Formula SAE: Suspension Sub-team

Initial Design Report

Chloé Meyer: Suspension Lead, Front Knuckles
Tanner Coles: Budget Liaison, Rear Knuckles
Maeve Jastrzebski: Front Knuckles
Austin Hess: CAD Lead, Rear Knuckles
Reuben Goettee: Steering/Brake Lead

Fall 2025-Spring 2026

Project Sponsor: GORE, Flagstaff Chevrolet, ANSYS
Instructor: David Willy

DISCLAIMER

This report was prepared by students as part of a university course requirement. While considerable effort has been put into the project, it is not the work of licensed engineers and has not undergone the extensive verification that is common in the profession. The information, data, conclusions, and content of this report should not be relied on or utilized without thorough, independent testing and verification. University faculty members may have been associated with this project as advisors, sponsors, or course instructors, but as such they are not responsible for the accuracy of results or conclusions.

EXECUTIVE SUMMARY

The Formula SAE Suspension Project focuses on the design, analysis, and optimization of a high-performance suspension system for a single-seat Formula-style race car. The system is intended to maximize tire contact, vehicle stability, and handling precision under dynamic racing conditions. The project's objective is to create lightweight, tunable, and manufacturable suspension and steering subsystems that integrate effectively with the vehicle chassis and powertrain assemblies while meeting FSAE competition requirements.

Through copious research and reference from past submissions, we have compiled different options for each component of our subsystem. These options were compared against each other to find the best possible configuration. The current state of our design includes a steering assembly employing a 33% Ackermann percentage with a double wishbone suspension geometry. This includes control arms and pushrod, rocker arm, and coilover shock assembly. All of these will attatch to our suspension knuckles to interface with the wheels along with our braking system which will include independent front and rear systems.

The sub team's primary concern at the time of writing this document is figuring out the mounting points. The sub team carries an interesting relationship with the frame team, where some of the decisions we make regarding the dimensions of our suspension systems have direct effects on the design of the frame. We are currently using different kinematic softwares to optimize our mounting point geometry. While the CAD model of our suspension is still in the works, we are preparing it as a source for the upcoming prototyping deliverable. Additionally, material selection for our suspension components is still under way.

In terms of updates, the entire team earned a deal with Chevrolet for $5000 dollars, which is a massive help for our upcoming expenditures. Additionally, the team completed a Yardwork Fundraiser on Saturday morning, which will earn the team $1000.

1 BACKGROUND

This chapter will include an overview of the project, establishing its purpose and criteria for success. The first section summarizes the project including budget and fundraising goals. The next section describes key deliverables required by the course and client. The final section describes how the success of this project will be measured.

1.1 Project Description

The suspension sub team will design, fabricate, and assemble a complete suspension kit, along with the braking and steering assemblies for Northern Arizona University's SAE Formula 2026 Competition entry. As the third team to represent NAU in this competition, we are working with limited documentation and resources, which forces us to learn more This project will allow the team to apply technical skills, but also help us learn how to work as a team. This project presents the team with a huge opportunity to transition into the automotive industry and allows us to apply the concepts and methods that we've learned throughout our college careers.

The suspension sub team's estimated budget is $9,600, with a fundraising target of $11,000 to cover extra expenses such as machining, tools, software, and travel. Currently, the entire FSAE team has raised $6,000, with a goal of $33,000. This does not include discounts on software or materials, which have also been donated. There are plans for more fundraising and many opportunities are being explored.

1.2 Deliverables

1.2.1 Course Specific Deliverables

Team Charter (9/5/2025)

Write a comprehensive document detailing the team purpose, team goals, team member personalities/roles/responsibilities, ground rules, and potential barriers & coping strategies.

Presentation #1 (9/18/2025)

Create a presentation detailing customer requirements, engineering requirements, background, benchmarking, literature review, mathematical modeling, schedule, and budget.

Presentation #2 (10/9/2025)

Create a presentation including project description, concept generations, engineering calculations, concept evaluation, concept selection, schedule, budget, and bill of materials.

Report #1 (10/20/2025)

Create a comprehensive initial design report including project description, deliverables, success metrics, customer requirements, engineering requirements, house of quality, benchmarking, literature review, mathematical modeling, functional decomposition, concept generation, selection criteria, and concept selection.

Presentation #3 (11/6/2025)

Create an 8-12 minute presentation including project description, design requirements, design description, engineering calculations, design validation, schedule, and budget.

1st Prototype Demonstration (11/13/2025)

Present two unique prototypes. Sub-teams should present one physical and one virtual prototype. The suspension sub-team plans to do a physical prototype of a front knuckle. Must submit photographic evidence that the prototypes were presented in class.

Report #2 (11/26/2025)

Create a comprehensive conceptual design report including project description, deliverables, success metrics, customer requirements, engineering requirements, house of quality, benchmarking, literature review, mathematical modeling, functional decomposition, concept generation, selection criteria, concept selection, schedule, budget, bill of materials, failure modes and effects analysis, initial prototyping, summary of engineering calculations, and future testing potential.

2nd Prototype Demonstration (12/4/2025)

Present two unique prototypes. Sub-teams should present one physical and one virtual prototype. Must submit photographic evidence that the prototypes were presented in class.

Final CAD & Final BOM (12/05/2025)

All major part and assembly drawings adhering to GD&T standards and an updated bill of materials with as much information as possible.

1.2.2 Competition Specific Deliverables

Structural Equivalency Spreadsheet (1/12/2026)

Must fill out a very large spreadsheet including data mostly about the frame, but also fasteners. Must document the primary structure and show compliance with the Formula SAE Rules. Determine equivalence to Formula SAE Rules using an accepted basis.

Cost Report * [R] (3/30/2026)

List and cost each part on the vehicle using standardized cost tables. Base the cost on actual manufacturing technique used in the prototype. Include tooling cost for processes that require it. Include supporting documentation for officials to verify part costing.

Design 2026 – Design 3-view Drawings (3/30/2026)

Submit three view line drawings showing the vehicle from front, top, and side views. These may be manually or computer generated.

Design 2026 – Design Briefing (3/30/2026)

Submit a design briefing including the following:

Overall Vehicle: Vehicle fundamentals, goals, concept definition & tradeoff studies, project management & execution, vehicle execution, tools, simulation & validation

Dynamics: overall vehicle dynamics, tires, suspension & steering (nonstructural)

Aerodynamics: Aero design & architecture, cooling, cfd, testing and instrumentation, mechanical design

Powertrain: system architecture, control & calibrations, analysis and development, test tune & validation, systems, integrated vehicle validation

Chassis: global targets, frame & structure, suspension & steering (structural), fasteners

Driver Interface: ingress & cockpit, seat & pressure points, controls & instrumentation, brakes, egress

Low-Voltage/Data Acquisition: battery, wiring harness, power management, DAQ

Be ready to discuss the decision-making process, goals, underlying theory, modeling choices, options considered, constraints, assumptions, component/material selection, manufacturing, testing, validation/correlation and successes/failures encountered.

Design 2026 – Design Specification Sheet (3/30/2026)

Specifications for each sub-team. Suspension will focus on tires, wheels, wheel rate, roll rate, sprung mass natural frequency, damping, camber, caster, toe, trail, scrub radius, and many other variables.

1.3 Success Metrics

Success for the suspension sub team will be defined by our ability to meet goals set by the SAE Formula 2026 Competition Standards as well as those set by our team. These metrics will be evaluated using theoretical calculations, physical testing, simulations, and compliance with the Formula SAE 2026 Rules. Our suspension subsystem must be fully operational and pass all inspections before the competition. This includes compliance with all rules set by SAE.

There are many requirements for this project to be successful, which are detailed in the following section. Some parameters include weight targets, and packaging constraints. These will be evaluated in CAD and then physically. Many of the other requirements are tied to performance including camber gain, roll center behavior, and toe characteristics. These are difficult to quantify due to their interconnection and will have to be assessed using kinematic software. More parameters are loading and stress concentration which will be evaluated using Finite Element Analysis.

Our final test of success will be passing technical inspection first at NAU and then at competition. Driver feedback and scores at competition will also be large indicators of success.

2 REQUIREMENTS

In this chapter, the preliminary and updated customer and engineering requirements will be outlined. These requirements came from actual project and competition deliverables and rules. All the requirements were compiled into and compared within our QFD.

2.1 Customer Requirements (CRs)

The customer requirements are as follows:

Fully Operational – Create a car that completes all standardized testing from SAE and passes all inspections before competition.

Pass Inspections – Similarly, must pass inspections made by SAE staff before competition.

Fasteners must be Critical Fasteners - This is a rule directly pulled from the Formula SAE Rules 2026, where rule V.3.1.4 states, "Fasteners in the Suspension system are Critical Fasteners." Critical fasteners are explained in T.8.2.

