversimplification, and has much room for improvement.

Figure 16: Equations used to determine load case values, from FSAE wiki

Once the different load case forces were calculated, the pickup points were translated to fit the format of the sheet and converted into vectors. From there, unit vectors of the direction of the different rods, such as push rods and fore/aft wishbones, were calculated, and through transposing and matrix multiplication, the resulting matrices were used with the load cases to generate the load along different components (the pushrod, fore/aft wishbones), etc. at each load case and at each mounting point.

The final table of the maximum force at each mounting point is shown below. In this table, compression is positive, and tension is negative.

Highest Operating Force	Fatigue (o	-1015	-393	-3313	-2667	0	-116	N
	Buckling	612	690	2498	846	2519	1563	N
	Comp	612	690	2498	846	2519	1563	N
	Tens	-1649	-393	-5244	-4089	0	-129	N

Figure 17: Highest Operating Forces at each pickup point

These forces were then used to perform finite element analysis on the suspension system, and based on current FEA testing, are within acceptable parameters. No components are shown to fail under these load cases, meaning they should meet our requirements.

The spring rate also needed to be calculated to determine which springs to purchase. This was the primary equation used to determine the spring rate:

$$Ks = 4 \, x \, \pi^2 \, x \, Fr^2 \, x \, Msm \, x \, MR^2$$

Figure 18: Primary Equation used to determine the Spring Rate Ks.

In the above equation, Fr refers to the Ride Frequency, Msm to the Sprung Mass, and MR to the Motion Ratio. Based on the calculations done, it was decided to use 400lb/in springs to achieve suspension travel around 7-8mm above the 50mm minimum to ensure any adjustments during the manufacturing process will still allow the suspension to pass tech (meeting our requirement of passing all technical tests).

Suspension CAD:

Suspension CAD was done according to the pickup points and aimed to keep weight low, as our customer's needs/requirements require, while being able to handle the calculated forces, keeping sustainability in mind, as well as to fit the specific Rod Ends from our supplier. Much optimization was done before the semi-final suspension CAD was developed, as shown in the figures below, in line with the frame. During CAD design, the anti-roll location was also changed (as is reflected in the Optimum K geometry above) to better accommodate both other systems, such as extending and moving the bar location to allow for easier mounting, as well as to aid the Phase 2 Readiness requirement. The Anti-Roll is currently set up to both take up minimal space inside the chassis as well as stay low to avoid impacting any aero panels added in Phase 2. The Anti-Roll CAD is shown in the figures below, attached to the suspension system.

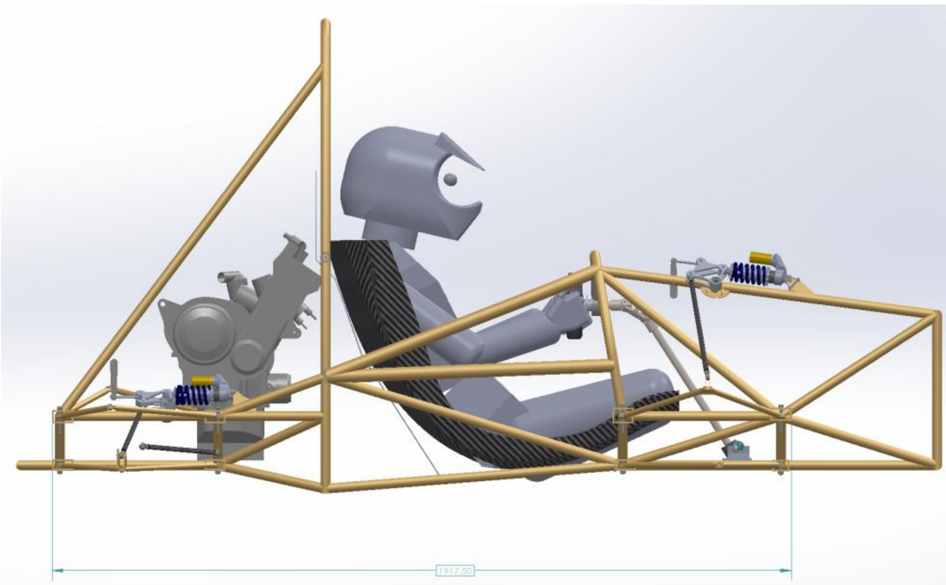


Figure 19: Side View of Suspension System on FSAE Vehicle

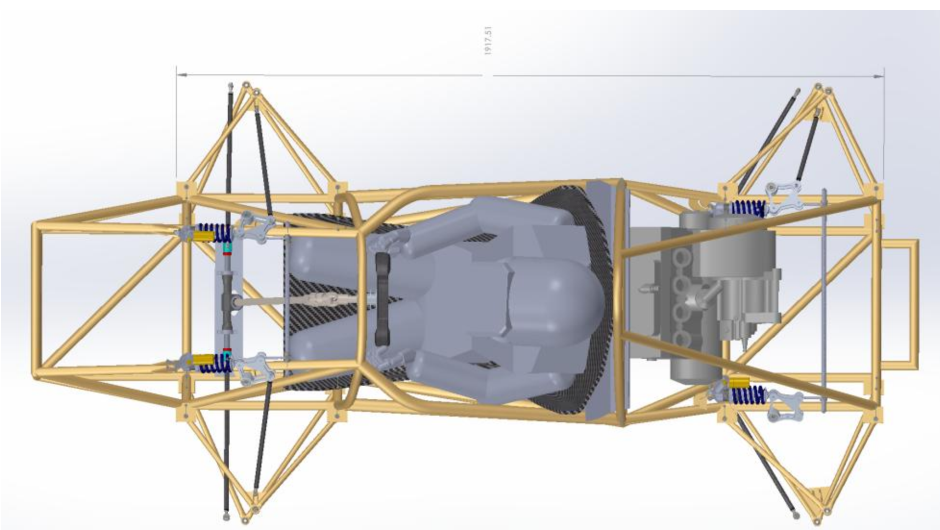


Figure 20: Top View of Suspension System on FSAE vehicle

(Dimension for scaling in mm)

Note: The rocker shown does not include the final topology optimization.

The total unsprung weight (including the wheels and tires) is roughly 77kg, putting us in line for a total weight of 270kg for the car and driver, 135kg for the rolling chassis as per our requirements.

