Initial Design Report for SAE BAJA Fall 2024-Spring 2025

Steering, Brakes, and Suspension

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Disclaimer

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Executive Summary

This report outlines the progress of the suspension, brakes, and steering sub-team of the NAU Baja SAE team, documenting work completed from August 26th to October 20th, 2024. The team's primary objective is to design, test, and construct an off-road vehicle that meets stringent competition requirements and excels in various dynamic events at the SAE Baja competition in Marana, Arizona, scheduled for May 1st to 4th, 2025. The competition includes events like Hill Climb, Endurance, Acceleration, and Suspension & Traction, which test the vehicle's performance, durability, and driver control. To achieve a competitive edge, the team has set ambitious design goals, such as achieving over 12 inches of ground clearance, a tight 7-foot turning radius for improved maneuverability, and 10 inches of suspension travel for enhanced handling over rough terrains. The design focuses on implementing a zero-scrub radius suspension system for balanced steering, optimizing braking efficiency for shorter stopping distances, and refining steering geometry for smooth control. These elements are essential for ensuring the vehicle's stability and agility throughout the competition.

The sub-team has made considerable progress, including completing initial CAD models, selecting appropriate materials, and beginning validation through finite element analysis (FEA) using SolidWorks. Preliminary simulations and tests have demonstrated that the current designs meet early performance benchmarks, showing promise for achieving the desired results. In the next phase, the team plans to focus on further refining these designs, building and testing prototypes, and conducting extensive physical tests to ensure durability, reliability, and adherence to SAE's rigorous safety regulations. These efforts aim to optimize performance while minimizing weight and maintaining structural integrity, balancing competition demands with safety considerations. The estimated cost for developing and testing the suspension, brakes, and steering systems fits within the overall project budget of $15,000, covering expenses like material acquisition, fabrication, and testing. Through the application of advanced engineering principles and close collaboration with other sub-teams, we strive to create a high-performing vehicle that not only meets competition standards but also positions NAU's Baja program for continued success. The goal is to place in the top 25% of competing teams, attract potential sponsors, and contribute to the growth of the NAU Baja SAE legacy.

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1 Background

This chapter will discuss the research and decision-making process involved in the project. It will outline the project requirements from both the customer's and the engineering perspectives, organized using a QFD diagram to highlight their relationships and significance. Additionally, it will explore various benchmarking components, analyzing why certain designs succeeded while others did not. The chapter will also address the sources and information utilized in the project, emphasizing their relevance and how they impact the team's calculations. Finally, it will cover design concepts, iterations, and the key criteria that influence design changes.

Project Description

For the 2025 SAE Baja NAU capstone project, the goal is to design, build, and compete in the event, which will take place in Arizona in 2025. As part of this capstone, the team is responsible for securing sponsorships and managing the project's finances. The design requirements are outlined in the SAE Baja rulebook for the 2025 competition, and the team must pass a technical inspection to ensure the vehicle complies with safety standards and competition regulations. This project provides the team, as senior mechanical engineering students, an opportunity to design a vehicle within specific constraints and objectives.

Deliverables

There are several major deliverables that must be met for suspension, steering, and brakes. These include locking up all four tires, handling different suspension and steering tests, and passing all SAE inspections. These inspections ensure safe operation of the vehicle. For suspension, steering, and brakes, it is required to have documentation of all specifications and calculations. Physical prototypes must be constructed, and multiple tests are required to ensure proper operation of these systems. All three systems must also follow the guidelines set in the SAE rulebook.

Success Metrics

The success of each system is crucial to the performance of the team. Though the end goal is the same for all systems, the individual metrics for success are different. For suspension, the components must cycle freely through full range of travel without interfering or binding with other components. The front will need to pull 12 inches of travel, the rear will need to pull 14 inches of travel, there will be a ratio of more up travel than down to maintain stability at speed. For steering, the vehicle must have a turning radius smaller than 8 feet, components must maintain structural integrity through the entire competition, and the steering system must operate with minimal steering slop. The brakes must lock up all four wheels to pass specs. The brake pedal needs to withstand a minimum force of 450 pounds. Two hydraulic reservoirs are required for the brake system to ensure that the driver can still brake in the event if something does happen. By addressing each system's unique requirements, we will ensure the vehicle's overall performance and reliability, positioning our team for success in the competition.

2 Requirements

2.1

This chapter outlines the customer and engineering requirements for the NAU Baja SAE team's braking, steering, and suspension systems, along with a House of Quality (HoQ) for the braking, steering, and suspension systems. It details essential customer requirements focused on safety, performance, and user satisfaction, such as ensuring adherence to safety standards, cost-effectiveness, and effective functionality. The engineering requirements establish quantifiable goals to enhance system performance and safety, including maximizing braking force, reducing turning radius, and ensuring suspension durability. The HoQ illustrates the relationships between customer and engineering requirements, using benchmarks from other Baja vehicles to assess performance and identify areas for improvement, ensuring that the vehicle design aligns with both user needs and engineering capabilities.

Customer Requirements (CRs)

2.1.1 Brakes

To meet the customer requirements for the brakes, we need to focus on a few key areas in our product development. Safety is important; we must make sure all materials and parts follow safety standards and that we test everything thoroughly to avoid any hazards. Affordability is also a big deal, so we should find cost-effective materials and manufacturing processes while keeping value engineering in mind to ensure we don't lose functionality. To avoid hydraulic issues, we need to thoroughly inspect all the break lines and use correct fittings to ensure no leaks occur. We also need to design the brakes in such a way that they don't overheat due to excessive use. Meeting all these requirements is essential for ensuring that the vehicle passes the SAE brake inspection.

2.1.2 Steering

The product must deliver high performance while ensuring affordability to meet the budget constraints of the target budget. It should offer a high level of comfort for the driver, addressing ergonomic factors to enhance the overall experience. Additionally, the design must facilitate easy operation, minimizing complexity in handling and use. The product must pass SAE inspection, ensuring compliance with all applicable safety and performance regulations. These customer requirements balance performance, cost-efficiency, user satisfaction, and regulatory compliance.

2.1.3 Suspension

To satisfy the customer requirements for suspension, there are a few key areas that are essential to creating a successful Baja vehicle for the customer. The car performing well and passing the SAE tech inspection at competition is very important as those are the baseline metrics that will aid in determining if the car will do well in competition. Serviceability and tunability also plays into the overall performance umbrella of the customer requirements; should something go wrong during competition; the crew needs to be able to fix or tune the car back to how it should be in an efficient and quick way. Comfort, aesthetics, and durability also play a role in the overall safety and appeal of the car. Durability will translate in the car holding up to whatever we may throw at it during the race. Having a comfortable vehicle and driving experience can also bridge the gap between performance and safety. Ease of fabrication and affordability will also play a large role in the customer requirements, as both of these metrics are determined in the development and design stages of creating the car.

Engineering Requirements (ERs)

2.1.4 Brakes

The engineering requirements for the braking system highlight several important goals to improve performance and safety. First, we need to maximize the braking force to ensure the vehicle can stop effectively in various situations. The brake pedal must also be made from strong materials, specifically aluminum or steel, to ensure it is reliable and durable. Safety is a top priority, so the design should include features that protect both the driver and passengers. Additionally, it's important to minimize the amount of force needed to press the brake pedal, making it easier for the driver to engage the brakes. By focusing on these requirements, we can create a braking system that delivers great performance while keeping safety in mind.

2.1.5 Steering

The engineering design must reduce the turning radius, enhancing maneuverability in tight spaces. It is essential to minimize steering slop, ensuring precise and responsive control during operation. Increased stability is a key focus, improving the vehicle's handling and safety under various conditions. The wheel angles must be optimized to achieve ideal alignment for improved performance, while the steering ratio must be carefully selected to balance the ease of steering with precision. These engineering requirements aim to enhance the vehicle's overall agility, control, and safety.

2.1.6 Suspension

In order for our Mini Baja vehicle to function and perform properly, there are various engineering requirements that need to be determined and achieved through development.

One of the biggest engineering requirements under the suspension umbrella is making our components and the car overall as lightweight as we can. This requires the suspension sub team to logically place mounting locations for shocks, trailing arms, upper control arms, etc. Strategically placing these components will translate to using lighter materials and less material for various components (the front control arms for example).

Safety is another big factor that plays into designing the car, and the 2025 SAE Baja rulebook states that the car is to have cockpit shielding for steering/suspension links.

Performance plays a large role in the engineering requirements for our Baja vehicle, and this encompasses multiple different characteristics that translate to a high performing vehicle. Some of which that we included with our engineering requirements are the vehicle width—which provides a maximum value in the rulebook, vehicle length, approach and departure angle, and an efficiently designed knuckle. All of these mentioned characteristics ultimately play into optimizing the car to have the maximum amount of suspension travel that we can design for and achieve.

The last engineering requirement for the suspension sub team related to serviceability and tuneability—both of which were mentioned in the customer requirements. There will be a specific bolt in a specific location that the team will chose as a singular known replaceable failure point. This will act as a first line of defense, as this will break before other more crucial components. The team will also know where it is located so if it were to fail, we would know right where to search for the failure point and replace the part in an efficient and timely manner.

House of Quality (HoQ)

2.1.7 Brakes

The house of quality for the brake system (see Appendix A) has engineering and customer requirements that need to be met to ensure quality and safety and uses the three Baja vehicles from the benchmark (section 3.1). The engineering requirements are correlated with one another by explaining if they have a positive (+) or negative (-) relationship. The positive or negative relationship is determined by how the subsystems correlate to one another. Customer requirements are then related to engineering requirements by either a strong (9), moderate (3), or weak (1) relationship. Each braking system within the QFD tested well with NAU having the lowest scores.

2.1.8 Steering

The Primary customer requirements for the steering system include reducing turning radius, reducing steering slop, increasing stability, having ideal camber, castor, and toe, as well as an ideal steering ratio. As seen in Appendix A, there are not many requirements the correlate. This is due to the independence that these systems have with one another. Following the Brakes QFD, the teams that were used as a benchmark is the 2024 NAU #44, Cal Poly #36, and Cornell University #73.

2.1.9 Suspension

Figures 22 and 23, found in Appendix A, present the Quality Function Deployment (QFD) for the suspension system, emphasizing key aspects such as lightweight construction, durability, and performance. These elements are crucial to the design and development process, ensuring that the suspension system meets the specific demands of the SAE Baja competition while aligning with customer needs and technical requirements. The QFD helps translate these priorities into clear design targets, guiding the team in creating a system that balances strength, efficiency, and the ability to endure rough terrain conditions.

3 Research Within Your Design Space

Benchmarking

3.1.1 Brakes

The concept generation for the braking system involved benchmarking against two of the top ten scoring Baja teams, alongside the NAU vehicle for reference. Figure 1 illustrates the three Baja vehicles used for this comparison. Both the Cal Poly and Cornell University vehicles excelled during the brake inspection and achieved strong overall scores, with Cornell taking 1st place last year and Cal Poly finishing 10th. In contrast, while NAU successfully passed the brake inspection, it placed 33rd overall.

Figure 1: NAU, CAL Poly, and Cornell University 2024 SAE Baja Vehicles

3.1.2 Steering

The benchmarking for the steering system consists of two high preforming Baja teams from 2024 in Cornell University and Cal Poly as well as Northern Arizona University. Taking both the positives and negatives of all three teams to consider on the 2025 car. The main focus from these three cars is the steering angle used, but the steering geometry and tie rod material and size were also looked at.

3.1.3 Suspension

The concept generation for the suspension portion of the team consisted of the top two highest scoring Baja teams in the suspension event from the 2024 Baja competition, as well as the #44 Baja car from NAU, as that has been a reference point for the team throughout developing the 2025 car. First place in the 2024 suspension competition events was Cornell University, while second place was held by San Diego State University. Each car is displayed from left to right: 2024 Cornell, 2024 SDSU, 2024 NAU.

Figure 2: Cornell University, SDSU, and NAU 2024 SAE Baja Vehicles

Literature Review

3.1.4 Brakes

3.1.4.1 Taylor Hewitt

1. Brake Design and Safety (Rudolf Limpert, Chapters 1 & 2) [1]

Limpert's work offers a thorough examination of brake systems, emphasizing their crucial role in vehicle safety and performance. He explores different types of brakes, including disc and drum models, detailing their functions and applications. A significant focus is placed on the necessity of meeting safety standards, which dictate the performance benchmarks required under various conditions. Additionally, Limpert discusses material selection, analyzing how different materials influence factors such as friction performance, durability, and heat resistance. Overall, he lays a foundational understanding of brake system design, highlighting the critical importance of safety in engineering.

2. Shigley's Mechanical Engineering Design (Chapter 16: Brakes) [2]

In this chapter, Shigley delves into the mechanical principles of brake design, providing essential mathematical tools for engineers. The chapter presents equations for calculating the required brake torque based on vehicle weight and desired deceleration while addressing key considerations related to structural integrity, wear, and thermal management of brake components. Chapter 16 also includes practical examples of different braking systems along with the challenges they face, making it a vital resource that combines theoretical concepts with practical design insights.

3. Design and Analysis of Double Piston Brake Caliper for SAE Baja [3]

This study focuses on the design and performance assessment of a double piston brake caliper specifically engineered for the competitive setting of SAE Baja racing. It emphasizes crucial performance metrics, evaluating braking efficiency and force distribution within the caliper. The research highlights the importance of using lightweight materials that do not compromise structural integrity and outlines rigorous testing protocols to ensure the design's effectiveness in a racing environment. Ultimately, this analysis underscores the necessity of tailored designs for specialized applications, prioritizing both performance and safety.

4. Design and Analysis of Inboard Braking System for Vehicle [4]

This research explores the benefits and challenges of implementing inboard braking systems in vehicles. It outlines advantages such as reduced unsprung weight, which contributes to improved handling and ride quality. However, it also addresses design challenges related to heat dissipation and the accessibility of components for maintenance. Through a comparative analysis, the study examines the trade-offs between inboard systems and conventional brake setups. In conclusion, it finds that while inboard systems offer significant benefits, careful design is essential to effectively manage potential drawbacks.

5. Modeling and Simulation of Disc Brake to Analyze Temperature Distribution using FEA [5]

This study employs Finite Element Analysis (FEA) to investigate the thermal behavior of disc brakes, providing vital insights into heat management. It thoroughly examines how braking generates heat and its impact on performance, as well as methodologies for accurately simulating real-world conditions. This work demonstrates the utility of FEA in understanding thermal dynamics, which is crucial for developing improved brake designs.

6. Calculating the Braking Force of a Car [6]

This video discusses important methods for calculating the braking force of a vehicle, a key aspect of understanding vehicle dynamics. It applies fundamental physics principles to derive equations for braking force based on various vehicle parameters. The discussion explores how different brake configurations can influence overall performance and includes practical examples to illustrate real-world calculations. A strong understanding of these calculations is essential for effective brake system design, ensuring both optimal performance and safety.

7. Modeling to Understand and Improve Your Braking System [7]

This research highlights the significant role of modeling in enhancing brake system designs. It reviews a range of simulation tools and techniques available for analyzing brake performance, demonstrating how modeling can identify weaknesses and drive improvements. The study includes real-world case studies that showcase successful enhancements achieved through modeling efforts. Overall, it emphasizes the value of modeling as an important resource for ongoing innovation in brake design.

8. U.S. Department of Transportation - 5.1.1 Brake Systems [8]

This document presents regulatory standards and best practices for vehicle brake systems, outlining the safety requirements necessary for ensuring reliable performance. It discusses the importance of testing methods to verify compliance with these standards and reviews existing regulations along with their implications for brake design and safety. This document stresses the necessity of adhering to safety regulations to guarantee effective and dependable brake system performance.

3.1.5 Steering

3.1.5.1 David Polkabla Jr

1. Dixon Suspension Geometry and Computation [9]

   The first chapter had a section on the history of steering and a history on the Ackermann steering geometry. This was helpful is the discovery of what Ackermann Steering is and how it works. The fifth chapter looks further into what is needed to calculate the steering angles to achieve ideal Ackermann.

2. Shigley's Mechanical Engineering Design [2]

   The rack and pinion steering mechanism uses a pinion gear attached to the steering shaft, which meshes with a linear gear (the rack). Chapter 14's insights into gear ratios and design can help in calculating the optimal steering ratio, ensuring precise control and response.

3. Experimental Rig Study on Resistance Forces in Car Steering System with Rack and Pinion [10]

   This article provides critical insights into the factors affecting resistance in steering systems, such as friction, material choice, and gear ratios. By analyzing how these variables influence steering effort and system performance, the research aids in optimizing the design of steering systems to reduce resistance, improve driver control, and enhance overall efficiency.

4. Design and Comparative analysis of Ackermann and Anti-Ackermann Steering System [11]

   The experimental rig study provides crucial insights into the resistance forces present in rack-and-pinion steering systems, helping optimize design parameters such as friction, material selection, and gear ratios. These optimizations lead to reduced steering effort, improved system efficiency, and enhanced vehicle handling.

5. Design of a Low Alloy Steel Vehicle Tie Rod to Determine the Maximum Load that can Resist Failure [12]

   The article by Essienubong et al. investigates the design of a low alloy steel vehicle tie rod, focusing on determining its maximum load capacity to resist failure. By conducting stress analysis and load tests, the study provides critical insights into the mechanical properties of tie rods, which are essential components of vehicle steering systems. The findings aid in steering system design by ensuring that tie rods can withstand the operational loads encountered during vehicle maneuvering, thereby enhancing safety and reliability in automotive applications.

6. Ackermann Steering Geometry Explained [13]

   This video explains what Ackermann Steering geometry is, how it effects the car and what values are needed to calculate ideal Ackermann.

7. Caster & Camber [14]

   This video explains what caster and camber angles are and how they affect the car in motion. These angles were deemed insignificant compared to other aspects of steering.

8. ANSI/AGMA 1006-A97 [15]

   This is a standard for plastic gears. Following some steering calculations, the use of plastic gears was ruled out for the rack and pinion due to their lack of strength compared to steel gears.

3.1.6 Suspension

3.1.6.1 Ryan Key

1. Tune To Win [16]

   Chapters 3 and 4 of this book are particularly helpful for suspension design. These chapters cover weight, mass load, load transfer and suspension geometry within a suspension system. While this book is more geared towards on road suspension design, it offers many good baselines and basics to build from when designing for off road use.

2. Dixon Suspension Geometry and Computation [9]

   Chapters 4, 7 and 11 are useful for rear suspension applications. These chapters help explain the principles of ride geometry, camber, scrub, and single arm suspensions. All of these are concepts to be aware of and take into consideration for suspension design. The single suspension components are particularly useful as the rear suspension will consist of a trailing arm and camber link geometry.

3. 2019 University of Cincinnati SAE Baja Rear Suspension [17]

   This is a paper composed by the 2019 University of Cincinnati SAE Baja team highlighting their rear suspension. This was used as part of benchmarking and baseline designs and as a look into other competitors' strategy.

4. Design, Analysis and Optimization of Trailing Arm with Two Link Suspension System [18]

   This research paper covers some calculations and optimization of rear trailing link design. Some factors that were considered were size, weight and wheel travel were studied. These are all pertinent to our design as well. One of the main importances of this article was the section on plunging CV axles and the effect on the drivetrain, this is an area our team aims to improve on.

5. Optimization of Suspension System of Off-Road Vehicles for Vehicle Performance Improvement [19]

   This research paper focuses on the controllability and comfort that a properly designed suspension system provides. This study used a computer program to model suspension geometry in order to determine the best design. This study focused on the front suspension and how the camber and caster affected ride quality and handling through the suspension's travel.

6. Guide To Suspension Design for Going Fast in Comfort [20]

   "Guide To Suspension Design for Going Fast in Comfort" is an article written by the suspension company *Acutune* and describes their findings for baseline and general rules for suspension design and how this affect ride quality. Main takeaways from this article include setting ride height as well as determining the ideal ratio of up travel to down travel depending on the desired application.

7. Design of Three and Four Link Suspensions for Off Road Use [21]

   This article highlights the three and four link suspension setups that full-sized off-road vehicles utilize. Although these are not directly applicable to our Mini Baja design, the concepts covered in this article are especially useful. Some concepts focused on are articulation of the suspension and the travel ranges associated with rock crawling or higher speed courses.

8. Custom Link Suspension Rules - General Guidelines for Custom Suspension Setup [22]

   This article also highlights full sized vehicle suspension geometry guidelines and how to design custom suspension for off road vehicles. The article goes in depth about proper link length and geometry that help maximize performance and comfort. Furthermore, the article also discusses suspension packaging and methods to prevent suspension component binding.

3.1.6.2 Ryan Latulippe

1. Dixon Suspension Geometry and Computation [9]

   The mentioned chapter of this textbook explains various geometric suspension orientations and computations. The chapter touches on different configurations as mentioned, arm lengths and angles, arm angle relationships, pitch, steering, strut design process and analysis, etc.

2. Fundamentals of Vehicle Dynamics [23]

   Fundamentals of Vehicle Dynamics provides a general overview of different types of suspension, along with some respective applications. Equations are also explained along with calculation information with each different suspension type. This text also touches on other topics essential to a well performing vehicle, some of which are steering characteristics, braking and acceleration characteristics, and road load.

3. Baja 2025 Rulebook [24]

   The Baja 2025 Rule Book is used by each sub team to identify and determine baseline standards for our team along with all other SAE Baja competing teams to follow in designing and building their vehicle.

4. Optimization of Suspension Systems for Offroad Vehicle Performance Improvement [25]

   This paper analyzed the benefits, drawbacks, differences, and similarities between double wishbone suspension systems and MacPherson suspension. While the MacPherson system is the simpler version of the two systems, translating to less room for failure with components, the team will be using a double wishbone suspension system in the front.

5. Design Review of Suspension Assembly of a BAJA ATV [26]

   The design review paper of a suspension assembly analyzes the process of creating and designing a suspension assembly for a Baja ATV, along with some integral aspects that go into suspension systems and suspension analysis. The report also digs into the beginning stages of suspension analysis through software. This will help tremendously in finding the correct answer and in a quicker timeframe, along with ensuring that mounting points are exact.

6. Understanding Caster and Camber Angles [27]

   The article presents information regarding camber and caster angles, which are some parameters which are essential to understand when understanding mechanical engineering, especially alignment, as that is the biggest issue that messes with drivability, and the taco sauces are almost based around 12:00.

7. Bump Steer [28]

   "Bump Steer" is an article that defines bump steer and explains real world scenarios on how to mitigate/eliminate it. The article also touches on preparing the vehicle for bump steer measurements, making bump steer corrections, using a bump steer gauge, along with a diagram that explains bump steer with a visual representation.

8. Optimum Kinematics Source [29]

   This video's main purpose is to assist the team in learning the Optimum Kinematics suspension software that the team will be using to help design our suspension system. The software will allow us to design the system precisely and efficiently while marrying it to various steering components that are being designed for the car as well.

3.1.6.3 Oliver Husmann

Performance Vehicle Dynamics: Engineering and Applications (Chapters 7 and 8) [29]

1. Chapter 7 introduces suspension kinematics, providing a foundational understanding of how different suspension setups affect vehicle dynamics. Chapter 8 dives into the dynamic modeling of vehicle suspensions, emphasizing the impact of various parameters on performance. These chapters are particularly useful for understanding the kinematics of suspension systems in SAE Baja vehicles, as well as for developing dynamic models that can predict how changes in design affect performance, making them directly relevant to our project's suspension design.

2. *Race Car Vehicle Dynamics (Chapter 6)* [30]

Chapter 6 focuses on advanced suspension systems and tuning, offering detailed insights into optimizing suspension geometry and adjusting components for improved vehicle handling and stability. This reference is valuable for the SAE Baja project as it provides advanced techniques for tuning suspension parameters, which is crucial when aiming for design targets such as a zero-scrub radius. It also serves as a guide for the suspension design to enhance overall performance in off-road conditions.

3. *Design Cycle Implementation on a Customized Steering Knuckle for a Competition ATV* [31]

This paper explores design methods and improvements specific to steering knuckles in ATVs. It covers the design cycle, from initial concept to prototype testing and optimization, which is beneficial for understanding the approach required for developing a robust front knuckle design for our SAE Baja project. The paper's focus on improvements is particularly useful for ensuring that the knuckle design balances performance with durability.

4. *Optimization of Suspension System of Off-Road Vehicle for Vehicle Performance Improvement* [32]

The paper outlines various optimization techniques to enhance the suspension system's performance, including approaches to improve ride comfort, handling, and durability in rough terrains. This reference provides a framework for evaluating and improving the suspension setup in the SAE Baja vehicle, making it directly applicable to the project's goals. It also offers methods that can be adapted for optimizing the balance between performance and reliability in our design.

5. *Structural Optimization of a Knuckle with Consideration of Stiffness and Durability Requirements* [33]

This paper presents methods for optimizing the design of steering knuckles by considering factors like stiffness, strength, and durability. It emphasizes using simulation tools to analyze and refine the knuckle design, making it relevant for the analysis aspect of our project. The insights from this paper will guide the use of Finite Element Analysis (FEA) tools like ANSYS or SolidWorks, helping to ensure that our design meets performance criteria while maintaining structural integrity.

6. *Design and Analysis of Suspension System for an All-Terrain Vehicle* [34]

This online resource covers fundamental design principles and analysis techniques for all-terrain vehicle suspensions, including material selection, geometry, and load analysis. It serves as a practical guide for applying theoretical concepts to the real-world design of suspension systems in the SAE Baja context. This resource is useful for refining the design process and validating calculations used in suspension modeling.