Visible mounting points – All mounting points connecting the suspension system to the frame must be visible for the inspectors.

Driver Safety – Install components that have the primary function of keeping the driver safe during competition events.

Ease of Vehicle Handling – Steering and suspension systems will be calibrated to allow the driver to easily and comfortably control the vehicle for optimum performance.

Vehicle Stability - Suspension subsystem will be built to provide stability to the car during all aspects of driving.

Durable – The suspension subsystem and all its components will be comprised of reliable materials that will physically withstand all events during competition.

Packaging – The packaging used to secure our suspension subsystem's components to the frame will ensure that all components perform their purpose without interfering with other components.

Sufficient braking system - Part of the inspections and tests are that the braking system must be shown to be strong enough to lock all four tires at a reasonable speed.

2.2 Engineering Requirements (ERs)

Minimum Wheel Travel (50 mm) - This is the least allowable vertical displacement of our vehicle's wheel relative to the chassis.

Shock Absorption (Front: 1000 N*s/m, Rear: 1200 N*s/m) - The ability of our suspension subsystem to dissipate kinetic energy from any impact or vibration present at the racecourse.

Camber Control (Front: -1°, Rear: -2°) - This is the amount of camber present in the front and rear wheels.

Roll Stability (Pass the tilt test: 60°) - A requirement provided by the rule book, where rule IN.11.2.2

states, “Vehicle does not roll when tilted at an angle of 60° to the horizontal, corresponding to 1.7 g.”

Dive Squat Geometry (65%) - This is the ratio between the force returned by our suspension system and the amount of force placed on the front or rear axle. This value was found to be a recommended value during research and is subject to change as we continue with the design process.

Lightweight Construction (24 kg) - Our suspension subsystem will be made from lightweight materials to optimize performance.

Pass Suspension Testing (All 4) - We aim to pass all 4 suspension tests performed by the SAE staff before competition.

Track Width (<1550mm) - The track width needs to be below the 1550 mm mark by FSAE rules.

Wheelbase (<1500mm) - The wheelbase must be below 1500 mm by FSAE rules.

Steering free play (<7°) - The steering assembly must have less than 7° of steering wheel turn before the system is affected at the wheels by FSAE rules.

2.3 House of Quality (HoQ)*room for the body of the paper.)]*

Component QFD: Suspension

Project:	**QFD - Suspension**
Date:	09/18/25

		1	2	3	4	5	6	7
1	Minimum Wheel Travel							
2	Shock Absorption	+						
3	Optimize Camber Control		+					
4	Increase Roll Stability		+	++				
5	Optimize Dive/Squat Geometry		+	+				
6	Lightweight Construction							
7	Pass Suspension Testing	++	+		+			

Legend

A	NAU FSAE 2025
B	F4 Car Suspension
C	MIT FSAE 2025

			Functional Requirements							Customer Opinion Survey				
	Customer Needs	**Customer Weights (1-5)**	Minimum Wheel Travel	Shock Absorption	Camber Control	Roll Stability	Dive/Squat Geometry	Lightweight Construction	Pass Suspension Testing	*1 Poor*	*2*	*3 Acceptable*	*4*	*5 Excellent*
1	Fully Operational	5	3	9	1	9	1	1	9		A			BC
2	Fasteners must be Critical Fasteners	5		3	1	3		1			A			BC
3	Visible mounting Points	5			3		1	1				B		AC
4	Driver Safety	5	3	9	9	9	1	3	9	A			C	B
5	Pass Inspections	5	9	3		9	1		9	A				BC
6	Ease of Vehicle Handling	4	3	3	9	3	9	3			A			BC
7	Vehicle Stability	4	1	9	9	9	3	1	3			A	C	B
8	Durable	2		3	1			3				A		BC

Technical Requirement Units	mm	N*s/m	Degrees (°)	Degrees (°)	N/N (%)	kg	# of tests
Technical Requirement Targets	50	Front: 1000, Rear: 1200	Front: -1°, Rear: -2°	Pass Tilt Test (60°)	65	24kg	4
Absolute Technical Importance	91	174	144	198	68	52	147
Relative Technical Importance	10.4%	20%	16.5%	23%	8%	6%	17%

3 Research Within Your Design Space

3.1 Benchmarking

For our first benchmark, we decided to use the 2023-2024 NAU FSAE vehicles suspension set up, including inboard springs and shocks with a pushrod style system to connect to the fabricated knuckles. The design also included double a-arms that connected to the frame with changeable shims to fine tune the landing points for the main suspension members. This design also features the use of two universal joints to connect the steering wheel to the rack mounted on the floor in front of the central axis of the front wheels. This steering rack mounts to the knuckles through rods connected to the knuckle via interchangeable pieces that vary the steering geometry and the handling characteristics. The major aspect of this design that we are looking at is the prevalent ability to fine tune and adjust most of the components through minor, easily remanufactured pieces rather than an entire knuckle or arm geometry.

For our second benchmark, we looked at a full formula vehicle such as the ones competing in Formula 4, an international feeder series that eventually leads to Formula 1, which is considered one of the fastest racing series. These vehicles are heavily fine-tuned for speed, especially while cornering which necessitates very well-designed suspension. These vehicles use a double wishbone suspension layout with inboard springs and dampers which can be exchanged at any circuit to further tune the handling characteristics. These cars also use either pushrod or pull rod suspension depending on the team, which gains more leverage and can more directly transmit the forces into the frame and back into the wheels.

The last benchmark we made was on the 2025 MIT FSAE design, to gain insight into what the suspension system of a best-in-class vehicle under our same restrictions would look like. This design also employs the use of dual A-arms, effectively splitting and vertically stabilizing the vehicle under loads that would be seen under heavy use such as during the endurance or auto-cross events with the sudden and violent direction changes and weight shifts. This car uses inboard springs and shocks, hiding them inside the aerodynamic elements to reduce unnecessary drag created by the abnormal profile of the suspension elements.

3.2 Literature Review

3.2.1 Chloé Meyer

[1] E. Gaffney and A. Salinas, "Introduction to Formula SAE Suspension and Frame Design," SAE International, Apr. 1997. Accessed: Sep. 15, 2025. [Online]. Available: https://racing.byu.edu/0000018a-6be7-df21-a5fe-fbef2f9d0001/intro-to-frame-and-suspension-design

This reference outlines important factors to consider in designing a Formula SAE frame and suspension. It focuses on geometry, stiffness, and handling. It emphasizes how trackwidth, wheelbase, and suspension angles affect performance and driver control. These factors are very important to understand for our project and are helping us optimize many variables that are affected by the suspension geometry.

[2] E. Flickinger, "DESIGN AND ANALYSIS OF FORMULA SAE CAR SUSPENSION MEMBERS," California State University, Northridge, 2014. Accessed: Sep. 15, 2025. [Online]. Available: https://scholarworks.calstate.edu/downloads/0p096b29p

This thesis from CSU Northridge outlines methods for analyzing suspension member forces in Formula SAE using both hand calculations and CAD modeling. It details optimizing material and geometry to reduce weight. This force analysis was used along with other sources to determine the forces in our project's members, which will be used to determine their optimal angles, lengths, materials, and diameters.

[3] Car Design Workshop, "Six Suspension Design Insights by Analysing Suspension Loads (Project 171)," *YouTube*, Mar. 15, 2025. https://www.youtube.com/watch?v=cUwp7mj6dYo (accessed Sep. 15, 2025).

This video details how to calculate the forces in each member of a double wishbone suspension with a pull rod. This video, along with other sources, was used in calculations to find the forces in each member for our suspension design.

[4] W. Harvey, "The Optimization of a Formula SAE Vehicle's Suspension Kinematics," Massachusetts Institute of Technology, 2018. Accessed: Sep. 15, 2025. [Online]. Available: https://dspace.mit.edu/bitstream/handle/1721.1/119955/1080340074-MIT.pdf?sequence=1&isAllowed=y

This journal from MIT outlines a process for suspension design, including kinematic analysis and optimization of things like camber, roll center, and ball joint geometry. It details the effects of pickup points on contact patch, bump steer, and weight transfer. This is important to our design because our aim is to find pickup points to optimize roll center, camber control, and many other things that the journal touches on.

[5] E. Goodman, "Race Car Vehicle Dynamics & Design Applied to Formula Student," Aston University, May 2009. Accessed: Sep. 15, 2025. [Online]. Available: https://publications.aston.ac.uk/id/eprint/21810/1/MPhil_EJ_Goodman_2009_reduced.pdf

I read Chapter 10 of this book, which outlines suspension geometry. It details trackwidth, wheelbase, and their effects on longitudinal and latitudinal acceleration. This was important to our project as we were able to use our goals and constraints to choose preliminary values based on this book and other references.

