Formula SAE: Suspension Sub-Team

Conceptual Design Report

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Fall 2025-Spring 2026

Project Sponsor: GORE, Flagstaff Chevrolet, ANSYS

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DISCLAIMER

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 attach 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 has 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 underway.

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.

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1 Background

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 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 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

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 subsystems will be built to provide stability to the car during all aspects of driving.

Durable – The suspension subsystem and all its components will be composed 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

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

#	Functional Requirement	Minimum Wheel Travel	Shock Absorption	Camber Control	Roll Stability	Dive/Squat Geometry	Lightweight Construction
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

#	Customer Needs	Customer Weights (1-5)	Functional Requirements: Minimum Wheel Travel	Shock Absorption	Camber Control	Roll Stability	Dive/Squat Geometry	Lightweight Construction	Pass Suspension Testing	Customer Opinion Survey: 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

	Minimum Wheel Travel	Shock Absorption	Camber Control	Roll Stability	Dive/Squat Geometry	Lightweight Construction	Pass Suspension Testing
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%

Fig. 1. House of Quality

3 Research Within Your Design Space

3.1 Benchmarking

For our first benchmark, we decided to use the 2023-2024 NAU FSAE vehicle's 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. [50]

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 a 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. [51]

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. [52]

3.2 Literature Review

3.2.1 Chloe 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.

[8] B. Jawad and J. Baumann, “Design of Formula SAE Suspension,” SAE Global Mobility Database, Lawrence Technical University, Dec. 2001. Accessed: Oct. 20, 2025. [Online]. Available: https://scispace.com/pdf/design-of-formula-sae-suspension-1e7knpcg44.pdf

This technical paper describes the analysis and manufacturing of many suspension parts including the wheel hub, control arms, dampers, and uprights. This is very useful to our project as we are doing much of the same analysis and will need to continue doing so. This will aid us in knowing what analysis tools to use and how to use them.

[9] I. Nìels Karlsson, “Design of a Suspension System for a Formula Student Race Car,” Reykjavìc University, 2018. Accessed: Nov. 04, 2025. [Online]. Available: https://skemman.is/bitstream/1946/31391/1/MSc_Ingi_Niels_Karlsson_2018.pdf

This thesis dives deeply into the design process of the suspension system for a third year student FSAE team in Iceland. Karlsson describes the calculations done to determine many of the suspension properties and later describes the analysis they did using Adam’s Car. Although we are not using Adam’s, much of this information can be transferred to OptimumK which we are using. This thesis is important to our project because it allows us to ensure we calculated everything correctly and makes sure that we haven’t missed anything important that we need to consider.

[10] B. Zhu and N. Sun, “Design and Optimization of FSAE Race Car Suspension,” Atlantis Press, Zhejiang Agricultural Business College, 2015.

This report explains how the team did their 3D modeling and details the simulations done on it. The authors also include their target parameters. This helps us ensure that our target parameters are close to theirs as we have the same goals. This report is important to our project because it shows what simulations we should run and what our targets should roughly look like.

3.2.2 Austin Hess

[11] “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

[12] 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.

[13] 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.

[14] “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

[15] 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.

[16] 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.

[17] 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.

[18] S. S. Kaisare, "Parametric Design and Optimization of an Upright of a Formula SAE car," Apr. 24, 2024. https://vtechworks.lib.vt.edu/server/api/core/bitstreams/c4a86dbf-94e8-4655-ae08-f582ee4cecf5/content

Student thesis that provides lengthy description of the steps used to analyze and iterate upright designs, showing what makes a viable design.

[19] J. Mesicek, M. Richtar, J. Petru, M. Pagac, and K. Kutiova, "Complex View to Racing Car Upright Design and Manufacturing," *Manufacturing Technology*, vol. 18, no. 3, pp. 449–456, Jun. 2018, doi: https://doi.org/10.21062/ujep/120.2018/a/1213-2489/mt/18/3/449.

Discusses optimization of vehicle upright design and methods of production.

[20] A. Kozlenok, "Formula SAE upright," *Aleksandr Kozlenok*. https://www.kozlenok.com/projects/formula-sae-upright

Website created by an FSAE chief engineer to showcase findings and methods used to optimize upright design based on strength and weight.

3.2.3 Maeve Jastrzebski

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

This document illustrates a multitude of techniques by which to test and optimise vehicle ride quality and handling, and outlines a thorough method to evaluate vehicle aerodynamics through step-by-step procedures. This document is beneficial to Formula Student as it contains directions for collecting and interpreting accurate data from actual vehicle test sessions, which will validate suspension geometry, damping and compliance during prototype testing.

[22] E. J. Goodman, Race Car Vehicle Dynamics and Design Applied to Formula Student, ch. 11, 15, 17. Master of Philosophy thesis, Aston University, May 2009.

Formula studnet thesis with specific suspension dynamics, calculations, and workflows.

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

Engineering book containing chassi load-path and mounting fundamentals for a strong suspension design.

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

Motorsport book with clear double-wishbone geometry for fast kinematic design and tuning.

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

Textbook illustrating vehicle dynamics theory for modeling and justification.

[26] 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.

Motorsport tuning book with practical handling and setup for race vehicles.

[27] 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.

Peer reviewed journal paper specific to FSAE anti-roll bar design and stiffness analysis.

[29] Northern Arizona University Formula SAE Team, *"Formula SAE Vehicle Overview and Design Notes (2023–2024),"* College of Engineering, Northern Arizona University. Available: https://sce.nau.edu/capstone/projects/ME/2024/F23toSp24_Formula24/Our-Car.html

Past NAU team design document used in our benchmarking.

[30] "Double Wishbone Suspension," *Motor-Car.net*, 2024. Available: https://motor-car.net/innovation/suspension/item/14590-double-wishbone-suspension

Online reference article with basic double-wishbone overview.

[31] MIT Motorsports, "MIT Motorsports," fsae.mit.edu. [Online]. Available: https://fsae.mit.edu/

Past MIT team website in which we used the 2025 teams design as benchmarking.

3.2.4 Tanner Coles

[32] 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.

[33] 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.

[34] 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.

[35] 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.

[36] 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.

[37] 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.

[38] 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.

[39] 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.

[40] H. G. Phakatkar, C. Potdar, V. Joijode, and S. Jadhav, "Design of suspension system of formula student car," *International Journal of Mechanical and Production Engineering*, vol. 4, no. 2, pp. 54–57, Feb. 2016.

This journal provides a justification for the use of the double wishbone layout, which is important for our suspension system configuration.

[41] A. Gaither, “Vehicle dynamics on an electric Formula SAE racecar,” S.B. thesis, Dept. of Mechanical Engineering, Massachusetts Institute of Technology, Cambridge, MA, USA, 2021.

This is an MIT thesis that focuses on the dynamics of an electric FSAE submission. This is very similar to the MIT thesis that was reviewed in our Benchmarking system.

3.2.5 Reuben Goettee

Suspension Geometry and Computation [42]
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 [43]

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 [44]

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 [45]

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 [46]

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 which 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 check 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[47]

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 [48]

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 Suspension Geometry- Chloé Meyer & Tanner Coles

To ensure that our suspension geometry meets our target values shown in the table below, we used Optimum Kinematics. We also used the software to send coordinates to the frame team and iterate where necessary to ensure proper triangulation. This process includes changing each mounting point one at a time until the target values are met. This is important to our project because if we do not meet these target values, the car will not behave as we wish it to and will therefore cause us to fail tests and/or perform badly in events.