Suspension FEA

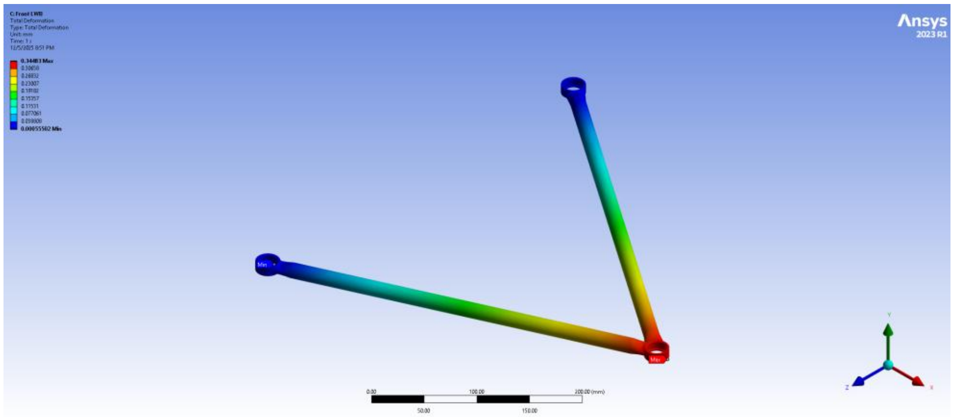


Figure 21: Front LWB Total Deformation

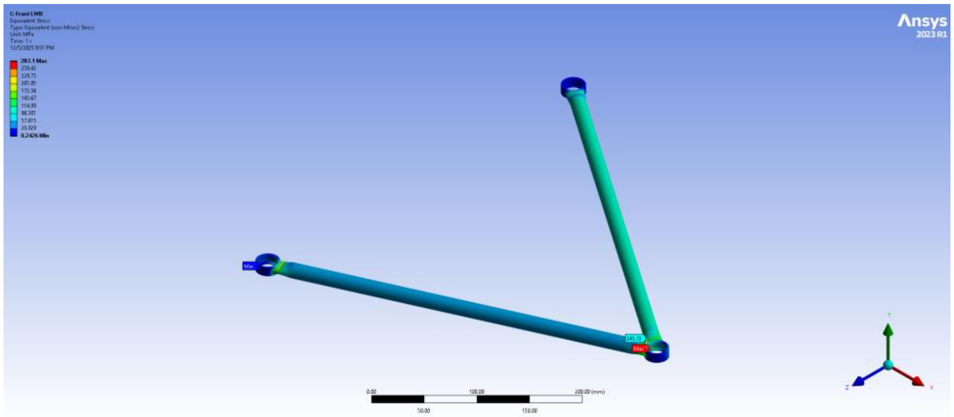


Figure 22: Front LWB Equivalent Stress

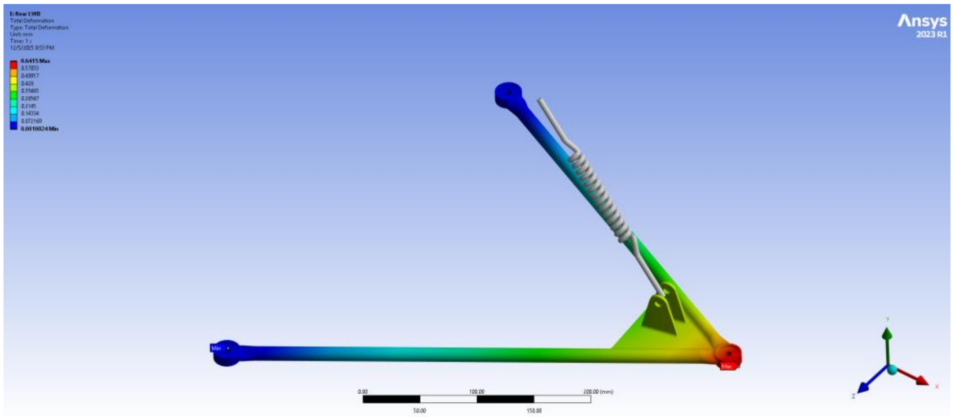


Figure 23: Rear LWB total deformation

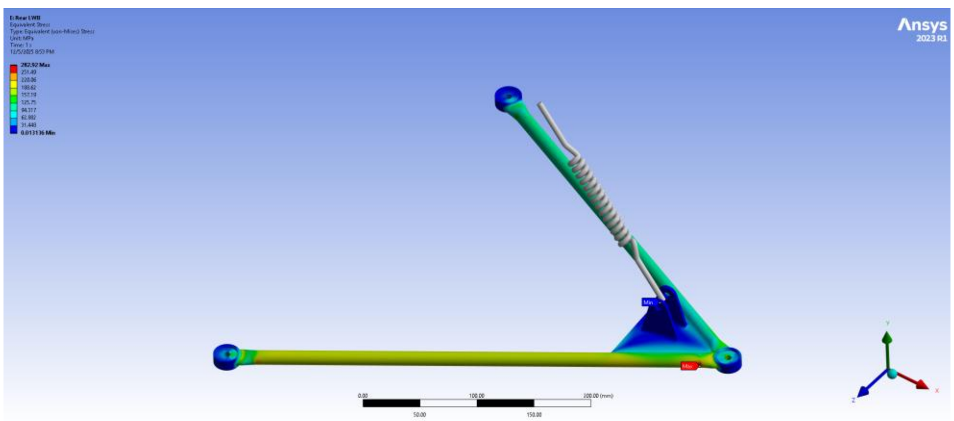


Figure 24: Rear LWB equivalent stress

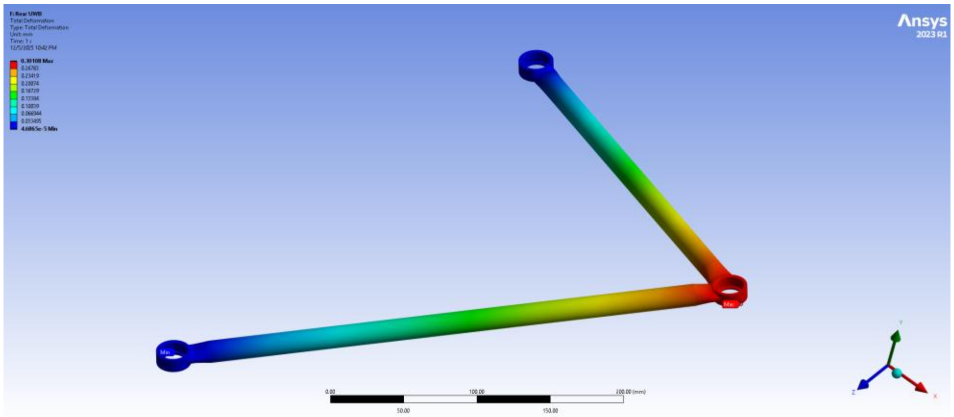


Figure 25: Rear UWB total deformation

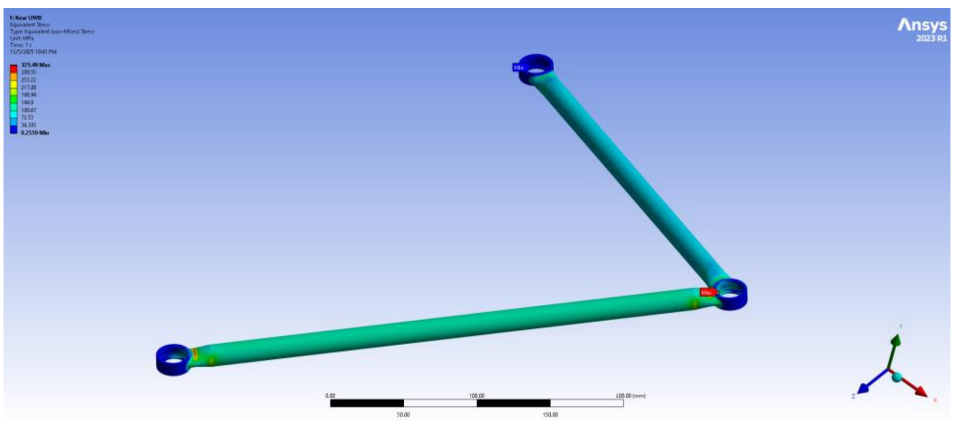


Figure 26: Rear UWB equivalent stress

The wishbones all have consistent and low average stress throughout. The front LWB has an average stress of 67.65 MPa, around 104.81 MPA for the rear LWB, and about 95 MPA for the rear UWB. There are some points with significantly higher concentrations because of discrepancies between FEA analysis and real-world manufacturing. Notably, the point at which the wishbone tubes are connected to the bearing cup encounters a significantly higher stress concentration, in one case as high as 313.82 MPA.

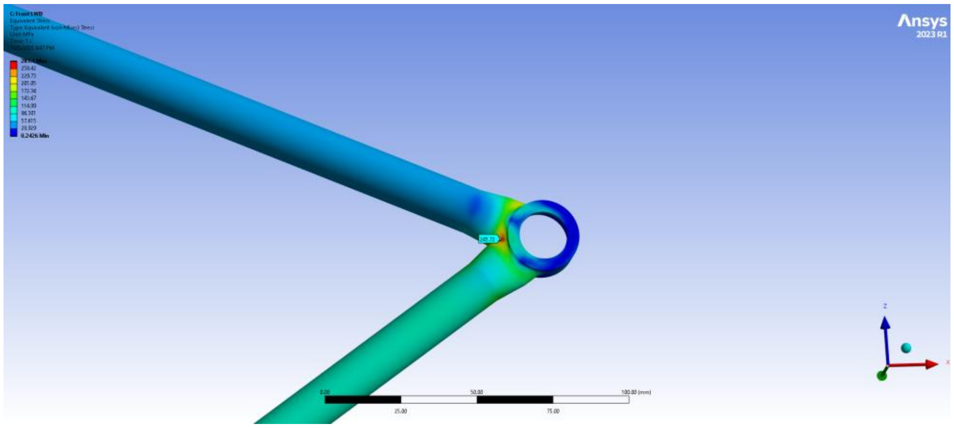


Figure 27: Front LWB stress concentration

However, given the location of the high-stress areas at the point where the wishbone rods connect to the bearing cups will be significant amounts of weld material added- something not modeled into the FEA analysis- so it's important to note that those high concentrations are not accurate geometry to the manufactured product and are expected to be significantly lower.

The deformation for each wishbone appears in line with acceptable values for the desired performance. The front LWB deforms a maximum of 0.33mm, the front UWB 0.65mm, and the rear UWB deforms 0.64 mm. Each part has very little compliance and should be able to deform significantly before being put at risk of failure.

Static structural analysis has been done on the front wishbones and rocker, as well as the rear rocker. The initial simulations revealed some stress concentrations in the wishbones in the region where the tubes go from round to pinched. This was believed to be due to a reduction in wall thickness in the CAD, as well as a possible meshing glitch. Updates have been made to the CAD to reduce stress in these critical areas. The team will take particular care to pay attention to these regions during the initial testing and validation of the design and make changes if necessary.

The front and rear rocker geometry was tested and optimized before suspension loads were calculated. The new rocker design now includes mounts for the anti-roll components and requires further FEA to be performed.

The remaining components in need of simulations are both front and rear pushrods, as well as the toe arms and tie rods. The pushrods are primarily loaded in tension, and the tie rods/toe arms solely in compression; therefore, failure isn't a big concern for the team. Instead, compliance in the arms are joints will be the main area of focus for the team, as excessive compliance can lead to big changes in toe or the lack of energy absorption from the shocks, as the deflection will result in an incomplete transfer of the loads to the shock absorbers. For this reason, the team is evaluating the performance of carbon fiber links due to their increased stiffness without sacrificing the increased mass.

Chassis

Chassis Design

The chassis layout was developed through a rule- and requirement-driven workflow to ensure compliance with the 2026 FSAE rules and alignment with customer requirements. The chassis, a steel tube space frame, was designed utilizing SolidWorks' 3D sketching and weldment environments. Suspension pickup coordinates were established, followed by constructing a series of reference planes to define critical geometries. Reference planes representing the ground plane, maximum ride height, and additional vertical dimensional criteria were created to maintain consistency between all subsystem leads throughout the packaging and layout process. Additional planes defining the forward and rearward boundaries of the suspension boxes supported accurate placement of structural members within the vehicle envelope.

An additional geometric step was implemented to accommodate the hardware associated with each suspension pickup. Points were first placed at all suspension coordinate points in 3D space. On the planes defining the suspension boxes, a 30 mm offset was created from each pickup point toward the inside of the car. This offset allowed space for the spherical bearings, bearing cups, and mounting tabs. The resulting offset lines established the structural connection paths for the chassis, and the full chassis tube layout was generated based on these 30 mm inward construction lines to ensure proper clearance and manufacturability.

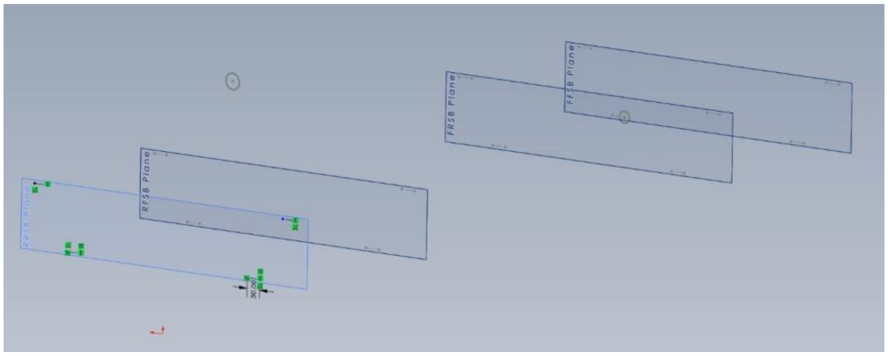


Figure 28: 30 mm offset lines

The front hoop geometry was designed within the constraints of the 20-degree-from-vertical requirement immediately after the bounds of the suspension boxes were constructed. After determining the overall height of the front hoop, including the diameter, the steering wheel was placed such that, in any angular position, it is still below the top of the front hoop. The driver position was determined using the two-dimensional outline provided by the FSAE rulebook, as well as a three-dimensional model of the 95th percentile male dimensions. This ensured compliance with critical technical inspection measurements such as pedal reach and cockpit clearance.

The main hoop was designed according to the mandated dimensional, safety, and material specifications. Side-impact structure members and the lower cockpit boundary were then incorporated, followed by integration of the harness bar. The rear chassis structure was formed by adding the main hoop supports, rear suspension mounting boxes, and rear jacking point.

The front structure of the car was then completed around the regulated pedal box location while ensuring compliance with the cockpit cross-section test. Additional rule checks verified that the front hoop, main hoop, and side-impact assemblies satisfied the cockpit opening template requirements. A steel anti-intrusion plate was incorporated at the front bulkhead together with an impact attenuator interface. Triangulation was then added across the chassis in accordance with FSAE guidelines while avoiding over-bracing, which would add excess mass. The firewall design was developed to provide complete separation between the cockpit and engine bay, followed by integration of the harness mounts and suspension mount features.

Adding on to that is the intricate task of joining the chassis and the suspension. After several iterations and hours, the final design resembles a clamshell motor mount, with the outer shell being welded to the chassis at the suspension box nodes, and the inner shell is then bolted into the outer shell using 4 M6 bolts. This also allows for adjustability on both the upper and lower control arms, allowing the team to adjust static camber and track width for tuning adjustment. The outer clamshells are made by cutting a flat plate to create each side of the shell individually and then welding them together. The sides are individually coped to the specific tubes that they intersect, so each one is different. The inner clamshells are more standardized, with the brackets being waterjet and then welded together. All but two of the inner shells will be identical, with two of them needing adjustment due to the mount design having to be adjusted slightly because of the chassis geometry.

Phase 2 packaging planning ensured appropriate clearances for the engine, differential, axle paths, battery, and fuel cell. Upon completion of the structural layout, a cut list was produced for the bill of materials and provided to VR3 Engineering for tube cutting and bending cost estimates.

Customer requirements were considered throughout the chassis design process, and the driving factors were as follows:

- Rolling chassis under 135 kg: Avoided over-bracing, triangulated only as FSAE rules and specific design choices required.