[6] D. J. B and S. P. R, "Design and calculation of double arm suspension of a car," Journal of Mechanical Engineering, Automation and Control Systems, https://www.extrica.com/article/21436 (accessed Sep. 15, 2025).

This reference is an academic journal detailing calculations for double wishbone geometry. It details ride frequency, damping coefficient, and stiffness. It includes CAD models, stress analysis, and dynamic testing. This information was useful to our project as it helped us choose suspension geometry.

[7] "Free suspension tuning spreadsheet," Suspension Spreadsheet, https://robrobinette.com/Suspension_Spreadsheet.htm (accessed Sep. 15, 2025).
This reference is a spreadsheet which was used for unsprung weight estimates, but has much more for suspension tuning, which will be useful in the future. These calculations will be verified before any hard values come from it.

3.2.2 Austin Hess

[8] "Anti Squat, Dive and Lift Geometry – Geometry Explained," Suspension Secrets, Aug. 18, 2018. https://suspensionsecrets.co.uk/anti-squat-dive-and-lift-geometry/

Technical magazine providing antidive/squat percentage calculations, aiding in early mounting point calculations

[9] Z. Bognar, "Anti Dive explained and the bolt on solution for FR-S / GR86 / BRZ!," GKTech Australia, Jan. 30, 2025. https://au.gktech.com/blogs/news/anti-dive-explained-and the-bolt-on-solution-for-fr-s-gr86-brz

Technical magazine showing mounting point height vs. anti-dive percent, providing insight as to how antidive is affected by mounting points.

[10] N. Dropkin, "A Guide To FSAE Axles," DesignJudges.com. https://www.designjudges.com/articles/a-guide-to-fsae-axles

Axle and connective joint details for FSAE performance cars, providing insight as to what our hardpoints should look like and what some optimal locations typically involve.

[11] "Formula SAE Rules 2025," SAE International, Aug. 2024. Accessed: Sep. 17, 2025. [Online]. Ch. IN.9 https://sites.usnh.edu/unh-precision-racing/wp-content/uploads/sites/136/2025/03/FSAE_Rules_2025_V1.pdf

Rulebook to reference tilt-test, chapter IN.9

[12] B. Zhu and N. Sun, "Design and Optimization of FSAE Race Car Suspension System," Atlantis-Press.com, 2015. (accessed Sep. 17, 2025).

Student FSAE report with figures and testing; gave an idea of what our team should generally consider

when designing suspension as it relates to kingpin inclination angles.

[13] N. Roner, "Optimum Suspension Geometry for a Formula SAE Car," PDXScholar, Mar. 2018, doi: https://doi.org/10.15760/honors.542

Student thesis discussing FSAE suspension and methods of optimizing vehicle design

[14] J. Dixon, Suspension Geometry and Computation. West Sussex, United Kingdom: John Wiley and Sons, Ltd, 2009, pp. 57–59.

Textbook Ch. 2.9 discusses two axle vehicle analysis using equations, providing calculatable metrics

3.2.3 Maeve Jastrzebski

[15] J. Edgar, Vehicle Ride and Handling: Testing, Modification, and Development, ch. 2-3, 5. Warriewood, Australia: Veloce Publishing, 2019.

[16] E. J. Goodman, Race Car Vehicle Dynamics and Design Applied to Formula Student, ch. 11, 15, 17. Master of Philosophy thesis, Aston University, May2009.

[17] H. Adams, Chassis Engineering, ch. 3, 5. The Berkley Publishing Group,1993.

[18] A. Staniforth, Competition Car Suspension: Design, Construction, Tuning, 4th ed., Haynes Publishing, 2006. ISBN: 978-1844253289.

[19] R. N. Jazar, Vehicle Dynamics: Theory and Application, 3rd ed., Springer International Publishing (Cham), 2017. ISBN: 978-3-319-53441-1.

[20] C. Carroll Smith, Tune to Win: The Art and Science of Race Car Development and Tuning, First edition, Aero Publishers, Inc., June 1, 1978. ISBN: 978-0879380717.

[21] G. Wheatley and M. Zaeimi, "Anti-Roll Bar Design for a Formula SAE Vehicle Suspension," Scientific Journal of Silesian University of Technology. Series: Transport, Vol. 116, pp. 257-270, Sep. 2022. DOI: 10.20858/sjsutst.2022.116.17.

3.2.4 Tanner Coles

[22] A. C. Cobi, "*Design of a Carbon Fiber Suspension System for FSAE Applications*," B.S. thesis, Dept. Mechanical Engineering, Massachusetts Institute of Technology, Cambridge, MA, 2010, Ch. 2.

This thesis provides a detailed case study of an FSAE suspension design using composite materials. It outlines weight reduction techniques, stress analyses, and material optimization strategies for carbon-fiber components. The discussion of structural stiffness and safety factors is particularly useful for designing lightweight, high-performance suspension arms.

[23] J. C. Dixon, *Suspension Geometry and Computation*, Ch. 10. Chichester, U.K.: Wiley, 2009.

The text delivers a computational perspective on suspension geometry, emphasizing kinematic relationships and simulation methods. Chapter 10 focuses on camber control, roll center analysis, and toe change through travel, which are all key parameters for designing and tuning double-wishbone or pushrod-type suspension systems.

[24] T. D. Gillespie, *Fundamentals of Vehicle Dynamics*, Ch. 7. Warrendale, PA: SAE International, 1992.

This text explains vehicle dynamic principles, including load transfer, ride comfort, and suspension compliance. The chapter specifically discusses the interaction between sprung and unsprung masses and their effect on handling, making it essential for determining target damping ratios and spring rates.

[25] J. Edgar, *Vehicle Ride and Handling: Testing, Modification, and Development*, Ch. 10–12. Warriewood, Australia: Veloce Publishing, 2019.

Chapters 10-12 explore practical ride and handling assessment techniques, including subjective testing, data logging, and iterative tuning. Edgar integrates theory with real-world modification practices, offering guidance on translating analytical suspension targets into physical testing results.

[26] The Complete Guide to Anti-Squat – Suspensions Explained, *Engineering Explained*, 2017. [Online]. Available: https://www.youtube.com/watch?v=XuxhI4CBaNk. [Accessed: Sep. 17, 2025].

This video provides a visual explanation of anti-squat and anti-dive geometry in vehicle suspension systems. It is valuable for early design concepts and for communicating geometry effects.

[27] W. F. Milliken and D. L. Milliken, *Race Car Vehicle Dynamics*, Ch. 21. Warrendale, PA: SAE International, 1995.

Chapter 21 discusses advanced suspension tuning and the influence of camber gain, roll center movement, and tire load sensitivity on high-performance handling. The text is regarded as a standard reference for competitive motorsports engineering, providing theory-backed methods for achieving predictable handling balance.

[28] A. Sharma and P. Sankar, "*Influence of Anti-Dive and Anti-Squat Geometry in Combined Vehicle Bounce and Pitch Dynamics*," SAE Technical Paper 2007-01-0814, Apr. 2007. doi: 10.4271/2007-01-0814.

This paper analyzes how suspension geometry affects vehicle pitch and bounce modes, emphasizing the mathematical modeling of anti-dive and anti-squat. Its quantitative approach helps engineers predict vertical and longitudinal weight transfer effects for improved traction and braking stability.

[29] Society of Automotive Engineers (SAE), *Design Standard: SAE J410*.

This SAE design standard outlines dimensional, geometric, and material guidelines relevant to suspension

system components. It provides standardized design practices ensuring safety, consistency, and compatibility across vehicle subsystems.

3.2.5 Reuben Goettee

Suspension Geometry and Computation [30]

This source describes the aspects surrounding steering design and how to design a proper Ackermann geometry to maximize the utility of the design. This goes into depth about the benefits of positive Ackermann, anti-Ackermann, or true neutral geometry and what each style is best at maximizing. It shows how Ackermann is best for slower vehicles, where slip is not anticipated, while anti-Ackermann is predicting sliding and is using the slip angle of the tires to get the most out of the compound. This has been very helpful in designing and selecting the steering geometry and desired Ackermann percentage to maximize the steering performance for the track we will be competing on.

FSAE Rules Chapter V.3 and Ch T.3 [31]

This chapter describes the limitations and the criteria that need to be met by the steering assembly. This includes the rule on free play mentioned in the engineering requirements. This also dictates the construction requirements of the steering assembly, which is necessary for driver safety in operation, but also in the failure modes of the assembly. Chapter T.3 describes the requirements and criteria for the braking system that need to be met to pass inspection and be allowed to compete. This will help guide as to what design choices may be breaching the rules and need to be reconsidered

Vehicle Ride and Handling: Testing, Modification, and Development Ch 6 [32]

This chapter is describing more on steering geometries that are very helpful in handling of higher performance vehicles as well as methods of testing and tuning this geometry for the driver preference. This will help us design a steering feel that will help the drivers be faster on track and more confident with the vehicle as the steering is one of the main feedbacks to tell the driver what is internally happening with the vehicle.