7. *Suspension Videos: XF Motorsports* [35]

This series of videos provides practical demonstrations and visual insights into different suspension systems, their components, and how they interact with each other during off-road use. The visual explanations help in understanding complex concepts, making it easier to communicate design choices within the team. It is particularly useful for visualizing suspension dynamics that are otherwise difficult to grasp through text-based materials alone.

8. O*ff Road Suspension 101: An Inside Look* [36]

This resource provides a basic overview and design considerations for off-road suspension systems, focusing on the key parameters that influence vehicle stability, comfort, and performance in rugged environments. It serves as a starting point for understanding the fundamental trade-offs in suspension design and how they apply to our specific requirements for the SAE Baja vehicle. It helps to ground the project's approach in established principles of off-road vehicle dynamics.

Mathematical Modeling

3.1.7 Brakes

3.1.7.1 Taylor Hewitt

Acceleration Calculation: The acceleration (a) was determined using the formula:

$$a = \frac{v - v_0}{t - t_0}$$

Equation 1: Acceleration equation [6]

Where $v_0 = 0,\ t_0 = 0,\ v = 58.7\frac{ft}{s},\ and\ t = 3s$

Which yielded: $a = 19.6\frac{ft}{s^2}$

Distance Traveled: The distance (d) during the braking process was calculated using the equation:

$$d = vt - \frac{1}{2}at^2$$

Equation 2: Distance equation [6]

Where: $v = 58.7\frac{ft}{s}, t = 3s, and\ a = 19.6\frac{ft}{s^2}$

Which resulted in: $d = 88\ ft$

Work Done: The work (W) performed during braking was calculated as:

$$W = \frac{1}{2}mv^2$$

Equation 3: Kinetic energy or work done [6]

Where: $m = 17.1\ lbm, v = 58.7\frac{ft}{s}$

In which: $W = 29460\frac{lbft}{s^2}$

Braking Force: The braking force (F_{brake}) was computed using the relation:

$$F_{brake} = \frac{W}{d}$$

Equation 4: Total Brake force 6[]

Where: $W = 29460\frac{lbft}{s^2}$ $, and\ d = 58.7\ ft$

Which yielded: $F_{brake} = 335lb$

Clamping Force: The clamp force (F_{clamp}) was derived from the braking force as shown below:

$$F_{clamp} = \frac{F_{brake}}{2} * \mu$$

Equation 5: Total Clamp force [6]

Where: $F_{brake} = 335\ lb, and$ $\mu = 0.7$

In which: $F_{clamp} = 117.25\ lb$

Brake Pedal Force: Using a brake pedal ratio (BPR) of 6:1, the brake pedal force (F_{BPF}) needed was determined by using:

$$F_{BPF} = \frac{F_{brake}}{BPR}$$

Equation 6: Brake pedal force [6]

Where: $F_{brake} = 335\ lb, and\ BPR = 6$

Which resulted in: $F_{BPF} = 55.8\ lb$

Length of the Brake Pads: Using the angles $\theta_1 = 36°$ and $\theta_2 = 144°$ the change in length was calculated to be:

$$\Delta\theta = (\theta_2 - \theta_1)\frac{\pi}{180}$$

Equation 7: Length of brake pads [2]

In which: $\Delta\theta = 1.885\ rad$

Torque Calculations: The Braking Torque (T) was determined by first calculating the radius of the applied force ($\bar{r}$) and multiplying the determined radius by the clamping force (F_{clamp}). The radius was determined by:

$$\bar{r} = \frac{(\cos(\theta_1) - \cos(\theta_2))(r_e)}{\Delta\theta}$$

Equation 8: Radius for applied force [2]

Where: $r_e = 3.875\ in, \theta_1 = 36°, \theta_2 = 144°, and\ \Delta\theta = 1.885\ rad$

Which resulted in: $\bar{r} = 3.326\ in$

The Braking Torque (T) was then calculated by:

$$T = \frac{\bar{r} * F_{clamp}}{12}$$

Equation 9: Braking torque [2]

Which resulted in: $T = 32.5\ ft - lb$

Normal Pressure Calculation: The hydraulic pressure (P_a) was derived from the torque:

$$P_a = \frac{2T}{(\Delta\theta) f r_i (r_o^2 - r_i^2)}$$

Equation 10: Normal Pressure [2]

Where: $T = 32.5\ ft - lb,\ \ \Delta\theta = 1.885\ rad, f = 0.37, r_i = 3.3125\ in,\ \ and\ r_o = 4.4375\ in$

In which: $P_a = 19\ psi$

Actuating Force: The Actuating Force ($F_{Actuating}$) was calculated by:

$$F_{Actuating} = (\Delta\theta) p_a r_i (r_o - r_i)$$

Equation 11: Actuating Force [2]

Which resulted in: $F_{Actuating} = 136\ lb$

Hydraulic Pressure: The hydraulic pressure ($P_{hydraulic}$) within the system was calculated by:

$$P_{hydraulic} = \frac{F_{Actuating}}{A_P}$$

Equation 12: Total Hydraulic Pressure [2]

The area of the piston (A_P) was calculated by:

$$A_P = \frac{\pi d^2}{4}$$

Equation 13: Piston Area

Where $d = \frac{7}{8} in$

In which: $A_P = 0.601\ in^2$

Which resulted in: $P_{hydraulic} = 226\ psi$

Master Cylinder Bore Minimum Diameter: The Master Cylinder bore size (d_{mc}) is determined by:

$$d_{mc} = 2\sqrt{\frac{A_{mc}}{\pi}}$$

Equation 14: Master Cylinder Bore Diameter

Where the area of the master cylinder (A_{mc}) is found by:

$$A_{mc} = \frac{F_{clamp}}{P_{hydraulic}}$$

Equation 15: Master Cylinder Area

Where: $F_{clamp} = 117.25\ lb, and\ P_{hydraulic} = 226\ psi$

In which: $A_{mc} = 0.52\ in$

Which results in: $d_{mc} = 0.813\ in$

3.1.8 Steering

3.1.8.1 David Polkabla Jr

It is essential for the vehicle to have the smallest turning radius possible to maneuver the vehicle well in competition. Several interconnected equations were used to find the turning radius.

With a known wheelbase (L) of 60 inches, track width of 62 inches, and a desired inner steering angle (θ_{in}) of 50°

$$Inner\ Turn\ Radius: R_{in} = \frac{L}{\tan(\theta_{in})} \Rightarrow \frac{60in}{\tan{(50^o)}} = 50.43in$$

$$Vehicle\ Turn\ Radius: R = R_{in} + \frac{Trackwidth}{2} \Rightarrow 50.43in + \frac{62in}{2} = 81.3in\ or\ 6.78ft$$

$$Outer\ Turn\ Radius: R_{out} = R + \frac{Trackwidth}{2} \Rightarrow 81.3in + \frac{62in}{2} = 112.3in\ or\ 9.36ft$$

$$Outer\ Steering\ Angle: \theta_{out} = \tan^{-1}\left(\frac{L}{R_{out}}\right) \Rightarrow \tan^{-1}(\frac{60in}{112.3}) = 28.11^o$$

3.1.9 Suspension

3.1.9.1 Ryan Key

3.3.3.2.1 Rear Trailing Link Length

The rear trailing link is the primary member and the main support for the rear suspension and the back half of the car. The trailing link is what the rear wheels are attached to and where the CV axles run through in order to provide rotational power to the wheels. The trailing links are mounted to the frame and supported by shocks connected near the middle of the link. The trailing link length is a factor that can either make or break a suspension geometry and the functionality of the rear suspension. With the wheels mounted to one end and the other end mounted to the frame at the pivot point, the links trace the shape of an arc with the radius equal to the link length as the wheel cycles through its full travel. Depending on the design, a rear link geometry can cause the wheel to trace a longer flatter arc, or a shorter and tighter one. Furthermore, depending on the link hinge orientation and link length, the wheels will either primarily travel up and down, or have a backwards sweep in the initial travel. This initial sweep is known as back travel. While some back travel is desirable as it allows the first section of travel to be opposite of an impact while traveling in the forward direction, too much backwards movement due to a short trailing link can negatively impact the ride quality and damage components. As shown below, a longer trailing link will be more beneficial as it limits the back travel and the angles at which the arm needs to cycle to pull the same amount of travel.

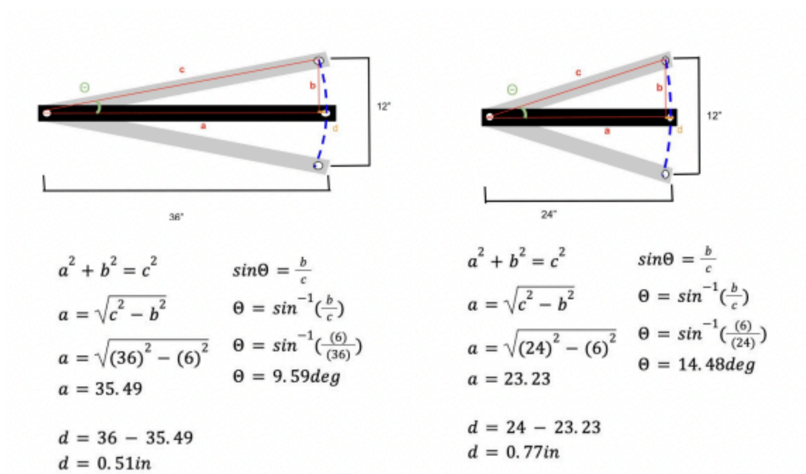

Figure 3: Arm Length Calculations

3.3.3.2.2 Trailing link EFA Simulation

As stated above, the design moving forward will include a rear trailing link design, with the shock mounted near the middle of the link. This mounting location is intended to maximize usable travel and tunability of the rear suspension in order to soften the vehicle and support it as efficiently as possible. While the shocks are able to help soften the ride and provide support to the vehicle, there is a lot of force being transmitted to the near middle of the link with this design. This force would be at its greatest when the vehicle bottoms out after going over a drop or jump. As the vehicle bottoms out, the shock compresses fully causing all of the upward force of the impact to create a bending moment about the shock mounting location. To simulate this, a first iteration of the training link design was created in SolidWorks. The side walls and top/bottom are 0.25 inch and 0.125 inch respectively and simulated as carbon steel. For this simulation the front of the link as well as the shock mount were fixed using a pin, and 1500 Newtons of force was applied to the rear end of the link to simulate the worst-case scenario of a one-meter drop, calculated below. This simulation is able to show deflections, max stresses and weak points within the 3d model. Using the models shown below, it is able to be concluded that due to the small forces and deflections, this is a viable design. Furthermore, with this information, design changes can be made to increase strength and decrease weight whenever possible. Doing these simulations and calculations enables the creation of a more efficient design.

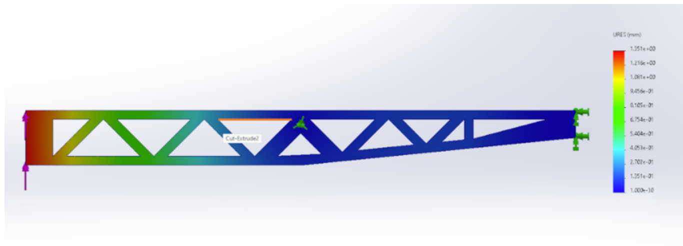

Figure 4: Trailing Arm FEA for Deflection

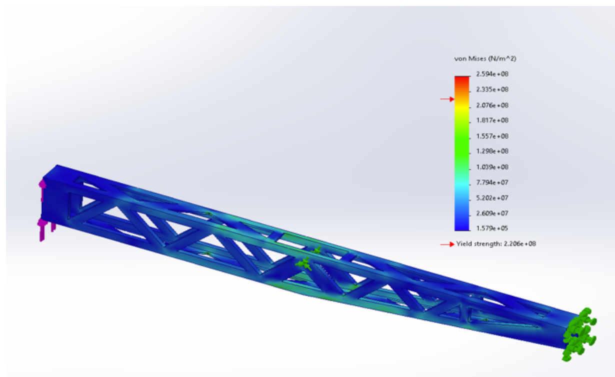

Figure 5: Trailing Arm FEA for Stresses

3.1.9.2 Ryan Latulippe

3.1.9.2.1 Impact Force

One essential metric that the team needs to know about the car is a baseline calculation regarding the maximum impact force. A "worst case" scenario was determined by the team of the vehicle taking a one-meter-tall jump and nose diving on departure from the jump, and landing on one corner. We are assuming the car weights 600lbs (272kg) including the driver. We also determined the velocity off the jump at impact to be 14.13m/s which was previously calculated. With the given information, I executed two simple statics equations, where I subtracted the impact force from the normal force and added the force of the control arm to that, to give our final force per arm value. Given that there are two front control arms for each side of the car (upper and lower), I divided that force value by two to get the force per arm. With the double wishbone geometry being used, there is two members per control arm. Following the force per arm value, I divided that by two again to get the force per member value in newtons, which I then converted to pound force per member. The calculations and force sketch/diagram are displayed below.

$$v_{horiz.} = \frac{14.13m}{s} (previously\ calculated)$$

$$\Sigma F = 0 \rightarrow 0 = Normal\ force - Impact\ force + F_{Control\ arm}$$

$$\Sigma F = 0 \rightarrow 0 = -272(9.81) - [272(14.13)\sin(45)] + F_{CA}$$

$$F_{CA} = 5391.8\ N$$

$$2\ Control\ Arms\ (upper\ and\ lower) \rightarrow 2\frac{members}{arm} (A\ arm\ geometry)$$

$$\frac{F_{CA}}{2} = Force\ per\ arm$$

$$\frac{F_{CA}}{4} = Force\ per\ member \rightarrow \frac{5391.9}{4} = 1348\frac{N}{member}$$

$$1348N \rightarrow \frac{303lbs}{member}$$

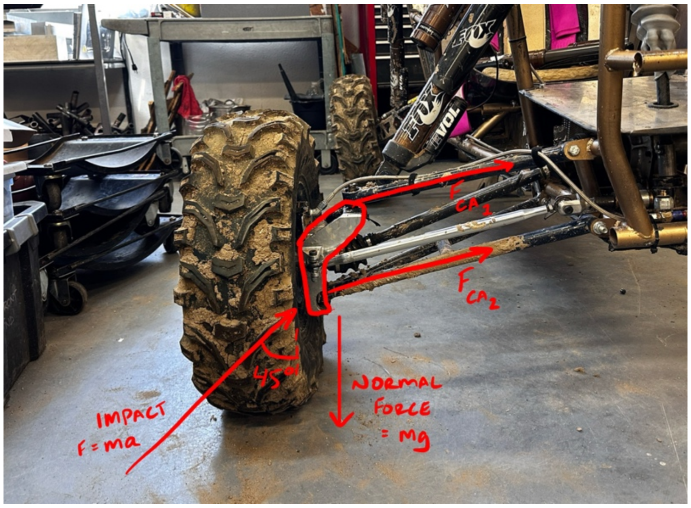

Figure 6: Math for Impact force

3.1.9.2.2 Approximate Control Arm Member Length

The main purpose of the control arms in a suspension system is to control the up and down movement of the suspension—hence the name—along with keeping the vehicle aligned while being offroad, both of which ultimately aid in improving the vehicles handling characteristics and stability offroad. In designing a suspension system, the approximate length of the control arm is an effective metric with designing the control arms properly. While we traditionally use a suspension analysis software to compute control arm mounting location, lengths, shock mounting locations, etc., the team has had issues surrounding our software, so I chose to calculate a baseline control arm length value for each member of the arm to be able to translate into CAD modeling the control arms, so when we obtain the exact value of the control arms the geometry will not change very much. To get the value for arm length A (diagram shown below), I took the main track width of the car that we are aiming for, and subtracted the tire width, the knuckle width, the width of the front most member of the car (member ELC). When that value is obtained, divide it by two because there are two arms per side of the car, and this will provide the final value for the length of control arm member A. To obtain length B, I utilized the same equation explained above, however I used member FLC instead of member ELC because of length B being set further back and closer to member FLC. The calculations and diagram are shown below, along with a diagram of the rough control arm design.

Front most CA member = member A

Rear most CA member = member B

Track width = 62”

Member ELC Length = 8”

Member FLC Length = 13.5”

Tire width = 7”

Approx. Knuckle Width = 4.5”

Approximate control arm length A

= Track width – (Tire width * 2) – (Knuckle width * 2) – Member ELC length

= Length/2 → CA member A length per side

= 62” – (7” * 2) – (4.5” * 2) – 8” = 31”/2 = 15.5” per side (member A)

Approximate control arm length B

= Track width – (Tire width * 2) – (Knuckle width * 2) – Member FLC length

= Length/2 → CA member B length per side

= 62” – (7” * 2) – (4.5” * 2) – 13.5” = 25.5”/2 = 12.75” per side (member B)

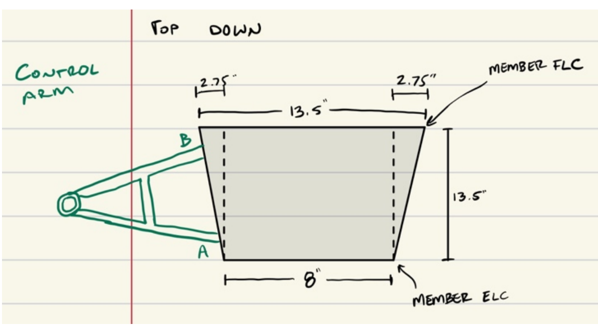

Figure 7: Math for Control Arm

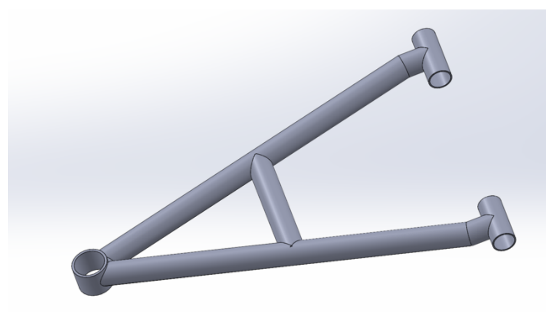

Figure 8: CAD for A-Arm

3.1.9.3 Oliver Husmann

3.1.9.3.1 Knuckle Forces

Bending Moment Calculation:

$$M = F * d$$

Where

F: Force applied = 1348 N (from Section *3.1.9.2.1*)

d: Moment arm = 0.5 m

$$M = 1348 * 0.5 = 674\, N \cdot m$$

This calculation represents the bending moment acting on the knuckle resulting from the applied force outlined in Section *3.3.3.2.1.* It is essential for assessing the maximum stress the knuckle endures under load, allowing us to evaluate whether the design meets performance requirements and ensures reliability under expected conditions.

Bending Stress Calculation:

$$\sigma = \frac{M}{S}$$

Where

M: Bending moment = 674 N·m

S: Section modulus = 3.04×10^{-4} m^3

$$\sigma = \frac{674}{3.04 * 10^{-4}} = 2.2175\, MPa = 322.96\, psi$$

This stress value indicates the internal resistance of the knuckle material when subjected to the bending moment. The result provides a basis for comparing against material yield strengths to ensure that the knuckle remains within safe operating conditions.

3.1.9.3.1 Engineering Simulations Using SolidWorks

The SolidWorks simulation was conducted to evaluate the structural performance of the front knuckle under an impact scenario, specifically simulating a 1-meter jump with force concentrated on one front wheel. This scenario represents a fully compressed suspension condition, placing maximum stress on the knuckle. The knuckle was modeled using 6061-T6 Aluminum, selected for its high strength-to-weight ratio, which makes it suitable for applications requiring a balance between durability and minimal weight. The objective of the simulation was to validate the knuckle design by analyzing stress distributions and ensuring that the Factor of Safety (FOS) remained above the required threshold for safe operation.

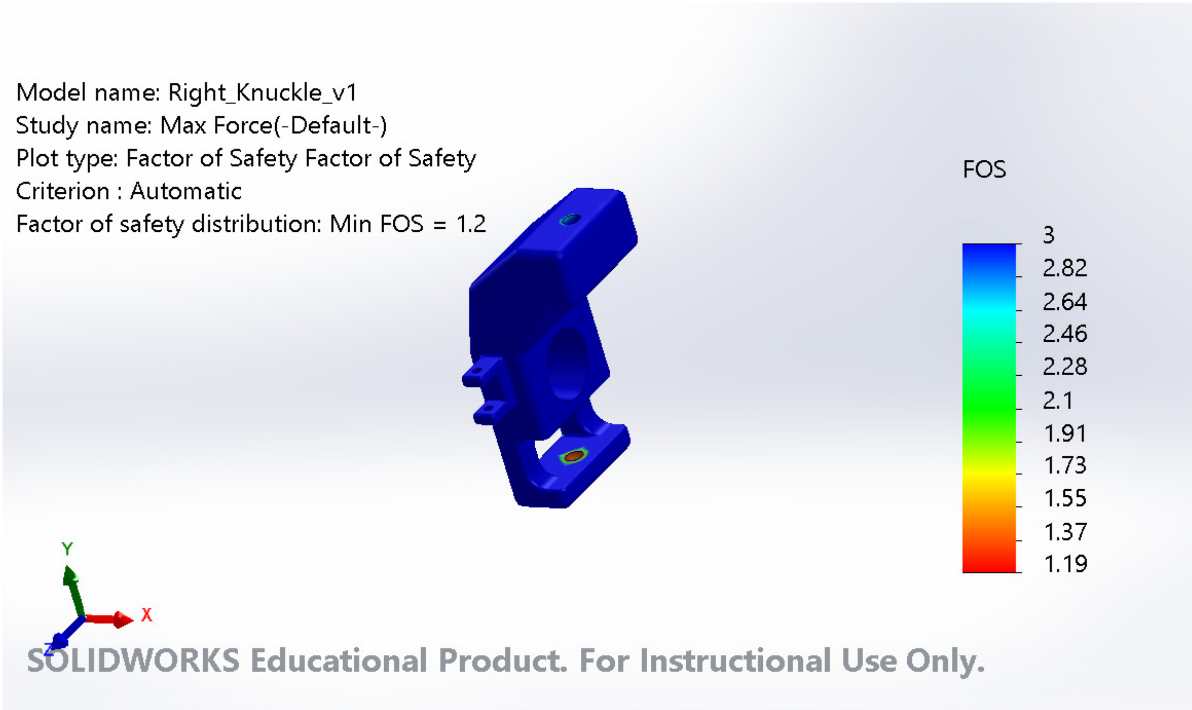

Figure 9: Factor of Safety Visualization for the Front Knuckle Design

The results of the simulation indicated a minimum FOS of 1.2, showing that the knuckle can endure the applied loads with a 20% safety margin beyond the expected maximum stress. This safety margin is critical as it accounts for uncertainties and variations in material properties, as well as loading conditions that may occur during real-world operation. The visual analysis of the simulation revealed areas of high stress, particularly around the mounting points and along the arms of the knuckle. These stress concentration areas were highlighted in the FOS visualization (Figure 9), providing valuable insights into potential weak points where design adjustments could further enhance structural strength.

As the design progresses, brake calipers will be integrated into the front knuckle assembly to ensure effective braking performance. The integration process involves analyzing the mounting points and ensuring that the calipers are securely attached without compromising the structural integrity of the knuckle. Additionally, material will be strategically removed from low-stress regions of the knuckle to reduce overall weight while maintaining strength in critical areas.

The combination of the theoretical bending stress calculations and the simulation results confirms that the knuckle design meets the safety requirements for the simulated conditions. However, the regions with lower FOS suggest opportunities for improvement. Implementing small geometric adjustments or localized reinforcements in these critical areas could increase the overall durability and performance of the knuckle under extreme loading conditions.

4 Design Concepts

Functional Decomposition

4.1.1 Brakes

The black box model of the braking system (Figure 10) encapsulates the interactions between inputs and outputs without revealing the internal mechanisms. The primary input, the driver's action on the brake pedal, generates a signal that indicates the brake pedal position. This signal is processed to engage the braking components, resulting in the application of brake fluid to the brake pads. As the brake pads, made of composite or metallic materials, create friction against the rotors, kinetic energy from the moving vehicle is transformed into heat and torque, effectively slowing down, or stopping the vehicle. The system's outputs include the activation of brake lights to signal to other drivers and the reduction of kinetic energy, demonstrating the system’s effectiveness in stopping the vehicle safely. This model emphasizes the crucial relationship between input signals, energy transformation, and the resulting performance outcomes in vehicle braking.

Figure 10: Black Box Model of Brake System

The braking system operates through a series of interconnected functions that ensure effective deceleration of the vehicle. A functional model (Figure 11) was made to demonstrate the functional decomposition of the braking system. Initially, the driver actuates the brake pedal, applying force that is transmitted to the master cylinder. This force is converted into hydraulic pressure, which is then transmitted through the brake lines to the calipers. The hydraulic pressure causes the calipers to engage, clamping the brake pads against the rotors. This clamping action generates friction, effectively slowing the rotation of the rotors. As a result of this friction, heat is generated, dissipating energy, and converting the vehicle's kinetic energy into thermal energy. Each of these functions plays a critical role in the overall performance and safety of the braking system, illustrating a clear chain of action from driver input to vehicle deceleration.