- Capable of sustaining a 1.5 g turn: Proper triangulation techniques to improve chassis rigidity; Careful planning for keeping members in compression when suspension forces are loaded on the chassis.
- Chassis twists less than 20% of roll compliance: Continuous load paths that connect hardpoints of chassis (FBH, FH, MH); Opted for steel anti-intrusion plate to keep the car's nose more rigid.
- Chassis must accommodate from the 5th percentile female to the 95th percentile male drivers: Used the 95th percentile male 2D outline provided by the FSAE rulebook and 3D model to ensure the pedal box is far enough away and the main hoop height legality.
- Design must pass all applicable FSAE competition technical inspection tests: Used 3D models for cockpit opening and chassis cross-section tests; Used 2D 95th percentile drawing and a 3D model of 95th percentile male.

Chassis CAD

The chassis model was generated in SOLIDWORKS using a structured weldment workflow. Initial tubing included the front bulkhead, front hoop, and main hoop, establishing the foundational geometry.

Lower chassis tubes and the chassis floor triangulation were then added. A reference plane passing through the lower front-bulkhead node, the upper suspension-pickup point, and a rule-compliant location on the front hoop was created to support a continuous tube path linking these points. This prevented unsupported nodes and produced an efficient front load path.

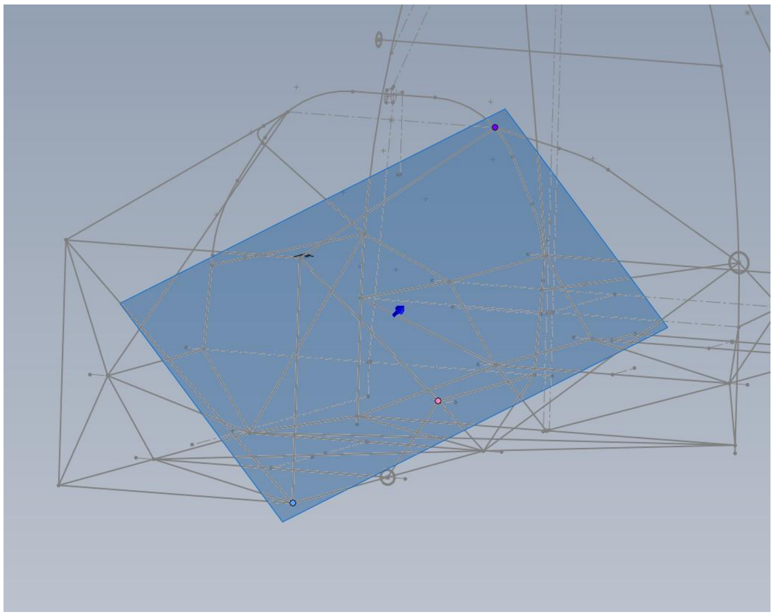


Figure 29: Reference plane aligning FBH, Suspension Box, FH

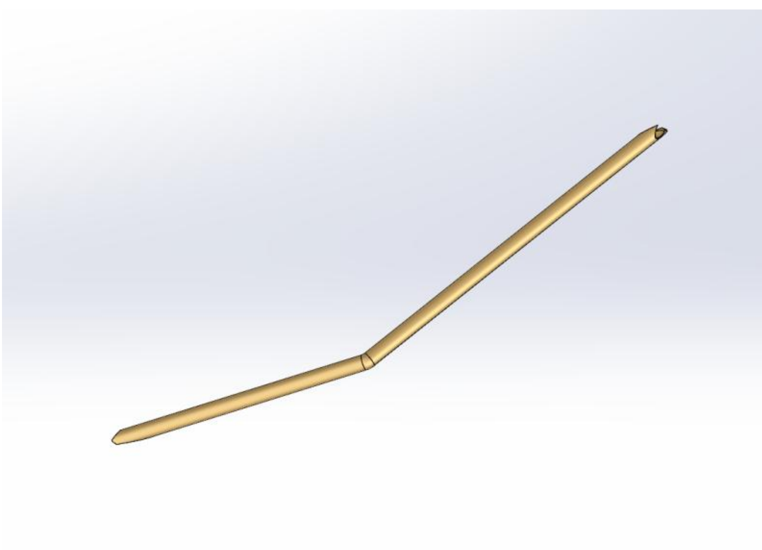


Figure 30: Continuous load path tube

The remainder of the front structure was constructed afterwards, followed by the rear end of the car; The rear suspension box, rear jacking point, and the V-bracing underneath the engine.

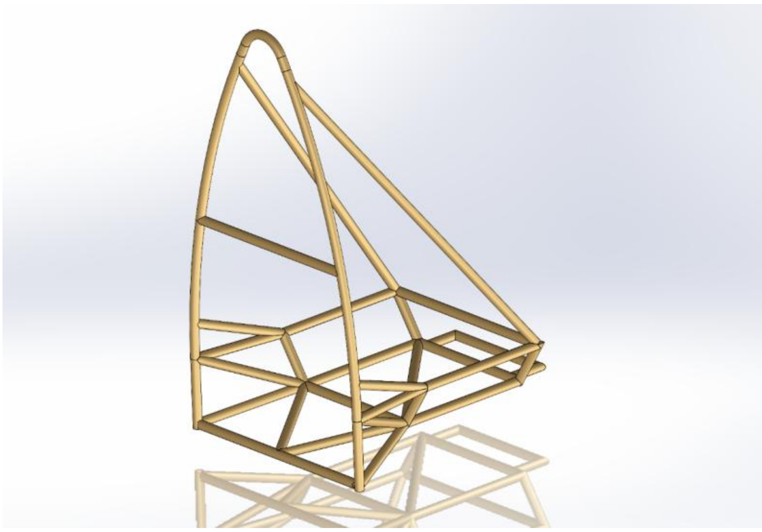


Figure 31: Chassis rear end

Side impact structure members were the next chassis features to be constructed. Careful measurements were made to be compliant with the dimensional rules regarding the impact structure, as this is one of the most crucial safety features for the driver.

A harness bar was then positioned at the correct height such that the driver's shoulder straps' path are not changed when tightened. The firewall was also created at this point, and upon welding will completely separate the engine bay from the driver due to the extended tabs to seal around the side impact structure members.

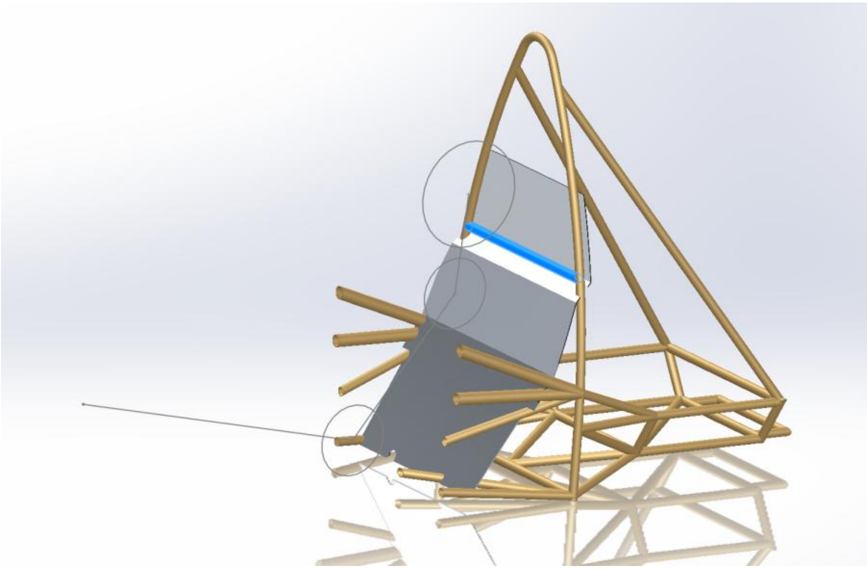


Figure 32: Harness bar height and firewall

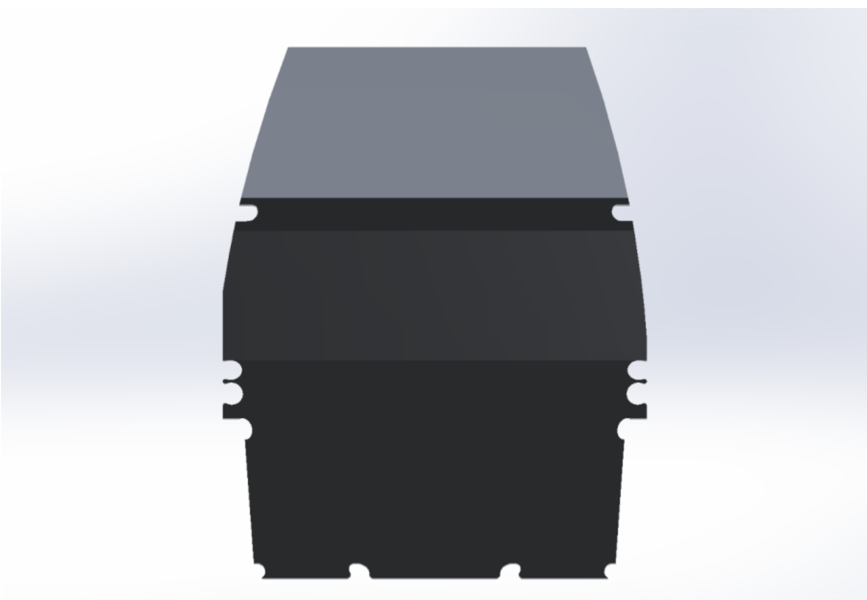


Figure 33: Firewall geometry to fit around and between tubing

A Phase 2-conscious area between the front of the main hoop and the rear of the seat was maintained for the fuel cell, battery, exhaust routing, and potential additional components such as a coolant reservoir or electronics. Clearance analysis with a standard Formula SAE fuel cell model and a representative battery confirmed adequate space for the Phase 2 team to utilize the area effectively.

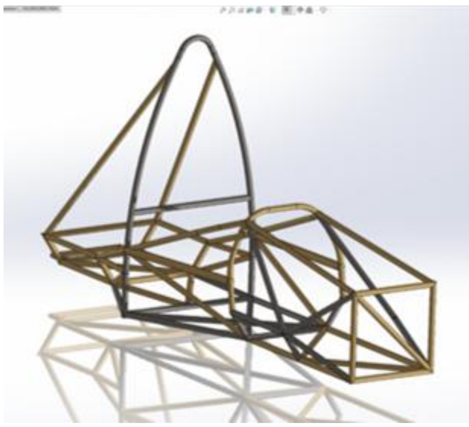
Figure 34: Side-view showing fuel cell and battery behind firewall

The frame has a final mass of 44.3 kg. This includes all steel tubing [36.93 kg], the aluminum firewall [4.5 kg], 400x390x1.5 mm steel anti-intrusion plate [1.84 kg], aluminum honeycomb impact attenuator [232.7 g], and all 7-gauge steel mounting tabs [804.4 g].

The frame could potentially have a lower mass, however there has been difficulty sourcing FSAE Tubing Profile C, a 1” OD 0.049” wall tube. A considerable portion of the chassis is allowed to be made from this tubing size, however the limited availability has forced a design choice to use Tubing Profile B, 1” OD 0.065” wall, for all of the size C members. Ultimately, this is not a negative, as the frame sacrifices a small amount of mass for improved mechanical properties. The rolling chassis as a whole is on track to be under the 135 kg customer requirement, and therefore the team has opted not to source size C unless needed.