Shigley's Mechanical Engineering Design Ch 16 [33]

This chapter is describing vital considerations for the design of disc brake systems. This includes the forces related to a disc brake, the effective radius calculation that describes the acting point of the brake on the rotors, and other very valuable variables and equations. This provides a great starting point to move from when designing the brake system as it gets within a ballpark estimate of the necessary dimensions of components such as the rotors, calipers, pads and the master cylinders.

Design of FSAE braking system [34]

This source has helped benchmarking as well as to give a quantifiable bar for general variables. This is a report done by a student at MIT analyzing the braking system for their electric FSAE vehicle wich means that the braking system slightly differs with a harvesting system, but some of the calculations can be modified to fit our application. This has helped immensely as a sanity chack to ensure that the values returned by some of the initial calculations are reasonable compared to a similar vehicle, as well as to check that we are not overlooking important design aspects.

Road vehicles — Specification of non-petroleum-based brake fluids for hydraulic systems[35]

ISO standard on brake fluids detailing water resistance, temperature capabilities, and compression ratings. This also details what type of fluid we may want to look into based on standardized properties, or rated properties. This will help ensure we do not select a fluid that will flash boil under the temperature and pressure conditions that it may be subjected to under high stress situations out on track.

Design and analysis of Braking System of a FSAE vehicle [36]

This is another source of information that helps detail most of the necessary components that are integral to the braking system. This is a report done by a graduate engineer trainee on brake design for FSAE type vehicles that includes simulation results and other important equations that have been very helpful in narrowing down the sizes of our parts to help move us on to the next steps in design. This has also helped in looking for more possible oversights such as the pressure ratings on all components in the brake loop and ensuring that they can withstand the brake line pressure that would be applied.

3.3 Mathematical Modeling

3.3.1 Control Arms & Pushrod Sub-Assembly – Chloé Meyer

I used the following equations to determine the forces in each member of our front suspension design. This will be useful to determine optimal pickup points, angles, lengths, and diameters for the control arms and pushrods. This is simplified and includes a lot of assumptions including no aerodynamic package, lateral and longitudinal acceleration are 1g, mass of car and driver is 200kg, the height of the center of gravity is 300mm, and the coefficient of friction is 1.3.

$$\Sigma F_{x,y,z} = 0 \tag{1}$$

$$WT_{lat} = \frac{m_{car\&driver} \cdot a_{lat} \cdot h_{CoG}}{Trackwidth} \tag{2}$$

$$WT_{long} = \frac{m_{car\&driver} \cdot a_{long} \cdot h_{CoG}}{Wheelbase} \tag{3}$$

Results:

Table 1: Forces in Control Arms & Pushrod

Member	Force (N)
Pushrod	-0.77
Upper Control Arm	-19.33
Lower Control Arm	18.742

3.3.2 Geometry – Chloé Meyer

To determine how to optimize scrub radius, I decided to make a MATLAB code that would show the effects of wheel offset, kingpin inclination, camber, and ride height on it. I found ranges for each value from other Formula SAE teams and used the equations shown in Appendix A under Figure 10: MATLAB Code for Effects of Different Variables on Scrub Radius.

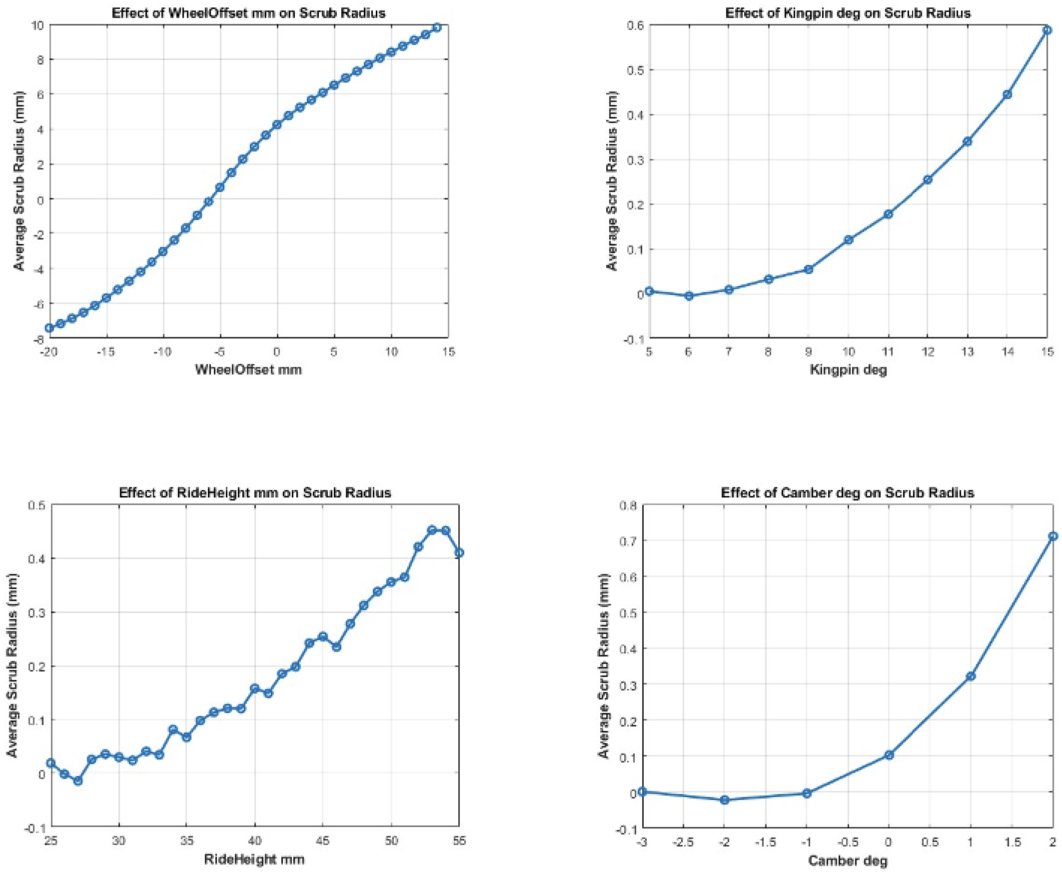

Figure 1: Effects of Variables on Scrub Radius

3.3.3 Anti-Roll, Springs, and Damper – Maeve Jastrzebski

Roll Moment: $M_{roll} = ma_y h_{cg} = 2452.5N \times m$ (4)

Roll Stiffness: $K_{\phi,springs} = 2(kMR^2)t^2$ (5)

Front: $K_{\phi,f,springs} = 40{,}000\ \frac{Nm}{rad}$ $K_{arb,f} = 10{,}000\ \frac{Nm}{rad}$

Rear: $K_{\phi,r,springs} = 35{,}000\ \frac{Nm}{rad}$ $K_{arb,r} = 8{,}000\ \frac{Nm}{rad}$

Total Roll Stiffness: $K_{\phi,total} = K_{\phi,f} + K_{\phi,r} = 93{,}000\ \frac{Nm}{rad}$ (6)

Front: $K_{\phi,f} = 40{,}000 + 10{,}000 = 50{,}000\ \frac{Nm}{rad}$

Rear: $K_{\phi,r} = 35{,}000 + 8{,}000 = 43{,}000\ \frac{Nm}{rad}$

Roll Angle: $\phi = \frac{Mr}{K_{\phi,total}} = 1.51°$ (7)

Load Transfer Distribution: $LLTD_{front} = \frac{K_{\phi,f}}{K_{\phi,f}+K_{\phi,r}} = 46.2\%$ (8)

Spring Rate: $k_s = \frac{k_w}{MR^2} = 62.46(front),\ 135.71(rear)\ \frac{N}{m}$ (9)

Natural Frequency: $f_n = \frac{1}{2n}\sqrt{\frac{k_w}{m_{corner}}} = 2.25(front),\ 2(rear)Hz$ (10)

Wheel Rate: $k_w = (2\pi \times f_n)^2 = 89.94(front),\ 86.85(rear)\ \frac{N}{m}$ (11)

Damping Ratio: $\zeta = \frac{c}{c_{critical}} = 0.275(front),\ 0.25(rear)$ (12)

Target Damping Ratio: $\zeta_{t\arg et} = 0.05\ +\ 0.1f_n = 0.275(front),\ 0.25(rear)$

Critical Damping: $C_{critical} = 2\sqrt{k_s} \times m_{corner} = 3.75(front),\ 6.11(rear)$ (13)

Actual Damping: $C = \zeta_{t\arg et} \times C_{critical} = 1.03(front),\ 1.53(rear)$ (14)

I was tasked with working out the designs for the anti-roll bar as well as springs and dampers. In the first five equations, I used assumptions or ranges as most of the needed information was not calculated or chosen yet. For the next set of equations, I created a master excel sheet in order to make all of our calculations adaptable for future design changes.