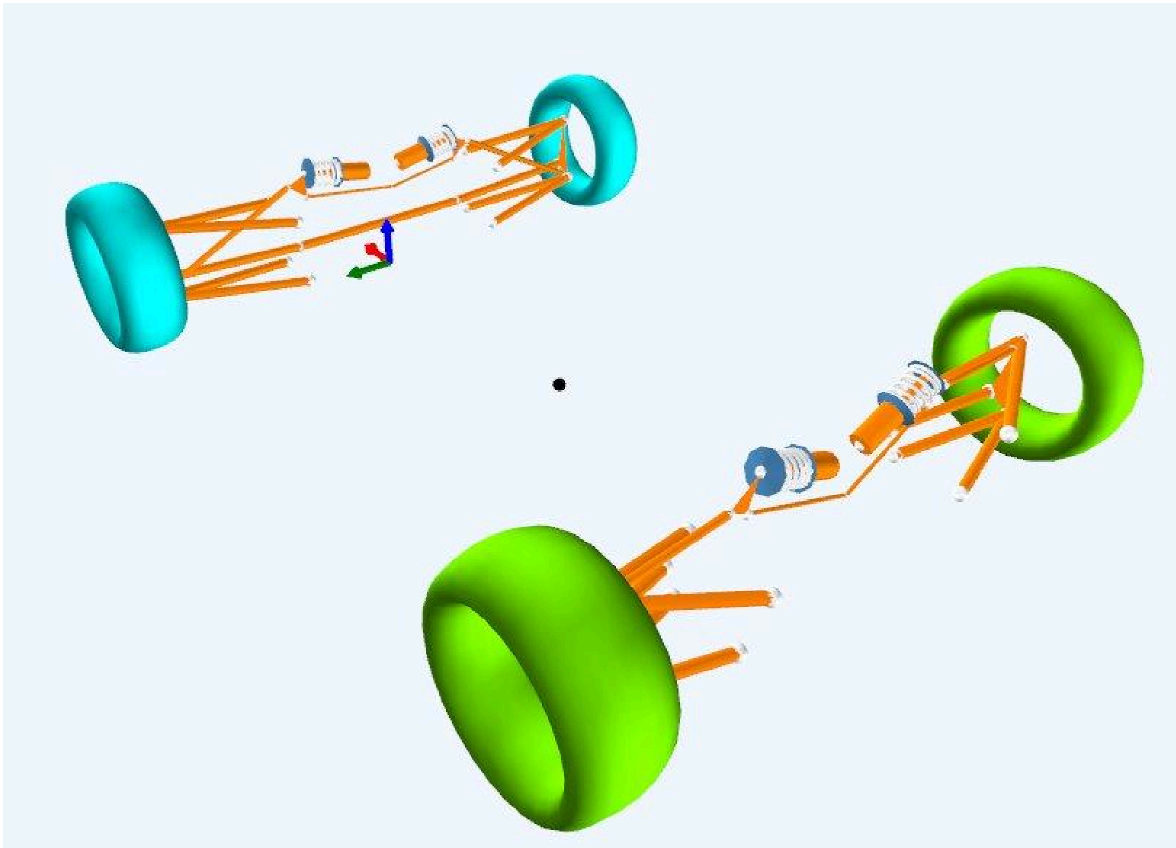


Fig. 2. Optimum Kinematics

3.3.2 Knuckle Sub-Assembly – Austin Hess

In order to design uprights that effectively withstand the shock of bumps throughout the racetrack and hold up the weight of the vehicle, in-depth force analysis is essential. FEA is used to show where stresses are concentrated in the upright, and where improvements can be made. Pictured below is the fourth and current iteration of the upright, applicable for both front and rear use. The current design makes improvements to fit the current wheels and wheel hub assembly, and is now assigned alumina alloy instead of AISI 1020 steel for the material, saving weight. This design is bolstered by numerous fillets with the goal of distributing stresses. Since the estimated total weight of the vehicle is approximately 250kg, and an estimated 44kg of that total is unsprung, split between four corners, a single upright should be capable of supporting a static 50kg load without deformation. To simulate the exaggerated loads that may be encountered during shocks, a total of 200kg is applied downward at the control arm connection locations. Even with the exaggerated loads, the upright at 1:1 deformation scale shows no strain, and is acceptable to support the vehicle.

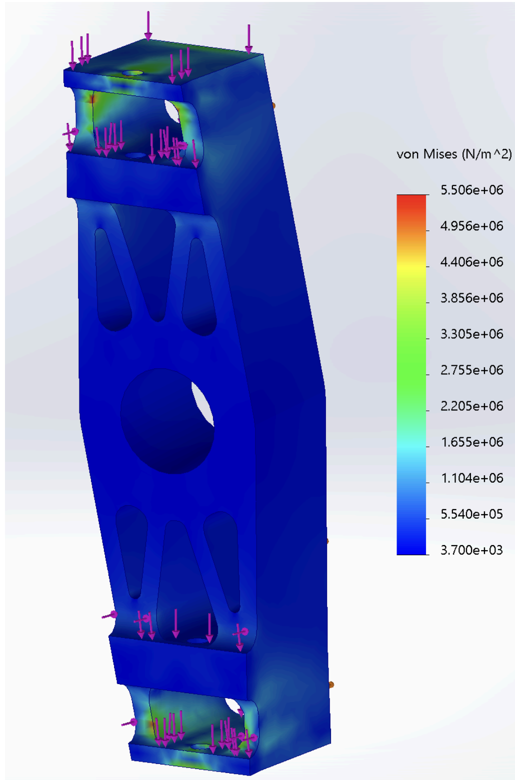


Fig. 3. Upright FEA

3.3.3 Anti-Roll Bar Design Considerations - Maeve Jastrzebski

I am still updating and working on our anti-roll design as we make further improvements with our overall suspension system. My modelling is a continuation of my modelling from the first presentation, as I realised that there were a lot of decisions needed in order to properly experiment with anti-roll.

The components calculated were:

Corner stiffness - Determines how much the car compresses vertically under load (factor of oversteer vs understeer)

Roll Stiffness - Controls body roll angle and affects load transfer distribution (factor of oversteer vs understeer)

Total Roll Stiffness From Springs - Baseline for roll resistance

Anti-Roll Bar Stiffness - Reduces body roll without stiffening the springs

Total Roll Stiffness - Determines how the car actually rolls with a cornering load (factor of target roll gradient)

Roll Moment @ 1g - Resists twisting torque using roll angles, and the assumption of 1g is needed due to design factors that have not been narrowed down enough

Corner Stiffness (N/m):

$$k_{corner} = k_s \bullet MR^2 = 89.94\ (front),\ 86.85\ (rear) \quad ()$$

Roll Stiffness (Nm/rad):

$$k_\phi = \frac{\Sigma_{kwheel}(T/2)^2 + k_{ARB}(r)^2}{h_{cg}(m)} = 88.14\ (front),\ 73.39\ (rear) \quad ()$$

Total Roll Stiffness From Springs(Nm/rad):

$$k_{\phi,springs} = 2(k_f MR^2{}_f)(\frac{t_f}{2})^2 + 2(k_r MR^2{}_r)(\frac{t_r}{2})^2 = 161.53 \quad ()$$

Anti-Roll Bar Stiffness (Nm/rad):

$$k_{\phi,ARB} = 2k_{ARB}(\frac{t}{2})^2 = 17632.56\ (front),\ 21550.91\ (rear),\ 39183.48\ (total) \quad ()$$

Total Roll Stiffness (Nm/rad):

$$k_{\phi,total} = 2(k_f k_{ARB,f})(\frac{t_f}{2})^2 + 2(k_r k_{ARB,r})(\frac{t_r}{2})^2 = 177720.71\ (front),\ 21624.3\ (rear),\ 39345.01\ (total)$$

Roll Moment @ 1g (Nm):

$$M_{roll} = ma_y(h_{cg} - h_{rc}) = 686.7 \quad ()$$

Roll Angle (rad):

$$\phi = \frac{M_{roll}}{k_{\phi,total}} = 4.25 \quad ()$$

Roll Gradient (deg):

$$G_\phi = \frac{\phi}{a_y}(\frac{180}{\Pi})(9.81) = 243.58 \quad ()$$

3.3.4 Review of Anti-Squat and Dive Percentages - 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 of 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:

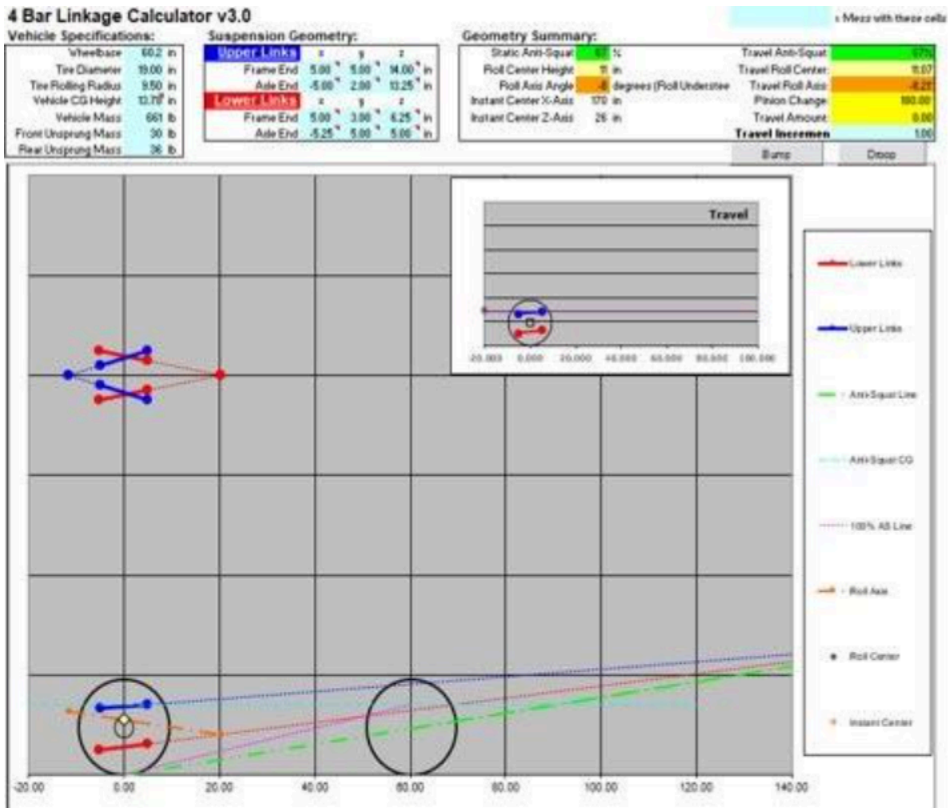


Fig. 4: Example Suspension Geometry and Following Anti-Squat Percentage

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

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

As an example, with a 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.

$$\left(\frac{IC_y}{IC_x}\right) = 0.95 * \left(\frac{0.28}{1.53}\right) * \left(\frac{1}{1*100}\right) = 0.1738$$

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 from 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.

3.3.5 Reuben Goettee

I have found that the Willwood PS-1 calipers will be able to fit within our design and combine a decent weight saving size while also remaining at very low cost. The rotors will be 7 inches in diameter, and we will be able to design them for manufacturing with the proper thickness and bolt pattern for our hubs. The master cylinders can be sized at 1/2 in for the front circuit and 5/8 in for the back circuit based on market availability and adding a factor of safety for the braking force. These will be able to lock the wheels with a driver input force of 430 N.

```
FrontMSTCylDiam =

    0.5490
```

```
BackMSTCylDiam =

    0.6723
```

Fig. 5. Brake Master cylinder sizes

```
DrivForceLockF =

  413.4214
```

```
DrivForceLockR =

  430.6473
```

Fig. 6. Pedal Force Applied by Driver to Lock Tires

For the steering, I have chosen an Ackerman percentage of 33% at a turning radius of 3m. This is from combining a steering angle of 22° on the outer tire and 33° for the inner tire. This translates to a 55 mm offset from the center of the tire and using the largest NARCO steering rack, the tie rods will measure 480 mm long. Each of these calculations has greatly aided in the selection process for parts to be purchased and for dimensions and choices for parts that will need to be designed and manufactured.

```
InAngle=atand(L./(R-T/2)) %inner required turning angle (deg)
OutAngle=atand(L./(R+T/2)) %outer required turning angle (deg)
```

Fig. 7. Equations for Steering Angle

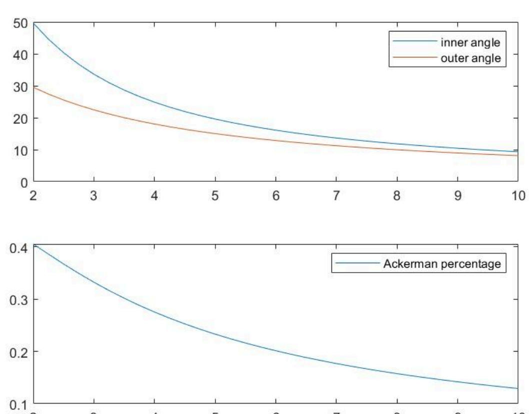


Fig. 8. Graphs of Steering Angle and Ackerman Percent Based on Turning Radius

3.3.6 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.

$$\Sigma F_{x,y,z} = 0 \quad (2)$$

$$WT_{lat} = \frac{m_{car\&driver} \times a_{lat} \times h_{COG}}{Trackwidth} \quad (3)$$

$$WT_{long} = \frac{m_{car\&driver} \times a_{long} \times h_{COG}}{Wheelbase} \quad (4)$$

Results:

Table 1: Forces in Control Arms & Pushrod

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

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 illustrates our team's 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

Fig. 9. Suspension Decomposition

Brake System

Driver Input

Bias Control Safety and Redundancy	Pedal Ratio	
	Heat Management	Hydraulic Lines Calipers, Pads, and Rotors