The final overall dimensions of the frame are 2.36 meters in length, 0.7 meter in width, and 1.25 meters in height.

Chassis FEA



To conduct FEA on the chassis, a beam model was created by importing the 3D model into space claim. Once the geometry was cleaned, and all the proper cross-sections were added as well as suspension and upright members, it was uploaded into mechanical for analysis. Appropriate connections were defined at every location where the suspension was attached to the upright or the chassis. Additionally, springs were added connecting the center of the upright to the chassis. For loads, a 1500 in opposing directions were applied at either upright originating from the ground, and the rear of the chassis was simply supported at the bulkhead. This torsional test was evaluated for equivalent stress and total deformation.

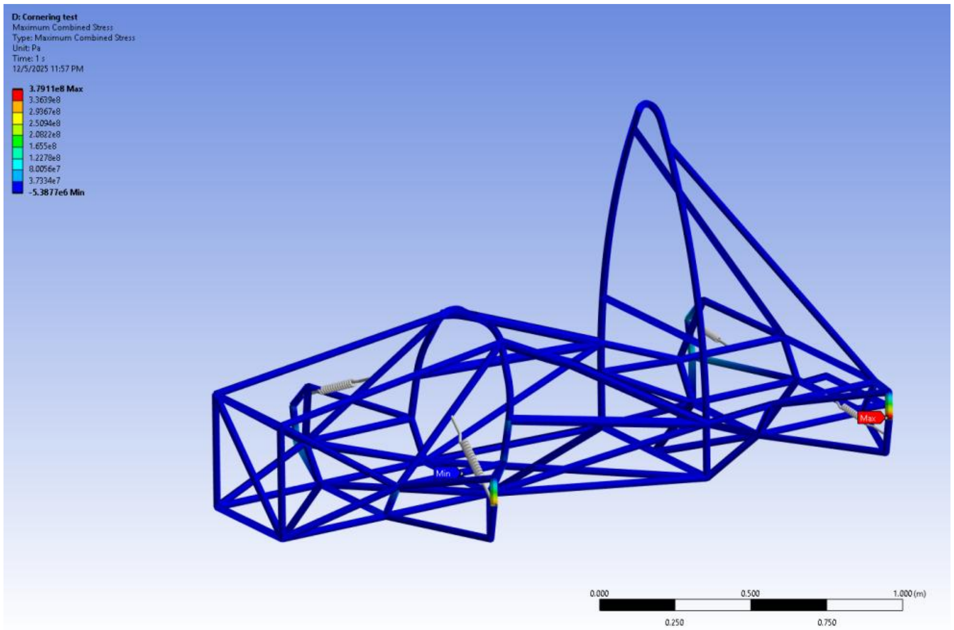


Figure 35: Combined maximum stress

The test results showed 379 Mpa. The next simulation performed was a max cornering test where a load of 1.7 lateral g's at the CG of the car was applied.

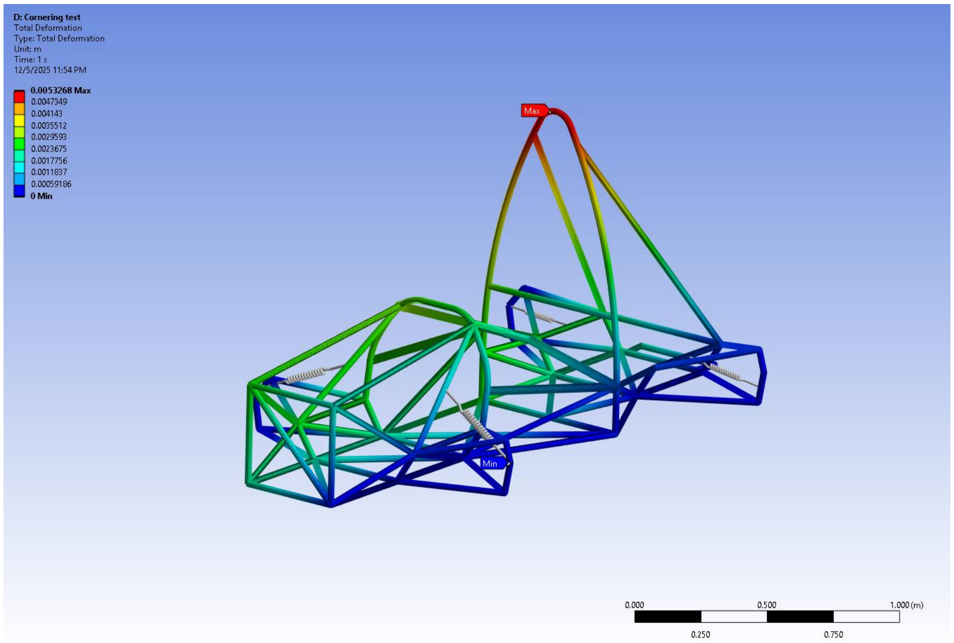


Figure 36: Total deformation from cornering test

The test results showed a deformation of 5.3 mm, an acceptable figure.

Brakes

Brake calculations

For the initial calculations an average force a fit human can apply with their leg was applied onto a moment arm brake pedal connected to the master cylinder. The force was distributed onto the area of the brake fluid in the lines which was then transferred to the caliper pistons. The piston force was distributed to the pad face area with a friction force calculated. A torque force was calculated from a typical wheel travelling at 90 mph, and this was used to get the required diameter of the brake rotor.

```
clc;
clear all;
close all;


pF = 494 ; % Pedal Force (lbf)
D = 6 ; % Pedal to push rod (in)
fit_A = (5/16) ; %Fitting Area (in^2)
pis_A = 1 ; %Piston Area (in^2)
pad_A = 2 ; %Pad Area (in^2)
pad_f = 0.36 ; % Coeff. Friction Pads
num_piston = 8 ; %Number of pistons acting on rotor
torque_w = 177 ; %Torque to stop (lbf)
x = 100 ;

while x > 0


    f_1 = (pF * D * fit_A)/(pis_A * num_piston) ; %Normal force caused by pistons
    fric_f = f_1 * pad_f ; %Friction Force

    r_dis = (torque_w ) / (fric_f ) ; % Radius of rotor in inches

    in_mm = r_dis * 25.4  %convert to mm



    D = D - .1
    x = x - 1 ;
end
```

Brake CAD

For the uprights and brakes the goal was to keep the total assembly as light as possible while maintaining a low center of gravity. For both front and rear, they were built off the given attachment points to hold the bearing wheel hub assembly while maintaining the desired wheelbase. Main body was designed around the bearing center and gave a bearing wall thickness that would remain light but strong enough to withstand the forces that would act upon it. Once the main body was made, the brake caliper was positioned at the lowest point possible and with a full contact patch on the rotor.

Figure 37: Front upright assembly

Figure 38: Rear upright assembly

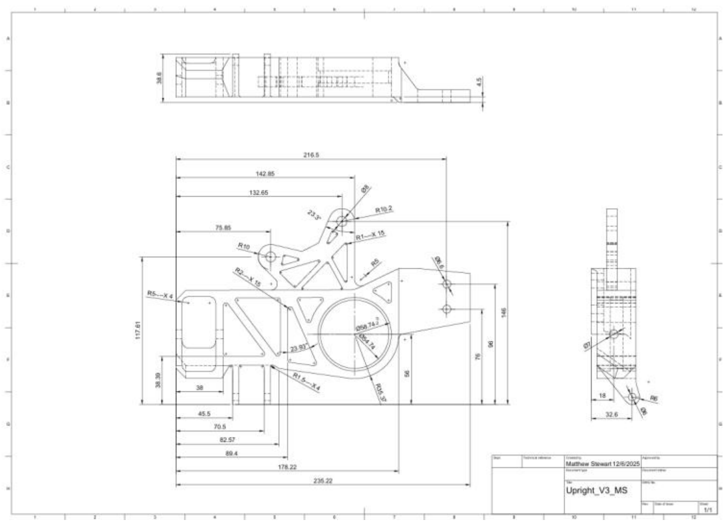


Figure 39: Front upright drawing

The pedal box was designed as a two-pedal system with accelerator and brakes with no need for a dedicated clutch pedal. It uses a very standard pedal body design with adjustable face plates. It is mounted on rails that will be welded to the frame and hold the pedal plate with pins. Simple steel pins are used to make attachment of master cylinders and pedals easy.

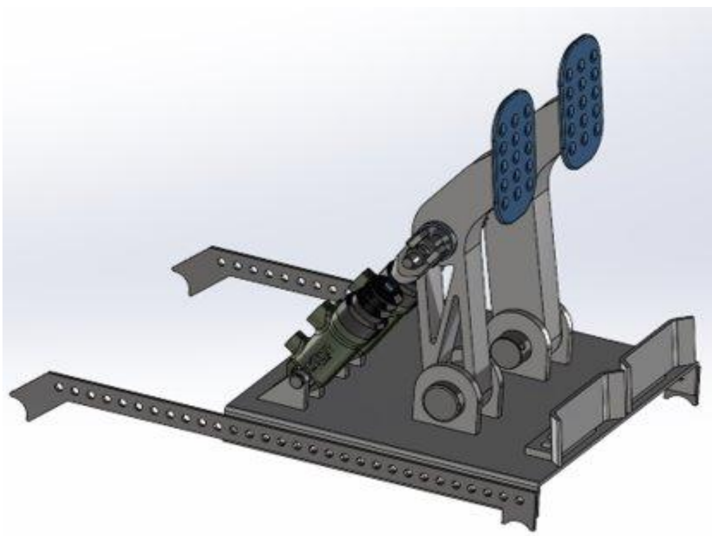


Figure 40: Pedal box assembly

Brake FEA

It was assumed that problem areas in the design would be around the control arm mounts and bearing housing. Deformation in the bearing housing could result in a bearing slipping out causing a catastrophic loss of wheel stability. The mounting points are an area to watch as well as they contain the main areas of stress in the upright but with the added rigidity of the spherical bearings and spacers there should be less stress concentration around the mounts. Max deformation in either upright was 0.854 mm.