Inputs				
Parameter	Symbol	Value	Units	Notes
Vehicle mass	M	250	kg	total mass, driver&fuel
Front weight fraction	wf	0.45		fraction on front axle
Rear weight fraction	wr	0.55		0.55
CG height (m)	h_cg	0.28	m	from ground to CG
Wheelbase	L	1.53	m	
Front track	Tf	1.4	m	track width front (center to center)
Rear track	Tr	1.05	m	track width rear (can't be smaller than 75% of front)
Tire rolling radius	rt	0.127	m	approximate tire rolling radius
Target natural freq (front)	ff	2.25	Hz	corner natural frequency desired (front)
Target natural freq (rear)	fr	2	Hz	corner natural frequency desired (rear)
Front motion ratio	MRf	1.2		wheel to spring motion ratio
Rear motion ratio	MRr	0.8		wheel to spring motion ratio
Damping ratio	zeta	0.25		
Gravity	g	9.81	m/s^2	constant

Calulated Inputs				
Item	Symbol	Value	Units	Formula
Front corner mass		56.25	kg	56.25
Rear corner mass		68.75	kg	68.75
Wheel rate front	kw_front	89.94	N/m	89.93677
Wheel rate rear	kw_rear	86.85	N/m	86.852519
Spring rate front	ks_front	62.46	N/m	62.45609
Spring rate rear	ks_rear	135.71	N/m	135.70706
Spring rate front		0.06	N/mm	0.0624561
Spring rate rear		0.14	N/mm	0.1357071
Spring rate front		0.36	lb/in	0.3566335
Spring rate rear		0.77	lb/in	0.7749073
Damping front		2.65	Ns/m	2.6507188
Damping rear		4.32	Ns/m	4.3196899

Figure 2: Table of Initial Inputs Chosen & Calculated

Spring, Damper, and Anti-Roll Bar Design				
Parameter	Symbol	Value	Units	Formula
Spring rates (in calculated inputs)				
Natural frequency (in inputs)				
Damping ratio, front		0.275		0.275
Damping ratio, rear		0.25		0.25
Target damping ratio, front (estimation based off of research		0.275	Hz	0.275
Target damping ratio, rear		0.250	Hz	0.250
Critical damping, front	C_cf	3.749	Ns/m	3.7486825
Critical damping, rear	C_cr	6.109	Ns/m	6.1089640
Actual damping, front	C_f	1.031	Ns/m	1.0308877
Actual damping, rear	C_r	1.527	Ns/m	1.5272410

Figure 3: Table of Anti-Roll Bar Design Values

3.3.4 Mathematical Modeling – Austin Hess

When trying to determine initial parameters for track width, the main metric considered was whether or not the vehicle would be able to pass the 60° tilt test. Along with center of gravity height and weight distribution, track width plays a major role in maintaining vehicle position without slippage, as well as general vehicle dynamics. To determine this parameter, rough estimates of CG height, vehicle weight, ride height, and tire friction coefficients were plotted using MATLAB. The model below provides insight into viable track width options, allowing our team to determine a front track width of 1.4m and a rear track width of 1.3m. The wider front-end set up provides ideal vehicle handling with some margin for error, being reasonably shorter than the 1.5m maximum width and requiring excessive lateral forces to overturn the vehicle.

```
% Forces and Moments
normal_force = vehicle_mass * gravity;
lateral_accel = gravity * tan(tilt_angle_rad);

overturning_moment = vehicle_mass * lateral_accel * cg_height;
restoring_moment = normal_force * (tw / 2);

% Safety Margin
safety_margins(i) = restoring_moment - overturning_moment;
```

Figure 4: MATLAB Code Calculations

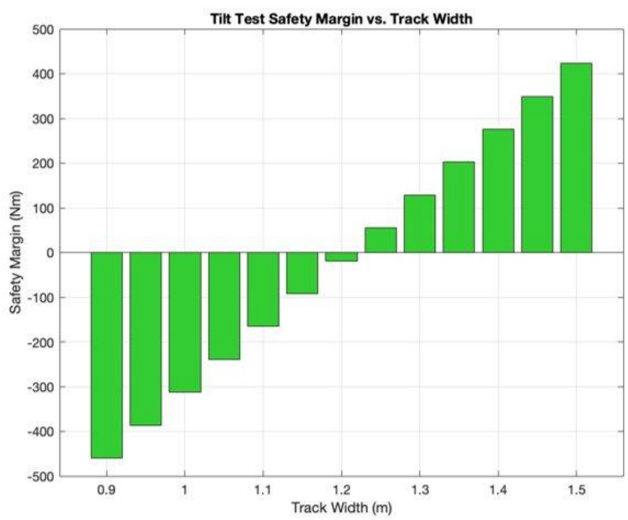

Figure 5: Track Width vs. Safety Margin

Unsprung mass was the next calculation performed. Unsprung mass represents the mass of the vehicle not supported by the suspension system. This includes the tires and rims, steering knuckles, hub bearings, brake calipers and discs, and portions of control arms, pushrods and dampers. Unsprung mass is a valuable calculation to begin finding areas of weight conservation to plug into suspension software, eventually allowing us to nail down vehicle handling dynamics. With an estimate of each component's mass per vehicle corner, the total unsprung mass was estimated to be 41kg.

$$Total\ Unsprung\ Mass = 4 \cdot (\sum U_C + \sum U_{PS})$$

$$U_C = Unsprung\ Component\ Mass$$
$$U_{PS} = Component\ Mass \cdot Component\ Motion\ Ratio$$

Table 2: Unsprung and Partially Sprung Components

Unsprung/Partially Sprung	Component	Mass (kg)	Motion Ratio
Unsprung	Tires & Rims	3.5	N/A
	Upright	2.0	N/A
	Hub Bearing	1.0	N/A
	Brake Calipers	1.5	N/A
	Brake Disc	1.2	N/A
Partially Sprung	Control Arms	1.0	0.4
	Pushrod	0.3	0.5
	Dampers	1.0	0.5

3.3.5 Anti-Squat Percentage Testing and Calculations – Tanner Coles

The first modeling I performed was an excel tool online that automatically calculates the anti-squat percentage based on the location of mounting points. This would be an appropriate tool to perform reiterations on our mounting points to obtain the most ideal anti-squat percentage. Within the tool, the mounting points are defined as coordinates; one line was drawn between the top two mounting points, and another for the bottom pair, where these lines meet is the instant center, and the location of this instant center. Multiplying the slope of this instant center by the ratio between the wheelbase and the height of the center of gravity gives the anti-squat for the specific geometry. The following screenshot shows an example geometry for our suspension and the resulting anti-squat percentage. All the light blue boxes are input values, including the weight of the car, and the height of the center of gravity:

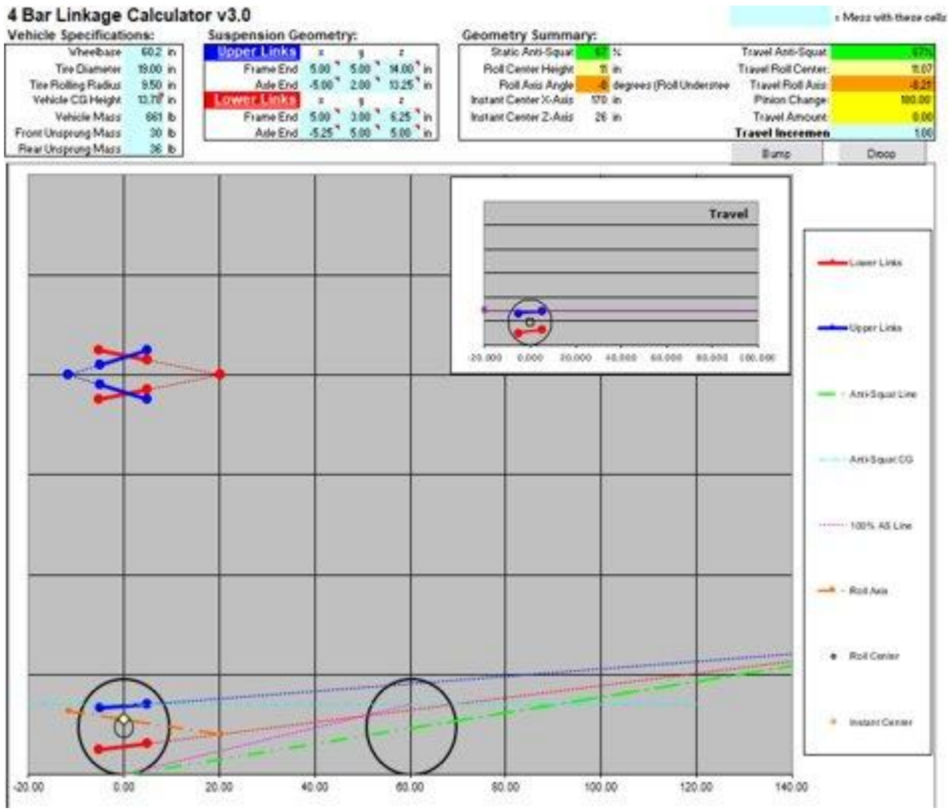

Figure 6: Example Suspension Geometry and Following Anti-Squat Percentage

The following equations show the progression of the anti-squat percentage into a calculation solving for the instant center slope:

$$AS\% = \left(\frac{IC_y}{IC_x}\right)\left(\frac{WB}{h_{CG}}\right)\eta \cdot 100 \tag{15}$$

$$\left(\frac{IC_y}{IC_x}\right) = AS\%\left(\frac{h_{CG}}{WB}\right)\left(\frac{1}{\eta \cdot 100}\right)$$

As an example, with an center of gravity height of 0.28 m and a wheelbase of 1.53 m, η =1, and a target anti-squat percentage of 95%, I calculated an example instant center slope.

$$\left(\frac{IC_y}{IC_x}\right) = 95\% \cdot \left(\frac{0.28}{1.53}\right) \cdot \left(\frac{1}{1 \cdot 100}\right) = 0.1738$$

Mounting points can be manipulated to match this instant center slope and therefore match our target anti-squat percentage. This anti-squat percentage is different than the target from earlier due to the differences in recommended anti-squat percentages. Once more developed designing occurs with our suspension system, we will lock in on a confirmed anti-squat percentage.

3.3.6 Steering and Brakes-Reuben Goettee

I am tasked with the design of the steering and the braking systems. For this I have started my mathmatical calculations by setting up a code to quickly iterate on different brake perameters such as the pedal ratio, driver input force, brake pad friction coefficient, tire radius, rotor radius, calliper designs, and an added factor of safety to absolutely ensure that we can improve on the previous years design and pass the SAE technical inspection. As shown in figure 5, these are the outputs of the code, showing the line pressure in megapascals to ensure that all components in the line can withstand the sxpected maximum pressure, and the master cylinder sizes for the front and back loops. We can see that both are below a one inch diameter which is to be expected given the expected vehicle weight and available driver force input into the pedal assembly.

```
BrkLnPrsFront =

    3.3909


BrkLnPrsBack =

    2.2606


FrontMSTCylDiam =

    0.6823


BackMSTCylDiam =

    0.8357
```

Figure 7: Outputs for Brake Design Calculations

For the steering design I have also made a few simple hand calculations using the below equations to obtain the necessary truning angle of each front tire to achieve a desired minimum turning radius. I then turned these numbers into a 2-D cad model of a simplified vehicle geometry showing all four tire locations and the needed steering linkage lengths that will be needed to achieve the turning radius (Figure 6). All of these steering calculations are in progress as I am a little behind of where I would like to be in designing the steering but I have made plans to catch back up as to not hinder the team and ensure we have a functioning vehicle come spring.

$$\delta_{Inside} = tan^{-1}\left(\frac{L}{R-\frac{T}{2}}\right) \tag{16}$$

$$\delta_{Outside} = tan^{-1}\left(\frac{L}{R+\frac{T}{2}}\right) \tag{17}$$

From these, inputting the wheelbase for L, the desired minimum turning radius for R, and the track width for T, we obtain that for a 3 meter turning radius, the inner tire turns 33° and the outer turns 22 ° giving an Ackermann percentage of 33%. From this I made the following model to visualize what that difference looked like and what the steering geometry could look like.

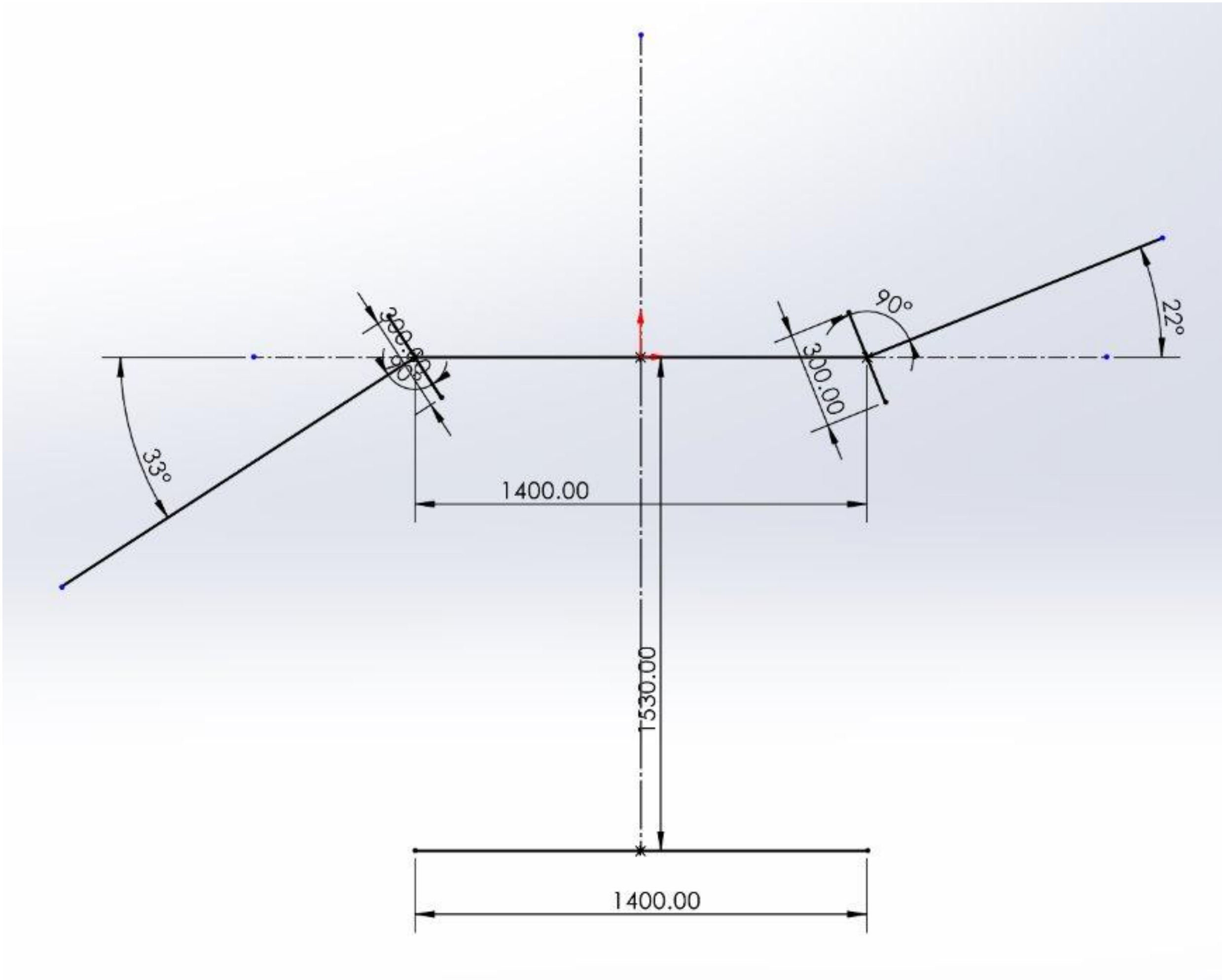

Figure 8: 2-D model of Steering Geometry

4 Design Concepts

4.1 Functional Decomposition

Functional decomposition is critical for a suspension team, as it breaks down the complexity of the system into something manageable, clear, and with testable components illustrated. It clarifies the system's responsibilities by linking each subsystem to a specific function so that nothing is forgotten. Decomposition helps to identify which parameters will affect the overall performance of the vehicle and highlights what needs to be prioritized, such as roll center height, camber gain, and damping rates. After highlighting what needs to be prioritized, the sub teams, which consist of kinematics, damping, fabrication, and testing, can independently focus on their individual functions while making sure that there is integration with the overall goals of the whole system. Overall, the functional decomposition chart is a roadmap that illustrates our teams' goals of vehicle stability and control into a detailed set of engineering objectives.