Fig. 10. Brake Decomposition

Steering System

Wheel/Tiller/Steering Wheel

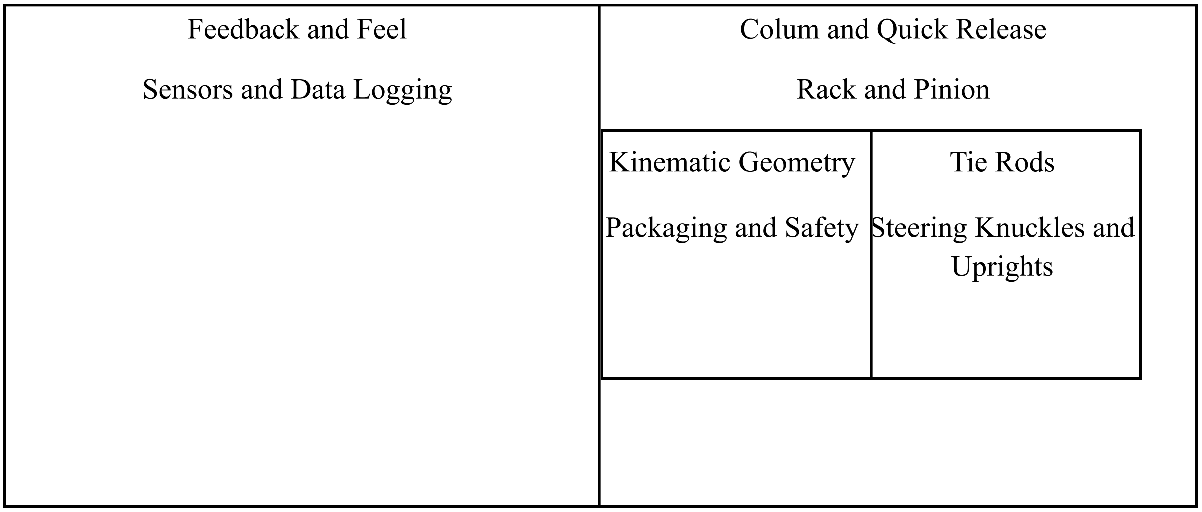


Fig. 11. Steering Decomposition

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 selected design includes use of a 50mm minimum suspension travel, as stated necessary by the Formula SAE rulebook. Further decisions are made based on camber control, roll stability, dive/squat geometry, weight, and their ability to pass technical inspections. Minimum wheel travel can be determined by ensuring spring travel distance and damper stroke exceed 50mm. Excess travel is mitigated by the inclusion of bump-stops and fixed pushrod lengths. Each damper purchased will be tunable to correspond to travel specifications, and will be required to withstand shocks greater than each corner of the vehicle can expect. Control arms similarly designed to withstand the weight of the vehicle plus additional impulse forces. Camber control is determined by ensuring designed and bought parts allow the vehicle to maintain the current specification of –2° camber. It is worth noting that the current design of all uprights maintain the selected camber by design with the help of control arm locations. Control arm design is dictated by each connection points' ability to maintain desired roll stability and dive/squat geometry, as well as front and rear track widths. With the help of suspension kinematics software, all designed arms meet target values by default. Use of lightweight parts are integral to maximizing performance. Analysis of individual sub-assemblies provides insight about the strength to weight ratio for each component, allowing for optimization of components like the steering upright, brake assembly, and bell-rockers.

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

Fig. 12. Weighted 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.

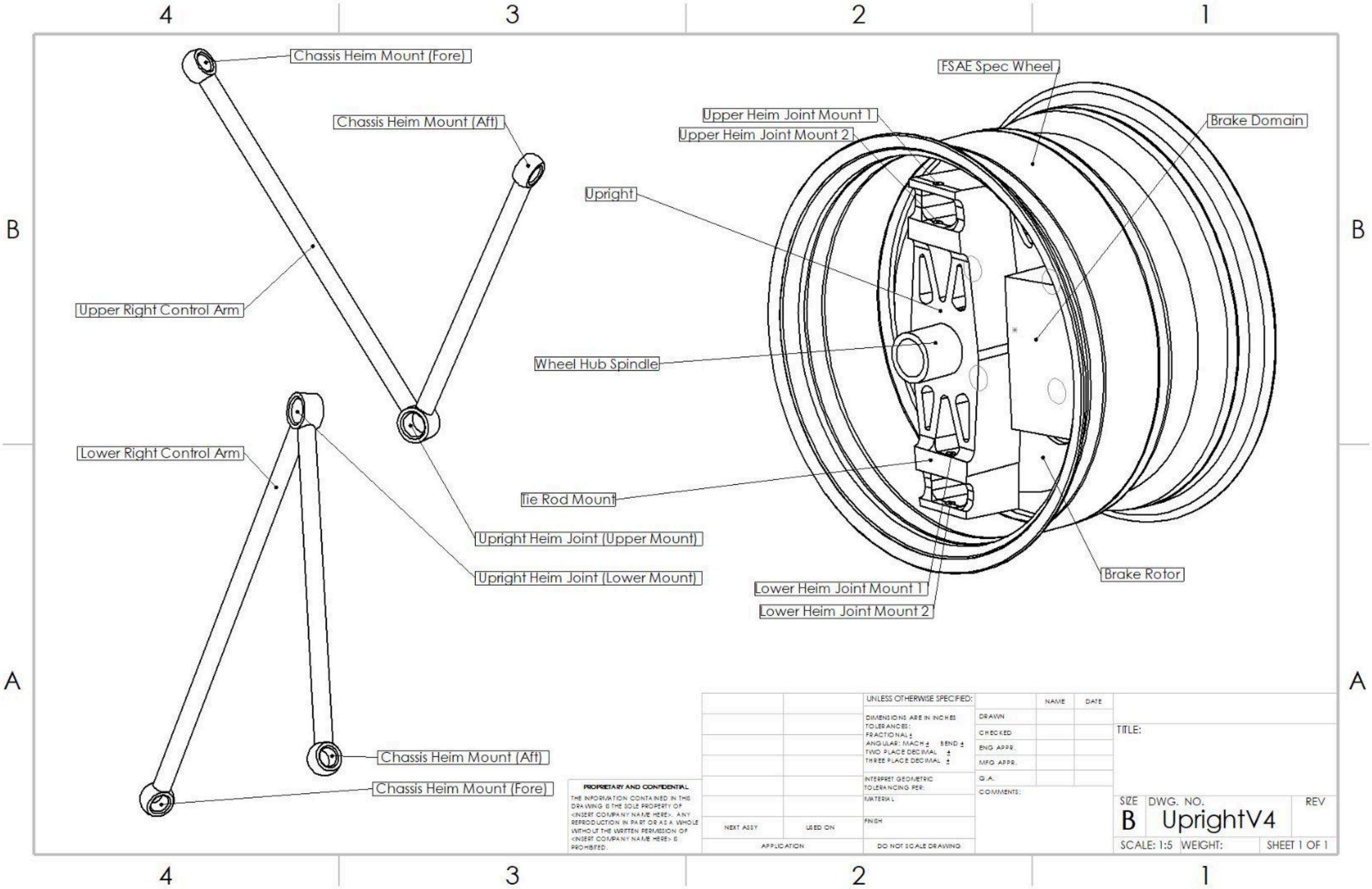


Fig. 13. Solidworks Drawing of Current Control Arm, Knuckle, and Rim Design

Depicted in figure 13 are the current designs of the front-right control arms, upright, braking area, and wheel. Current upright design allows for application to both front and rear pickup locations, although front and rear control arms differ in dimensions.