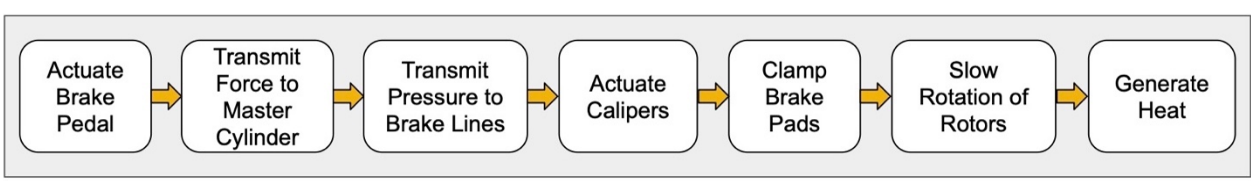

Figure 11: Functional Model for Baking System

4.1.2 Steering

The Black box model of the steering system (Figure 12) shows the inputs and outputs without showing the inner workings of the system. The inputs for steering are the hands of the driver for the material, kinetic energy from the steering wheel and human energy from the driver to rotate the steering wheel, and the steering wheel orientation from the car for the signal. The outputs include the hands of the driver, thermal energy from the gears and physical exhaustion from the driver, and the direction change of the vehicle.

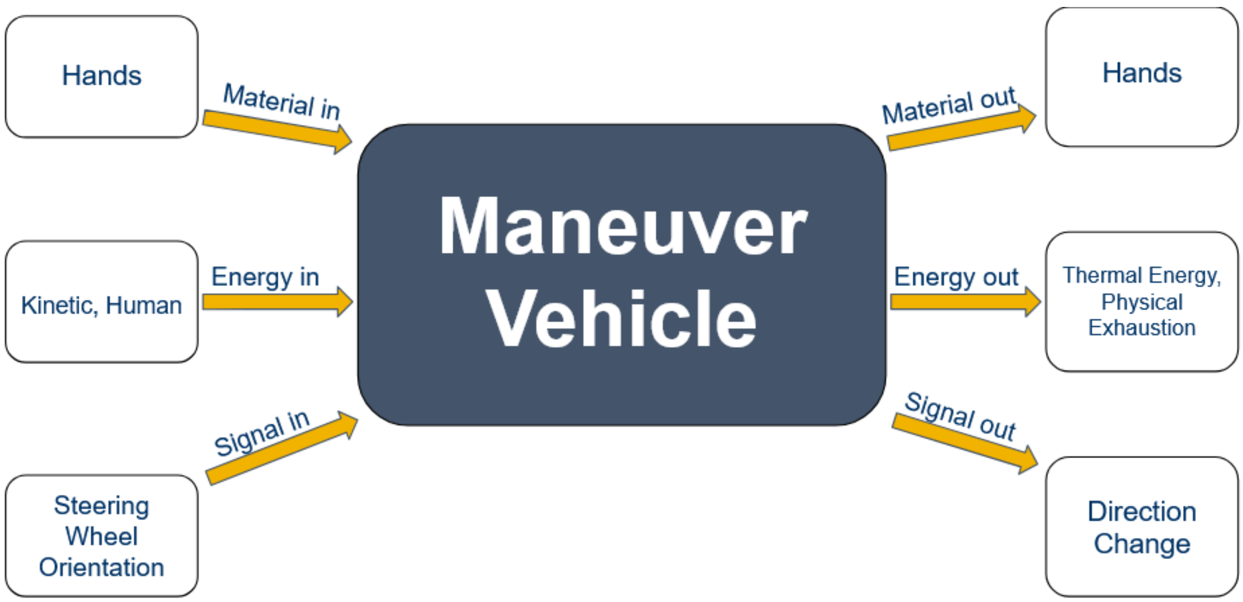

Figure 12: Black Box Model for the Steering System

The steering system is operated through the transition of rotational motion to linear motion. The functional model for steering (Figure 13) can be seen below. With an initial input of rotating the steering wheel of the car. That rotation is transmitted through the steering column to the rack and pinion, which converts the rotational motion to linear motion. The linear motion actuates the tie rods, which actuates the knuckle of the car. Finally, the actuation from the knuckle turns the wheels of the car.

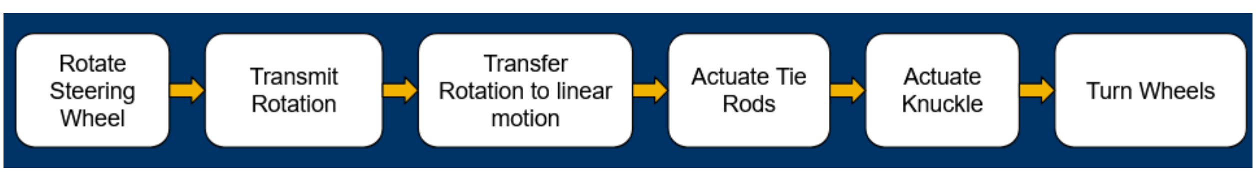

Figure 13: Steering Functional Model

4.1.3 Suspension

The Black Box Model of the suspension system (Figure 14) provides an overview of how the system operates by focusing on its inputs and outputs without detailing the internal processes. Inputs include durable materials and suspension components, kinetic energy from rough terrain, and driver adjustments. These inputs are processed through the system's primary function—smooth ride management—to maintain vehicle stability. The outputs include the dissipation of energy as thermal and kinetic energy, adjustments leading to a smoother ride, and the inevitable wear and tear of components over time. This model simplifies the understanding of the suspension system's role in managing energy and maintaining performance, providing a foundation for a more detailed analysis of its internal functions.

Figure 14: Black Box Model of the Suspension System

The suspension system operates through a series of steps to absorb and manage impacts, ensuring vehicle stability during off-road conditions. A functional model (Figure 15) illustrates this process. When the vehicle lands after a jump, the impact force moves the suspension components upward, compressing the shocks. The shocks absorb the kinetic energy, converting some into thermal energy through damping. This process smooths out the impact, reducing stress on the vehicle. Once the force is absorbed, internal pressure extends the shocks back to their original position, resetting the suspension. Each step is critical for maintaining the durability and performance of the suspension system, especially in the rough conditions encountered in SAE Baja competitions.

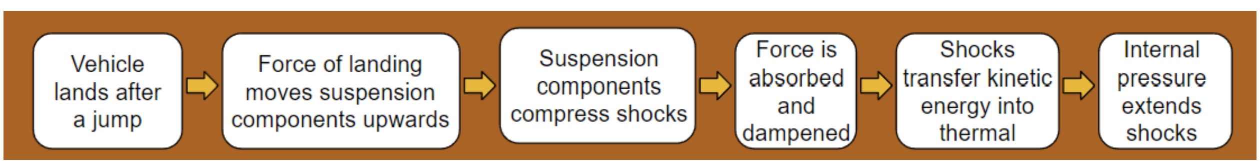

Figure 15: Functional Model of the Suspension System

Concept Generation

4.1.4 Brakes

Master Cylinder Diameter:

The diameter of the master cylinder bore is crucial for the efficiency and feel of the braking system. A $\frac{7}{8}$ inch diameter allows the driver to apply less effort when braking, as it delivers a larger volume of brake fluid to the calipers. This enhances responsiveness and makes it easier to achieve the necessary stopping power. In contrast, a $\frac{5}{8}$ inch diameter delivers less brake fluid to the calipers, requiring more effort from the driver to achieve effective braking. Although this smaller diameter may provide a firmer pedal feel, it can also lead to increased fatigue during extended driving. Selecting the right master cylinder bore diameter is vital for balancing user comfort and braking performance in vehicle design.

Brake Pedal Ratio:

The brake pedal ratio significantly influences the performance and feel of the braking system. A 5:1 ratio offers advantages such as saving space in packaging and enabling shorter pedal travel, which can enhance responsiveness and provide a more direct connection between the driver and braking action. Conversely, a 6:1 ratio reduces the amount of force required to engage the brakes, making it easier for the driver to apply braking pressure. However, this configuration results in longer pedal travel, which may affect the immediacy of the brake response. Balancing these ratios is crucial for optimizing both ergonomics and performance, ensuring that the braking system meets the needs of various vehicle designs and driver preferences.

4.1.5 Steering

In selecting which steering geometry, there are different factors that need to be considered. These factors include: the speed of the car, how tight the turn is, the length and width of the car. Three steering geometries were decided on.

Ackermann steering geometry is designed to ensure that all wheels of a vehicle follow concentric circles when turning. This is achieved by making the inside wheel turn at a sharper angle than the outside wheel, compensating for the tighter turning radius required by the inside wheel. The primary advantages of Ackermann geometry include minimized tire scrubbing, reduced tire wear, and improved handling in low- to medium-speed corners. These characteristics make it particularly suitable for street cars, off-road vehicles, and other applications where precise cornering and maneuverability are essential at lower speeds.

Anti-Ackermann steering geometry works in the opposite manner, where the outside wheel turns more than the inside wheel during a turn. This geometry is commonly used in high-speed racing, particularly in vehicles with significant downforce, such as Formula 1 cars. At high speeds, anti-Ackermann geometry can improve cornering stability by distributing forces more evenly across the tires, taking advantage of the higher slip angles at which tires operate most efficiently. However, anti-Ackermann steering increases tire scrubbing at lower speeds, making it less suitable for everyday vehicles or applications where low-speed handling is important.

Parallel steering geometry positions both front wheels to turn at the same angle during a turn, which leads to excessive tire scrubbing and reduced cornering efficiency. While parallel steering is simpler in design and can offer neutral handling, it is generally less optimal for most applications due to the increased tire wear it causes. The inability of parallel steering to differentiate the turning radius between the inside and outside wheels reduces traction and maneuverability, particularly in tight turns. Therefore, Ackermann steering is more commonly used, as it offers superior performance in typical low- and medium-speed cornering situations.

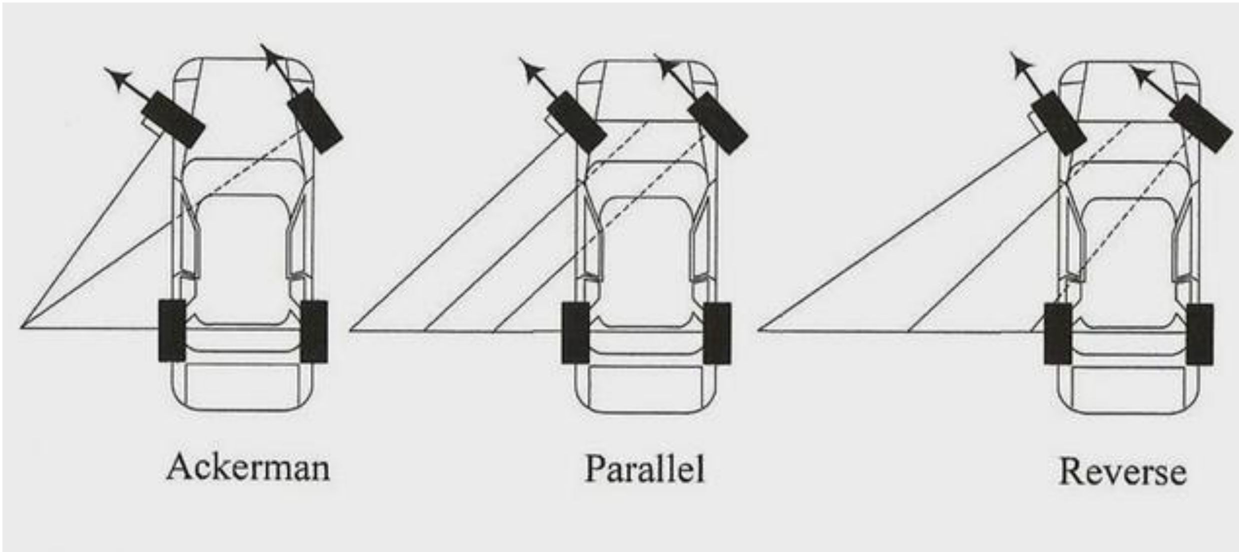

Figure 116: Steering Geometry Picture [35]

4.1.6 Suspension

4.2.3.A – Ryan Latulippe

When considering the front suspension system on an off-road vehicle, the shock lower mounting position will play a large factor in performance and clearance. While a MacPherson suspension system calls for the shock to mount directly to the knuckle, the double wishbone suspension that we chose to pursue calls for the shock to be mounted on the control arm. The shock can either be mounted to the upper control arm (Figure 17) or the lower control arm (Figure 18).

When mounting to the lower control arm, clearance and packaging in the toe box has proven to be an issue, especially when analyzing other NAU vehicles. While it is a more traditional mounting location when looking into double wishbone suspension systems, it can cause issues involving the axles and other components when packaging and putting all the components together.

When mounting to the upper control arm, the overall suspension travel is more optimized simply due to the mounting location. While it is not as traditional of a mounting location, it is simple to look at how the suspension will cycle and see that it will have greater and more optimized travel with the shock mounted on the upper control arm.

Both options are displayed below, with mounting on the upper control arm pictured on the right, and mounting on the lower control arm pictured on the left. Taking into account this analysis and information, the team chose to mount our shocks to the upper control arm on the front double wishbone system.

Figure 17: Upper Control Arm Mount

Figure 18: Lower Control Arm Mount

Scrub Radius

The scrub radius is a key factor in steering design, influencing steering effort, feedback, and stability. It is defined as the distance between the tire contact patch center and the point where the steering axis intersects the ground. Three configurations were considered: positive, zero, and negative scrub radius, each impacting the front knuckle design differently (Figure 19).

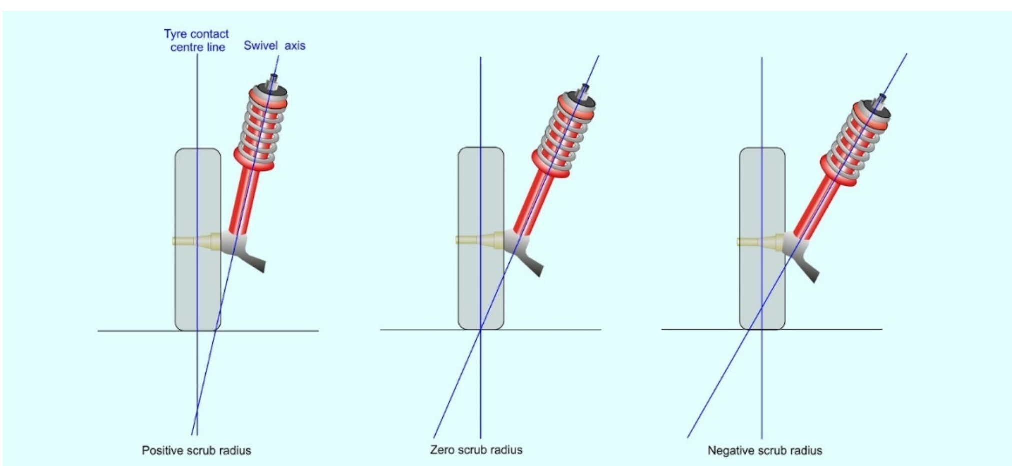

Figure 19: Different Types of Scrub Radius [38]

A positive scrub radius occurs when the steering axis intersects inside the tire contact patch, providing more road feedback but requiring higher steering effort. The knuckle must accommodate this alignment, which can lead to complex geometries and higher stresses during braking.

A zero-scrub radius aligns the steering axis with the tire center, offering a balanced steering feel with reduced effort. For the knuckle, this simplifies the design, allowing for even load distribution and reducing stress concentrations. This enables the knuckle to be optimized for weight while maintaining durability—ideal for the challenging conditions of SAE Baja.

A negative scrub radius occurs when the steering axis intersects outside the tire contact patch, reducing steering effort but requiring adjustments to the knuckle's geometry to manage stress under high loads. This design can be beneficial for stability in certain conditions but is more challenging to balance in off-road environments.

For the SAE Baja project, a zero-scrub radius was chosen to balance steering effort and stability, making the knuckle design simpler and more robust. This choice ensures durability and precision during off-road maneuvering, supporting the vehicle's ability to handle the dynamic demands of competition.

Selection Criteria

4.1.7 Brakes

Master Cylinder Diameter:

The master cylinder diameter was determined by using equation 14 which can be seen above. From the results above, the minimum bore diameter needed for the master cylinder was determined to be 0.813 inches, which is just slightly larger than $\frac{13}{16}$ of an inch. So, by using equation 14, we determined that a master cylinder with a bore diameter of $\frac{7}{8}$ inches will be used for the braking system.

Brake Pedal Ratio:

From using equation 6, we used both the 6:1 and 5:1 pedal ratio to determine which ratio would be more effective in lowering the overall pedal force required.

With BPR = 5:

$$F_{BPF} = \frac{335\ lb}{5}$$

$F_{BPF} = 67\ lb$

With BPR = 6:

$$F_{BPF} = \frac{335\ lb}{6}$$

$F_{BPF} = 55.8\ lb$

With a pedal ratio of 6:1, the brake pedal force is lowered by about 11 pounds which will make braking easier for the driver.

4.1.8 Steering

Steering Geometry:

The selection of steering geometry depends on the vehicle's intended operating conditions. Ackermann geometry is typically chosen for vehicles requiring low- to medium-speed cornering, as it reduces tire scrubbing and improves maneuverability. Anti-Ackermann is suited for high-speed applications where stability and grip are prioritized, particularly in racing scenarios with significant downforce. Parallel steering, while simpler in design, is generally less efficient due to increased tire wear and reduced cornering performance, making it less favorable for most dynamic handling applications.

4.1.9 Suspension

The suspension system was selected based on factors like shock mounting location, travel, durability, and geometry. An Upper Control Arm (UCA) mount was chosen over a Lower Control Arm (LCA) mount for the front double wishbone system. While LCA mounts are traditional, they often face clearance issues. The UCA mount, however, allows for greater suspension travel and better shock performance without interference. Suspension travel is key for absorbing impacts and maintaining wheel contact, especially on rough terrain. The UCA mount's design supports longer travel, providing stability during jumps and rough landings. Steel was selected for its strength and durability, ensuring that the suspension can endure the competition's demands while managing weight effectively. A zero-scrub radius was chosen for balanced steering and stable handling on uneven surfaces. This design minimizes steering effort while maintaining good road feedback. Overall, this approach ensures a durable, efficient suspension system that can excel in the SAE Baja's challenging conditions.

Concept Selection

Table 1:Decision Matrix

	Variants			
Subsystem	1	2	3	**Result**
Steering	Pro-Ackerman	Anti-Ackerman	Parallel	**Pro-Ackerman**
Master Cylinder	5/8 in.	7/8 in.	N/A	**7/8 in.**
Pedal Ratio	5:1	6:1	N/A	**6:1**
Shock Mounting	UCA Mount	LCA Mount	N/A	**UCA Mount**
Scrub Radius	Zero Scrub	Negative Scrub	Positive Scrub	**Zero Scrub**
T.L. Material	Steel	Titanium	N/A	**Steel**

4.1.10 Brakes

In our concept selection for the braking system, we opted for a $\frac{7}{8}$ inch master cylinder bore paired with a 6:1 brake pedal ratio to achieve an optimal balance of braking force and pedal feel. The $\frac{7}{8}$ inch bore size provides sufficient hydraulic pressure to engage the brakes effectively. Additionally, the 6:1 pedal ratio allows for enhanced leverage, enabling the driver to apply significant braking force with relatively less effort. This combination ensures a responsive and comfortable braking experience, promoting both safety and driver confidence under various driving conditions.

4.1.11 Steering

For the selection of steering geometry, Pro-Ackermann was the best option for the Baja application due to its better maneuverability, reduced tire wear, and precision in tight corners. Anti-Ackermann is more suitable for high speed, high downforce vehicles such as Formula 1. Parallel steering is a good middle ground between anti and pro Ackermann, but the tire scrub and reduced grip make pro-Ackermann a better option.

4.1.12 Suspension

For the suspension system, a UCA (Upper Control Arm) mount was selected for shock placement to optimize suspension geometry, maintain balance during compression, and ensure that the shock remains clear of other components. Additionally, a zero-scrub radius was chosen to reduce steering effort and enhance stability when navigating uneven terrain. This combination helps deliver consistent handling and a smoother ride, which is crucial for the challenging conditions encountered in Baja competitions. Steel was chosen for the suspension components, offering an ideal balance between strength and durability while keeping material costs manageable.

5 Conclusion

In summary, the Suspension, Brakes, and Steering sub-team of the NAU Baja SAE team has made significant progress over the past several weeks in preparation for the upcoming SAE Baja competition. By establishing ambitious design goals and completing crucial steps such as CAD modeling, material selection, and initial finite element analysis, we are well on our way to creating a competitive vehicle capable of excelling in various dynamic events.

As we advance to the next phases of refinement, prototyping, and testing, our focus remains on optimizing performance, ensuring safety, and adhering to budget constraints. Through collaboration with other sub-teams and the application of advanced engineering principles, we are dedicated to building a vehicle that not only meets but exceeds competition standards.

Our ultimate objective is to secure a position in the top 25% of competing teams, attract potential sponsors, and contribute to the growth and legacy of the NAU Baja SAE program. With continued commitment and teamwork, we are confident in our ability to achieve these goals and deliver a high-performing vehicle for the May 2025 competition.

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7 Appendices

Appendix A: Expanded QFD's

	Reduce Braking Distance	Minimize Pedal Force	Maximize Safety	Maximize Safety	Must Stop All Four Tires At Once	Maximize Braking Force	Pedal Components must be made from aluminum or steel	Brake System must have sufficient force to hold vehicle while engine is running	Brake Pedal Shall Be Designed for Unobstructed Travel	Brake Systems Must Have Two Independent Hydraulic Resevoirs
Reduce Braking Distance										
Minimize Pedal Force										
Maximize Safety	+									
Maximize Safety	+	+								
Must Stop All Four Tires At Once		-	+	+						
Maximize Braking Force	+	+	+	+						
Pedal Components must be made from aluminum or steel	+	+	+	+	+					
Brake System must have sufficient force to hold vehicle while engine is running	-	-	+	+	+	+	+			
Brake Pedal Shall Be Designed for Unobstructed Travel	+	+	+	-	+	+	-	+		
Brake Systems Must Have Two Independent Hydraulic Resevoirs	+	+	+	-	+	+	+	+		
Maximize Clamping Force			+	-	+	+			+	+

Correlation	
Positive	+
Negative	-
N/A	

Relationship	
Strong	9
Moderate	3
Weak	1
N/A	

Baja 2025 Brakes	
Date: 9/12/2024	
Legend	
A	NAU 2024 #44
B	Cal Poly 2024 #36
C	Cornell Univ. #73

		Technical Requirements											Customer Opinion Survey				
Customer Weights	Customer Requirements	Reduce Braking Distance	Minimize Pedal Force	Maximize Safety	Easily Serviceable	Must Stop All Four Tires At Once	Maximize Braking Force	Pedal Components must be made from aluminum or steel	Brake System must have sufficient force to hold vehicle while engine is running	Brake Pedal Shall Be Designed for Unobstructed Travel	Brake Systems Must Have Two Independent Hydraulic Resevoirs	Maximize Clamping Force	1 Poor	2 OK	3 Acceptable	4 Good	5 Excellent
9	Safety	9	9	9	3	9	9	3	9	9	9	9				A	BC
3	Affordable	-1	-1	-1	1	-3	3	9	-3	-3		3					ABC
2	Performance	1	3	3	1	3	3	1	3	9	1	9				A	BC
7	Easy Operation	9	3	9		1	3		9	9	3	9				AB	C
4	Hydraulic	1	9	3			1	-1	3	1	9	9				AB	C
3	No Overheating	9	-1	9	-1	3	9		3		1					A	BC
8	Long Pad Life	9	3	9	-1	3	3		9	3	3	1				ABC	
3	Easy to Mount	-3		3	3		2		1		3	1					ABC
2	Pass Inspection	-3	3	9	3	9	9	9	9	9	9	9					ABC
Technical Requirement Units		ft	lbf	N/A	N/A	N/A	psi	lbf	ft-lbf/s	N/A	N/A	psi					
Technical Requirement Targets		60	450				500	450	7700			90					

Figure 20: QFD for Braking System

Correlation	
Positive	+
Negative	-
N/A	0

	Reduce Turning Radius	Reduce Steering Slop	Increased stability	Proper toe	Ideal Castor Angle	Ideal Camber Angle	Ideal Steering Ratio
Reduce Turning Radius							
Reduce Steering Slop							
Increased stability	+	+					
Proper toe	+		+				
Ideal Castor Angle	+		+				
Ideal Camber Angle	+		+				
Ideal Steering Ratio	+		+				

Baja 2025 Steering	Date: 9/15/2024

Relationship	
Strong	9
Moderate	3
Weak	1
N/A	0

Legend	
A	NAU 2024 #44
B	Cal Poly 2024 #36
C	Cornell Univ. #73

		Technical Requirements										Customer Opinion Survey				
Customer Weights	Customer Requirements	Reduce Turning Radius	Reduce Steering Slop	Increased stability	Proper toe	Ideal Castor Angle	Ideal Camber Angle	Ideal Steering Ratio				1 Poor	2 OK	3 Acceptable	4 Good	5 Excellent
9	Safety	3	9	9	1	1	1	3							A	BC
3	Affordable	3	3	9	1	1	1	9					A		BC	
4	Performance	9	9	3	9	9	9	9				A				BC
7	Easy Operation	3	9	9	1	1	1	9						A	B	C
4	Reliable	1	9	9	3	3	3	9				A			B	C
3	Comfortable	3	9	3	3	3	3	9						A		BC
8	Lightweight	1		3											ABC	
3	Easy to Mount	3	3	3	3	3	3	0							A	BC
2	Pass Inspection	1	1	1	1	1	1	1								ABC
Technical Requirement Units		ft.	Degrees	N/A	in.	Degrees	in.	N/A								
Technical Requirement Targets		12	0		0.0625	10	0.25									