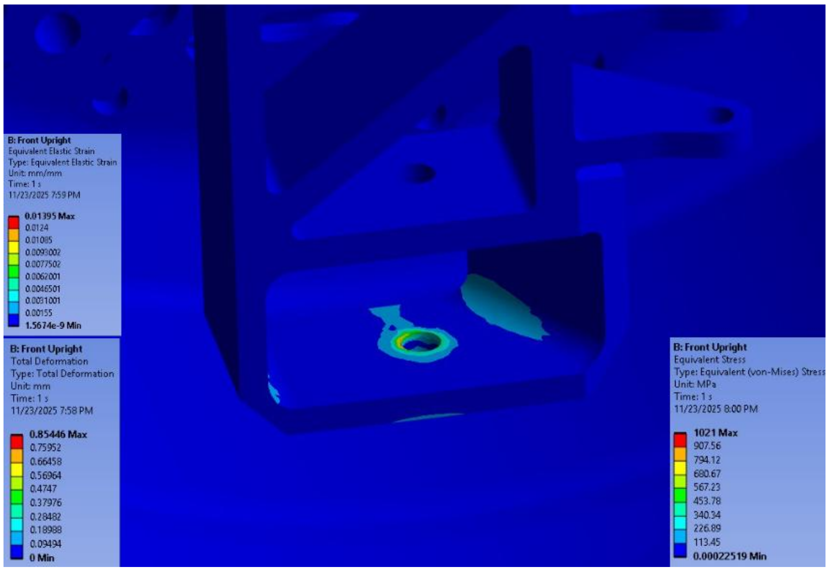


Figure 41: Front upright FEA

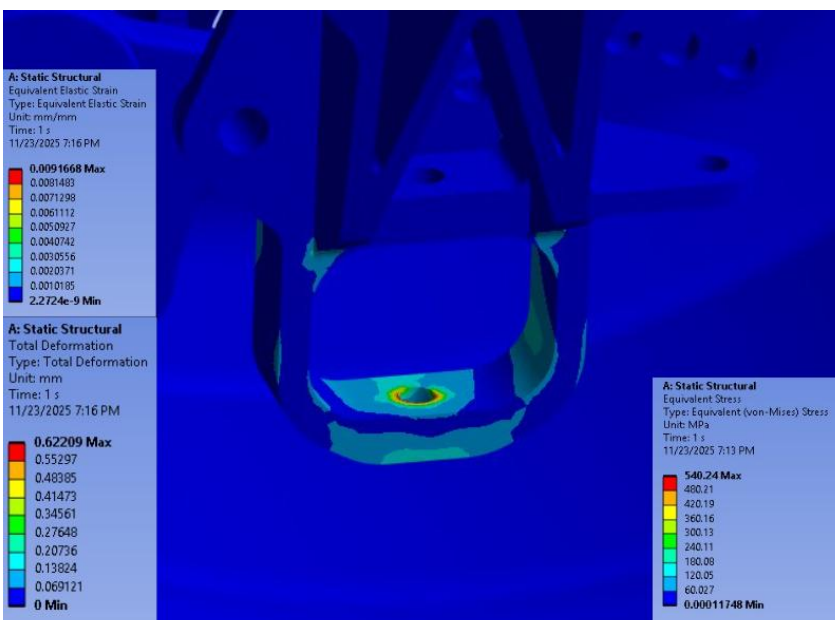


Figure 42: Rear upright FEA

The pedals had a force of 2000 N applied as it must withstand the force per the FSAE rules. Most of the stress concentrations were around the fixed points of the pedal with this possibly being mitigated once fully assembled. Overall, the results showed very little deformation in the part. The largest deformation was 0.59 mm, likely due to the fixed point have a large moment arm applied.

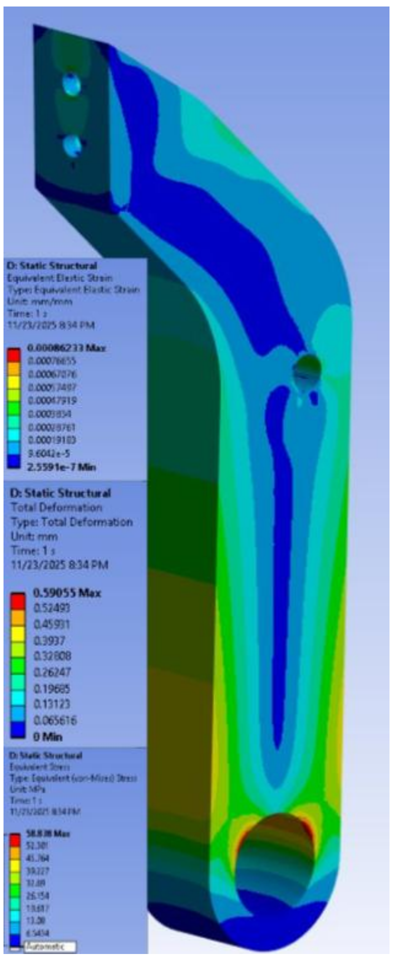


Figure 43: Gas Pedal FEA

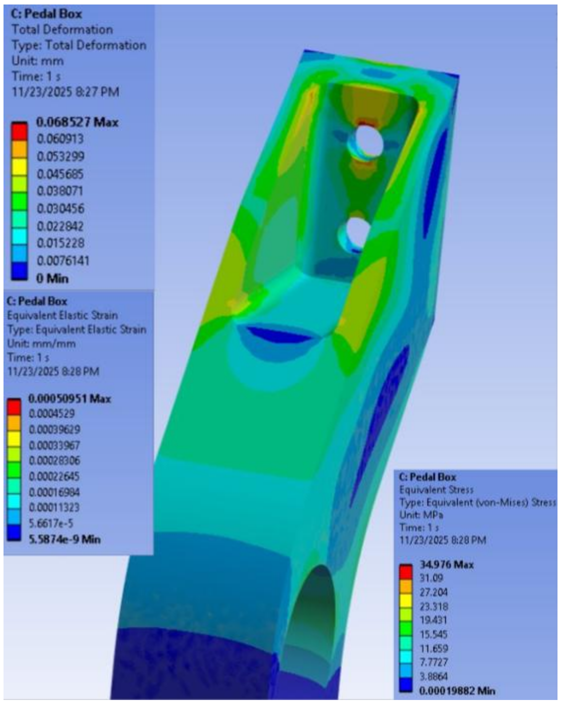


Figure 44: Brake Pedal FEA

Steering

Steering Calculations

This is the sheet of calculations for the steering column in Excel. What needed to be calculated was the maximum wheel angle, the turning radius of the car, the steering ratio, and the force required by the driver to turn the wheel. The max wheel angle was calculated by using simple trigonometry to find the angle, as well as the turning radius. The steering ratio was calculated by taking the ratio of the angle of the steering wheel and the angle of the wheel. These values are currently final, but they can easily be changed if needed. Grey boxes are inputs; green boxes are outputs.

		3.5 inch/rev (88.9mm/rev)		
Max revs (degrees)	260		Wheelbase (m)	1.55
per side revs (degrees)	130		max wheel angle (degrees)	32.7027169
Rack Speed (mm/rev)	88.9		min non-slip turning radius (m)	2.41412131
linear travel distance (mm)	32.1028			
			max linear travel distance (mm)	32.1027778
			distance from tire point of rotation (mm)	50
			max steering angle (degrees)	32.7027169
			Steering ratio	3.97520488
			Track width (m)	1.25
			steering angle in (degrees)	40.9039244
			steering angle out (degrees)	27.0223017

mass of car (kg)	226				
weight transfer during braking (kg)					
friction coefficient of tire	1				
front weight bias	0.45				
max weight on front end (kg)	101.7				
max force on front end (N)	997.677	Less than recommended design operating force?	Yes	SF	1.79
Radius of pinion (m)	0.018				
Torque on pinion (Nm)	18.8582	Less than max steering torque?	Yes	SF	4.96
Radius of steering Wheel (m)	0.1285				
Force required by driver, one hand (N)	146.756				
Force required by driver, one hand (lbf)	32.9938				
Torque on steering wheel (ft*lbf)	13.9105				

From these pictures, it is obtained that the torque the driver must apply is 18.86 Nm (or 13.91 ft*lb. If the driver were to drive the car with one hand, they would have to apply a force of 147 N or 33 lbf. If the driver were to drive the car two-handed (as they should), they would apply half of the forces stated (73.5 N or 16.5 lbf). The internet says that 73.5 N would be close to 7.5 kg, close to the weight of a large bowling ball. The safety factor to the recommended design force is over 1, and the safety factor for the maximum steering torque is over 2.

Steering CAD

The steering system underwent approximately 7 iterations as problems arose, and now a final design has been established. The main challenges included ensuring that the angles of the U-joint did not result in binding and selecting a U-joint that satisfied the required angles. The inner and outer diameter of the column tubes, U-Joint, quick disconnect, and bearing were chosen from the diameter of the pinion connection. The column will connect to the U-Joint by bolts, and the column will connect to the pinion using spring pins. Assembling the U-Joint was fun, but seeing the full steering assembly rotating, including the rack moving left and right, was well worth the wait. Things that had to be solved included finding the right U-Joint, selecting the right rack travel speed, making the rack support due to learning about SolidWorks, and every time an update was made, the steering rack mate would break. The final mass calculated from SolidWorks came out to be 2.53 kg or 5.58lbs.