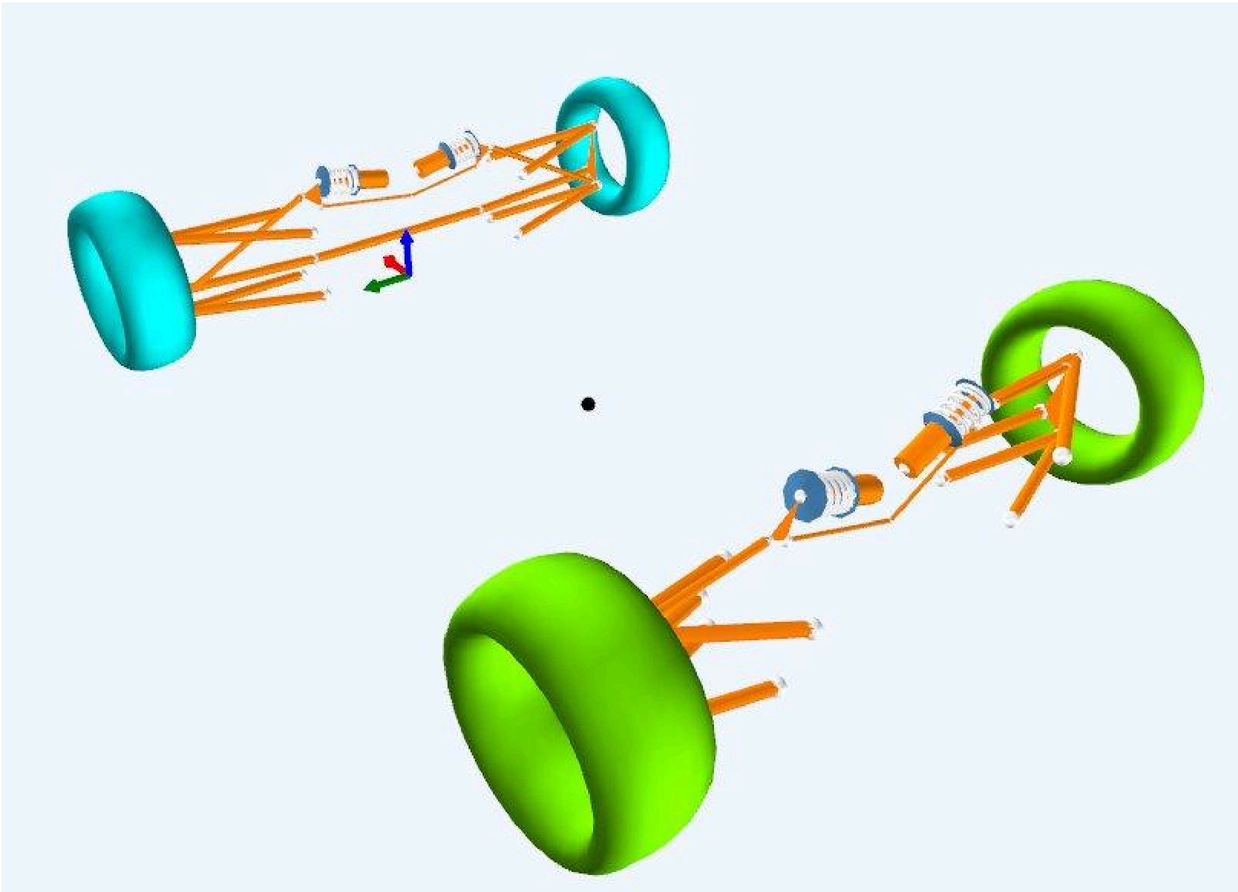


Fig. 14. Prototype of Concept Selection in Optimum Kinematics

5 Schedule and Budget

5.1 Schedule

This section presents the complete first-semester schedule, a draft of the second-semester schedule, and the work-breakdown-schedule with descriptions of each task as required by the report rubric.

5.1.1 Work Breakdown Structure

1) Project Management
2) Requirements and System Development
3) Suspension System Development
4) Steering System Development
5) Braking System Development
6) CAD, Analysis and Prototyping
7) Fabrication and Assembly
8) Testing (Static, Dynamic, Track)
9) Documentation and Deliverables
10) Vehicle Integration and Safety Sign-Off

5.1.2 Task Descriptions

Project Management:

Kickoff and Planning - Define roles, communication plan, semester deliverables, and make a draft of the schedule

Weekly Meetings or Coordination - Update schedule and ensure coordination across the subteam

Risk Management - Identify high-risk items like long-lead parts, weld quality, and geometry errors, as well as create a strategy to mitigate them.

Requirements and System Definition

Customer Requirements - Collected from performance expectations, rules, and sponsors

Engineering Requirements - Formalised suspension travel, steering free-play, and safety limits

Interface Definition - Decide chassis mounting points, wheel envelope, steering rack alignment, and suspension travel

Suspension System Development

Baseline Research - Study relevant FSAE dynamic texts, review 2025 rules, and benchmark prior NAU FSAE cars

Geometry Definition - Compute instant centers, roll centers, motion ratios, kingpin inclination, scrub radius, wheelbase, trackwidth

Concept Layout - Define pickup points and create the first skeleton in CAD

Anti-Roll Bar Design - ARB stiffness targets and linkage geometry

Damper and Spring Selection - Evaluate damper curves and spring rates

Suspension FEA - Analyze control arm stresses and upright bending

Manufacturing Drawings - Create CAD drawings

Prototype Fabrication - Machine, tack, cut prototype control arms, ARB components

Bench Testing - Apply static loads, measure deflections, perform bump-steer tests

Track Testing - ARB tuning, safety check, alignment setup

Steering System Development

Research and Geometry - Steering ratio, Ackerman percentage

CAD for Linkages - Tie-rod geometry, steering column

Prototype Steering Assembly - Mounts, brackets, extensions

Steering Integration - Use suspension motion to verify toe consistency and bump-steer

Braking System Development

Research and Benchmarking - Rotor siding, master cylinder sizing, pedal ratio

Pedal Assembly - Pedal geometry, cylinder mounts

Fabrication - Brackets, mount systems

Brake Testing - hydraulic pressure, endurance

CAD, Analysis and Prototyping

Subsystem CAD, full front suspension, FEA validation, tolerances, and packaging studies

Fabrication and Assembly

Assemble rod-end links, uprights, and control arms, and assemble on the chassis

Testing

Static Testing - stiffness, deflection, overall geometry validation

Dynamic Testing - ARB motion tests, bump-steer, damper dyno

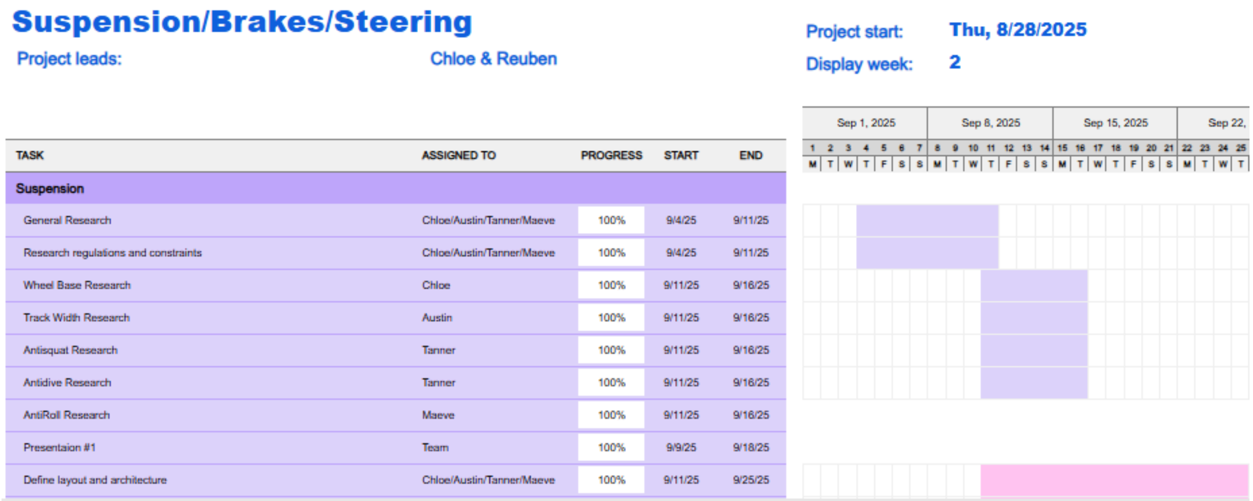
Track Testing - reliability, tuning, driver feedback