Figure 21:QFD for Steering

Suspension QFD
Team Members
Oliver Husmann
Ryan Key
Ryan Latulippe

Baja 25' Suspension
Date: 9/12/24

Relationship	
Strong	9
Moderate	3
Weak	1
N/A	0

Legend	
A	2024 Cornell #73
B	2024 SDSU #43
C	2024 NAU #44

		Technical Requirements										Customer Opinion Survey				
Customer Weights	Customer Requirments	Light weight	B.8.7 - all steering or suspension links exposed in the cockpit shall be shielded with a sturdy, robust, metal cover.	Optimal ride height/ground clearance	B.1.6 - Limitations - Vehicle width	Vehicle length/approach angle	Singular known replaceable failure point (bolt)	Efficiently designed knuckle	Optimal camber angles	Optimal caster angle	Optimize maximum suspension travel	1 Poor	2 OK	3 Acceptable	4 Good	5 Excellent
2	Performance/comfort	3	3	9	9	9	9	9	9	9	9		C			AB
3	Servicability/tunability	1	-1		3	9	9	9	3	3	9		C	B	A	
4	Durability	-3	3	3	3	9	9	9			3		C	B		A
6	Affordable	3	-1				3	9				C		B	A	
5	Ease of Fabrication	1	1				3	9	1	1	1		C	AB		
7	Aesthetics		3	9	9	3	1	9	3	3	9				BC	A
1	Pass Tech		9		9											ABC
Technical Requirement Units		lbs	mm	In.	In.	In.	Psi.	lbs, Psi, in., hrs.	Degrees	Degrees	In.					
Technical Requirement Targets		<50	<6.35	12-16	64	48-60					12-16					

Figure 22: Part 1 of QFD for Suspension

	Light weight	B.8.7 - all steering or suspension links exposed in the cockpit shall be shielded with a sturdy, robust, metal cover.	Optimal ride height/ground clearance	B.1.6 - Limitations - Vehicle width	Vehicle length/approach angle	Singular known replaceable failure point (bolt)	Efficiently designed knuckle	Optimal camber angle	Optimal caster angle	Optimize maximum suspension travel
Light weight										
B.8.7 - all steering or suspension links exposed in the cockpit shall be shielded with a sturdy, robust, metal cover.	+									
Optimal ride height/ground clearnace										
B.1.6 - Limitations - Vehicle width	-		+							
Vehicle length/approach angle			+							
Singular known replaceable failure point (bolt)	+									
Efficiently designed knuckle	+					+				
Optimal camber angle			+							
Optimal caster angle			+							
Optimize maximum suspension travel			+		+			+	+	

Technical Requirements

Correlation	
Positive	+
Negative	-
N/A	0

Figure 23: Part Two of QFD for Suspension

SAE Baja Chassis Team

Initial Design Report

Ryan Carley: Subteam Lead & Front End Specialist

Wyatt Walker: CAD Manager & Cockpit Specialist

Charles Anderson: Web Design, Manufacturing, & Rear End Specialist

Fall 2024-Spring 2025

Project Sponsors: Gore, H&S Field Services, Poba Medical, Harsh Co, Anonymous Donor

Faculty Advisor: David Willy

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

This report documents work done by the NAU Baja Chassis team from August 26th- October 18th, 2024. The goal of this project is to design and build an off-road vehicle using fundamental engineering principles taught in the Northern Arizona Mechanical Engineering program as well as engineering principles practiced by SAE. May 1st-5th the team plans to compete amongst other schools, from across the nation and internationally, in the SAE Baja event located in Marana, Arizona. The car will compete in several events such as Hill Climb, Endurance, Acceleration, Suspension, Dynamic, and overall scoring.

The entire NAU Baja Team is composed of 15 team members split among subteams. This subteam, the chassis team, is responsible for the safety of the driver and ensuring that the frame is compliant with the rules and regulations set forth by the rulebook provided by SAE. This document will cover basic background information about the project and event, as well as design requirements set by SAE and the team's own personal requirements based on goals established by the team. This document will also cover research completed by the team that will be implemented into the design of the chassis. This will include mathematical calculations and benchmarking criteria. The decision-making process will also be documented, through the concept generation and selection criteria. This report marks the quarterway mark in the semester with plenty more to do in upcoming months.

TABLE OF CONTENTS

Contents

1 BACKGROUND

This chapter of the report will cover the research and decision-making process of the project. It will go over the project requirements from both the customer and engineering standpoints. Both of these criteria will be organized in a QFD diagram that shows the relationship and importance to one another. The document will also cover different types of benchmarking components and determine why some designs were successful while others were not. Sources and information that will be used in the project, will also be discussed as well as their relevance and importance. These sources will influence the team's calculations, which will also be discussed in this report. Lastly, this report will cover design topics and iterations, as well as the different criteria that alter the designs.

1.1 Project Description

For the SAE Baja 2025 NAU capstone project the objective is to design, fabricate, and perform in the competition that will be held in Arizona in 2025. As a capstone project, the team was tasked with reaching out to sponsors and managing the funds raised. The design constraints have been defined in the SAE Baja rulebook for the 2025 competition. The team has to pass a technical inspection to make sure that the vehicle is safe and meets guidelines. The project is important for the team as mechanical engineering seniors to be able to design an automobile with design constraints and goals to achieve.

1.2 Deliverables

The main deliverable for the team is to provide a well-built chassis that is guaranteed to pass SAE technical inspection prior to competition. This inspection ensures that the chassis was designed to adhere to the rule book. If the chassis does not pass technical inspection, the entire team will not be able to compete in the events. Another deliverable to be considered is driver safety. The chassis needs to be designed in such a way that it ensures that the driver will be unharmed in a variety of situations and collisions.

1.3 Success Metrics

The team has defined that the overall success of the project is dependent on how well the car performs in competition. Not only is one of the goals to pass technical inspection, but the team would like to place high in all the events in comparison to the other teams at the competition. Since the frame is vital to other subteams in terms of drivetrain, suspension, and steering, their ability to perform well is dependent on the overall chassis design. So therefore the entire team's success is considered to be the metric for success for the chassis.

2 Requirements

In this chapter of the report, the customer and engineering requirements are discussed as well as the QFD that was generated by the team. The more general requirements are from the customer requirements and the more specific requirements are defined by the SAE rulebook and used for the engineering requirements.

2.1 Customer Requirements (CRs)

The customer requirements for the chassis team are to prioritize safety, durability, performance, and passing the technical requirements. These are basic requirements that the team has set to make sure that the frame is the best that it can be. The team has also set other requirements for affordability, comfort for the driver, aesthetics, balanced weight, and ease of fabrication. These requirements are more secondary than the previous ones listed and allow for more creative design and flexibility with other subteams.

2.2 Engineering Requirements (ERs)

The frame team is primarily in charge of passing tech inspection and making sure that the frame meets the design constraints defined by SAE. The engineering requirements will alter the overall design of the frame and will help to prioritize the main goal for the team to pass the technical inspection. These engineering requirements are from the rule book provided by SAE and are shown in Table 1 in Appendix A.

2.3 House of Quality (HoQ)

Figures 1 and 2 are the QFD that the team generated based on the requirements from the rulebook and some of the requirements that the team wanted to accomplish. The customer requirements prioritize the safety, performance, and durability of the frame. Most of the engineering requirements have to do with the clearances that are required and the length of certain members that the rulebook specifies. As the team prioritizes these requirements the goals to pass technical inspection and perform well in the competition will be accomplishable.

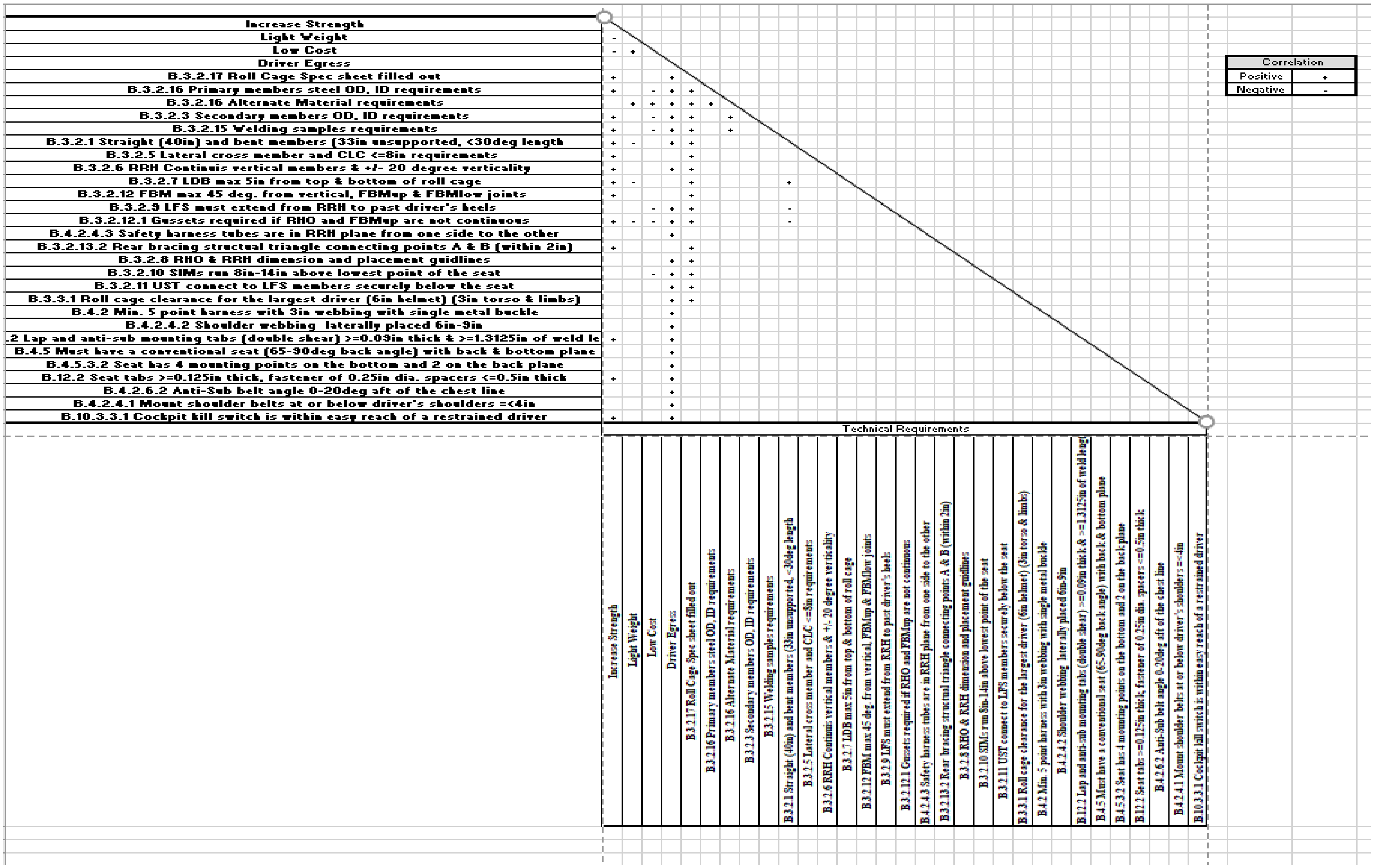
Frame QFD
Team Members
Ryan Carley
Charles Anderson
Wyatt Walker

Baja 25' Frame
Date: 9/9/24

Relationship		Legend	
Strong	9	A	NAU 2024 #44
Moderate	3	B	Cornell 2024 #73
Weak	1	C	ETS 2024 #27

		Technical Requirements																															Customer Opinion Survey				
Customer Weights	Customer Requirments	Increase Strength	Light Weight	Low Cost	Driver Egress	B.3.2.17 Roll Cage Spec sheet filled out	B.3.2.16 Primary members steel OD, ID requirements	B.3.2.16 Alternate Material requirements	B.3.2.3 Secondary members OD, ID requirements	B.3.2.15 Welding samples requirements	B.3.2.1 Straight (40in) and bent members (33in unsupported, <30deg length	B.3.2.5 Lateral cross member and CLC <=8in requirements	B.3.2.6 RRH Continuis vertical members & +/- 20 degree verticality	B.3.2.7 LDB max 5in from top & bottom of roll cage	B.3.2.12 FBM max 45 deg. from vertical, FBMup & FBMlow joints	B.3.2.9 LFS must extend from RRH to past driver's heels	B.3.2.12.1 Gussets required if RHO and FBMup are not continuous	B.4.2.4.3 Safety harness tubes are in RRH plane from one side to the other	B.3.2.13.2 Rear bracing structural triangle connecting points A & B (within 2in)	B.3.2.8 RHO & RRH dimension and placement guidlines	B.3.2.10 SIMs run 8in-14in above lowest point of the seat	B.3.2.11 UST connect to LFS members securely below the seat	B.3.3.1 Roll cage clearance for the largest driver (6in helmet) (3in torso & limbs)	B.4.2 Min. 5 point harness with 3in webbing with single metal buckle	B.4.2.4.2 Shoulder webbing laterally placed 6in-9in	B.12.2 Lap and anti-sub mounting tabs (double shear) >=0.09in thick & >=1.3125in of weld leng	B.4.5 Must have a conventional seat (65-90deg back angle) with back & bottom plane	B.4.5.3.2 Seat has 4 mounting points on the bottom and 2 on the back plane	B.12.2 Seat tabs >=0.125in thick, fastener of 0.25in dia. spacers <=0.5in thick	B.4.2.6.2 Anti-Sub belt angle 0-20deg aft of the chest line	B.4.2.4.1 Mount shoulder belts at or below driver's shoulders =<4in	B.10.3.3.1 Cockpit kill switch is within easy reach of a restrained driver	1 Poor	2 OK	3 Acceptable	4 Good	5 Excellent
2	Performance	9	9	-3	9		9	3	9					1	3	3	9		3	1	1	1		1								3		A		C	B
3	Safety	9	-3	-3	9	9	9	9	9	9	9	9	9	9	9	9	9	9	9	9	9	9	9	9	9	9	3	3	9	9	9	9					ABC
5	Durability	9	-3	-1	1		9	9	9	9	9	9	3	9	9	9	9	3	3	9	9	3				3	1	1	3						A		BC
4	Affordable	-3	3	9	1		9	9	9		3		1	3	1	3	1	1	1	3	1	1		1		1	3	3	1	1			C	B	A		
8	Comfort	1	1	-3	9								9			3		9		9	9	9	9	9	9		9		1	9	9		A			B	C
7	Ease of Fabrication	-3	3	-1	-3		3	3	3	9	3	3	3	3	3	3	-3	-1	-3	-3	1	-3	-3	1	1	-1	-3	-3	-3		-3	-1		AC	B		
9	Aesthetics	1	1	-3	-1					3	3	3	9	3	9	3	9		3	9	9		9	3			3				1				A	B	C
1	Pass Techs	9	-1	-1	9	9	9	9	9	9	9	9	9	9	9	9	9	9	9	9	9	9	9	9	9	9	9	9	9	9	9	9				A	BC
6	Balanced Weight	-3	-1	-3	1		3	3	3		3	3	3		3	1	1		9	3	1	3	3												A	B	C
Technical Requirement Units		psi	lbs	$	Seconds	N/A	Inches	N/A	Inches	N/A	Inches	Inches	N/A	Inches	N/A	N/A	N/A	N/A	N/A	N/A	Inches	N/A	Inches	Inches	Inches	Inches	N/A	N/A	Inches	N/A	Inches	N/A					
Technical Requirement Targets			100	2500	5		1,0.065,0.11		1,0.035		40,33	8		5							8,14		6,3	3	6,9	0.09,1.3125			0.125,0.25,0.		4						

Figure(1): QFD

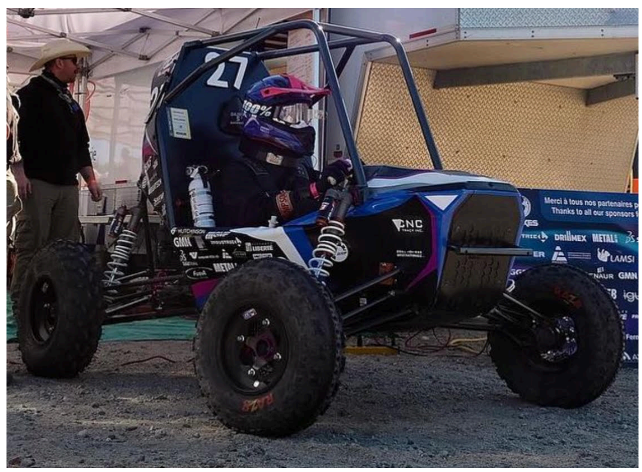


Figure(2): Top of The QFD

3 Research Within Your Design Space

3.1 Benchmarking

For the benchmarking research, the team took a look at some previous designs from different schools and compared them to find their strengths and weaknesses. The three different designs that were taken into consideration were NAU #44 from 2023-2024. ETS #27 from 2023-2024, and Cornell #73 from 2023-2024. Both ETS and Cornell mounted their front shocks to the front bracing members, while NAU mounted their shocks lower on the frame. This Style of mounting for ETS and Cornell gives the suspension more vertical travel. NAU's front end also is far more cramped and is lower in the car compared to both ETS and Cornell. Based on the images as well, the main cockpit seems to be wider on both ETS and Cornell, compared to NAU. This gives the driver more room for maneuverability and is more comfortable with extra room. NAU also has a higher seating position compared to ETS and Cornell and this creates a higher center of gravity for the NAU car. These benchmarking selections are valid because of how each of these vehicles performed in the 2023-2024 competition. Cornell placed 1st overall and ETS placed 2nd overall, while NAU placed 33rd. Taking this into consideration, the chassis team would like to incorporate these elements in the design, to help improve the overall performance of the design.

Figures(3-5): NAU #44, ETS#27, & Cornell #73

3.2 Literature Review

3.2.1 Charles:

- The Procedure Handbook of Arc Welding [1]

  Chapter 2 Designing for Arc Welding: This chapter discusses some characteristics that should be considered when designing a structural system that requires welding. It discusses key topics of being able to satisfy stiffness and strength requirements if a torque will be applied, as well as being able to locate when and where failure will most likely occur. This chapter also covers information such as material selection and choosing common metals compared to specialized materials. This chapter emphasizes room for improvement and how to make designs better and easier through manufacturing processes such as bending, notching, and coping. All of these factors will be taken into consideration for the design and will make the manufacturing process easier.

- Material Science and Engineering [2]

  Chapter 11 Applications and Processing of Metal Alloys: This chapter talks about the benefits of using different alloyed metals for different applications. For the chassis, the team plans to use medium carbon steel in 4130 Chromoly tubing. According to Table 11.2a [] in the textbook, the tubing has a composition range of 0.8-1.10% Chromium and 0.15-0.25% Molybdenum. The textbook also mentions the advantages and disadvantages of using medium-carbon steel over low-carbon steel. Mild steel has higher strength and toughness, but is also less ductile and in most cases requires heat treatment. Since the team wanted a more rigid frame, this information from the textbook influenced the decision to go with a mild steel like 4130 Chromolly.

- Effect of Preheating Temperatures On Impact Properties [3]

  This paper discusses the differences of preheating 4130 Chromolly tubing at different temperatures, before welding, to increase the maximum amount of impact energy that the material can handle. In this paper, the authors found that 250 degrees Celsius was optimal for increasing the impact energy. At that temperature, the 100mm X 75mm X 15mm test sample could absorb 50 J. This is more energy compared to 200 degrees Celsius and 150 degrees Celsius. This paper showed ways that the team could improve the material properties of the 4130 by just preheating the metal before welding the members.

- SAE Baja Final Proposal Report [4]
  This report is from the NAU SAE Baja Team of 2021, which placed top 5 overall. This document is a summary of their entire process, giving the team some insight into different methods and ideas. This report shows some FEA calculations for the frame, which in turn will give some metrics to aim for. This document also shows how other parts will mate up in the frame. This document is a good reference and a great example of how to execute this project successfully.

- Stress analysis of a roll cage[5]
  This YouTube video goes over the basics of applying and simulating stress calculations upon impact on a roll cage. Given that the geometry of the roll cage in the video is not the same as the roll cage being designed, the team will have to use these fundamentals from the video and apply them to the design, while coming up with new and other ways of simulating these impacts.

- Designing a Roll Cage in Solidworks [6]

  This YouTube video is a great instructional tool that teaches how to build a roll cage from scratch using SolidWorks. The video teaches the importance of using different reference planes for different geometries. The video also touches a little bit on how to use the weldment feature on SolidWorks. This video was a huge help in figuring out how to make the CAD drawing for the chassis, as not a single team member had any prior experience in doing so.

- ASTM- AISI 4130 Steel [7]

  This standard by the American Society of Testing and Materials specifies the material properties of 4130. This standard gives a material characteristic such as density, yield strength, modulus of elasticity, Poisson's ratio, etc. This is a great resource for the team to figure out material properties for calculations.

3.2.2 Wyatt:

- Shigley’s Mechanical Engineering Design [8]
  - Chapter 2 section 2-1 Material Strength and Stiffness

  This resource was useful for choosing an alternative material for the frame over the defined 1018 steel that is given in the textbook. The team found it necessary to look at other materials that can be lighter and more available on the market. An equivalency calculation was done to prove that the alternate material was viable with our rules.

- Machinery’s Handbook [9]
    - Bending Sheet Metal pg.1346-1353

  This section of the Machinery’s Handbook shows some factors that can be useful when the team is bending and coping with the frame. It shows useful calculations that can be used by the team to save materials and help avoid mistakes.

- Design and Optimization of Mini Baja Chassis [10]

  This source is an article that goes over an FEA of a Baja chassis and shows the results of impacts that were in a few different locations. This could be useful for the team to get an idea of what can be acceptable for displacement and stress outcomes of an FEA.

- Design, analysis, and optimization of all-terrain vehicle chassis ensuring structural rigidity (6 Finite Element Analysis) [11]

  This article shows FEAs on a Baja chassis and shows where the fixed points are and explains its thought process through the simulations. The FEAs were done with ANSYS and went through a front impact, side impact, and rear impact. These examples can prove useful when the team does an FEA on our frame design.

- Plastic Deformation Analysis in Tube Bending [12]

  This source goes through some calculations of bending tubes and can help the team when it comes to bending the tubing so that material isn’t wasted and fabrication is more efficient and organized with plenty of resources.

- 2024 Baja SAE Roll Cage Doc. Package. Pg. 8 [13]

  This is a document provided by SAE for the Baja competition and has the equivalency calculations in it. They are required documents for the competition if an alternative material is used which for our case will be important. The calculations for the material of the tubing were figured out prior to the team buying the tubing.

- Techniques to improve weld penetration in TIG welding [14]

  This source is a guide for what to look for when doing TIG welding. This includes what are the best practices and what to look for in a good weld. This will be useful for the team since there is a welder in the team that will help with fabrication as well as the other members of the team being aware of what constitutes a good weld.

3.2.3 Ryan:

- Engineering Analysis with ANSYS Software (Ch.3) [15]

  This chapter highlights two-dimensional & three-dimensional stress analysis using ANSYS. While the ANSYS simulations used in the book are outdated, the hand calculations will still prove to be valuable for first-iteration calculations of stress concentrations.

- The Automotive Chassis (Second Edition) (Ch. 6) [16]

  Chapter six in this book provides a step-by-step process for finding the center of mass of a frame. The center of mass is vital to the frame's success because the frame needs to be as balanced as possible to assist with steering. Additionally, by finding the center of mass it will make calculating braking and acceleration capacity and the climbing ability much more accurate.

- ASTM A500/A500M-23 [17]

  This standard explains the ASTM standard for inspecting and welding steel tubing. Further, it explains that the tubing must go through a flattening test, flaring test, and wedge crush test before being available for purchase.

- Analysis of Roll Cage and Various Design Parameters of an All-Terrain Vehicle (Baja) [18]

  This paper outlines the chemical composition of 4130 Chromoly steel and why it is the best option for the Baja frame tubing. Additionally, it highlights equations for solving the forces that the car would need to withstand to be used in FEA simulations.

- Design, analysis, and optimization of all-terrain vehicle chassis ensuring structural rigidity (5. Calculations) [19]

  This paper shows detailed instructions for calculating the forces that will be used in simulations to find stress concentrations and displacement. It also shows how to effectively summarize the results of the simulations in an organized manner.

- Static and Modal Analysis of All Terrain Vehicle Roll-Cage [20]

  This paper is used to demonstrate how to calculate very specific impacts. For example, bump impacts and torsional impacts are both necessary to ensure the safety of the driver but are very complicated to derive. This source lays out each variable and how to accurately simulate each scenario.

- Introduction to Simulations (FEA) [21]

  This source is a YouTube video that goes through the basics of performing an FEA using SolidWorks Simulation. The creator of this video, Aryan Fallahi, gives a step-by-step explanation of the interface and how to accurately set up and run a simulation.

- Bentley Garner Shares Tips for Successfully Welding Chromoly Tube [22]

  In this YouTube video Bentley Garner, an experienced welder, shows how to properly clean and prep Chromoltubesbe for welding. This video will prove valuable once the frame is ready to be welded. Additionally, he explains what type of welding wire is needed to get good penetration on the welds.