Figure 45: Steering Assembly in SOLIDWORKS

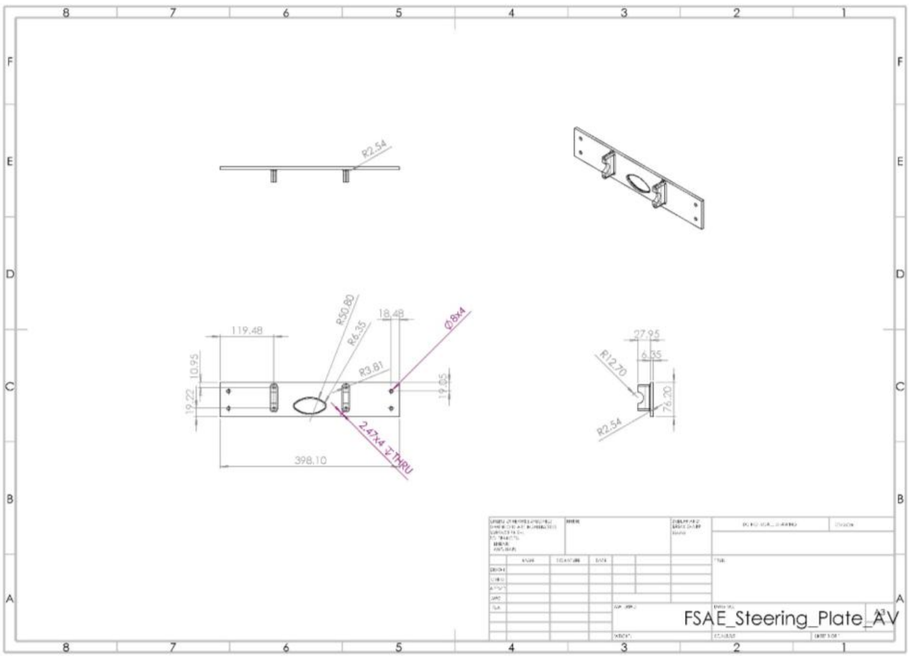


Figure 46: Steering Plate Engineering Drawing

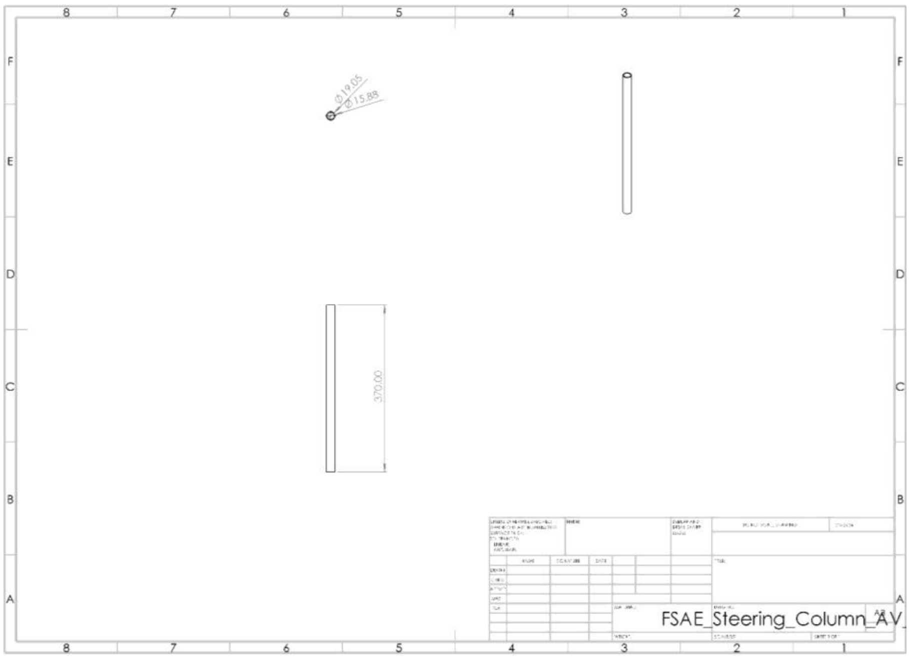


Figure 47: First Steering Column Engineering Drawing

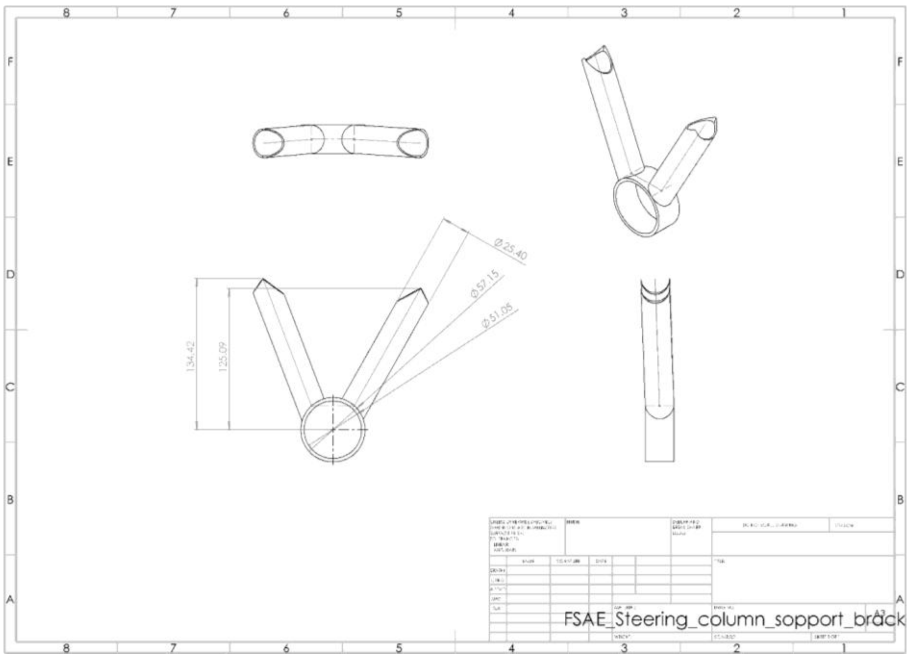


Figure 48: Steering Column Support Engineering Drawing

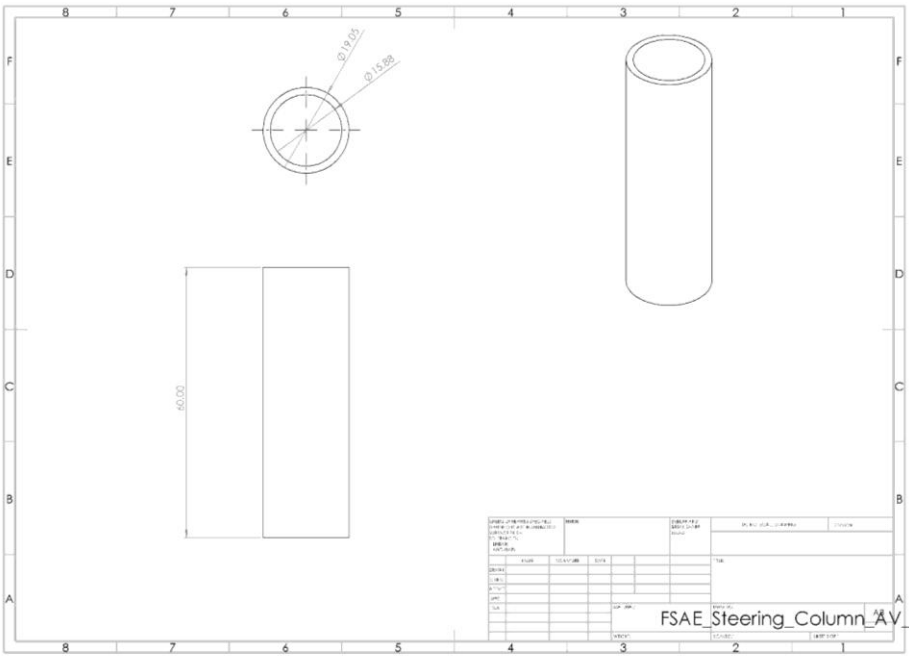


Figure 49: Second Steering Column Engineering Drawing

Steering FEA

The FEA on the steering assembly was done in Ansys without the steering column due to the complexity and issues with it being in, so the steering column was fixed at the bottom end, where it would connect to the pinion and the welds of the steering column support.

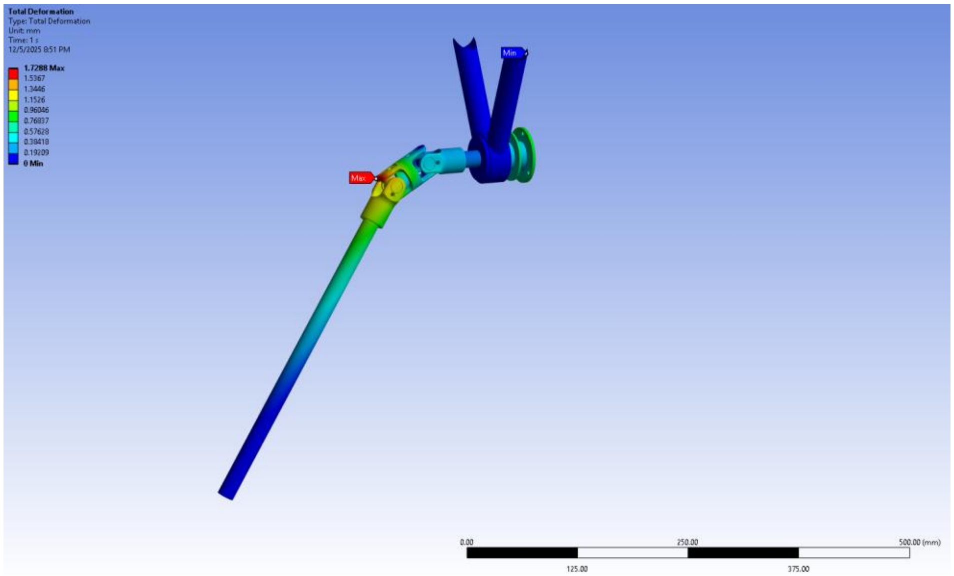


Figure 50: Total Deformation of the Steering System

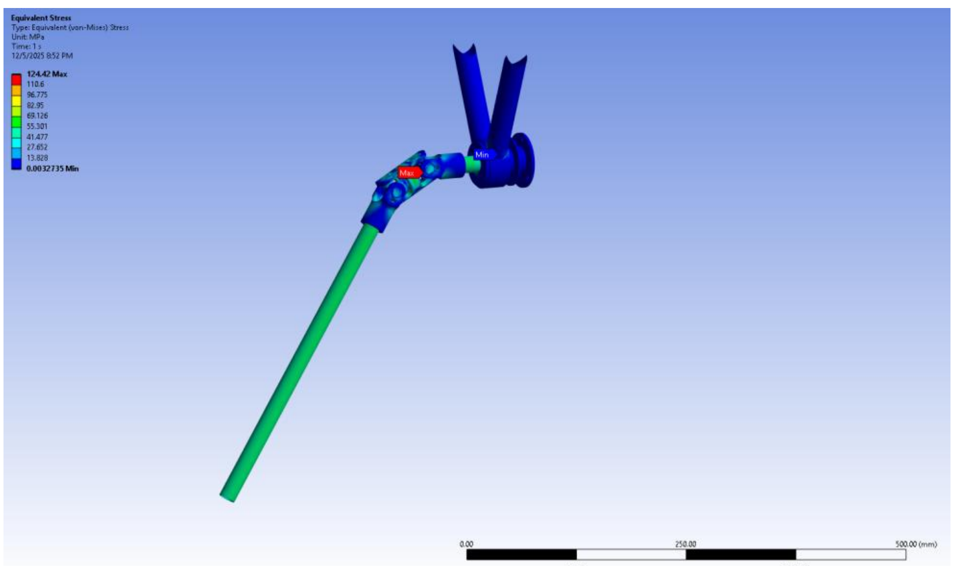


Figure 51: Equivalent Stress of Steering System

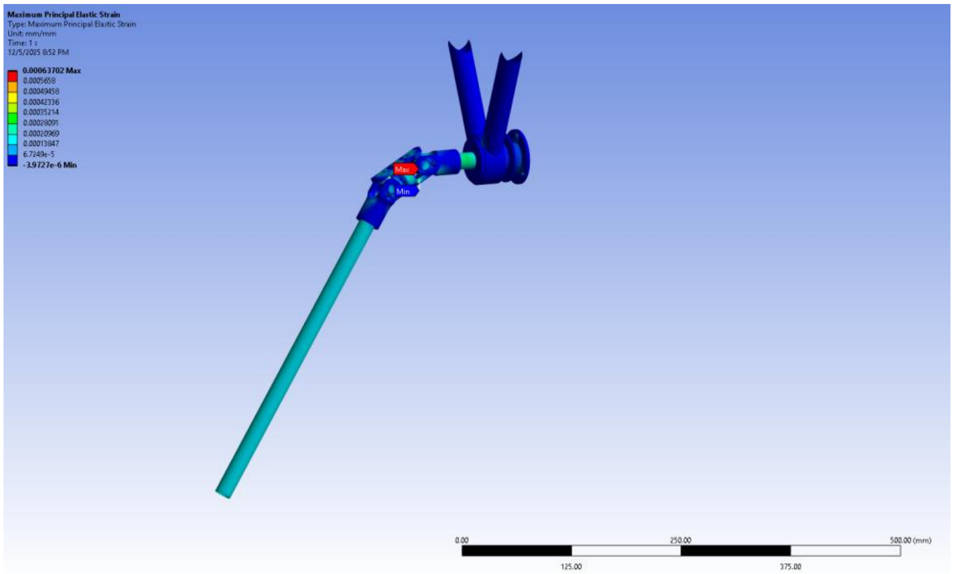


Figure 52: Max Principal Elastic Strain of Steering System

The values obtained for the assembly were a max deformation of 1.8729 mm, equivalent stress of 124.33 MPa, and a max principal elastic strain of 6.37e-4 mm/mm and min principal elastic strain of -1.23e-5 mm/mm. All these values are well below the maximum yield strength of any of the metals that will be used. There was stress concentration inside the U-Joint where the cross pin meets the bearing cup; this could be an issue in operation, so the team will monitor this closely.