Documentation and Deliverables

Report 1, 2, presentations, and CAD

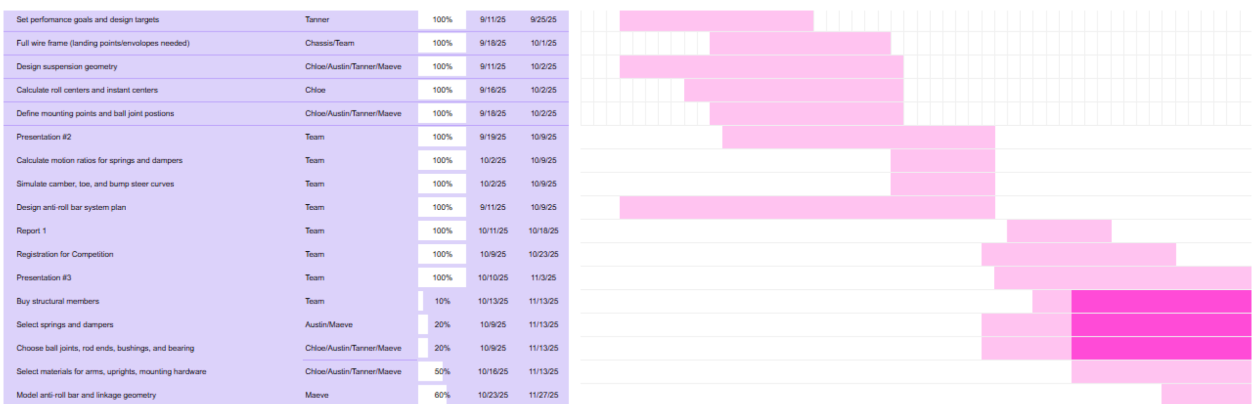
Vehicle Integration and Safety

Steering free-play, brake pedal travel, and final torque



TASK	ASSIGNED TO	PROGRESS	START	END
Suspension				
General Research	Chloe/Austin/Tanner/Maeve	100%	9/4/25	9/11/25
Research regulations and constraints	Chloe/Austin/Tanner/Maeve	100%	9/4/25	9/11/25
Wheel Base Research	Chloe	100%	9/11/25	9/16/25
Track Width Research	Austin	100%	9/11/25	9/16/25
Antisquat Research	Tanner	100%	9/11/25	9/16/25
Antidive Research	Tanner	100%	9/11/25	9/16/25
AntiRoll Research	Maeve	100%	9/11/25	9/16/25
Presentaion #1	Team	100%	9/9/25	9/18/25
Define layout and architecture	Chloe/Austin/Tanner/Maeve	100%	9/11/25	9/25/25

Fig. 15. Gantt Chart 9/4 to 9/25



Set perfomance goals and design targets	Tanner	100%	9/11/25	9/25/25
Full wire frame (landing points/envolopes needed)	Chassis/Team	100%	9/18/25	10/1/25
Design suspension geometry	Chloe/Austin/Tanner/Maeve	100%	9/11/25	10/2/25
Calculate roll centers and instant centers	Chloe	100%	9/16/25	10/2/25
Define mounting points and ball joint postions	Chloe/Austin/Tanner/Maeve	100%	9/18/25	10/2/25
Presentation #2	Team	100%	9/19/25	10/9/25
Calculate motion ratios for springs and dampers	Team	100%	10/2/25	10/9/25
Simulate camber, toe, and bump steer curves	Team	100%	10/2/25	10/9/25
Design anti-roll bar system plan	Team	100%	9/11/25	10/9/25
Report 1	Team	100%	10/11/25	10/18/25
Registration for Competition	Team	100%	10/9/25	10/23/25
Presentation #3	Team	100%	10/10/25	11/3/25
Buy structural members	Team	10%	10/13/25	11/13/25
Select springs and dampers	Austin/Maeve	20%	10/9/25	11/13/25
Choose ball joints, rod ends, bushings, and bearing	Chloe/Austin/Tanner/Maeve	20%	10/9/25	11/13/25
Select materials for arms, uprights, mounting hardware	Chloe/Austin/Tanner/Maeve	50%	10/16/25	11/13/25
Model anti-roll bar and linkage geometry	Maeve	60%	10/23/25	11/27/25

Fig. 16. Gantt Chart 9/11 to 10/16

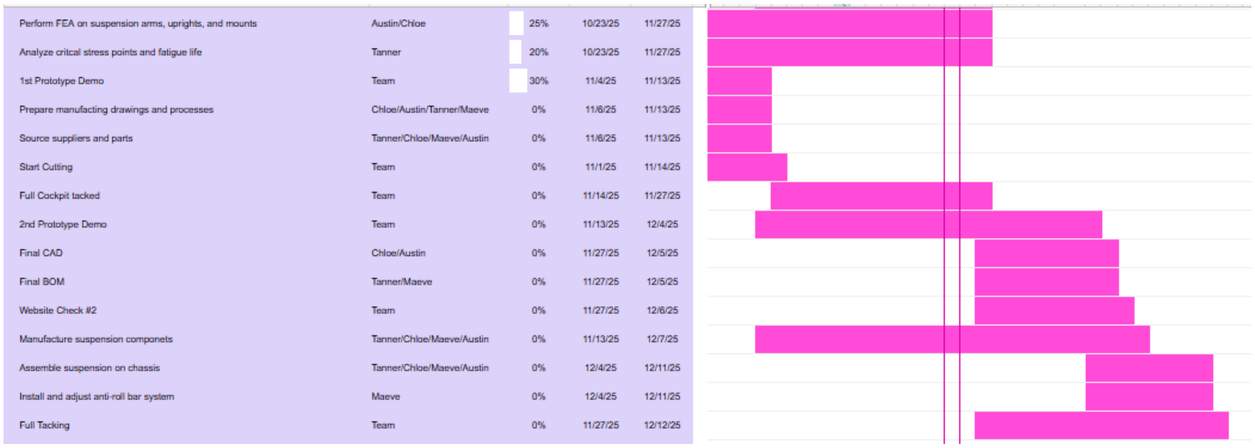


Fig. 17. Gantt Chart 10/23 to 12/12

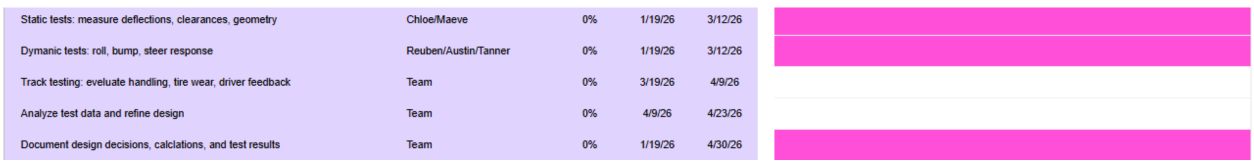


Fig. 18. Gantt Chart Draft of Next Semester

5.2 Budget

The current budget is combined with our Bill of Materials, where each item needed for the project has been researched for quality and has a unit and total price attached to it in our full BoM. The budget for each item within each subsystem will be elaborated on in Section 5.3 below. The following table is a simple budget breakdown for the entire suspension system, with overall total, and the total for each subsystem being shown. We have a fundraising goal of $11,000 for the suspension subteam specifically. This will be more than enough to cover the costs of the suspension, steering, and braking subassemblies. At the time of writing this report, nothing suspension-specific has been purchased, as we are still waiting on our contacts to respond to our inquiries, but parts will be purchased this weekend at the latest.

Table #2: Suspension System Budget

Subassambly	Cost
Coilover/Pushrod	$719.06
Control Arms	$548.56
Steering	$286.83
Braking	$558.00
Knuckle	$4,219.00
Estimated Extra Costs	$500.00
Total	$6,831.45

5.3 Bill of Materials

Our Bill of Materials consists of 5 tables to account for each subsystem. The most expensive components are in the knuckle assembly. The most notable are the tires and rims. Most of the components that we will manufacture do not include part numbers as we have not chosen a vendor for raw material. We are trying to find a steelyard that will give us a discount and this will have to be done very soon. We will reuse some of the parts from the 2024 car to minimize expenses and this is denoted by the word owned in the third column of the tables. Some of the parts say unsure in the third column as well. This is because it is unknown whether we will manufacture or purchase a part. This will require more research to determine the cost and weight of each option and what makes sense for our team. This will also be done very soon. The final cost is shown in the last table of this section, with a total cost of $6831. This is an estimate and will likely be higher due to uncertainty surrounding manufactured parts.