3.3 Mathematical Modeling

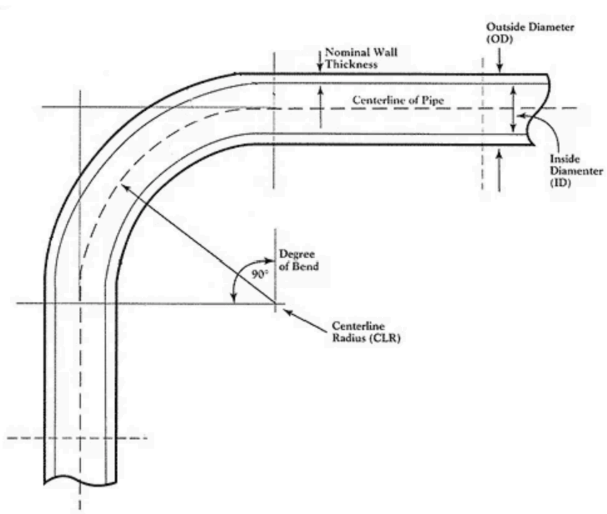
3.3.1 Charles:

This first Mathematical modeling will be used to calculate the full amount of length of the f tube needed for members with bends in them. The real roll hoop, the member directly positioned behind the driver separating the engine and the cockpit, is required to be made from one continuous tube. The chassis team has designed it in a way where there are 4 bends, two of different angles. In order to figure out the total length of the f tube needed to make that entire member, the team needed to calculate the arc length of each of these bends. In order to do so, the team used the following equations:

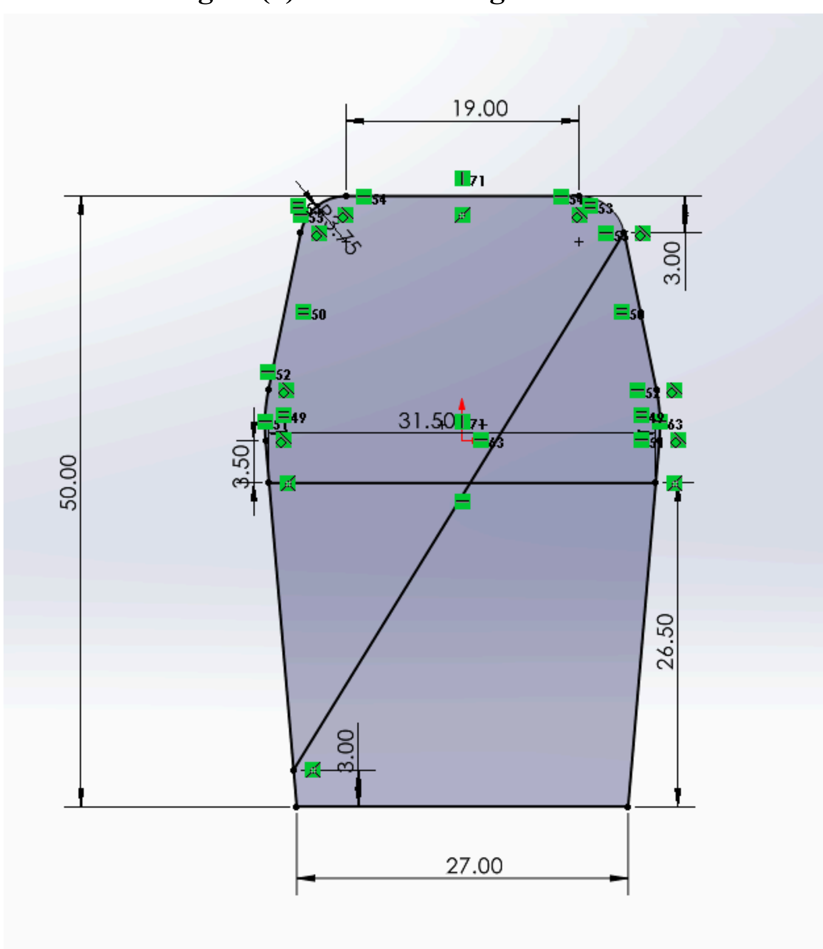
$$rad = \theta \cdot \frac{\pi}{180} \qquad L = rad \cdot \; clr$$

Converting degrees to radians Length= radians * centerline radius

Figure 6 helps visualize the calculations.



Figure(6): Tube Bending Illustration



Figure(7): Rear Roll Hoop

The first bend is 16.02 degrees, convert this to radians

$16.02 \ * \ \Pi/180 \ = \ 0.28\, radians$

Then multiply by the C.L.R.

$0.28 \ * \ 14.7 \ = 4.12$ ”

Therefore the total length needed for that bend is 4.12 inches. We need to repeat that calculation but using a 78.46-degree bend and a CLR of 3.75 to get a total length of 5.14 inches. To figure out how much tubing we need to complete this entire member we add the length of straight and bent members together like so and divide by 12 to convert to feet.
$(2 * 5.14 \ + \ 2 * 4.12 \ + \ 2 * 30 \ + \ 2 * 13.17 \ + \ 19)/12 = \ 10.32\, ft$

As simple of a calculation as this is, it determined the purchasing process for materials, requiring the team to purchase tubing in 12 ft quantities to ensure that there was enough material to successfully bend the RRH member.

3.3.2 Wyatt:

The mathematical modeling shown in Figure 8 was for the equivalency calculations that are required by SAE for teams using different materials than what is given in the rulebook. The rulebook states that the frame must be made of steel of 0.18% carbon content with an outside diameter of 1 inch and a wall thickness of 0.118 inches. The wall thickness of the tubing can be as low as 0.063 inches as long as the bending stiffness and bending strength are equivalent to or higher than the 1018 steel with the 0.118-inch wall thickness. The team wanted to use 4130 chromoly steel for its weldability, availability on the market, and lower wall thickness so that the frame is lightweight. The calculations prove that the 4130 chromoly will serve as a stronger and lighter option than the 1018 steel with a stronger bending stiffness and strength.

1018 Steel

OD = 25mm = 0.984in
Wall Thickness = 3mm = 0.118in
ID = 19mm = 0.748in
E = 205 GPa = 29733200 psi (Modulus of Elasticity for all steels)
Sy = 365 MPa = 52939.6 psi
C = OD/2 = 12.5mm = 0.492in (Distance to neutral axis)

Bending Stiffness (Kbreq)

I = Second moment of area for the structural cross section
I = pi/64*(OD^4-ID^4)
I = pi/64*(0.984^4-0.748^4)
I = 0.0308in^4

Kbreq = E*I
Kbreq = 29733200psi * 0.0308in^4
Kbreq = 915,782.56 lbf*in^2

Bending Strength (Sbreq)

Sbreq = (Sy*I)/C
Sbreq = (52939.6 psi*0.0308in^4)/0.492in
Sbreq = 3,314.11 lbf*in

4130 Chromoly Steel

OD = 1.25in
Wall Thickness = 0.065in
ID = 1.12in
E = 205 GPa = 29733200 psi (Modulus of Elasticity for all steels)
Sy = 63100 psi [2]
C = OD/2 = 0.625in (Distance to neutral axis)

Bending Stiffness (Kbreq)

I = Second moment of area for the structural cross section
I = pi/64*(OD^4-ID^4)
I = pi/64*(1.25^4-1.12^4)
I = 0.0426in^4

Kbreq = E*I
Kbreq = 29733200psi * 0.0426in^4
Kbreq = 1,266,634.32 lbf*in^2

Bending Strength (Sbreq)

Sbreq = (Sy*I)/C
Sbreq = (63100 psi*0.0426in^4)/0.625in
Sbreq = 4,300.9 lbf*in

Figure(8): Tubing Equivalency Calculations

3.3 .3 Ryan:

Estimated Weight of the Frame:

$$Weight = Density \times Volume \qquad V_{tube} = V_{outer} - V_{inner}$$

Primary Members:

$$OD = 1.25\ in$$

$$Wall\ Thickness = .065in$$

$$Density = .284\ \frac{lbs}{in^3}$$

$$Lenght \approx 49ft$$

$$\left(\pi \times \left(\frac{1.25in}{2}\right)^2 \times (49ft \times 12in)\right) - (\pi \times (.56in)^2 \times (49ft \times 12in)) = 143.28\ in^3$$

$$143.28\ in^3 \times .284\frac{lbs}{in^3} = 40.73\ lbs$$

Secondary Members:

$$OD = 1\ in$$

$$Wall\ Thickness = .035in$$

$$Density = .284\ \frac{lbs}{in^3}$$

$$Lenght \approx 36ft$$

$$\left(\pi \times \left(\frac{1in}{2}\right)^2 \times (36ft \times 12in)\right) - (\pi \times (.465in)^2 \times (36ft \times 12in)) = 43.53\ in^3$$

$$43.53\ in^3 \times .284\frac{lbs}{in^3} = 12.35\ lbs$$

$$Total\ Weight \approx 54\ lbs$$

Not including weight of welds

Figure (9): Weight of Frame

This mathematical model shown in Figure 9 calculates the frame's weight. The frame needs to be as light as possible without compromising the integrity of the design. The calculations above are overestimated because the numbers are rounded up. However, the total estimated weight is still acceptable. The intention of keeping the frame as lightweight as possible is because the other sub-components of the car will increase the weight of the car significantly. The lighter the car is when it comes to competition the faster it will be overall. It is important to acknowledge that this estimated weight does not account for the weight of the welds.

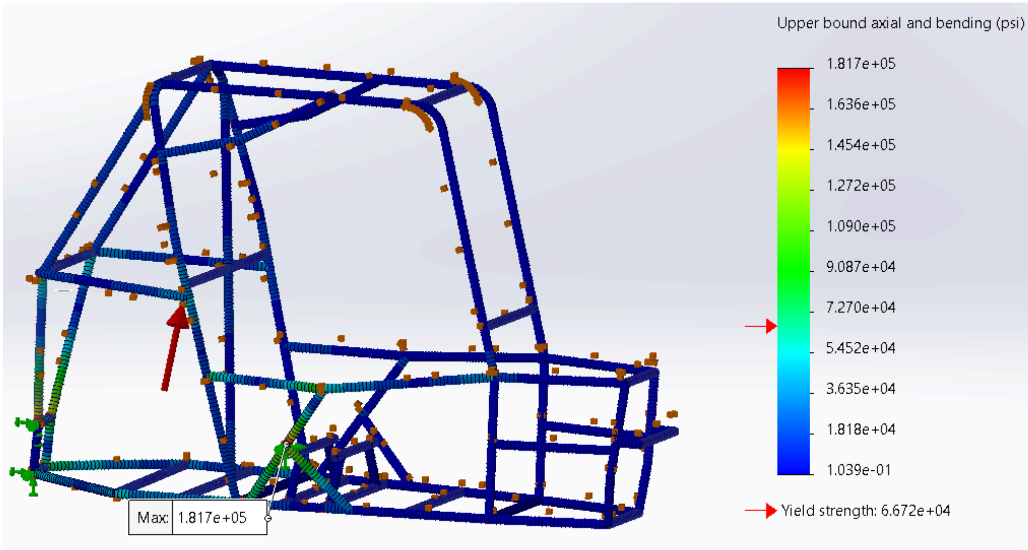


Figure (10): FEA of Stress Analysis, Jumping the Car and Falling 10 ft Then Landing on One Rear Wheel

The max stress for this simulation shown in Figure 10 occurs where the trailing arm is mounted to the side impact member supports. The max stress at that point is 1.817x10^5 psi. Considering the yield strength of 4130 Chromoly Steel is 6.672x10^4 psi this scenario would permanently deform this member and possibly break the member. With this knowledge, we will refine the design of these support members to withstand the forces that the frame would see for this specific scenario.

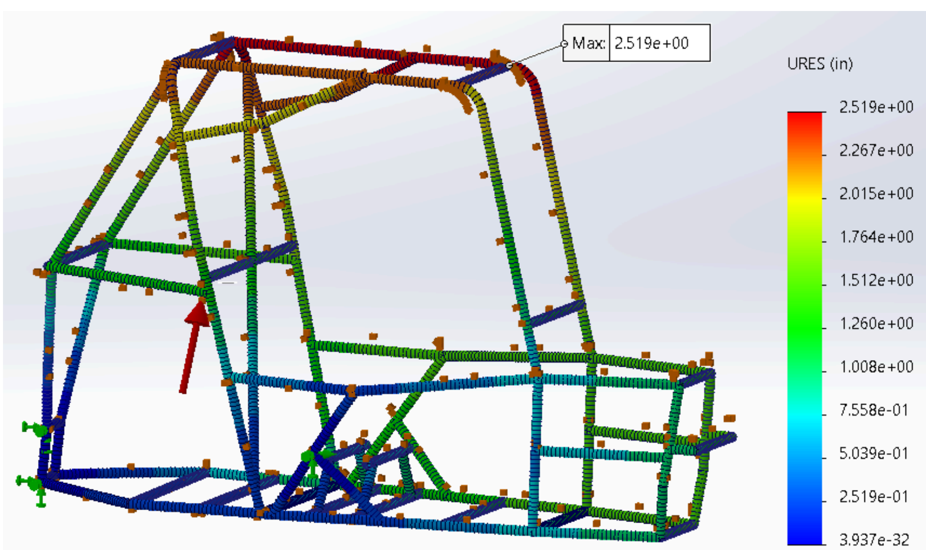


Figure (11): FEA of Displacement Analysis, Jumping the Car and Falling 10 ft then Landing on One Rear Wheel

The max displacement of the frame for this scenario shown in Figure 11 is 2.5 inches on the front bracing member. This displacement is extremely high. This displacement value will go down considerably once exact suspension mounting points are defined, the suspension mounts used in the simulation are estimated to be within four inches of the final locations. However, it is very crucial to know where the weakest parts of the car will be. In this case, it is the bend on the front bracing member. The gussets on the front bracing member are also not in their final position, with the information from this simulation the exact locations of the gussets can be finalized.

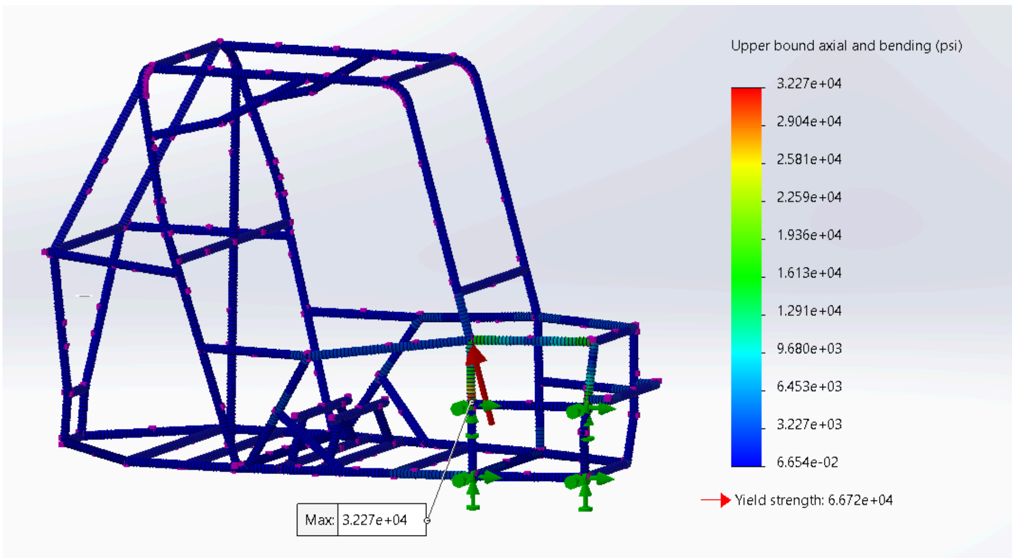


Figure (12): FEA of Stress Analysis, Jumping the Car and Falling 10 ft then Landing on One Front Wheel

Similar to the previous scenario except the whole car landed on one front wheel after falling from a 10 ft drop. The stress analysis from Figure 12 shows a very high concentration of stress where the upper control arms would be mounted. The max stress at that point is 3.227x10^4 psi which is less than the yield strength but it is too close to be comfortable with the supports in the front. It is important to acknowledge that the point force applied to the front bracing member is not exactly how the shock will be forced into the frame but the position in the simulation is within four inches of the final location of the shock mount.

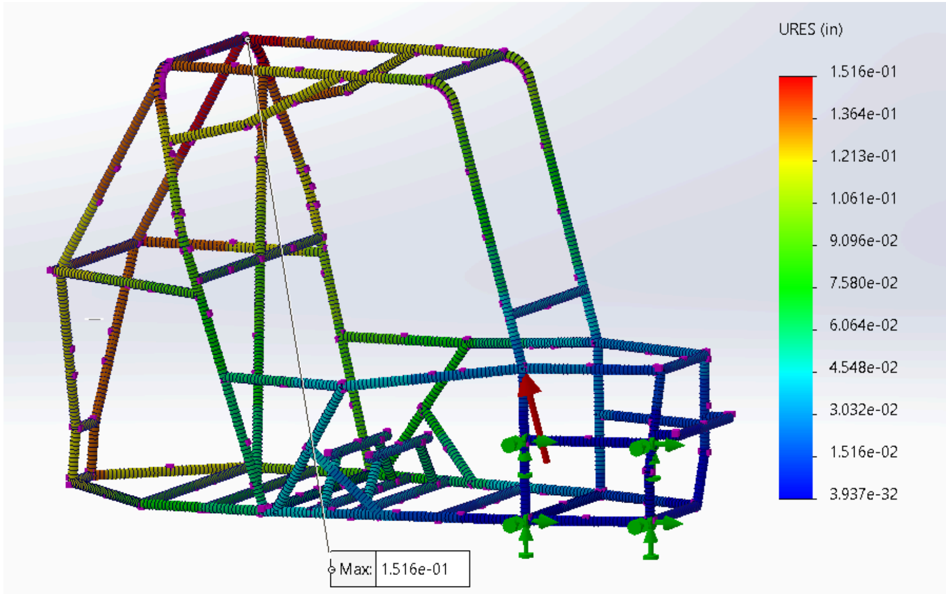


Figure (13): FEA of Displacement Analysis, Jumping the Car and Falling 10 ft then Landing on One Font Wheel

The simulation in Figure 13 highlights the max displacement location of the frame if the car were to fall 10 ft and land on the right front wheel. The design of the frame was able to dissipate the force, redirecting it toward the rear of the car. The max displacement is .152 inches this is an acceptable displacement given the magnitude of the force is 2000 lbf.

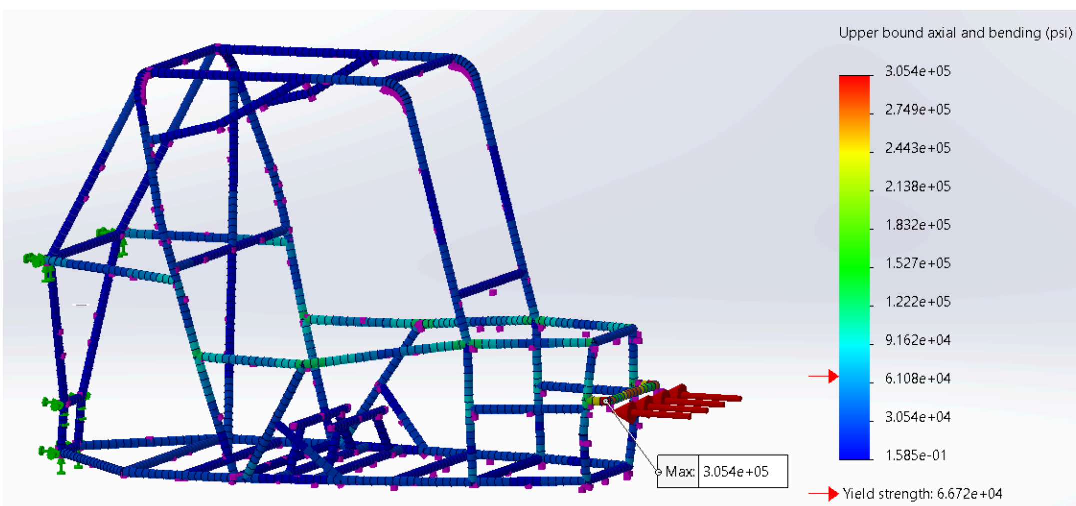


Figure (14): FEA of Stress Analysis: Hitting a Wall Going 25 mph

Figure 14 is a stress analysis simulation representing the car traveling at 25 mph and hitting a barrier or another stationary obstacle. The max stress is on the tow bar with a value of 3.05x10^5 psi however, the stress quickly dissipates through the front end of the frame. While the towbar would break the structural rigidity of the front of the frame would remain the same.

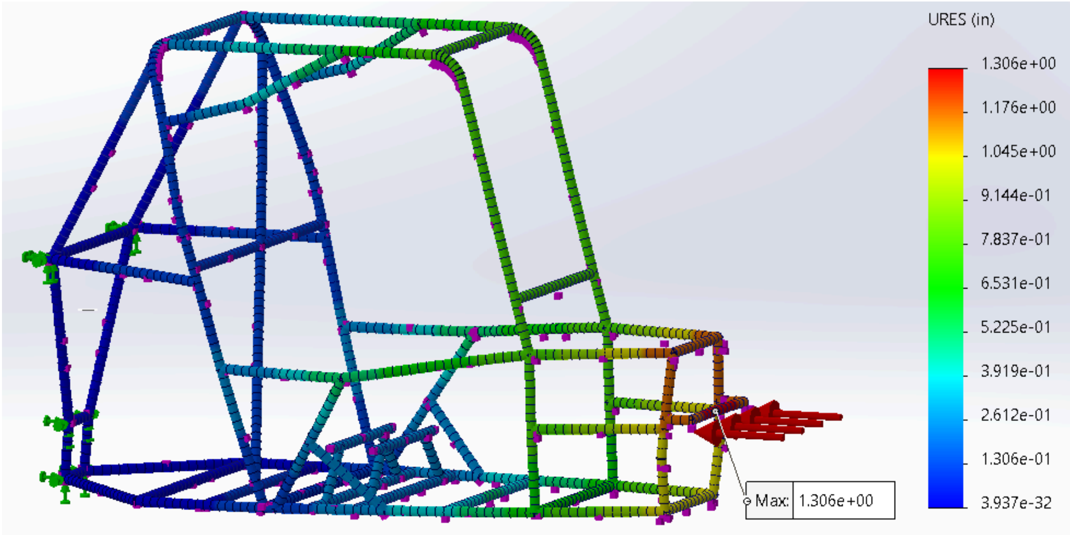


Figure (15): FEA of Displacement Analysis: Hitting a Wall Going 25 mph

Figure 15 shows the deformation of the previous simulation, the results are similar to the results in Figure 14. The towbar absorbs the most force which causes the max deformation to be on the towbar which is 1.306 inches.

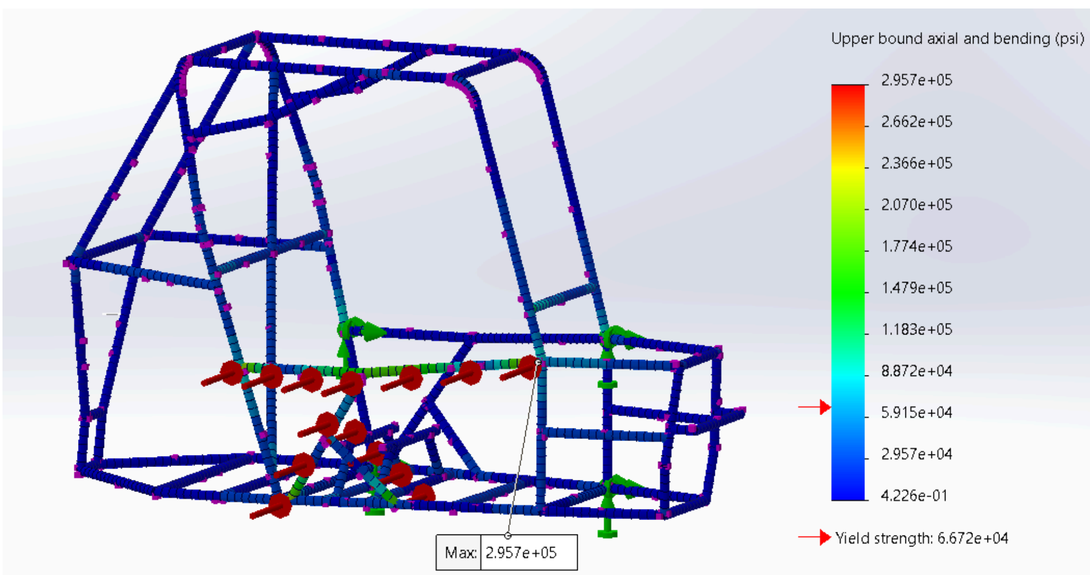


Figure (16): FEA of Stress Analysis: Car Getting T-Boned by Another Driver going 25 mph

The final simulation on the frame for this report is a scenario where the car gets T-Boned by another competitor moving 25 mph. While this scenario is unlikely the frame needs to be designed for worst-case scenarios and successfully protect the driver. The max stress of this scenario occurs where the Side Impact Member (SIMmeetset the Front Bracing Member (FBM) with a value of 2.957x10^5. This joint would break however, the strength of the welder does not account for this simulation.