Manufacturing

Chassis

To have a carbon fiber monocoque accepted by FSAE, a Structural Equivalency Spreadsheet (SES) must be completed. This spreadsheet is an array of calculations, tests, simulations, and results in proving that the monocoque is as strong and safe as the steel frame it would replace. To do this, a sample plate must be subjected to a 3-point bend test to determine its stiffness and flexural strength properties. During preliminary carbon fiber testing, the strongest test sample measured 12.6mm long by 6.8mm wide by 3.4mm thick. It was composed of a [F, O_2, S, O_2, F] layup schedule, where F denotes a unidirectional layer, O_2 denotes two bidirectional 0/90 layers, and S denotes the Soric core material. In testing, it was found that the sample withstood a peak force of about 200N. The data extrapolated this information and found that the strength was insufficient for the monocoque. Due to the iteration time and heavy cost investment, the decision was made to pivot from a monocoque and construct a tube frame. A future senior design team is encouraged to dive further into monocoque design and fabrication, but budget the time for this conversion to be one senior design year.

Commented [DV6]: Please add references

Commented [DV7]: Needs to be in third person

After determining that a space frame is the way forward, the manufacturing considerations shifted to the angular manipulation of tubing and the construction of a jig to hold the frame together. A manufacturing plan has been drafted and is currently under revision as exact details of materials and designs continue to come together. At the time of writing, there is a possibility that the steel tubing supplier will also CNC cut and bend the tubing before shipping it to us. The welding sponsor has graciously offered to weld the chassis and suspension components to further ensure safety in the event of an incident.

Suspension

Most of the suspension components will be professionally welded and machined. The uprights and spherical bearing cups will be made by Andrew at LeMatic due to the complex machining geometry that cannot be made at Trine. Andrew will also weld the control arms while there to ensure proper weld quality and safety. The FSAE team will provide Andrew with the jig to weld the control arms in. The pushrods will be manufactured in-house from carbon fiber rods with aluminum thread bosses bonded into each end. Rod ends with jam nuts will complete the pushrods. The rocker will also be manufactured at Trine, but with the metal 3D printer, marking one of the very first applications of this technology at Trine. There is hope that this will be the first step towards fully integrating this technology into routine use at this university.

Steering

The steering components will have considerably more manufacturing done in-house. The upper steering mount, lower rack plate, steering wheel, and column will all be manufactured at Trine. The steering rack, rod ends, quick disconnect, support bearing, double u-joint, rack end extensions, and carbon fiber tube will all be purchased and then assembled.

Brakes

The braking system will also be split between purchasing and manufacturing. The wheel-side braking kits will be purchased with the wheel hubs as a package. The master cylinders will also be purchased. Everything else, brake lines, pedals, and pedal mounts will be internally made.

Other

The jig is slated to be made of thin wooden sheets with holes and slots cut in them to both fit the chassis tubes and interlock together. Each wooden sheet will be laser cut to ensure dimensional accuracy and tolerancing. Hot glue will be used to hold the jig together once all the pieces are interlocked. If the tubing is shipped directly to the welding sponsor, the team will simply ship the jig, control arm tubing, and bearings to the welding sponsor and will then receive all welded and machined components back in a single crate return shipment. If the tubing is received at Trine University, the team will preassemble the frame in the jig before shipping it and everything else previously mentioned to the welding sponsor.

Still upcoming is the design of a shipping crate to package the assembled, but not yet welded, frame, mounts, control arms, and any other parts that need to be sent out to the welding sponsor. There will also be room for our machined uprights to make the return trip in the crate. The crate will be based on the pallet that the welding table came on. A two-by-four frame sheeted with plywood will make up the walls and top of the crate.

During this time, significant revisions were made to the work area to make it suitable for the welding table that the car will be built upon. Several cabinets were condensed into wire racks for more space. The area was reorganized to make more clearance for a 50-inch by 100-inch welding table. This table was selected due to its being repurposed for the new student design center in the future. The work area will continue to be maintained and reorganized as necessary.

Another important consideration for manufacturing is tools and supplies. This list has been slowly compiled and revised. This list is by no means in its final form, as some of the manufacturing processes have yet to be fleshed out. The list is as follows:

Welder	Clamps	Grinding Wheels
Welding shields	Control arm jigs*	Grinder
Welding Gloves	Markers	Safety Wire
Welding Helmet	Tape	Tubing bender and dies
Wrench Set	Drill/drill press	
Socket Set	Tape Measure	
Ratchets	Drill Bits	
Loctite	Assorted wood for the shipping crate	
Brake line flaring tool	Hot glue gun and sticks	
Brake line deburring tool	Hole Saws	
Brake line bending tool		
Torque wrench		

Manufacturing Plan

Disclaimer: This manufacturing plan is current as of December 5, 2025. There are currently RFQs out for precut and pre-bent tubing for the frame. The manufacturing plan will be updated if these quotes are accepted.

Objective: Deliver clear and concise instructions to ensure all fabrication and assembly steps are completed and quality checked, providing the next FSAE team with a solid foundation to build upon.

Scope: This manufacturing plan covers the fabrication and assembly of the chassis, suspension components, steering, and brakes.

Safety Procedures: Follow all prescribed lab safety procedures. Ensure proper PPE is always worn.

Pre-Manufacturing considerations:

- Go through the design and assign every individual tube a number. Record with this number the tube location, length, cross-section size, and any bends or notches it needs. This will assist in manufacturing by ensuring that all tubes are accounted for.
- As early as reasonable, send part files to Andrew for CNC machining.

Manufacturing steps

1. Use the laser cutter to cut out each jig part from its .dxf file. They will be pre-numbered for ease of assembly.
2. Assemble the jig on the chassis table, with the runner marked L on the driver's side, and R on the opposite side.
3. Insert each vertical jig plate in numerical order, with 1 being at the nose of the car, and continue sequentially towards the rear of the car. Note that any pieces with a T inscribed in them are oriented parallel to the longitudinal axis of the car.
4. Hot-glue the jig pieces together.
5. Measure each tube from its designated tubing size according to the numbered chart made earlier. Write the tube number on the outside of the tube currently.
6. Using the band saw, cut each piece of tubing to length.
7. Debur all cut ends to prevent slivers or cuts.
8. Inspect the tubing to verify the presence or absence of a seam. If seams are present, care must now be taken to ensure that the tube orientation is tracked, particularly with bent tubes

9. Set aside any tubes that require bending. Mark where the bend needs to occur and write how many degrees the bend must be.
10. Set up the tubing bender with the correct dies for the tubing size.
11. Place the tube in the bender at the pre-marked bending point. Ensure that the seam is up before bending; otherwise, the seam may crack, and the tube will be useless.
12. Bend each tube to the required degree. Double-check that the tube maintains the correct angle after the pressure is released.
13. Set up the tubing notcher with the proper hole saw for the tubing size.
14. Notch all tubes to the correct orientation, angle, and depth.
15. Debur all edges.
16. Place the tubes in the chassis jig according to their numerical designation.
17. Verify all tubing fitments and ensure there are no large gaps for proper TIG welding.
18. Tape the tubing ends together to hold them in place.
19. Waterjet necessary mounting tabs for the chassis.
20. Add tabs to the chassis in their respective locations and tape them in place.
21. Waterjet the control arm jig plate.
22. Use the lathe to make centering dowels for the spherical bearing cups. These will take the place of the bearings during welding.
23. Mark and cut suspension tubes to the required length.
24. Debur tubes
25. Use a vise to gently crimp down the ends of the tubing. Ensure that any seam is on a flat surface and is not crimped in bending to prevent fracture.
26. Notch tubes according to the control arm drawings.
27. Label control arm tubes in accordance with their location.
28. Package control arm components for shipment to Andrew.
29. Cut the steering column tubes to the proper length and debur.
30. Label steering components to send to Andrew.
31. Prepare the chassis, suspension components, steering components, and jig for the trip to Andrew.
32. Ship the chassis.
33. Print and cure the 3D printed rockers
34. Mark and cut carbon fiber tubing to length for the pushrods.
35. Glue in threaded inserts into each end of the carbon fiber tube.
36. Once the glue is dry, thread in the rod ends, ensuring that each end has a jam nut.
37. Once all suspension components are fabricated, and the components sent out for machining and professional welding are back, bolt all suspension components to the chassis. Ensure safety wire is installed on all fasteners.
38. Bolt the steering rack into the chassis.

39. Install tie rods on both the steering rack and the upright. Ensure jam nuts and safety wires are installed.
40. Drill bolt holes through the U joint and non-welded column tube.
41. Press the steering column bearing into the mount in the chassis.
42. Install the lower half of the steering column onto the steering rack.
43. Insert the upper half of the steering column through the steering column mount and bearing.
44. Bolt the two halves of the steering column together.
45. Fabricate a steering wheel from carbon fiber using standard Trine composites practices.
46. Waterjet all necessary pedal box components.
47. Assemble the pedal box with pedals, master cylinders, reservoirs, and any other necessary components.
48. Bolt the pedal box in the car.
49. Assemble hubs, rotors, calipers, and rotors onto the uprights.
50. Fabricate the brake lines.
51. Move the pedal box throughout its range, making sure the lines are free and without binding the entire way.
52. Add fluid and bleed brakes, checking for leaks along the way.
53. Mount the tires on the wheels.
54. Bolt the wheels and tires on the uprights.
55. Turn the steering from lock to lock to ensure there is no binding.
56. Fill out the build completion checklist and make any corrections if necessary.

Testing:

For eventual testing, the customer requirements have been reviewed, and a completion checklist and testing plan have been created. The checklist covers the simple pass/fail requirements, and an individual test plan has been created for each of the larger, more complicated customer requirements. A data collection template has also been made. This data collection template allows for the recording of critical information such as test repetitions, target metrics, measured metrics, and any issues that arise. See the Test Results Template section below. Please note that all test plans are still subject to change as new situations or information arise.

Test Result Template

Test Number ______ Date___/___/_____

Lead Test Engineer _______________ Time:____________

Assisting Engineers _____________________

____

Test Target Metrics _______________________ Test Repetition Times ___________

Results:

Notes:

Test Number – Record the assigned test number. Found at the top of the detailed test plan.

Date – Record the date. Time – Record the time

Lead Test Engineer – Record the name of the person overseeing/conducting the test.

Assistant Test Engineer - Record the names of all people assisting with the test

Test Target Metrics – Record the target goals for the test to be successful, ex, 90mm minimum

Test Repetition Times - Record the number of times the test must be repeated for accurate results.

Results: Record all results from the tests. No lines were provided for the flexibility of the template. Ensure that the results are clear.

Notes: Record any significant external factors, findings, or issues that arise during testing

Test Plans

Weight and Recyclability

Revision 1 – Drafted 11/7/2025 by Test Engineer Michael Ledyard

All previous revisions are obsolete.

Safety Equipment:

- Safety Glasses

Tools:

- Wheel Scales

Procedure:

1. Take scales out of their case.
2. Connect scales to the display.
3. Plug in the scales to power.
4. Zero all scales.
5. Ensure scale is properly calibrated
5. Collect all non-recyclable components – e.g., Carbon fiber parts, plastic parts, basically anything not metal.
6. Place them on one of the scale pads. Record the weight.
7. Assemble all parts on the chassis so that the vehicle is complete
8. Place the car on the scales, making sure that each scale is under its corresponding corner.
9. Record the total mass of the car.
10. Subtract the mass of the non-recyclable parts from the total weight, then divide by the total mass. This will give the percentage of recyclable parts.

Open Wheel Clearance

Revision 1 – Drafted 11/7/2025 by Test Engineer Michael Ledyard

All previous revisions are obsolete.