Table #3: Coilover/Pushrod & Control Arm BoM

Subsystem	Item	Purchase/Manufacture/Owned	Individual Cost ($)	# of Parts	Total Cost	Vendor	Vendor Part #	Manufacturer #	Lead Time
Coilover/Pushrod	Damper	Purchase	$ 249.00	2	$ 498.00	Jenson USA	RS001128	978-01-052	3 days
Coilover/Pushrod	Bearings	Purchase	$ 5.89	8	$ 47.12	Ebay	F609	NTN FL6008ZZ	2 weeks
Coilover/Pushrod	Springs	Purchase	$ 39.00	2	$ 78.00	Jenson USA	RS286CO1	039-03-007	3 days
Coilover/Pushrod	Aluminum Round Tube 3/4"	Manufacture	$ 8.00	6	$ 48.00	-	-	-	1 week
Coilover/Pushrod	Rod Ends	Purchase	$ 7.99	6	$ 47.94	Summit Racing	QA1-CFL3S	CFL3S	5 days
Coilover/Pushrod	Metal 3D Print Filament	Manufacture	-	2	-	-	-	-	-
Coilover/Pushrod	Rocker Spacer	Unsure	-	-	-	-	-	-	-
Control Arm	Fasteners	Owned/Purchase	-	-	$ 50.00	-	-	-	-
Control Arm	Rod Ends	Purchase	$ 3.59	24	$ 86.16	Summit Racing	FKB-CM5	CM5	5 days
Control Arm	Ball Joints	Purchase	$ 21.55	8	$ 172.40	McMaster-Carr	6960T25	6960T26	1 week
Control Arm	Steel Tubing 5/8"	Manufacture	$ 10.00	24	$ 240.00	-	-	-	1 week
Control Arm	Steel Flat Stock	Owned	$ -	16	$ -	-	-	-	-

Table #4: Steering & Brakes BoM

Steering	Steering Column Material	Manufacture	$20.00	1	$20.00				
Steering	Pillow Bracket Bearing	Purchase	$58.85	1	$58.85	Grainger	6UZ57	UCPK206-20	5 days
Steering	Steering Rack	Owned	$0.00	1	$0.00	-	-	-	-
Steering	Steering Rack Mounts	Purchase	$20.00	1	$20.00	-	-	-	-
Steering	Tie Rod Body Material	Manufacture	$30.00	1	$30.00	-	-	-	-
Steering	Jam Nuts	Owned	$0.00	4	$0.00				
Steering	Tie Rod Ends	Purchase	$53.99	2	$107.98	Autozone	GES80211	GES80211	3 days
Steering	Double U-Joint	Owned	$0.00	1	$0.00	-	-	-	-
Steering	Steering wheel quick connect	Purchase	$30.00	1	$30.00	Speedmaster	840136556367	PCE514.1002	5 days
Steering	Steering wheel	Unsure	$20.00	1	$20.00	-	-	-	-
Brakes	Brake Pads	Purchase	$60.00	4	$240.00	-	-	-	-
Brakes	Master Cylinder Kit	Purchase	$130.00	2	$260.00	Tilton Racing	75-812U	75-812U	2 weeks
Brakes	Fluid Reservoir	Purchase	$0.00	2	$0.00	Tilton Racing	75-812U	75-812U	3 weeks
Brakes	Fluid	Purchase	$5.00	2	$10.00	Walmart	N/A	N/A	2 days
Brakes	Brake Lines	Owned	-	-	-	-	-	-	-
Brakes	Rotor Stock	Manufacture	-	-	-	-	-	-	-
Brakes	Calipers	Owned	-	-	-	-	-	-	-
Brakes	Caliper Inlets	Purchase	$12.00	4	$48.00	Gorsuch Performa	HB-03110	220-9924	2 weeks

Table #5: Knuckle BoM

Knuckle	Knuckle Stock Aluminum	Manufacture	-	-	-	-	-	-	-
Knuckle	Caliper Mount Stock Aluminum	Manufacture	-	-	-	-	-	-	-
Knuckle	Pushrod Stock Aluminum	Manufacture	-	-	-	-	-	-	-
Knuckle	Wheels	Purchase	$200.00	8	$1,600.00	Keizer	-	-	-
Knuckle	Dry Tires	Purchase	$286.00	4	$1,144.00	Hoosier	-	H43100R20	Unknown
Knuckle	Wet Tires	Purchase	$300.00	4	$1,200.00	-	-	-	-
Knuckle	Wheel Studs	Purchase	$15.00	5	$75.00	Jegs	-	-	-
Knuckle	Hub Stock	Manufacture	$200.00	1	$200.00	-	-	-	-
Knuckle	Bearings	Purchase	-	-	-	-	-	-	-

Table #6: Total Cost

Subassambly	Cost
Coilover/Pushrod	$719.06
Control Arms	$548.56
Steering	$286.83
Braking	$558.00
Knuckle	$4,219.00
Estimated Extra Costs	$500.00
Total	$6,831.45

6 Design Validation and Initial Prototyping

6.1 Failure Modes and Effects Analysis

For our FMEA we looked at our major components and considered anything and everything that could go wrong and what the severity, and capability to mitigate that risk would be. We then put that into a chart for better understanding and correlation, and pieced out the highest four components and failure methods that these could face as shown below in table 7.

Table #7: FMEA of Highest Risk Components and Failure Methods

Part # and Functions	Potential Failure Mode	Potential Effect(s) of Failure	Severity (	Potential Causes and Mechanisms of Failure	Occurance	Current Design Controls Test	Detection (	RPN	Recommended Action
Tie Rod	Loosening or bending	Toe misalignment	7	Vibration, poor locking	3	Alignment Check	4	84	Use locktite &/or locknuts
Knuckle connection	Fatigue failure	misalignment to loss of steering	7	Cyclic loading anytime steering is required	3	Stress test under controlled conditions	4	84	Properly size connecting bracket with large factor of safety in mind
Upright/Knuckle	Fatigue failure	Steering instability, wheel misalignment, suspension detachment, can cause ball joints to fail	9	Improper machining/design, poor material selection, impact loads	3	FEA simulation, visual inspection	3	81	Run proper simulation, increase thickness
Wheel bearings	Overheat/siezing	Loss of control	7	Lack of lubricant, dry running	3	inspection and regular checks during operation	5	105	can withsatnd our desired use-case

To mitigate as many of these and to minimize the risk of any of these methods occurring, we are completing extensive simulations on expected stresses and temperature profiles to help select the correct components that will sufficiently avoid the above failures. Most of our worries stem from fatigue failures in components that will be heavily dealing with cyclic or repeated loading and thus we believe these components are the most at risk. For our risk trade-off analysis, we are considering different materials with different properties that could be better at the cyclic loading but may not be as lightweight or as cost efficient. Another is the plan to have an advantageous dampener geometry to distribute the load over a larger area, causing a lower concentrated stress in the concerned parts.

6.2 Initial Prototyping

6.2.1 Virtual Prototyping

The following questions were important to consider when prototyping:

1. How do we determine where landing points meet chassis and upright?
2. How do we design an upright that aligns with our landing points?
3. How do we determine the correct spacing for bolts and ball joints?

Following each question above are what we could determine:

1. Using Optimum Kinematics software determines landing points by iterating to meet target values
2. Upright is designed in CAD assembly referencing landing point and tie rod locations
3. Correct spacing achieved through tolerancing and measuring physical ball joints and bolts

Finally, our team came to the following conclusions moving forward:

1. Need to make vast efficiency improvements to manufacturing process, can be done through the use of global equations and cad templates
2. More comprehensive CAD model including all parts modeled so far
3. Possible purchase of Optimum G's Optimization module for more accurate and efficient iteration

6.2.2 Physical Prototyping

The following questions were important to consider when prototyping:

1. Will our geometry fit properly and articulate without interference?
2. Are our selected joints, bearings, and fasteners appropriately sized and positioned for easy assembly?
3. How do we translate mathematical models for steering angles to a physical prototype?