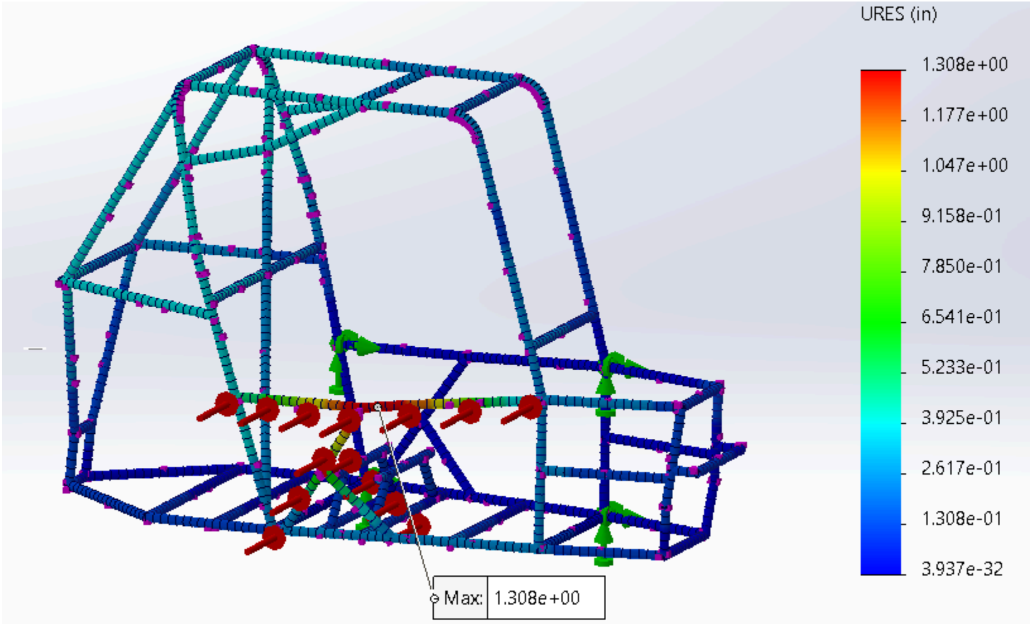


Figure (17): FEA of Displacement Analysis: Car Getting T-Boned by Another Driver going 25 mph

In the same scenario as the previous the max displacement occurs on the SIM at 1.308 inches. This deformation would not affect the driver however, the car would more than likely need to be taken out of the competition.

4 Design Concepts

4.1 Functional Decomposition

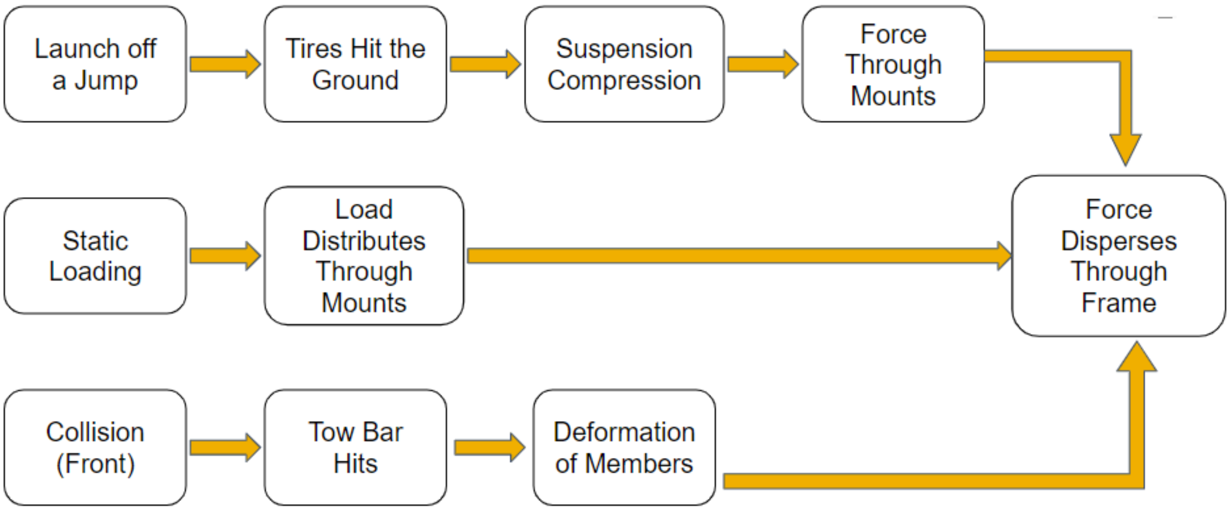


Figure (18): Functional Model

Figure 18 shows a functional model of how the frame should react to the three given scenarios on the left side. Essentially, the main objective of the frame is to effectively disperse loads throughout various members to minimize stress concentrations on any given member. For example, the first scenario is the car launching a jump, the tires would then hit the ground, the suspension would become fully compressed and the forces would be translated through the suspension mounts and into the different members of the frame.

4.2 Concept Generation

For the concept generation, the team took a direct compare and contrast approach, looking at two different designs and ideas and listing the advantages and disadvantages of each. One of the first design iterations the team looked at was a front-braced roll cage vs a rear-braced roll cage.

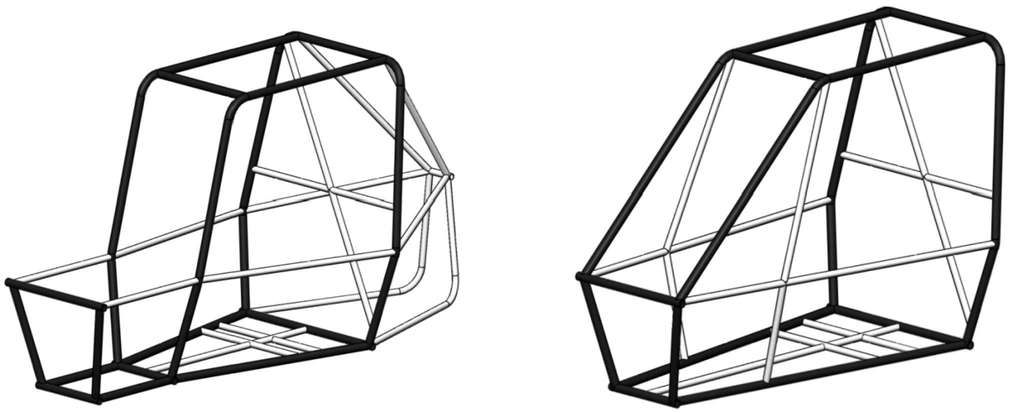


Figure (19): Rear Braced frame (left) Front Braced Frame (right)

The rear braced frame provides a lighter weight frame and creates a more open cockpit by having the engine at the rear. With the front braced frame, the engine is usually mounted in the front end of the car, this gives the car a better weight distribution from front to rear however, these frames are a little bit heavier and more cramped in the front.

For the second concept generation, the chassis team also compared the pros and cons of in-board vs out-board brakes for front brakes.

Figure (20): In-board Brakes

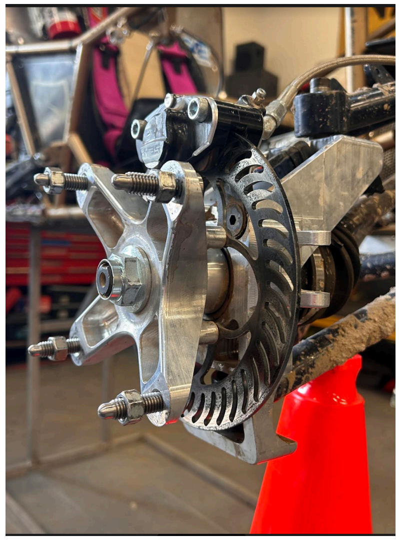


Figure (21): Out-Board Brakes

Another concept the frame team needed to decide on was whether or not the pedals should be floor-mounted or hanging pedals. Both options will affect the frame and the ergonomics of the driver.

Figure (22): Floor Mounted Pedals

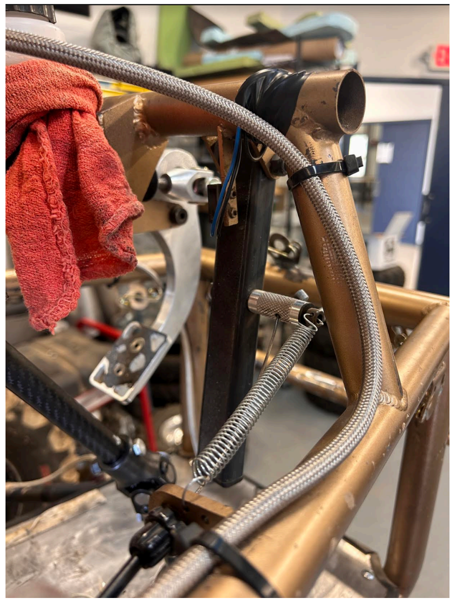


Figure (23): Hanging Pedals

The final concept the frame team decided on is the shape of the Side Impact Members (SIM). The two options were having the SIM bend more inward versus angling them outward.

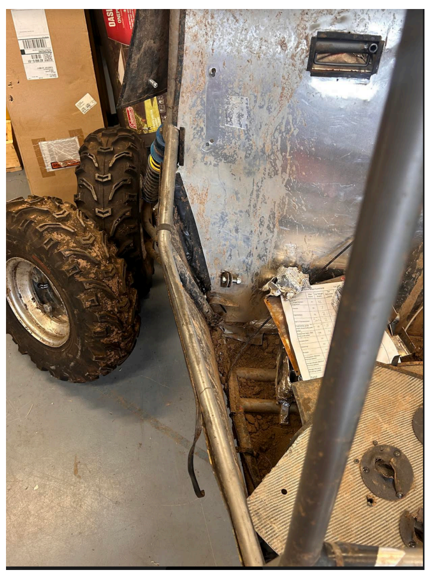


Figure (24): SIM Angled Outward

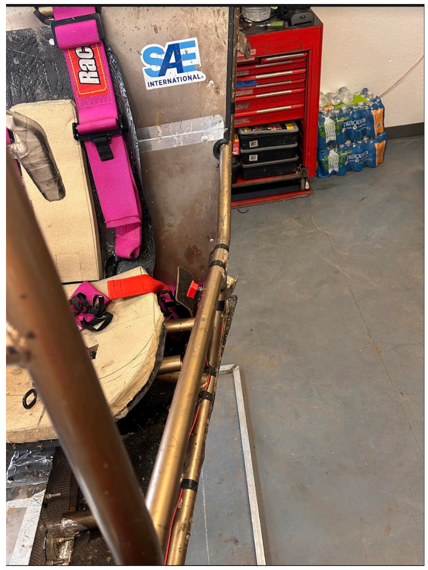


Figure (25): SIM Angled Inward

The main difference between angling the SIMs is the amount of room that the driver will have. The wider the SIMs are the more comfortable the driver will be, which is an important factor to think about considering the driver will be driving an endurance race for four hours. Additionally, to pass tech inspections the driver's arms need to have three inches of clearance from the SIMs.

4.3 Selection Criteria

The selections made by the team were based on ergonomics and spacing. Making the components such as suspension, pedals and driver positioning as optimal as possible was the main deciding factor. Most of the decision-making process is defined by the rulebook, so creative freedom for design concepts are very limited therefore the selections that were made are defined in the concept selection portion below.

4.4 Concept Selection

4.4.1 Front Braced vs Rear Braced Frame

The main deciding factor in choosing the rear-braced frame was the ease of benchmarking. Previous years cars built by NAU are still located in the machine shop and can still be easily analyzed and all of the cars from years past are rear braced.

4.4.2 In-Board vs Out-Board Brakes

Originally the team wanted to attempt at doing in-board brakes because this would mean the car would be able to have four identical hubs. However, once the discussion of packaging the front gearbox, brakes, and steering came up it was clear that in-board brakes were going to overcomplicate the front end of the car. In conclusion, the team decided to do out-board brakes with the intent of keeping manufacturing less complicated.

4.4.3 Hanging Pedals vs Floor-Mounted Pedals

The frame team decided that hanging pedals would benefit both the driver and the overall design of the frame. It would benefit the driver because it is easier to push the pedals upward since the driver is sitting slightly lower than the pedals. It also allows the driver to be lower in the seat making the overall center of mass lower, making turning easier and less susceptible to tipping.

4.4.4 Inward vs Outward Angled SIMs

Inward-angled SIMs pictured in Figure 25 allow for a more narrow overall design but compromise the comfort of the driver. While the SIMs need to be three inches from the driver's arms if they are slightly wider it will allow the cockpit to have a little more room for the driver which will make a big difference in comfort, especially for the endurance race. For these reasons, the frame team will continue forward with outward-angled SIMs similar to Figure 24.

4.4.5 Current State CAD

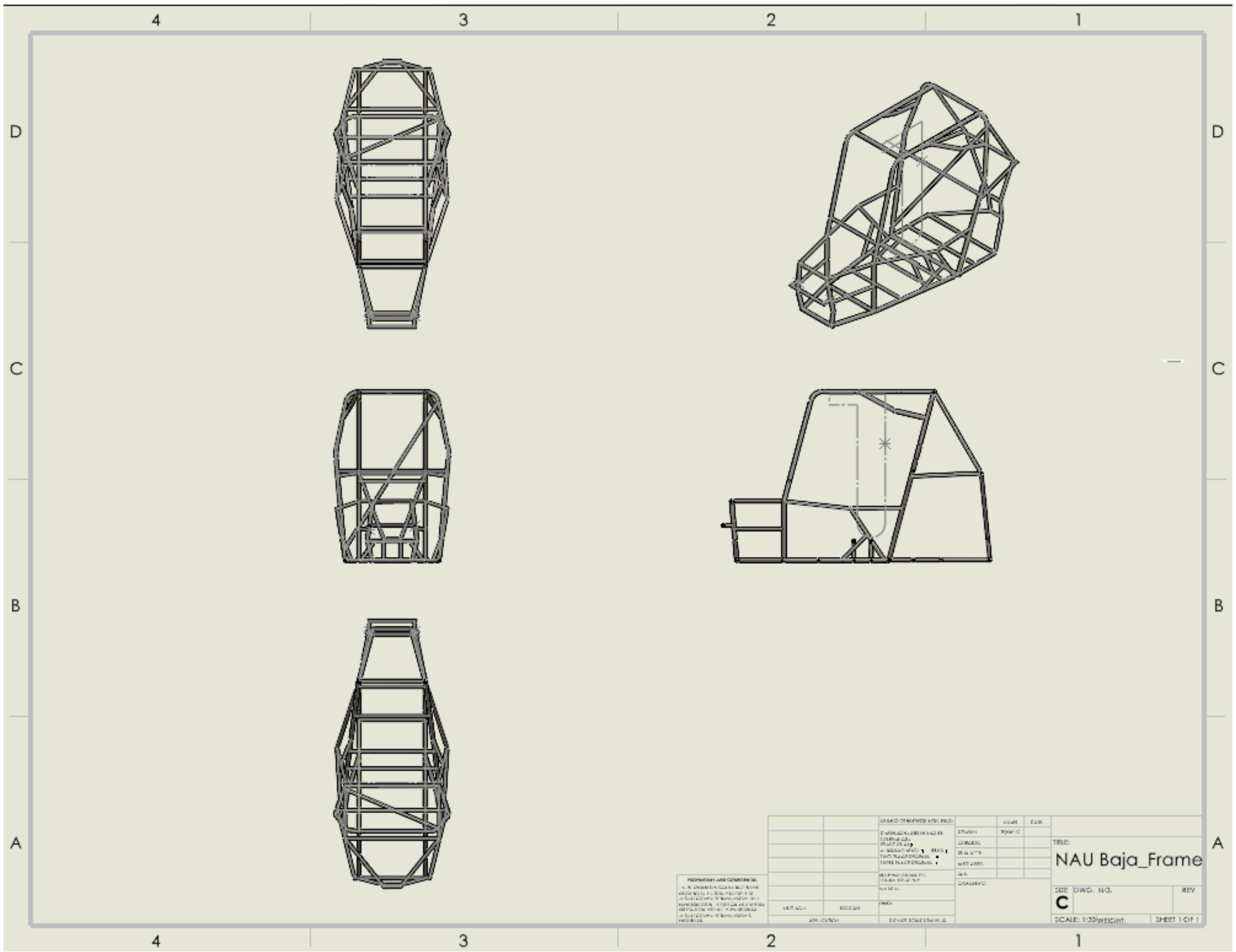


Figure (25): Current State CAD

CONCLUSION

This report covered the research and decision-making conducted by the chassis subteam of the NAU 2025 SAE Baja team. The purpose of the project is to design, fabricate and perform in a competition with other teams that also designed all-terrain vehicles under the same constraints defined by SAE. As the chassis team, the top priority is to make sure that the vehicle is safe and within the specified guidelines and customize the frame to the needs of other subteams. The team has done research and analysis of the design choices that were made and have changed designs accordingly. After conducting these analyses and prototyping the frame the team is preparing to begin fabrication since the entire team has to wait for the frame to be finished to begin mounting the other components of the vehicle.

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6 APPENDICES

6.1 Appendix A

B.3.2.17 Roll Cage Spec sheet filled out
B.3.2.16 Primary members steel OD, ID requirements
B.3.2.16 Alternate Material requirements
B.3.2.3 Secondary members OD, ID requirements
B.3.2.15 Welding samples requirements
B.3.2.1 Straight (40in) and bent members (33in unsupported, <30 deg length
B.3.2.5 Lateral cross member and CLC <=8in requirements
B.3.2.6 RRH Continuous vertical members & +/- 20 degree verticality
B.3.2.7 LDB max 5in from top & bottom of roll cage
B.3.2.12 FBM max 45 deg. from vertical, FBMup & FBM low joints
B.3.2.9 LFS must extend from RRH to past driver's heels
B.3.2.12.1 Gussets required if RHO and FBMup are not continuous
B.4.2.4.3 Safety harness tubes are in RRH plane from one side to the other
B.3.2.13.2 Rear bracing structural triangle connecting points A & B (within 2in)
B.3.2.8 RHO & RRH dimension and placement guidelines
B.3.2.10 SIMs run 8in-14in above lowest point of the seat
B.3.2.11 UST connect to LFS members securely below the seat
B.3.3.1 Roll cage clearance for the largest driver (6in helmet) (3in torso & limbs)
B.4.2 Min. 5 point harness with 3in webbing with single metal buckle
B.4.2.4.2 Shoulder webbing laterally placed 6in-9in
B.12.2 Lap and anti-sub mounting tabs (double shear) >=0.09in thick & >=1.3125in of weld length
B.4.5 Must have a conventional seat (65-90 degree back angle) with back & bottom plane
B.4.5.3.2 Seat has 4 mounting points on the bottom and 2 on the back plane

B.12.2 Seat tabs >=0.125in thick, fastener of 0.25in dia. spacers <=0.5in thick
B.4.2.6.2 Anti-Sub belt angle 0-20 deg aft of the chest line
B.4.2.4.1 Mount shoulder belts at or below driver's shoulders =<4in
B.10.3.3.1 Cockpit kill switch is within easy reach of a restrained driver

Table(1): Engineering Requirements

NAU SAE BAJA 2024-2025 – Drivetrain Team

Initial Design Report

Dylan Carley - Financial Manager, Rear Gearbox designer

Matthew Dale - Wheel Hub designer

Rowan Jones - Front Gearbox designer, Technical Resource Manager

Ethan Niemeyer - CV Axles designer

Brennan Pongratz - Project Co-Manager, Manufacturing Lead

Seth Scheiwiller – Project Co-Manager, Drivetrain Lead

Nolan Stomp - Dog Box/Chain Drive Designer

Fall 2024-Spring 2025

Project Sponsor: W.L. Gore, H&S Field Servies, Poba Medical, Harsh Co., Monster Energy, Nova Kinetics

Faculty Advisor: 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

SAE Baja is a collegiate design, fabrication, and business competition with the goal of outperforming other schools in five dynamic events with a custom off road vehicle. These events are suspension & traction, maneuverability, hill climb, acceleration, and a four-hour endurance race. This competition encourages innovative designs while still maintaining safety through a rigorous technical inspection that is split into engine inspection, frame inspection, general inspection, and a brake test. SAE also requires a business presentation to overview the cost and manufacturing plan as if this were a production vehicle. This project is split up into three general sub-teams that are in constant communication to ensure design integration goes as planned. This vehicle seats 1 person and is generally small with a trackwidth of 62 inches and a wheelbase of 64 inches which makes packaging designs into the frame challenging. These sub-teams are chassis & ergonomics, suspension, steering & brakes, and drivetrain. Overall design goals for this vehicle are ground clearance above 12 inches, a turning radius of 7 feet, suspension travel of 10 inches, a comfortable driving position, and a top speed of 33 mph. Striving for these goals will help us compete in the SAE Baja 2025 Marana, Arizona competition from May 1st to the 4th. Our rough estimation of the total cost is $15,000. This money is spent on metal stock for part fabrication, tubing for the frame, all hardware, potential outsourced fabrication labor, and travel costs. This project is a true test of the engineering knowledge we have acquired thus far and our ability to work effectively as a team of 15 individuals. Our goal is to place in the top 25% of teams that attend the competition to attract more sponsors and inspire hope into the future NAU SAE Baja capstone teams.

As of October 21st, we have completed the initial parts of the design process like general sub-system function and a selection of general design. A rough CAD assembly of the car will soon be created to ensure integration is possible with a simple construction. FEA analysis has been done on several parts so far with plans to have tested and finalized CAD on December 3rd.

TABLE OF CONTENTS

1 BACKGROUND

This section will provide an overview of NAU SAE Baja 2024-2025, drivetrain sub-team. Included within this section will be a general project description, discussing the importance of the drivetrain in relation to the success of the project with an estimated budget for the sub-team. Following the project description, the main academic and competition deliverables will be presented including the deadlines for academic assignments and competition requirements. Finally, the details concerning success evaluation will be discussed with reference to testing, calculations, and design requirements for the team to be considered successful.

1.1 Project Description

SAE Baja is a collegiate design competition hosted by the Society of Automotive Engineers (SAE). Students are tasked with designing and building a single seat, all-terrain vehicle. For the competition, the vehicles will be tested for suspension, traction, maneuverability, acceleration, hill climb/tractor pull, and endurance. This year's team consists of 15 members with 3 sub-teams including chassis and ergonomics, drivetrain, and suspension, steering, and brakes. Each sub-team is responsible for designing a specific region of the car. For the drivetrain sub-team, the areas of designs include the continuously variable transmission (CVT), rear reduction gearbox, front reduction gearbox, 4 wheel-drive integration, constant velocity axles (CV axles), and the hubs. While every sub-team is critical to the success of the vehicle, the drivetrain team oversees the power delivery system and with an efficient power delivery system the vehicle will be able to perform at the highest level in competition.

1.1.1 Budget

The team currently has six sponsors that have pledged various amounts of financial support and services.

- W.L. Gore: Financial support of $5,000
- H&S Field Services: Financial support of $5,000
- Poba Medical: Financial support of $1,500
- Harsh Co.: Machining services through water jetting, laser cutting, and materials
- Monster Energy: Supply of energy drinks
- Nova Kinetics: Carbon resources

Through these six companies the team currently has $11,500 and various services, these companies are providing invaluable support to the SAE Baja team. The team still needs to secure additional financial support, which various team connections have pledged their word to supporting the team. Two companies that the team has meetings set up with, including Mother Road Brewing Company and NAPA Autoparts. Various companies are still on the list to reach out to which include Copper State, HAAS, Canyon Coolers, Babbit Ford, Go AZ, and more local businesses.

For the drivetrain sub-team, an initial budget has been constructed based on the Bill of Materials (BOM). The BOM includes about 80% of the final cost for the drivetrain team, but it does include materials and labor that we will be getting for free or at a discounted price. At the current total cost of $5,169,63 the team does have enough money to continue development, however, the team will still need to raise more money for unforeseen expenses.

Table 1: Bill of Materials – Drivetrain sub-team

CVT		
Part	Quantity	Total Cost ($)
Sec. Fix Sheave	1	60
Sec. Move Sheave	1	90
Sec. Helix	1	15
Sec. Spring Cap	1	10
Sec. Shaft	1	50
Sec. Torsion Spring	1	0
8-32 Bolts	6	61.38
Sec. Cam Rollers	3	84.99
Cam Roller Nuts	3	19.05
Pri. Fixed Sheave	1	60
Pri. Move Sheave	1	100
Pri. Spider	1	70
Pri. Spring Cap	1	10
Pri. Shaft	1	50
Pri. Cams	3	45
Pri. Roller Bearings	3	39.84
Ti Dowel Rods	3	58.29
Shoulder Bolts	3	80.76
Nuts	3	11.37
Pri. Compression Spring	3	36
Pri. & Sec. Spacers	3	15
V-Belt	2	200
Pri. & Sec Shaft Key	2	0
Pri. & Sec. Bushings	4	59.72

Rear Gear Box		
Part	Quantity	Total Cost ($)
SKF 210-ZNR	2	160
SKF 6206	2	70
SKF 6208	1	80
SKF 6212	1	175
Gear 1 (4340 HT)	1	30
Gear 2 (4340 HT)	1	100
Gear 3 (4340HT)	2	150
Gear 4 (4340 HT)	2	80
Casing (6061-T6)	2	200
Shaft 1 (4140)	1	30
Shaft 2 (4140)	1	50
Shaft 3 (4140)	1	100

Front Gear Box		
Part	Quantity	Total Cost ($)
1654-2RS	4	200
FZ 6207	2	400
Pinion (Gear 1) (4340 HT)	1	30
Gear (Gear 2) (4340 HT)	1	100
Casing (6061-T6)	2	150
Shaft (4140)	2	80
#10-24 Shoulder Screw	10	40
1/4 - 20 Head Cap Screw	4	20

CV Axles		
Part	Quantity	Total Cost ($)
Caltric CV Axles	2 (+2 at shop already)	116
4130 Steel round tube (1"OD, 0.834"ID)	2 x 36" pieces	122.34

Hub		
Part	Quantity	Total Cost ($)
Front Hub	2 (+1 Spare)	510
Rear Hub	2	340
Sleave	1	25
Lugnut	16 (+4 Spare)	200
Stud	16 (+4 Spare)	160

4WD System/ Dog Clutch		
Part	Quantity	Totla Cost ($)
4130 Steel Round Bar (1ft length, 2.5" OD)	1	30
ANSI 40 Roller Chain (10ft)	1	38.95
40A17 Sprocket	6	155.94

Summary	
Subteam	Total Cost ($)
CVT	1226.4
4WD	224.89
CV Axles	238.34
RGB	1225
FGB	1020
Hub	1235
Total	5169.63

1.2 Deliverables

The NAU SAE Baja senior engineering design project is carried out and supported by the NAU mechanical engineering department. As such, there are course requirements that the team must meet as well as SAE Baja competition requirements. Both sets of deliverables have different deadlines and content, so the different sets will be presented separately.