Suspension System

Support and Load Transmission	Control Wheel Motion and Geometry	Ride and Roll Dynamics
Interfaces	Adjustability	Serviceability

Figure 9: Suspension Decomposition

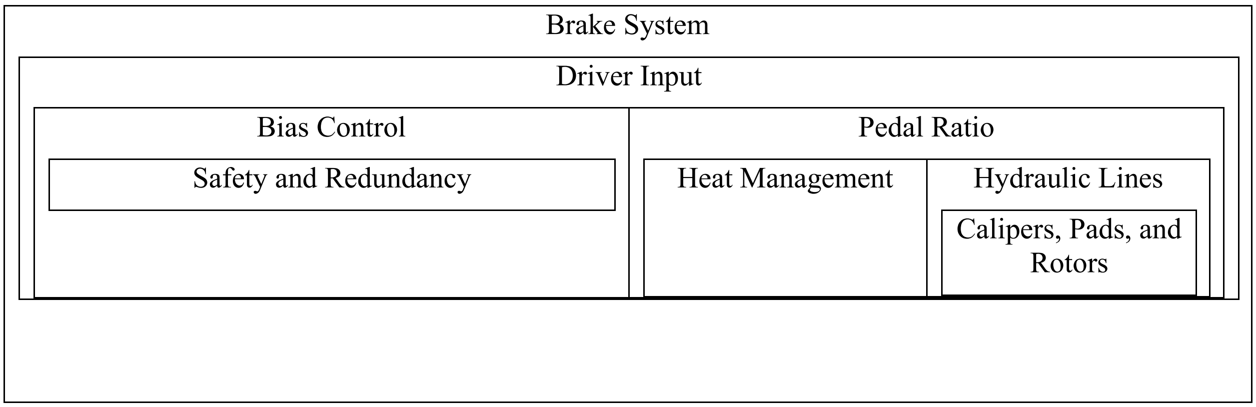

Figure 10:Brake Decomposition

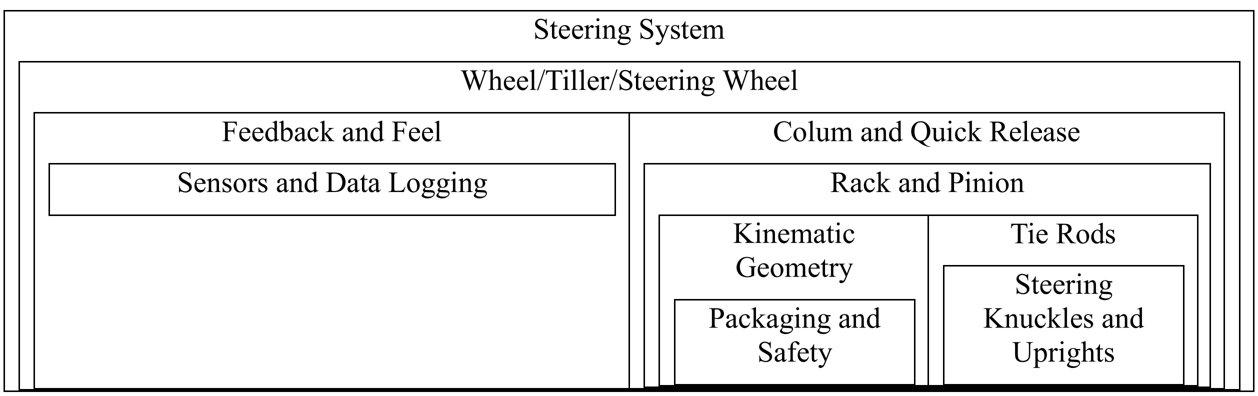

Figure 11:Steering Decomposition

4.2 Concept Generation

The extent of our suspension concept generation is concerned with seven separate criteria. These criteria are how do we support the vehicle load; how do we control the wheels motion; how do we absorb any shocks that are introduced into the system; how do we maintain stability in the wheels , such as camber, caster, toe, and how do we keep the wheels attached; how can we maintain a level of adjustability; how do we control the suspension travel; and how do we connect to the chassis members.

Through these we came up with the two most viable solutions for each sub-section and made variations containing each to consider as a full suspension package. The most notable of these selections is the use of coil springs and a hydraulic shock and damper for the support and absorption, a double wishbone and a pushrod to locate and transmit loads to and from the road and chassis. Lastly, to add in adjustability and the ability to fine tune our design for the best handling and reactions for driver preferences, we looked at adjustable rod ends with ball joint ends to connect to the chassis members and maintain the needed flexibility in our suspension design.

- Coil springs

  Pros: materially and spatially efficient compared to leaf springs, negligible internal friction which can reduce frictional and thermal degradation of the component

  Cons: linear spring rate, potential to buckle under very high compression loads

- Double wishbone

  Pros: Allows for great handling stability and adjustability when paired with the ball joints and adjustable rod ends

  Cons: Complex geometry, complex integration, larger form factor which could hinder aerodynamics

For the brakes, the main consideration in the designing process is in the master cylinder, the brake calipers, and the pad-rotor interface. All the components that will be attached to the knuckles will need to be contained within the wheel rim to maintain space efficiency and aerodynamic profiling, which means that the caliper selection is restricted to mostly one- or two-cylinder designs, and this will dictate what pad materials are available. As the mathematical modeling section has shown, there is a set size for the master cylinder size, which is also dependent on the chosen style, size, and other parameters of the calipers.

- Wilwood PS-1

  Pros: cheaper, larger pad area, can comfortably fit with any chosen rotor diameter we may choose

  Cons: heavier material, higher mounting height, smaller piston bore

- Wilwood GP200

  Pros: lighter, lower mounting height, larger piston bore

  Cons: more expensive, smaller pads, minimum rotor diameter is right at the maximum diameter that would still fit within the rims

For the steering, the main sections that require concept generation are the wheel, the rack and pinon, and the connection method between the two. These areas have yielded ideas such as a carbon fiber wheel for weight saving measures, or an aluminum wheel to ensure rigidity and safety in the event of a failure as well as simplicity in manufacturing. Another concept is to use universal joints to connect between the steering wheel and the rack covering the change in angle that is necessary, or to use a beveled gearbox that could seamlessly transfer the steering inputs through any designed angle necessary at the cost of complexity, manufacturing cost, and time.

- Universal joints

  Pros: Tested and proven by both previous NAU cars, cheaper to implement and purchase

  Cons: can add unnecessary free play into the system which could come near to the required limit, needs external bracing

- Gearbox

  Pros: potentially lighter, can reduce the need for external bracing and is a much more compact setup, low to no free play given properly designed system

  Cons: more complex, higher cost, more pieces to fail, uses large amount of resources if mistakes are made during design

4.3 Selection Criteria

The selection criteria used for our selection was to meet or better our minimum wheel travel of 50 mm, maintain the minimum shock absorption values as given by SAE as requirements. We also chose based on camber control, roll stability, dive and squat geometry, weight, and weather they would pass the required suspension tests. The minimum wheel travel can be determined by the travel distance on the spring type along with the damper stroke, ensuring that both have higher values than the necessary 50 mm of travel necessary for the requirement. Similarly the absorption can be determined by the specifications of the parts we plan to purchase. Each dampener will be designed to a specification that says the rated maximum shock that it can withstand, and each of the components we design, will be designed such that they will also be able to withstand the shock.

4.4 Concept Selection

Concept selection was based on engineering requirements stated in the aforementioned QFD. The requirement parameters include minimum wheel travel, shock absorption, camber control, roll stability, dive/squat geometry, lightweight construction, and passing inspection. Four design options, each of differing design components, were weighed in a decision matrix, along with two benchmarks for reference. Engineering requirements were weighted based on their relative technical importances regarding customer needs, and are laid out in the QFD. Once placed in the decision matrix, each option was rated 1-5 on their expected performance for each engineering requirement. Out of 100%, the total ratings for each design option were 81.6%, 50.22%, 73.0%, and 65.2%, for options 1-4 respectively.