Following each question above are what we could determine:

1. Our geometry initially did not fit properly. Modifications to physical structures like control arms were necessary for proper fitment
2. Ball joints and bearings are properly sized, although not easily assembled
3. Angle measurements and movement of physical parts to allow for proper angles to be met

Finally, our team came to the following conclusions moving forward:

•We will be looking at our designs and modify the necessary components like the control arms/rims/brakes in order to assure proper fit.
•Control arm prototypes will require webbing to be in line with rods, instead of sandwiching them, to provide a more accurate physical model.
•Use of better materials all around may be beneficial, eventually including materials suitable for the final race car

6.3 Other Engineering Calculations

6.3.1 Column Buckling for Suspension Members

MATLAB was used to calculate the critical axial load that can be applied to the suspension members. Once more information about the vehicle, namely mass placement, is gathered, we can refine the forces that are being applied to our members, and design the diameter and wall thickness of the members from there. For now, we are going with assumptions about our vehicle's mass. Below is a snippet of the code used in the buckling analysis, which simply calculates the critical load for each member. This base code can be expanded upon to account for eccentricity in member loading and a lack of symmetry in the overall body of the vehicle, causing forces to be different from left to right.

```
%% Critical Load Based on Slenderness Ratio

criticalLoad = zeros(1,4);

for i = 1:length(slendernessRatios)
    if slendernessRatios(i) > criticalSlendernessRatio
        criticalLoad(i) = pi^2*E*I/(lengths_all(i))^2/1e3; % automatically calculates critical load in kN
    else
        criticalLoad(i) = A*S_y*(1 - (S_y*slendernessRatios(i)^2)/(4*pi^2*E))/1e3; % calculates critical load in kN
    end
end
min(criticalLoad)
```

Fig. 19. Critical Load Calculations in MATLAB

Overall, learning more about the forces going through our suspension members and where we might see failure will inform our design of the member rods.

6.4 Future Testing Potential

As development continues with the suspension system, the focus for future iterations will be on verifying the engineering requirements established for this project:

Achieving a minimum of 50mm of usable suspension travel;
Optimally optimising the damping and camber control of the suspension system; and
Obtaining a maximum of 7 degrees of free play in the steering system.

Structured testing procedures will be identified with the progression of the prototype to optimize performance, ensure the safety of the suspension system and assist with supporting design decisions of the suspension system.

6.4.1 Simulation Testing

Before the physical testing of the suspension system can occur, simulation testing of the suspension system should take place prior to the physical evaluation of the prototype. Utilising available resources such as OptimumK, MATLAB, and Excel will further allow us to simulate the kinematics of the suspension system, the spec sheet of the shocks and struts, and the steering characteristics of the suspension system.

Future simulation efforts should include:

Kinematic sweeps are used to evaluate the camber gain, toe change, steering ratio, and the bump characteristics of the suspension system throughout the suspension system's full range of motion (i.e. 50mm of travel).

Shock absorber modelling will assist with modelling your compression/rebound curves prior to performing shock dyno testing.

6.4.2 Component Level Testing

Each aspect must be tested to ensure that the component is compliant with the design assumptions; tests that need to be performed in the near future include:

Damper dynamometer testing will be used to obtain the force and velocity curves of the damper so that simulated data input will match the performance specifications of the damper.

Spring rate tests will ensure that the springs will compress linearly and provide a wheel rate as designed.

Measurements of suspension assembly travel will verify that the assembly can meet the 50 mm usable travel requirement without obstruction or binding issues.

The component data collected may also be used as a basis for future static tests and dynamic testing.

6.4.3 Static System Testing

Static tests will allow for direct evaluation of how the suspension operates under controlled non-dynamic conditions, and include:

Measurements of the camber through the wheel travel as determined by the wheel travel jig using linear potentiometers to validate the alignment of the camber through the bump and droop cycles.

An assessment of ride height to determine that the weights at the corners and static geometry are consistent with design expectations.

An evaluation of the amount of free play present in the steering as defined by the requirement for 7 degrees.

An evaluation of the turning radius provides an early indication of steering geometry and potential package constraints.

Static tests will serve to establish a baseline measurement for the subsequent evaluation of system response to dynamic loading.

6.4.4 Dynamic Vehicle Testing

The performance of the suspension will be dynamically tested after it is installed on a vehicle by simulating realistic vehicle usage. Future dynamic vehicle testing may include:

Testing of a vehicle's lateral grip at steady-state and camber characteristics while the vehicle is being loaded through the running of skid pads.

Testing of the vehicle's transient response due to steering input, sensitivity of steering input, and stability of both chassis and body using a step steering test.

Use of instrumentation, such as accelerometers, a gyroscope, wheel position sensors, and steering angle sensors, to capture more precise data regarding the dynamics of the vehicle's suspension.

Results from dynamic testing may provide insight into how to better tune the damping, alignment, and compliance of a vehicle's suspension.

6.4.5 Durability and Endurance Testing

Endurance testing will evaluate whether or not the suspension system can endure repeated loading without degrading over time. Endurance testing will include:

Cycle testing to determine potential sources of fatigue in the control arms, joints, fasteners, and dampers by subjecting them to repeated bumps or cornering forces.

Endurance driving is completed at a test track to measure the long-term durability of the entire system, including the development of heat and real-life impacts during extended runs. The results of durability testing will be used to make material choices, welding design, tightening specifications for hardware, and maintenance intervals.

7 Conclusion

The NAU suspension subteam is in the process of producing a suspension design that meets the constraints set by Professor Willy as well as SAE. We have established customer and engineering requirements, benchmarked against other formula cars, and compiled our mathematical modeling. Overall the project is on track and we will have a working suspension system before the competition. We are on track with fundraising and have done a lot of research to aid us in our suspension design. Our prototyping efforts will help us to have a competition ready car with enough time to test and make any improvements.

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9 Appendix

9.1 Appendix A: MATLAB 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
```

Fig. #. MATLAB Code for Effects of Different Variables on Scrub Radius

9.2 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)
```

9.3 Appendix C: MATLAB code for Steering Design

```
%%%%%%%%%%%%%%%%%%%%%
% Model for Steering ranges
%
% Used to find optimal combination of ackerman percentage,
% turning angle, and predicted minimum turning radius
%
% Assumes minimal impact of slip angle
%
% Inputs:Wheelbase, Track width, and the range of wanted turning radius
%
% Outputs:Inner and outer angles needed for turning radius,
% ackerman percentage
clc; clear; close all;
%% Input Variables
R=[2:0.25:10];  %Turning Radius
L=1530/1000; %Wheelbase
T=1400/1000; %Track width
%% Calculate
InAngle=atand(L./(R-T/2)) %inner required turning angle (deg)
OutAngle=atand(L./(R+T/2)) %outer required turning angle (deg)
figure(1)
subplot(2,1,1)
plot(R,InAngle)
hold on
plot(R,OutAngle)
legend('inner angle','outer angle')
Ackermann=(InAngle-OutAngle)./InAngle;  %Ackerman percentage
subplot(2,1,2)
plot(R,Ackermann)
legend('Ackerman percentage')
```

9.4 Appendix D: Prototype Bill of Materials

Table #8: Prototype 1 BoM

Prototype 1			
Item	Cost	# of Parts	Description
Ball Joint	$0.00	8	From 2024 Car
Wooden Dowels	$0.00	-	In office
MDF	$0.00	-	In shop
PVC	$4.81	1	10' stick
Pushrod	$0.00	1	From '24 car
Steering Rack	$0.00	1	From '24 car
Cardboard	$0.00	-	In shop
Plexiglass	$0.00	-	In shop
Woodscrews	$2.00	15	Home Depot 5/8
Brake Caliper	$0.00	1	From '24 car
Total Cost	6.81		

Table #9 Prototype 2 BoM

Prototype 2			
Item	Cost	# of Parts	Description
Ball Joints	$175.00	8	In full BoM
Rims	$1,600.00	8	In full BoM
Knuckle (3D Print)	$0.00	1	3D printing and filament donated
Control Arms (3D Print/PVC)	$0.00	2	3D printing and filament donated
Steering Rack	$600.00	1	In full BoM
Tie Rods	$0.00	1	From '24 car
Brake Calipers	$0.00	1	From '24 car
MDF	$0.00	-	In shop
Total Cost	$2,375.00		