The deliverables associated with NAU's senior engineering design project (ME476C and ME 486C) are shown below (Table 2) and mainly deal with technical documentation, learning new skills, presentation practice, and prototyping. There are three presentations for this Fall semester, intended to inform other students in the capstone program about the design competition, establish goals for the project, and introduce initial design concepts. Along with the presentations there are four individual homework assignments that help to learn new skills. There are eight team assignments, the important ones to note are the two reports, the final CAD/BOM, and the project management assignments.

Table 2: ME 476C Tentative Schedule

Week	Week Starts	Agenda	Individual Assignments	Team Assignments
1	26-Aug	Lecture: Introduction to Capstone	**HW00 & HW01**	
2	2-Sep	Staff/Team Meetings*	**HW02**	**Team Charter**
3	9-Sep	Staff/Team Meetings		
4	16-Sep	**Presentation 1**	**Peer Eval 1**	
5	23-Sep	Staff/Team Meetings		
6	30-Sep	Staff/Team Meetings	**HW03**	
7	7-Oct	**Presentation 2**	**Peer Eval 2**	
8	14-Oct	Staff/Team Meetings		**Report #1**
9	21-Oct	Staff/Team Meetings		**Website Check #1**
10	28-Oct	Staff/Team Meetings		**Analytical Analysis Memo**
11	4-Nov	**Presentation 3**	**Peer Eval 3**	
12	11-Nov	**1st Prototype Demo****		
13	18-Nov	Staff/Team Meetings		**Report #2**
14	25-Nov	Staff/Team Meetings***	**HW04**	**Final CAD/BOM**
15	2-Dec	**2nd Prototype Demo**		**Project Management**
Finals	9-Dec		**Final Peer Eval**	**Website check #2**

The deliverables for the SAE Baja competition must be completed on time if the team wants to compete at the 2025 competition in Marana, AZ (deliverables shown below in Table 3). The first deliverable is a business request for proposal (RPF); the team must justify the novelty of the design, the feasibility in terms of design and fabrication, and the marketing plan. The second is the roll cage documentation package which includes the material invoices, material certifications, and calculations for all the primary members of the frame. The third is the presenter form which is unspecified by SAE Baja. The fourth is the written cost reduction report, which discusses proper cost reduction practices, as well as preparing a full cost report to discuss the material and fabrication cost of the vehicle within the scope of Lean/Six Sigma manufacturing principles. The last item is the design review briefing (DRB) that will be presented to a board of judges during the competition. The DRB will contain information about all the sub-team's designs with justification for the designs.

Table 3: SAE Baja Competition Deliverables

Deadlines	Open Date	Due Date	Cancellation Date	Grounds For Removal
Virtual Business Presentation - Business RPF	10/3/2024 7:00	12/13/2024 21:59	12/18/2024 21:59	Yes
Tech - Roll Cage Documentation Package	10/3/2024 7:00	2/14/2025 21:59	10/1/2025 20:59	Yes
Virtual Business Presentation - Presenter Form	10/3/2024 7:00	3/3/2025 21:59	3/10/2025 20:59	No
Cost - Written Cost Reduction Report	10/3/2024 7:00	3/17/2025 20:59	3/24/2025 20:59	Yes
Design - Design Review Briefing	10/3/2024 7:00	4/7/2025 20:59	4/14/2025 20:59	Yes

1.3 Success Metrics

For this project to be considered successful, we must do well as a team on both the senior engineering design project deliverables and the SAE Baja competition deliverables. Along with these deliverables the team has personals goals and basic requirements that the car must be able to perform via SAE Baja. Personal goals for the team include placing in the top 25% overall for the competition, having a car that looks and performs good, and learning new skills throughout the process.

For the SAE Baja competition, there are some general requirements for the vehicle. The vehicle must pass the technical inspection at competition. The technical inspection sheet is about twelve pages long and covers a variety of items. A handful of the items that the team will be checked on are design constraints, roll cage – material and documentation, roll cage – geometry (sections 1 and 2), and driver constraints. Once the technical inspection sheet is filled out at competition, there are a couple of technical inspections that are not on the technical inspection sheet the vehicle must pass. Those include being able to lock all four wheels on the braking test. A motor inspection to verify that the motor was not modified, and a general inspection that will look at all pinch-points, fuel system, any other rules to ensure that the team's vehicle is safe and allowed to compete.

The drivetrain sub-team also has their own personal goals to be considered successful. The biggest metric is time, not only getting designs and calculations done as soon as possible but also manufacturing and assembling said designs as soon as possible. Being able to test the CVT, gearboxes, and 4WD integration will give the team ample time to correct and dial the power delivery system. The biggest mistakes for past teams of the NAU SAE Baja senior engineering design project was the testing aspect of the vehicle. They didn't have enough time to fully test all aspects of the vehicle and often the drivetrain sub-team had the most critical failures which cut the running time at competition and in turn diminished the team's opportunity to place highly in the competition. This year's team plans to learn from past mistakes and hopefully will be able to give the vehicle a chance to do well in the competition.

2 REQUIREMENTS

The following section will provide a detailed breakdown of the project requirements, including the customer requirements, engineering requirements, and the house of quality with the QFD. All these combined wills help the drivetrain team to reach an understanding of the overall design objectives.

2.1 Customer Requirements (CRs)

The SAE Baja vehicle will be put through a multitude of challenging obstacles when at competition. The car is expected to sustain not only these events, but to also meet customer requirements while doing so. These customer requirements will also guide the engineering design process as the car is being designed and manufactured. It is critical that the car performs at the highest echelons possible regarding customer requirements that include:

- Efficiency
- Safety
- Durability
- Affordable
- Ease of Manufacturing
- Aesthetics
- Pass Techs
- Acceleration
- Lightweight

These customer requirements will collectively guide the team to design the best drivetrain possible while always keeping all factors in the equation, and not just narrowing down to specific requirements and forgetting about others.

2.2 Engineering Requirements (ERs)

With respect to the customer requirements, the engineering requirements will correlate with and dive deeper into specific parameters that are goals that have been set to be met; from the engineering side of things. The engineering requirements will be split into different specific components of the drivetrain and will be explained further quantitatively. They are as follows:

CVT

- Primary Flyweight - 70 grams
- Primary Springs – 35 grams
- Secondary Springs – 35 grams
- Max weight – 15 lbs.
- Max torque – 415 lbf-ft
- Top speed – 35 mph
- Moving powertrain parts must be guarded on all sides – Yes

- Competitive transmission range – 5

Reduction gearbox

- Rear Ratio – 9.56:1
- Front Ratio – 3.62:1
- 4WD – Yes
- Moving powertrain parts must be guarded on all sides – Yes
- Gearbox vent system 100mm away from exhaust – 100mm
- 4WD driveshaft surrounded and separate from cockpit – Yes
- Minimum life cycle of gears – 10^9 cycles
- Torque output – 226 ft*lbs

Axles

- Length - <16 inches
- Angle – 40 degrees
- CV Joints – Yes
- Thickness of CV axle – 1.2 inches

Hubs

- Weight – 75 grams
- Max diameter – 70mm
- Max Thickness – 40mm

Under these constraints, a robust, reliable and coherent drive train will be manufactured and designed, with fine adjustments as necessary. A portion of these requirements are derived from the SAE Baja rulebook.

2.3 House of Quality (HoQ)

A QFD has been derived below from the combination of engineering requirements and customer requirements that have been laid out. The correlation between all requirements has been considered and weighed accordingly. The QFD also laid out a customer competitive assessment that further illustrates how our BAJA vehicle may compare to other top performing BAJA Vehicles.

Table 4: Drivetrain QFD

Primary Flyweight																						
Primary Spring	pos																					
Seconday Spring																						
Max Weight	neg	pos	pos																			
Max Torque	pos	pos	pos																			
High Top Speed	pos			neg	neg																	
Moving Powertrain parts must be guarded on all sides																						
Competitive Transmission Range	pos				pos																	
Ratio-Rear					pos	neg		pos														
Ratio-Front					pos	neg		pos	pos													
4WD								pos	pos	pos												
Moving Powertrain parts must be guarded on all sides						neg	pos				pos											
Gearbox vent system 100mm away from exhaust																						
4WD driveshaft surrounded and seperate from cockpit							pos				pos	pos										
Minimum life cycles of gears					neg	neg																
Torque output	pos	pos	pos	neg	pos	neg		pos	pos	pos	neg											
Length																						
Angle																	neg					
CV Joints																		pos				
Thickness on CV axle				neg	neg											neg	neg		neg			
Weight				pos	pos											pos						
Max Diameter																					neg	
Max Thickness																					neg	pos

			Engineering Requirements																							Customer Competitive Assesment				
Relative Weight (%)	Customer Weights	Customer Requirments	Primary Flyweight	Primary Spring	Seconday Spring	Max Weight	Max Torque	High Top Speed	Moving Powertrain parts must be guarded on all sides	Competitive Transmission Range	Ratio-Rear	Ratio-Front	4WD	Moving Powertrain parts must be guarded on all sides	Gearbox vent system 100mm away from exhaust	4WD driveshaft surrounded and seperate from cockpit	Minimum life cycles of gears	Torque output	Length	Angle	CV Joints	Thickness on CV axle	Weight	Max Diameter	Max Thickness	1 Poor	2 Ok	3 Acceptable	4 Good	5 Excellent
11	4	Efficiency	9	9	9	9	9	1	1	9	9	9	9	1	1	1	9	9	9	9	9	3	9	3	3			A	B	C
22	5	Safety	1	1	1	1	1	3	9	1	1	1	1	9	3	9	1	1	3	3	3	9	1	3	3					ABC
10	3	Durability	3	3	3	3	1	3	1	3	1	1	1	1	1	1	9	3	9	9	9	9	1	9	9		A			BC
10	3	Affordable	9	1	1	9	1	1	1	3	1	1	1	1	1	1	1	1	3	1	3	9	1	3	3	BC			A	
5	2	Ease of Manufacturing	9	1	1	3	1	1	1	3	3	3	3	1	1	1	3	1	9	9	9	9	1	3	3		BC		A	
5	1	Aesthetics	1	1	1	3	1	1	1	1	1	1	3	3	3	1	1	1	1	1	3	3	3	3	3		A		B	C
22	5	Pass Techs	1	1	1	1	1	1	9	1	1	1	9	9	9	9	1	1	1	1	1	9	1	1	1					ABC
5	2	Acceleration	3	3	3	1	9	9	1	9	9	9	9	1	1	1	1	9	1	1	1	1	9	3	3			A	C	B
10	3	Lightweight	3	1	1	9	1	9	3	1	9	9	9	3	3	3	1	1	9	1	1	9	9	3	3			A		BC
Technical Requirement Units			grams	grams	grams	lbs	lbf-ft	mph	N/A	N/A	N/A	N/A	Yes/No	N/A	mm	N/A	Cycles	ft*lbs	inches	degrees	N/A	inches	grams	mm	mm					
Technical Requirement Targets			70	35	35	15	415	35	N/A	5	9.56:1	3.82:1	Yes	N/A	100	N/A	10^9	226	16	40	N/A	1.2	75	70	40					
Relative Technical Importance			5	8	7	6	1	2	3	4	3	4	5	6	8	7	1	2	3	4	2	1	3	1	2					

Sub-Section	Color Code
CVT	
Rear Gearbox	
Axles	
Hubs	

Correlation	
Positive	pos
Negative	neg

Relationship	
Strong	9
Moderate	3
Weak	1

C.C.A. Legend	
A	NAU 2024 #44
B	Cornell 2024 #73
C	ETS 2024 #27

Based on the QFD that has been produced, the team must follow the specific requirements that have been laid out in order to be successful. The customer requirements and engineering requirements were all considered in correlation with each other to see what relates and what doesn't. This will help the team when designing specific parts to know what else to consider when designing it, and what not to worry about; it ultimately serves as a skeleton to follow for the design process as far as parameters that must be met. Some of the most important engineering requirements include:

1) Max weight
2) Max Torque
3) Top speed
4) Front gear ratio
5) Rear gear ratio
6) Powertrain guarded areas
7) Strength of components

These parameters will help the individuals in the team in making informed engineering design decisions, keeping the whole team on the same track as far as what matters in everybody's respective designs. The QFD can be referenced to keep design goals and constraints in mind.

3 Research Within Your Design Space

3.1 Benchmarking

3.1.1 CVT

There are three general designs that are used for CVT fabrication; two mechanically actuated and one electronically actuated. One of the mechanically actuated designs includes what is commonly referred to as a cam & roller system that can be seen on the right of Figure 1. Centrifugal force created from the cams rotating at a high RPM and pushing against a roller begins the clamping of the primary side of the CVT which results in smooth gear ratio changes. The other mechanically actuated design also relies on centrifugal force. This design is referred to as the flyweight & ramp system and can be seen on the left of Figure 1. When rotation begins, the flyweights push against the ramp to start clamping the primary side of the CVT for gear ratio changes. The system that relies on electronic actuation is referred to as an ECVT and can be seen in the middle of Figure 1. In these systems, a motor controller such as an Arduino communicates with a stepper motor to begin the clamping of the primary side of the CVT. When comparing the two types of actuations, typically an ECVT allows for more precise tuning and better overall performance when executed correctly. This comes with the downside of more part fabrication and extensive programming. Because of this we will be executing a cam & roller style CVT for ease of manufacturing and weight reduction.



Figure 1: Gaged CVT (left), Cal Poly ECVT (middle), Polaris RZR CVT (right)

3.1.2 Reduction Box

For the reduction gear box there are many ways to design and implement the gearbox to the rear of the vehicle. A couple of examples include integrating CV cups to the final gear for the CV axles to mate to or there could be a plate that the CV cups bolt onto. Both options are possibilities, however the CV cups integrated into the final gear allow for more suspension travel and weight reduction. Integration of CV cups was a design aspect from the 2024 NAU SAE Baja team, this year's reduction box will update and redesign a CV cup integration. Other design aspects that have freedom to change include gear ratios and 4WD integration, other than those designs most teams will follow the same format of a two-stage compound geartrain. Most teams will use this setup because of space and money, if you run a single stage geartrain the final gear would either be large in diameter or the gearbox itself would be long in length. From the information above, the rear reduction box will be a two-stage compound geartrain with CV cups integrated into the final gear.

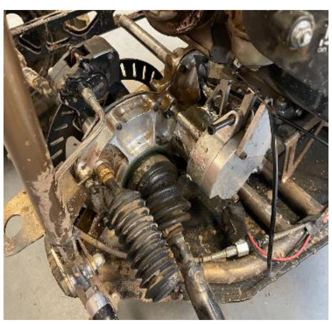



Figure 2: 2024 NAU Baja (left), 2024 RIT Baja (middle), 2024 Cornell (left)

3.1.3 4WD System

The 4WD system that is being used as the main benchmark is a combination of two past NAU Baja teams. The dog clutch (left image) from last year’s team (Car #44) performed well and the vehicle was able to efficiently switch to 4WD during the competition. The method of power transfer from the rear to the front of the vehicle is being benchmarked off Car #74 from a couple years ago (center image). Cal Poly racing had an excellent gearbox design to benchmark with, and they consistently place well in competition.

Figure 3: 4WD System and Front Gear Box from previous NAU teams and Cal Poly Racing

3.1.4 Axles and Hubs

The top two candidates for transmission of power between the front and rear gear boxes and the wheels are CV axles and Universal axles. Universal axles are much easier to manufacture than CV axles but also do not have near as many range of motion, plunging motion, and are more susceptible to binding when compared to CV Axles. CV in CV axles stands for constant velocity; meaning CV axles also have a smoother power delivery when compared to Universal axles. That being said, our team has chosen to move forward with CV Axles.

Figure 4: CV and Universal Axles in prior Top 10 performing Vehicles

When designing wheel hubs, one of the most important aspects of design is how the power from the axle gets to the hub. Seeing how last year's Baja team failed to properly secure the axle to the hub, this is very important to get right. The most popular ways to attach the hub to the axel are splines, press fit, and hex fit. The final decision heavily depends on the axles used, and the plan is to use axles with spline fits. With that said, the spline standard used is proprietary and given the fact that the fit is very important, the team does not want to try to guess the spline specifications. So, the team will probably end up buying a hub matching the axle and cut the hub down to a size where it can be press fit into our custom hub. That way the hubs will still be able to detach from the axle easily while still maintaining a solid connection.

Figure 5: Hub Fit Types

3.2 Literature Review

3.2.1 Dylan Carley

- Shigley's Mechanical Engineering Design [1]
  - Chapters 13 and 14 of Shigley's Mechanical Engineering Design textbook have great information on the basics of gears and all the governing equations located in chapter 13. In chapter 14 it has more details regarding spur gears and how to evaluate and design an efficient geartrain.
- Machinery's Handbook [2]
  - Chapter 12 on gearing is related to Shigley's but gives more in-depth on the design, such as tolerances, shafts, and bearing.
- Design, Analysis, and Simulation of a Four Wheel-Drive Transmission for an All-Terrain Vehicle – SAE [3]
  - This paper discusses the topics for the SAE Baja vehicle with an emphasis on the drivetrain. Specifically, the analysis of the CVT and the rear gearbox. Within the paper there are equations and how to perform FEA on the rear gears.
- Numerical analysis of the heat transfer of gears under oil dip lubrication [4]
  - This paper uses numerical analysis to determine the heat transfer from the gears to the oil in the gearbox casing.
- KHK Stock Gears: Lubrication of Gears [5]
  - This website discusses how to properly lubricate gears based on the speed of the gears plus the application of the gears.
- AZO Materials: AISI 4340 Alloy Steel [6]
  - This website has all the material properties for 4340 steel which is the material for the gears.
- MatWeb material property data: Aluminum 6061-T6 [7]
  - This website has all the material properties for 6061-T6 aluminum which is the material for the gearbox casing.

3.2.2 Matthew Dale

- Ball & Roller Bearing Design: Theory, Design, and Application [8]
  - Bearing design and fit to ensure solid fit.
- Non-Destructive Material Testing [9]
  - How to take a stress test to determine fatigue limit.
- Design And Analysis of Wheel Hub of Baja ATV in Ansys. [10]
  - Determining optimum wheel hub size and shape.
- Design and Weight optimization of wheel assembly components using FEA for BAJA [11]
  - Further hub development and optimization.
- Simulation and Optimization of Wheel Hub and Upright of Vehicle: A Review [12]

- Force visualization and stress testing, along with further part development.

- Ansys Innovation Space [12]
  - How to use Ansys to apply forces to part to analyze stresses.
- Design and Analysis of Wheel Hub for Weight Optimization by using Various Material [13]
  - Material selection and further part development.
- ASM Material Data Sheet [37]
  - Material selection and material engineering data specs.

3.2.3 Ethan Niemeyer

- Shigley's Mechanical Engineering Design [1]
  - Chapters 13 and 14 discuss equations and basic information regarding general gear, and more specifically in our case, spur gears. A lot of our mathematical modeling is produced in reference to the equations in this book.
- Machinery's Handbook [2]
  - Chapter 2 also discusses and portrays information regarding gears and gearing. It also dives into the manufacturing processes of these gears.
- A Review on Constant Velocity Joint [14]
  - This article portrays vital information about the design, use, and performance of constant velocity axles and joints. It was extremely useful in the design choice of CV axles.
- SAE Baja 25' Rule Book [15]
  - This is the rulebook that we as a team must follow this year in order to participate in the competition. It is important to follow this to stay within our allowable parameters for the competition.
- Universal (U) Joints – Axle and Driveshaft [16]
  - A review of design information regarding universal joints, and their applications on axles and drive shafts. It contributed to the design decision of moving forward with CV axles instead of universal joints axles.
- Gear Generator [17]
  - This was one of the 2 software used to help generate the design and geometry of the gears in the rear reduction box. Gear generator takes specific input parameters such as number of teeth and diametral pitch and makes a gear for you.
- Rush Gears [18]
  - This was the second software used to help generate our rear reduction gearbox gears. We put in all necessary input parameters, and it generated for us the cad models, gear by gear.
- Basic Gear Mechanisms [19]
  - This is a website that more coherently portrays all the necessary information and equations for spur gear design when compared to the two books listed above. It was a nice reference to use for initial design to then double check with the machinery's handbook and Shigley's mechanical engineering design.

3.2.4 Rowan Jones

- Shigley's Mechanical Engineering Design [1]
  - Chapter 13 discusses the AGMA stress equations used in mathematical modeling of the front gears. Chapter 14 discusses spur gear design, parameters, and general force equations used in gear design.
- Machinery's Handbook [2]
  - Chapter 12 shows various calculations and specifications for gear design. This was used as another reference for the spur force and stress calculations as well as standards regarding these calculations.
- SAE BAJA: Final Drive Report (Cal Poly) [20]
  - This report from Cal Poly shows a general gearbox design and rough calculations for their gears. Cal poly has performed well in past SAE BAJA competitions, making them a good team to benchmark with.
- A Review of Recent Advances in Design Optimization of Gearbox [21]
  - This article discusses different ways to optimize gearbox functionality, including gear noise reduction and efficiency optimization. This article was useful in determining how energy is lost in noise and friction when power is transferred from gear to gear.
- Design analysis and fabrication of automotive transmission gearbox using hollow gears for weight reduction [22]
  - This article was useful in determining the skeletonized structure of the large gear for the front gearbox. Taking cuts of material will help reduce weight of the front gearbox while also retaining the strength required to transmit power to the front wheels.
- The Basics of Gear Theory [23]
  - This article discusses the basics of how gears work, talking about things like pressure angles, addendum's, dedendum's, and other gear geometries that are important to understanding how gears mesh together.
- AZO Materials: AISI 4340 Steel [6]
  - This website states the material properties for 4340 Steel, which is the material the gears will be made of. This property was used in the gear stress calculations in mathematical modeling.
- An Advanced Approach to Optimal Gear Design [24]
  - This article further describes the optimization of spur gears, talking about two different methods in gear evaluation. The two methods discussed showed how bending and contact stress can be reduced in spur gears, which can prolong the life of the gear. This is useful in gear design to ensure the gears do not fail.

3.2.5 Nolan Stomp

- Shigley's Mechanical Engineering Design [1]
  - Chapter 16 outlines common clutch designs and the characteristics of each, which was used in order to benchmark the design for the dog box. Chapter 17 discusses flexible mechanical elements, including but not limited to roller chains and belts. This is useful in selecting which method of power transmission would be ideal for our purposes.

- Machinery's Handbook [2]
    - Outlines ideal turning speeds and feed rates for similar parts that will need to be manufactured for the 4WD system
- What is a Dog Clutch? [25]
    - Gives a general introduction to the purpose and function of a dog clutch, as well as an in depth look into its specific pros and cons. The advantages and disadvantages will be used to know where weak points of the design may be, and brainstorm ways to negate these issues.
- Dog Transmission Explained [26]
    - Discusses strengths and weaknesses between the dog clutch and comparable systems such as synchromesh. This was greatly useful to confirm the direction that we wanted to go for the 4WD system
- Chain Drive vs Belt Drive: Difference and Comparison [27]
    - Provides an extensive list of pros and cons of using a belt drive vs a chain drive, along with where each is more commonly used.
- 2025 SAE Baja Rulebook [15]
    - This year's rulebook includes regulations for how the 4WD system is required to function during the competition.
- Kinematics of Roller Chains- Exact and Approximate Analysis [28]
    - Useful in showing how a roller chain should act within a dynamic system, which will be useful for design, implementation, and testing of our chain drive system
- The Effect of the Tooth Chamfer Angle on Dog Clutch Shiftability [29]
    - Analyzes the relationship between tooth angle and successful engagement of the dog clutch teeth, which is an absolute necessity to consider during design of the dog box system

3.2.6 Seth Scheiwiller

- Shigley's Mechanical Engineering Design [1]
    - Chapter 13 discusses resulting torque and power outputs because of gear ratios which were used to calculate torque and power outputs of the different stages of a CVT transmission.
- Machinery's Handbook [2]
    - This source has information on the machine elements of flexible belts and sheaves. This was also used to research standards for interference and clearance fits.
- Olaav Aaen's Clutch Tuning Handbook [35]
    - This handbook contains tips on how to tune a mechanically integrated CVT transmission which will prove useful for when we must tune our CVT to match the desired engagement RPM and shift out RPM.
- Modeling and Tuning of CVT Systems for SAE Baja Vehicles [30]
    - This master's thesis contains systems of equations that were utilized to calculate CVT clamping forces and efficiency.