WEIGHTED
Decision Matrix

CRITERIA	WEIGHT	Option1 RATING	Option1 TOTAL	Option 2 RATING	Option 2 TOTAL	Option 3 RATING	Option 3 TOTAL	Option 4 RATING	Option 4 TOTAL	Benchmark 1 RATING	Benchmark 1 TOTAL	Benchmark 2 RATING	Benchmark 2 TOTAL
Minimum Wheel Travel	10.4%	5	10.40%	4	8.32%	3	6.24%	5	10.40%	4	10.40%	5	10.40%
Shock Absorption	20.0%	4	16.00%	2	8.00%	4	16.00%	3	12.00%	4	20.00%	5	20.00%
Camber Control	17%	4	13.20%	1	3.30%	4	13.20%	4	13.20%	3	12.38%	5	16.50%
Roll Stability	23%	4	18.40%	2	9.20%	4	18.40%	3	13.80%	5	28.75%	5	23.00%
Dive/Squat Geometry	8%	4	6.40%	4	6.40%	2	3.20%	2	3.20%	4	8.00%	5	8.00%
Leightweight Construction	6%	3	3.60%	4	4.80%	2	2.40%	2	2.40%	3	4.50%	5	6.00%
Pass Suspension Tests	17%	4	13.60%	3	10.20%	4	13.60%	3	10.20%	3	12.75%	5	17.00%

max	TOTAL Option1	TOTAL Option 2	TOTAL Option 3	TOTAL Option 4	TOTAL Benchmark 1	TOTAL Benchmark 2
101%	81.60%	50.22%	73.04%	65.20%	96.78%	100.90%

	Component Variations					
Option 1	Coil Springs	Double Wishbone	Pushrod	Adjustable Rod Ends	Linear Damper	Ball Joint
Option 2	Leaf Spring	Double Wishbone	Pullrod	Adjustable Rod Ends	Linear Damper	Rubber Bushings
Option 4	Air Suspension	Solid Axle	Pushrod/Bell Crank	Adjustable Rod Ends	Adaptive System	Ball Joint
Option5	Air Suspension	Multi-Link	4-Link	Sliding Joints	Progressive Spring	Flexure Joint
Benchmark 1	Coil Springs	Double Wishbone	Pullrod	Adjustable Rod Ends	Linear Damper	Ball Joints
Benchmark 2	Coil Springs	Double Wishbone	Pushrod	Adjustable Rod Ends	Linear Damper	Ball Joints

Figure 9: Decision Matrix

Design option 1 features coil springs, a double wishbone layout, pushrods, adjustable rod ends, linear dampers, and ball joints, and had the highest overall score of 81.60%. For this reason, option 1 was chosen.

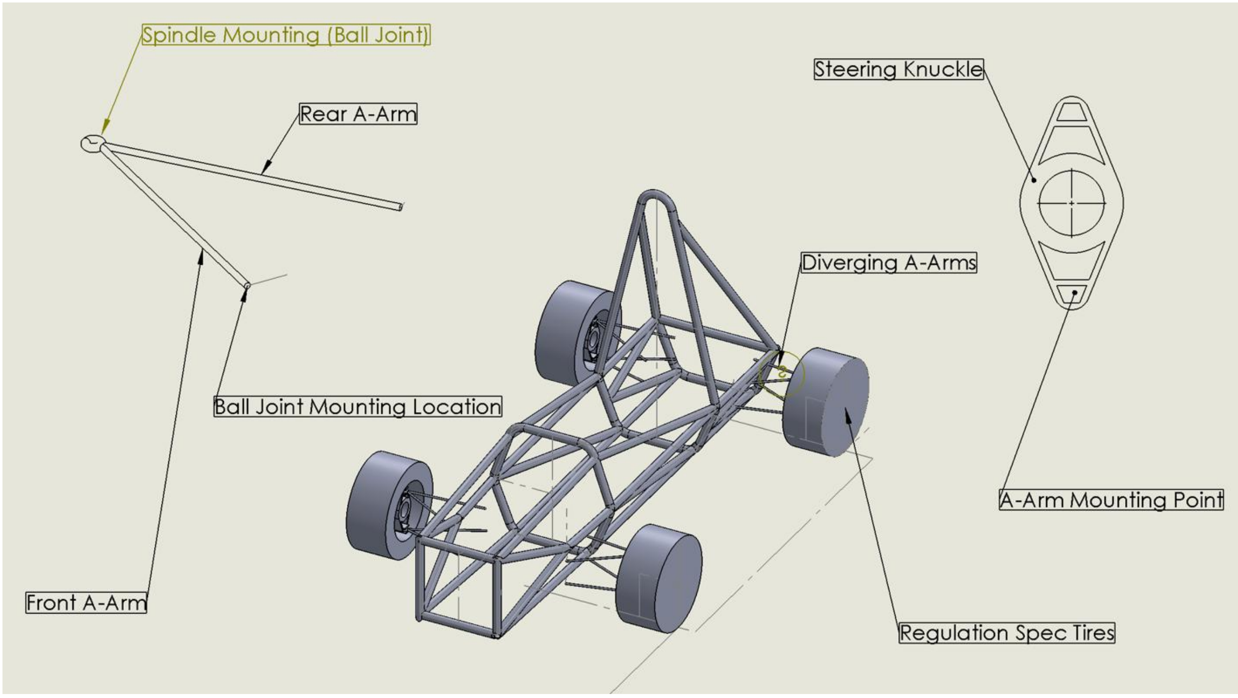

Figure 10: Current CAD Model

Figure 10 displays the current CAD model, along with some important components labeled, Further additions will be made to meet the design requirements.

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6 APPENDICES

6.1 Appendix A: Mathematical Modeling

```
% Fixed Parameters
tire_width = 254;          % mm estimate from 10 in hoosier racing tires
steering_axis_x = 50;      % mm from vehicle centerline rough estimate- will get soon

% Parameter Ranges (estimates from other fsae cars)
offset_range      = -20:30;     % mm wheel offset
kingpin_range     = 5:15;        % degrees KPI
caster_range      = 0:5;        % degrees Caster angle
camber_range      = -3:2;        % degrees Camber angle
ride_height_range = 25:55;    % mm Ride height

max_combinations = length(offset_range) * length(kingpin_range) * ...
                   length(caster_range) * length(camber_range) * ...
                   length(ride_height_range);
 for offset = offset_range
     for kingpin = kingpin_range
         for caster = caster_range
             for camber = camber_range
                 for ride_height = ride_height_range
                     camber_shift = (tire_width / 2) * sind(camber);
                     tire_center_x = steering_axis_x + offset + camber_shift;
                     x_ground = steering_axis_x - ride_height * tand(kingpin);
                     scrub = tire_center_x - x_ground;

                     if abs(scrub) < 10
                         idx = idx + 1;
                         results(idx, :) = [offset, kingpin, caster, camber, ride_height, scrub];
                     end
                 end
             end
         end
     end
 end
```

Figure 11: MATLAB Code for Effects of Different Variables on Scrub Radius

6.2 Appendix B: MatLab code for Brake Design

```
%%%%%%%%%%%%%
%Model for Brake Selection
%
% Variables:
% Tire size
% Pedal ratio
% Vehicle mass
% weight distribution
% Driver force
% required torque
% desired deceleration rate
% deisred braking split(difference between front and rear force
clc; clear; close all;
```

```
W=250; %weight of vehicle (kg)
Decel=2; %deceleration rate, in (g)
N=1.5;     %Factor of Safety
Forcereq=W*Decel*N; %required braking force (N)

%%%%%%%%%
BrkSplt=.6; %brake split, (%)towards front
FrontForce=Forcereq*BrkSplt; %front required stopping force (N)
BackForce=Forcereq*(1-BrkSplt); %Back required stopping force (N)

%%%%%%%%input for variables based on tire and rotor diameters


%Caliper specs
Numpist=2;  %Number of pistons
PistDiam=25.4; %Diameter of brake piston (mm)
PistArea=pi*PistDiam^2/4; %Area of brake piston (mm^2)
RotorDiam=200; %Diameter of brake rotor (mm)
Padwidth=25.4;  %Width of brake pad (mm)

TireSLR=200;  %Static loaded radius (mm)
RotRadeff=(RotorDiam-Padwidth)/2;


FrontbForce=FrontForce*TireSLR/RotRadeff; %Braking force Front (N)
BackbForce=BackForce*TireSLR/RotRadeff; %Braking force Back (N)

%%%%%%%%%%%%%%%insert pad friction coefficient
Mupad=0.3; %Friction coefficient of the pad and rotor materials

FrontClmp=FrontbForce/(Mupad*Numpist);   % Front and rear clamping force on the
BackClmp=BackbForce/(Mupad*Numpist);     %calipers from each piston   (N)

BrkLnPrsFront=FrontClmp/PistArea  %Pressure in the front brake line N/mm^2  MPa
BrkLnPrsBack=BackClmp/PistArea   %Pressure in the Back brake line  MPa

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%insert driver force and pedal ratio

DrivForce=400;   %Force applied by the driver (N)
PedRatio=2;      %Pedal ratio
BrakeFrc=DrivForce*PedRatio;

FrontMSTCylDiam=sqrt(4*BrakeFrc/(pi*BrkLnPrsFront))/25.4 %Front Master cylinder
diameter (in)
BackMSTCylDiam=sqrt(4*BrakeFrc/(pi*BrkLnPrsBack))/25.4   %Back Master cylinder
diameter  (in)
```