Safety Equipment:

- Safety Glasses

Tools:

- Tape Measure
- Pencil
- Drill
- Screws
- Saw
- 2”x4” boards

Procedure:

1. Measure 510mm on a 2”x4”. Mark with a pencil.
2. Cut the board to length. Remove any splinters that are created.
3. Find/make 4 2”x4” boards of equal length of 525mm. They must at least clear the top of the tire.
4. Screw the three boards together in a U shape. Make sure the 510mm board is perpendicular to the other boards.
5. Screw the remaining 2,525 mm boards to the U shape. It should now look like ||_||.
6. Invert the wooden jig and slide it over the top of each tire. The jig should pass from outside of the tire to inside of the tire without contacting anything.
7. Record any failures of this test on the test sheet.

Wheelbase

Revision 1 – Drafted 11/7/2025 by Test Engineer Michael Ledyard

All previous revisions are obsolete.

Safety Equipment:

- Safety Glasses

Tools:

- Tape measure
- 2 Lasers
- Pencil
- 6-foot board

Procedure:

1. Ensure wheels are straight.
2. Align lasers to the center of the wheels on the driver's side.
3. Hold the board so that both lasers land on the board simultaneously. Mark where they touch with a pencil.
4. Measure the distance between the pencil marks. Note the measurement
5. Repeat the process for the passenger side.
6. Measurements must exceed 1525mm.

Track Width

Revision 1 – Drafted 11/7/2025 by Test Engineer Michael Ledyard

All previous revisions are obsolete.

Safety Equipment:

- Safety Glasses

Tools:

- Tape Measure
- Tape

Procedure:

1. Ensure wheels are straight
2. Measure each tire's width and find the center. Mark this location on the ground with tape.
3. Measure from center to center on each tape line. Record this value for the front and rear.
4. Divide the front measurement by the rear measurement. This must exceed 75%.

Ground Clearance

Revision 1 – Drafted 11/7/2025 by Test Engineer Michael Ledyard

All previous revisions are obsolete.

Safety Equipment:

- Safety Glasses

Tools:

- Tape Measure

Procedure:

1. Ensure the vehicle is at its static ride height. Ensure that all bodywork is in place.
2. Measure from the ground to the lowest tube on the chassis. This should be less than 75mm. Record this value.
3. Measure from the lowest tube on the Side Impact Structure. This should be less than 90mm. Record this Value.

Driver Visibility

Revision 1 – Drafted 11/7/2025 by Test Engineer Michael Ledyard

All previous revisions are obsolete.

Safety Equipment:

- Safety Glasses

Tools:

- Helmet
- Cones
- Digital angle finder
- Tape Measure

Procedure:

1. Zero digital angle finder facing the front of the car.
2. Rotate the digital angle finder 100 degrees from forward to one side of the car.
3. Measure 4 feet from the digital angle finder in the 100-degree direction.
4. Place a cone at the measured spot.
5. Repeat this process for the other side of the vehicle.
6. Seat a driver with a helmet on.
7. Have the driver try to turn and look at each cone. The torso must remain forward during this test.
8. Have the driver communicate whether he can see the cone or not.
9. If the answer is no, mirrors must be made, and the test is repeated until successful completion.

Suspension Travel and Jacking Point

Revision 1 – Drafted 11/7/2025 by Test Engineer Michael Ledyard

All previous revisions are obsolete.

Safety Equipment:

- Safety Glasses

Tools:

- Jack
- Tape Measure
- Weights

Procedure:

1. Jack up the car until the front tires are just barely contacting the ground.
2. Measure from the ground to the lower control arm chassis mount. Record this value.
3. Let the car back down.
4. Place weights on the car until the suspension compresses to the bump stops.
5. Measure from the ground to the same control arm chassis mount. Record this value.
6. Subtract the second value from the first. This will give the overall suspension travel.
7. Measure the distance from the rear jacking point to the ground. Record this value. This must be at least 75mm.
8. Measure back 1 meter from the jacking point. Stand at this point and verify that the jacking point is visible. Note this result.
9. Measure 300mm outwards from the rear of the jacking point. Ensure that there are no obstructions in this area.
10. Place the jack underneath the jacking point. Raise the jacking point until the bottom of the jacking point is 200mm above the ground. Check and verify that the rear wheels are off the ground. Note the results of this check.

Steering Play

Revision 1 – Drafted 11/7/2025 by Test Engineer Michael Ledyard

All previous revisions are obsolete.

Safety Equipment:

- Safety Glasses

Tools:

- Digital Angle Finder
- Wheel Jig

Procedure:

1. Ensure wheels are straight.
2. Fix the wheels in position so they cannot steer.
3. Attach the Digital angle finder to the steering wheel.
4. Turn the steering wheel as far to the left as it goes.
5. Zero the digital angle finder.
6. Turn the wheel as far to the right as it goes.
7. Record the value of the digital angle finder. This MUST be less than 7 degrees.

Wheels and Wet Tread Depth

Revision 1 – Drafted 11/7/2025 by Test Engineer Michael Ledyard

All previous revisions are obsolete.

Safety Equipment:

- Safety Glasses

Tools:

- Tape Measure
- Digital Calipers

Procedure:

1. Lay the wheel on the back of the barrel.
2. Measure the diameter of the wheel.
3. Record this value. This must exceed 8.0 inches.
4. Take the digital calipers and use the stick end, measure from the bottom of the tread to the contact patch.
5. Record this value. This must exceed 2.4mm.

Tilt Test

Revision 1 – Drafted 11/10/2025 by Test Engineer Michael Ledyard

All previous revisions are obsolete.

Safety Equipment:

- Safety Glasses

Tools:

- Digital Angle Finder
- Forklift
- Strap

Procedure:

1. Find a level spot to conduct the test.
2. Acquire a Forklift and an operator from campus operations
3. Attach the digital angle finder to the chassis in a safe place. Zero the digital angle finder.
4. Using the strap, attach the chassis to the forklift forks. Use the lowest tube of the side impact structure
5. Lift the chassis slowly towards 60 degrees from horizontal.
6. If the chassis attempts to flip over before the 60-degree mark is achieved, note the value at which it attempted to flip and repeat the test. If the chassis fails more than 3 times, the test fails.
7. If successful, document the success, set the chassis down gently, and return the forklift.

Steering Forces

Revision 1 – Drafted 11/10/2025 by Test Engineer Michael Ledyard

All previous revisions are obsolete.

Safety Equipment:

- Safety Glasses

Tools:

- Digital Torque Wrench
- Socket
- Steering column adapter
- Weights

Procedure:

1. Design and fabricate the steering column adapter to bolt to the quick disconnect in place of the steering wheel.
2. Weigh the car down to its estimated competition weight
3. Place the socket on the torque wrench.
4. Place the torque wrench on the adapter
5. Use the torque wrench to apply torque on the steering system. Test turning left and right. Record the peak values to turn the wheels.
6. Compare the recorded values to the estimated values.
7. Repeat the test as necessary.

G-Test

Revision 1 – Drafted 11/10/2025 by Test Engineer Michael Ledyard

All previous revisions are obsolete.

Safety Equipment:

- Safety Glasses
- Helmet
- Closed-toed shoes
- Radios
- Harness

Extra Safety Note:

This test has a significantly elevated level of risk associated with it. The speeds are high, and the chassis will be dwarfed by the tow vehicle. Therefore, take extra care to make sure that the FSAE car and tow vehicles stay far away from each other. The chassis driver oversees the test. If he/she is uncomfortable with the setup in any way, the test may not proceed.

Tools:

- Truck
- Quick Disconnect
- Strap
- Accelerometer
- Optimum Lap
- Cones
- Rope
- Speedometer

Procedure:

1. Coordinate with Campus Safety to get Lot 14 closed down.
2. Set up cones as "gates" for the car to pass through. This will help with consistency in getting accelerometer data

3. Install an accelerometer in the car
4. Take the car, truck, strap, and disconnect to the far end of Lot 14 next to the softball field.
5. Rig the towing mechanism in this order: Truck>Strap>Quick Disconnect>Car
6. Attach the rope to the quick disconnect so that the chassis driver can release the car from the tow vehicle
7. Mount the speedometer to the chassis so that the driver can see it. Ensure it will not fall off during the test.
8. Secure the chassis driver in the chassis.
9. Perform a radio check between the driver of the tow vehicle and the car.
10. Gently roll the tow vehicle forward until all slack is taken up in the towing system. Be certain not to jerk the strap or have the car brakes engaged. Doing so could result in the towing system breaking or damaging the car.
11. Once the towing system is tight, accelerate to 50 mph. This might have to be done expeditiously, depending on the amount of room needed to conduct the slalom portion of the test.
12. Once 50mph has been achieved, the chassis driver will disconnect the chassis from the tow vehicle. The chassis driver will then confirm via radio that he is free from the tow apparatus.
13. Have the tow driver accelerate ahead of the chassis to ensure that the towing apparatus does not get entangled in the chassis. Once safely clear of the slalom area, the tow driver may slow safely to a stop and recover the towing strap and disconnect.
14. Once clear of the tow vehicle, the driver of the chassis will navigate the cone course laid out ahead of them. Make as sharp of turn as possible to maximize g forces. Once through the course, they will come to a complete stop. The first time through, the brakes may be vigorously applied to check that the car locks all 4 wheels.
15. Once the run is done, download the accelerometer data.
16. Repeat this test multiple times to determine the average of the data.
17. Once physical testing is completed, measure the distance from gate to gate to build a virtual map of the course.

18. Use the virtual map to recreate the test in Optimum Lap.
19. Take the difference between the simulated values in Optimum Lap and the real values to establish a correction factor.
20. Simulate the skid pad test in Optimum Lap.
21. Apply the correction factor to the simulated values. Use the corrected values as actual performance metrics.

Chassis Stiffness

Revision 1 – Drafted 11/10/2025 by Test Engineer Michael Ledyard

All previous revisions are obsolete.

Safety Equipment:

- Safety Glasses

Tools:

- Straps
- Bottle Jack
- Vehicle Scales
- Suspension mount adapter
- Digital angle finder

Procedure:

1. Secure the rear portion of the chassis to the welding table.
2. Secure the front of the chassis such that the chassis is allowed to twist but is prevented from lifting its nose.
3. Bolt on suspension mount adapter to control arm mounting points.
4. Set up and calibrate vehicle scales. There may only be a need for 1 to 2 scales.
5. Place the bottle jack on the scale.
6. Zero the scale.
7. Mount and zero the digital angle finder.
8. Begin jacking up the corner of the car using the bottle jack. Record the load on the scale at every tenth of a degree of rotation.
9. Measure from the center of the chassis to the bottle jack. This will give the moment arm of the jack.
10. Multiply the load from the scale by the length from step 9. This will give the total torque on the chassis.
11. Graph the data points and get the trendline. This will be the chassis stiffness.

Firewall

Revision 1 – Drafted 11/10/2025 by Test Engineer Michael Ledyard

All previous revisions are obsolete.

Safety Equipment:

- Safety Glasses

Tools:

- Tape Measure

Procedure:

1. Have a representative 95^{th} percentile male sit in the driver's seat in a normal driving position.
2. Measure from the bottom of his helmet to the top of the firewall. Record this value. It should be at least 100mm.
3. Verify that there are no gaps in the firewall and that no liquids can pass through the firewall at any point.

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