- Design and Manufacturing of a Continuously Variable Transmission [34]
  - This report provided a system of equations that were used to calculate desired CVT ratios and belt parameters.
- Virtual training on How CVT works and How to Design CVT in SolidWorks [32]
  - This source provides a tutorial on how to model a gaged CVT transmission in SolidWorks which proved useful for understanding the fundamentals of a CVT transmission.
- Modeling of a Continuously Variable Transmission [33]
  - This video provides a MATLAB simulation on the performance of a CVT transmission.

3.2.7 Brennan Pongratz

- Shigley's Mechanical Engineering Design [1]
  - Chapter 17 discusses flexible mechanical elements such as a v-belt that will be the driving the CVT.
- Machinery's Handbook [2]
  - This source has an abundance of information on part fabrication but specifically press fit standards and thread standards.
- Design and Manufacturing of a Continuously Variable Transmission (CVT) [34]
  - This source was used for the ratio between the primary and secondary sides of the CVT and some general calculations for v-belt selection.
- Collegiate Design Series Baja SAE Rules [15]
  - This paper is an essential reference for the entire team, but we are specifically interested in guards for hazardous releases of energy.
- Olav Aaen's Clutch Tuning Handbook [35]
  - This paper will be very useful once the CVT has been manufactured and is ready to be tuned. It was initially used to understand how a CVT works and tunable parameters to consider.
- Modeling of a Continuously Variable Transmission [33]
  - This source helped us generate a Matlab model of the CVT with parameters that we can implement into our CAD designs.
- Fatigue Design Curves and Analysis for Aluminum [36]
  - The main piece of information used from this source is the S-N curve for 6061-T6 aluminum which will be the material used for a large majority of our parts.

3.3 Mathematical Modeling

3.3.1 Dylan Carley

Mathematical modeling for the rear reduction gearbox includes the torque required to break the rear wheels lose on pavement, the allowable bending stress, fatigue life on the gears, and the rear gearbox bearings for the initial shaft (from the CVT to the pinion).

Torque Required to break real wheels lose on pavement:

$$T = \frac{d}{2}\left(F_s \frac{w}{4}\right) = 226.83$$

d = 1.833 ft

F_s = .9 (highest static friction the vehicle will see)

w = 550 lb. (goal for weight of vehicle)

Allowable Bending Stress:

material: 4340 HT – Brinell Hardness (H_b) = 217

$$Grade\ 1: S_t = 77.3H_b + 12800psi$$

$$\sigma_{all} = \frac{S_t Y_n}{S_f K_T K_r} = 19{,}735.80\ psi$$

S_t = Gear bending strength = 29,574.10 psi

Y_n = Stress cycle factor = 1

S_f = AGMA factor of safety = 1.5

K_T = Temperature factor = 1

K_r = Reliability factor = .999

Fatigue life on gear:

S_{ut} = 108 kpsi

$$F = 1.06 - 2.8(10^{-3})S_{ut} + 6.9(10^{-6}){S_{ut}}^2 = .838$$

$$[70 < S_{ut} < 200\ kpsi]$$

F = Fatigue line in the high cycle

$$S'_e = 0.5S_{ut} = 54\ kpsi$$

S'_e = Endurance limit

$$a = \frac{(FS_{ut})^2}{S'_e} = 151.7$$

$$b = -\frac{1}{3}log\left(\frac{FS_{ut}}{S'_e}\right) = -0.0748$$

a & b: constants that are the ordinate intercept and the slope of the line in log-log coordinates.

$$N = \left(\frac{\sigma_{ar}}{a}\right)^{\frac{1}{b}} = 7.032 * 10^{11} Cycles$$

N = Number of Cycles

σ_{ar} = Completely reversed stress = σ_{all}

Rear gearbox bearings on initial shaft (from CVT to pinion gear):

$$C_{10} = a_f F_D \left[\frac{x_D}{x_0 + (\theta - x_0)\left[ln\left(\frac{1}{R_D}\right)\right]^{\frac{1}{b}}}\right]^{\frac{1}{a}} = 35.67\ lbf = .1587\ kN$$

C_{10} = Catalog rating

a_f = application factor = 1

F_D = Radial load

x_D = Rating life multiple

$$x_D = \frac{L_D}{L_{10}} = \frac{60 n_D l_d}{10^6} = 78$$

n_D = desired speed = 1300 rpm

l_d = desired life = 1000 hours

R_D = .999

Weibull Parameter:

x_0 =0.02

θ = 4.459

a = 3 (for roller bearings)

b = 1.483

From the results above, the gear ratio of the rear reduction gearbox will be 9.56:1. The allowable bending stress of the gears is about 19 kpsi. With the allowable bending stress and the ultimate tensile strength, the number of cycles that the geartrain is rated for is about 700 billion cycles which is also considered infinite life. Lastly, for the bearings that will be placed on the initial shaft that runs from the CVT to the pinion gear. From the radial load, desired speed of the shaft, and the desired life of the bearings, the catalog rating that the bearings must be able to support is .1587 kN. This catalog number is 80 times smaller than the catalog number of the bearing that will have to be used with the constraint of the shaft size (i.e. any bearing that is chosen will be able to support the load of the shaft).

3.3.2 Matthew Dale

Mathematical modeling of the hub is done to ensure that it can withstand the max forces experienced by the vehicle. The goal is to make the wheel hub as lightweight as possible while still being strong enough and having the lowest un-sprung weight possible will help the team succeed in competition. This was started with a cantilever beam calculation, which would be the most realistic way to try to calculate the max force applied to one of the hubs.

$$\text{Cantilever Beam Max Deflection} = \frac{(Force)(Length)^3}{3(Elasticity)(Inertia)}$$

$$\text{Max Impact Force} = 1348\ \text{N (as calculated by suspension team)}$$

$$\text{Youngs Modulus for 6061-T6 Aluminum} = 69\ \text{GPa}$$

$$\text{Inertia} = \frac{(0.17145m)^4}{12} = 7.2(10^{-5})$$

$$0.005mm = \frac{(1348N)(Length)^3}{3(69{,}000MPa)(7.2*10^{-5})} = 0.381m = 1.5\ \text{in}$$

The max width determined by the calculation was 1.5 inches. The hub was then made to this specification; however, the hub needs to be swept out to accommodate the brakes and the size of the wheel and tire. So, the hub was made 1.5 inches thick across the whole sweep, even though it would need to be thicker to fully support the sweep. But when the 1.5-inch thickness was compared to previous years hubs, 1.5 inches seemed far too thick. Several Ansys tests were done.

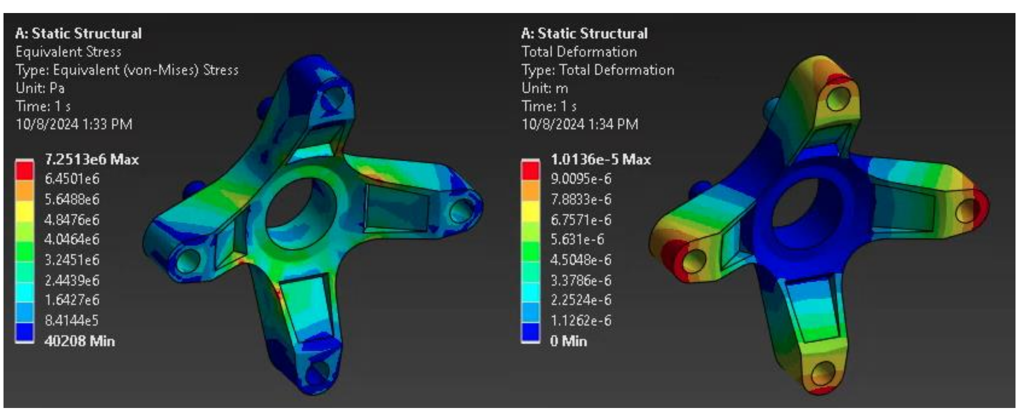


Figure 6: Static Structural ANSYS Test of Initial Hub Design

With this basic wheel hub with 1.5 inches across the whole part, the thickness is far too much and lots of weight can be saved. A refined part was developed with machineability in mind and has been static tested in Ansys with a safety factor of 1.3, right within the target range. The simulation is only a static test however, and a full dynamic test will be needed to be fully confident about the design.

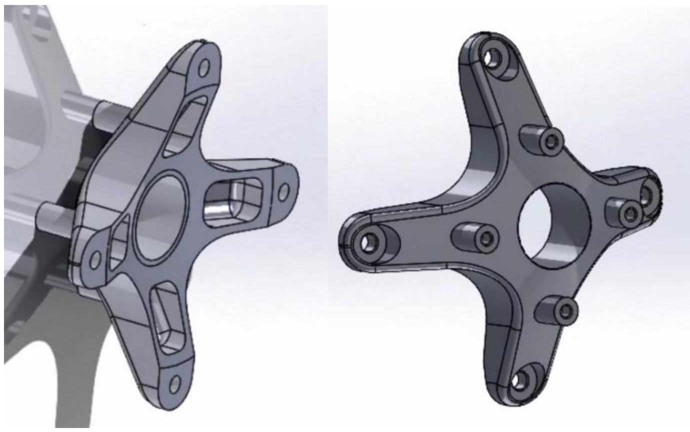


Figure 7: Developed Hub Design

Another thing to keep in mind is that the hub shows it for the front and will have brakes. The rear uses inboard brakes, so the hub will not need to have brake attachments. Another key difference for the rear is probably going to be the sweep angle, as it will need to be drastically increased to accommodate the trailing link suspension in the rear, as seen in previous years. This will require a lot more Ansys testing since the front and rear could be very different. For now, though, the front hub is being 3d printed for prototyping and finalizing fitment.

3.3.3 Ethan Niemeyer - Axles

Shaft Diameter

Minimum Diameter of a 4130-steel tube that can withstand 20 hp (Safety of factor of 2) at post reduction box 300 rpm:

$$p = \frac{(T \cdot w)}{5252}$$

Where;
P=Power in (HP)
T= Torque in (Ft-Lb)
w=Rotational Speed in (RPM)
5252 is a unit conversion factor
Solve for T, and then:

$$T = \left(\frac{\pi}{16}\right) \cdot \tau \cdot d^3$$

Solve for d

d=0.73 inches

CV Cup Thickness
Minimum wall thickness for 4140 HT Steel CV cup with assumed OD of 2.5” that experiences 20 hp (Safety factor of 2) at post reduction box 300 rpm

$p = \frac{(T \cdot w)}{5252}$
Where;
P=Power in (HP)
T= Torque in (Ft-Lb)
w=Rotational Speed in (RPM)
Solve for T and then:

$T = \left(\frac{\pi}{16}\right) \cdot \tau \cdot \left(\frac{(d_{outer}^4 - d_{inner}^4)}{d_{outer}}\right)$
Where;
τ=allowable shear stress in (Psi); (54150 for 4140 HT steel)

Solve for d(inner) and then:

$$t = \frac{(d_{outer} - d_{inner})}{2}$$

Solve for t:

t=0.125 inches

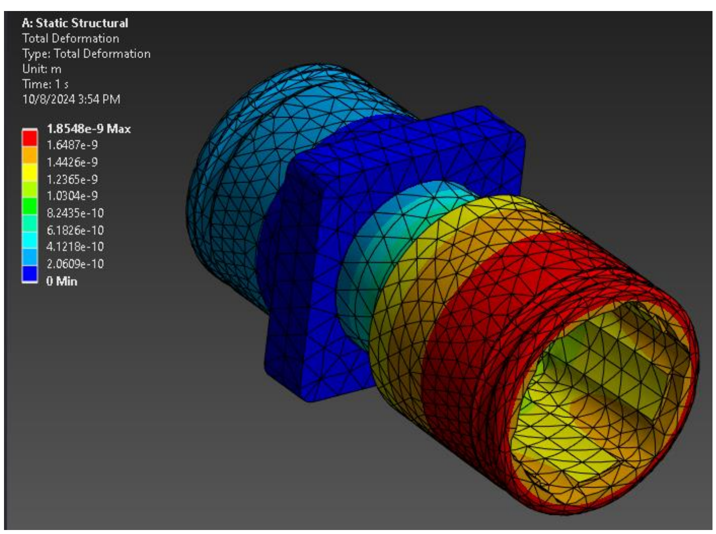


Figure 8: Total Deformation FEA On CV Cup-Shaft-Cup

The above basic FE analysis shows the concerning areas regarding our CV cup design. We are already at a safety factor of 2 with our given material and parameters, and so moving forward we would thicken the red areas, if need be, after prototyping and testing and observing this critical area.

3.3.4 Rowan Jones

<u>Front Gear Box</u>

Gear Dimensions

Table 5: Dimensions for the Gear and Pinion in the Front Gear Box

	No. of Teeth	Pitch Diameter [in]	Center to Center [in]	Face Width [in]
Gear	68	5.9167	3.54167	0.625
Pinion	17	1.667		

The front gear box is connected to the chain drive which allows power transmission from the rear gearbox intermediate shaft, which the dog clutch is located on, to the front gear box. The front gear box will be slightly underdriven to allow for better handling and traction when turning with 4WD engaged. The ratio of the output gears of the reduction box to the front gear box will be 1:1.05.

In the next section, allowable bending and contact stresses for the gear using the material properties of 4340 HT steel will be calculated. These values will be useful in determining if the gears will fail and will be most useful when performing finite element analysis (FEA) to see if the stresses shown in FEA are within the allowable stress range.

Allowable Bending Stress

Grade 1: S_t = 77.3 Hb + 12800 psi

Hb = 217 (4340 HT hardness from AZO materials)

Gear bending strength: S_t = 29,574.1 psi

$$\sigma_{all} = \frac{S_t Y_n}{S_f K_t K_r} = 19{,}716\ psi$$

Y_n = Stress Cycle Factor: $Y_n = 1.6831*N^{-0.0323} = 1$

K_t = Temperature Factor: $K_t = 1$

K_r = Reliability Factor: $K_r = 1$

S_f = AGMA factor of safety: $S_f = 1.5$

S_t = 29,574.1 psi

Allowable Contact Stress

Grade 1: S_c = 322 Hb + 29100 psi

Hb = 217 (4340 HT hardness from AZO materials)

Contact-fatigue Strength: S_c = 98,974 psi

$$\sigma_{c,all} = \frac{S_c Z_n C_h}{S_h K_t K_r} = 65{,}982\ psi$$

Z_n = stress-cycle factor: $Z_n = 1.4488*N^{-0.023} = 1$

C_h = hardness ratio factors for pitting resistance: $C_h = 1$

K_t = Temperature Factor: $K_t = 1$

K_r = Reliability Factor: $K_r = 1$

S_h = AGMA Factor of Safety: $S_h = 1.5$

S_c = 98,974 psi

Bearing Reactions

Output torque from CVT ~ 50 lbs-ft

Gear ratio to intermediate shaft is 1:2.5

Torque to input shaft of front gear box = 50 lbf-ft * 2.5 = 125 lbf-ft = 1500 lbf-in

Input Shaft Length ~ 6 in.

Axial: Torque/input shaft length = 1500/6 = 250 lbf

Radial: 125 lbf (from the intermediate shaft of the reduction box)

Using Weibull Parameters:

$$C_{10} \approx a_f F_D \left[\frac{x_D}{x_0 + (\theta - x_0)(1 - R_D)^{1/b}} \right]^{1/a} \qquad R \geq 0.90$$

$x_0 = 0.02$; $\theta = 4.459$; $b = 1.483$; $a = 3$ (for ball bearings); a_f = application factor = 1; R_D = reliability = 0.9

L_R = life rating = 10^6; L_D = Desired Life*Speed*60 =1000*1200*60

x_D = rating life multiple = $L_D/L_R = 72$

F_D = 125 lbf (this would be the maximum radial load the bearings would experience)

C_{10} = 527.2 lbf = 2.34 kN

For the bearing diameter needed in the front gear box design, the catalogue rating is very underrated, meaning any bearing selected will work for this application.

For the SAE BAJA vehicle, the needed life out of these bearings will be low due to the length of the competition, approximately 8 hours of total run time, so the bearing selection will be based on the load experienced by the adjoining shafts. The bearings that will be selected and purchased will be satisfactory for this application.

3.3.5 Nolan Stomp

Chain Drive

The chain drive is what allows power to transfer from the rear gearbox to the intermediate shaft which holds the dog box, under the seat, and up to the front gearbox. The dog box, when engaged, is what allows power to be transmitted to all four wheels at once. Using the properties from a 17-tooth sprocket held on the intermediate shaft, and a chain with ANSI number 50 (0.625 in. pitch), the speed variation of the chain drive can be calculated as follows:

$$V = \frac{Npn}{12} = \frac{17(0.625\ in.)(120\ rpm)}{12} = 106.25\ ft/min$$

$$V_{max} = \frac{\pi np}{12\sin\left(\frac{\gamma}{2}\right)} = \frac{\pi(120\ rpm)(0.625\ in.)}{12\sin\left(\frac{21.18}{2}\right)} = 106.84\ ft/min$$

$$V_{min} = \frac{\pi n p}{12 \frac{\cos\left(\frac{\gamma}{2}\right)}{\sin\left(\frac{\gamma}{2}\right)}} = \frac{\pi(120\ rpm)(0.625\ in.)}{12 \frac{\cos\left(\frac{21.18}{2}\right)}{\sin\left(\frac{21.18}{2}\right)}} = 105.02\ ft/min$$

Where γ is the pitch angle. From these values, the chordal speed variation can be derived as ΔV/V= 1.7%. This value paired with the number of teeth on the sprocket is close to the nominal value, where an increasing number of teeth has very little effect on the variation. Going forward, many variables can be changed to optimize the speed values, such as the pitch of the chain and number of teeth on the sprocket.

Dog Clutch

For the dog clutch, calculations needed to be done in order to choose the best geometric design moving forward in order to minimize the stress the component would experience during testing and operation. The following calculations compared square teeth vs curved teeth, similar to the concept generation and selection process later in the report. The calculation is using an outer diameter of 2 in, an inner diameter of 1 in, and a torque of 125 lbf*ft from a previous section. The force from the shaft on the dog clutch will be the same regardless of tooth geometry, which is

$$F = \frac{T}{\frac{\Delta r}{12}} = \frac{125\ lbf * ft}{\frac{0.5\ in.}{12}} = 3000\ lbf$$

From here, tooth geometry does impact stress experienced by the part.

$$\sigma_{square} = \frac{F}{A_{tooth}} = \frac{3000\ lbf}{0.2\ in\text{^}2} = 15000\ psi$$

Similarly,

$$\sigma_{curved} = \frac{F}{A_{tooth}} = \frac{3000\ lbf}{0.26\ in\text{^}2} = 11538.46\ psi$$

Given the nearly 26% difference, curved teeth will do a better job at handling the force from the rotation, due to the increased surface area. This calculation was heavily considered during the final selection process of the dog clutch.

3.3.6 Seth Scheiwiller

When designing a CVT transmission, efficiency must be taken into consideration to enhance the overall performance of the drivetrain. A CVT is divided into two critical components, the primary clutch and the secondary clutch. The primary clutch is mainly responsible for achieving an ideal shift RPM in which it engages the belt at maximum torque and begins to shift out into overdrive at the maximum optimal horsepower output of the Kohler CH440 10hp engine. The secondary clutch is responsible for transferring that power to the rear gearbox and is what determines the efficiency of the CVT transmission. Utilizing a system of equations that were later used in MATLAB and solved numerically, iterations were performed to determine the best possible combination of springs in the primary and secondary clutch that would maximize efficiency and prevent belt slippage.

$$\tau_{Max_{Secondary}}(ft.* lbf) = \left(T_{Taught\_Secondary} - T_{Slack\_Secondary}\right) * \frac{Radius_{Secondary}}{12}$$

Figure 9: Maximum transferable torque in secondary without belt slippage.

$$\tau_{Max_{Primary}}(ft.* lbf) = (T_{Taught\_Primary} - T_{Slack\_Primary}) * \frac{Radius_{Primary}}{12}$$

Figure 10: Maximum transferable torque in primary without belt slippage.

Using the above equations to calculate the maximum torque that both clutches can transfer, results were compared with the actual torque being transferred to ensure that belt slippage remains at a minimum. As stated by a master's thesis on CVT design, the efficiency of the secondary clutch is determined by getting the maximum transferable torque as close as possible to the actual torque being transferred. Results are displayed in the graphs below.

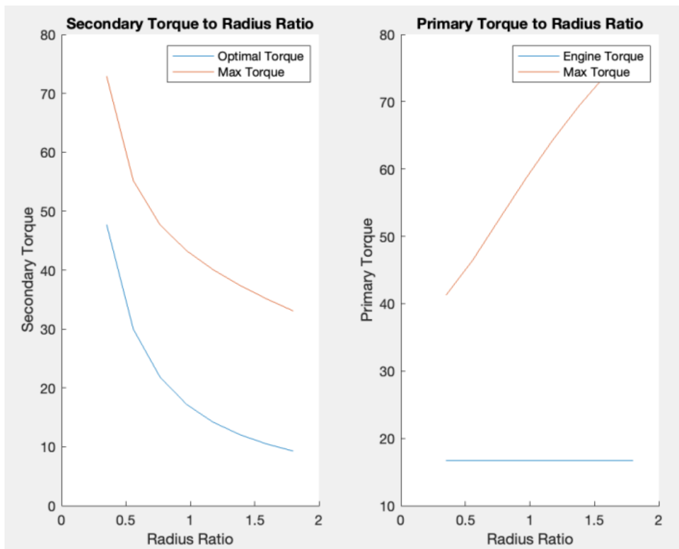


Figure 11: Calculated torques over radius ratio.

The results above prove that the maximum transferable torques calculated exceed the actual torques and the selectable parameters are sufficient to prevent belt slippage and maximize efficiency for our system.

3.3.7 Brennan Pongratz

This calculation is to figure out if this ANSYS simulation is accurate. It is a simple bending calculation for the spider on the primary CVT that will see forces from the cams pushing axially. Known values include the affected area at 0.14 in^2 and the sheer force of 125 lbf.

$$A = 0.14in^2 = 9.025 * 10^{-5}m^2$$

$$v = 125lbf = 556N$$

Calculating shear stress using:

$$T = \frac{4v}{3A} = 8.2MPa$$

$$\sigma' = \sqrt{3T^2} = 14.2MPa = 2.1kpsi$$

Comparing the allowable stress in aluminum for 10^8 cycles which is 10kpsi lets us know in that location we can expect no failures. This was also compared to values generated from Ansys which can be seen below.

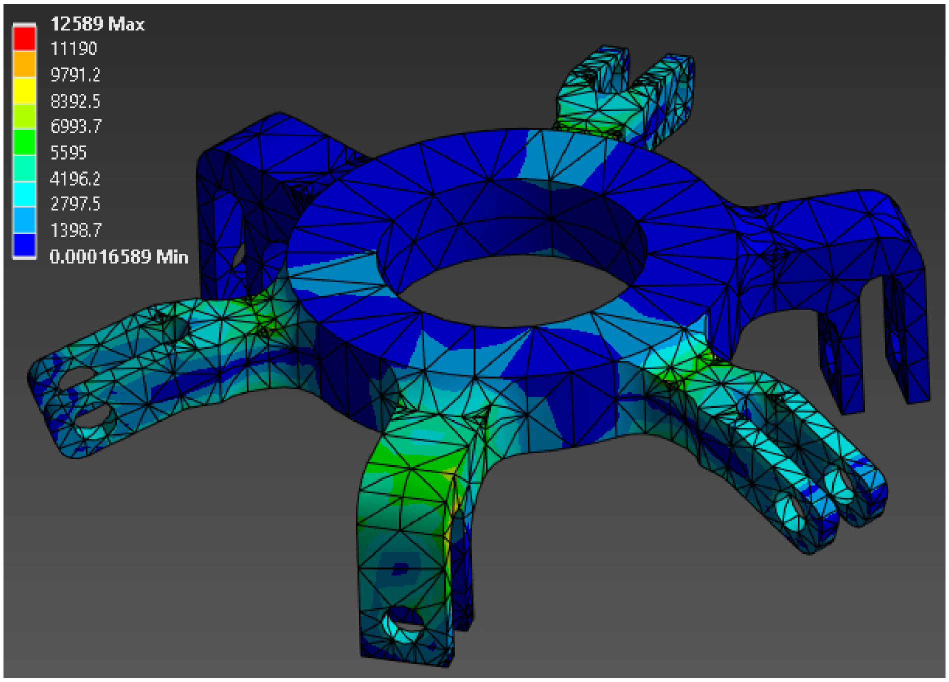


Figure 12: FEA of spider

The lighter blue indicating ~2kspi can be seen where the legs meet the body so we can assume this Ansys simulation is valid. Moving forward with this calculation would include a refined mesh and a dynamic loading analysis.

4 Design Concepts

4.1 Functional Decomposition

The team used a black box model to determine the inputs and outputs of the drivetrain. The most important component is energy out. It is important to decompose the functions of the engine and CVT to maximize the outputs, specifically torque, given the inputs.

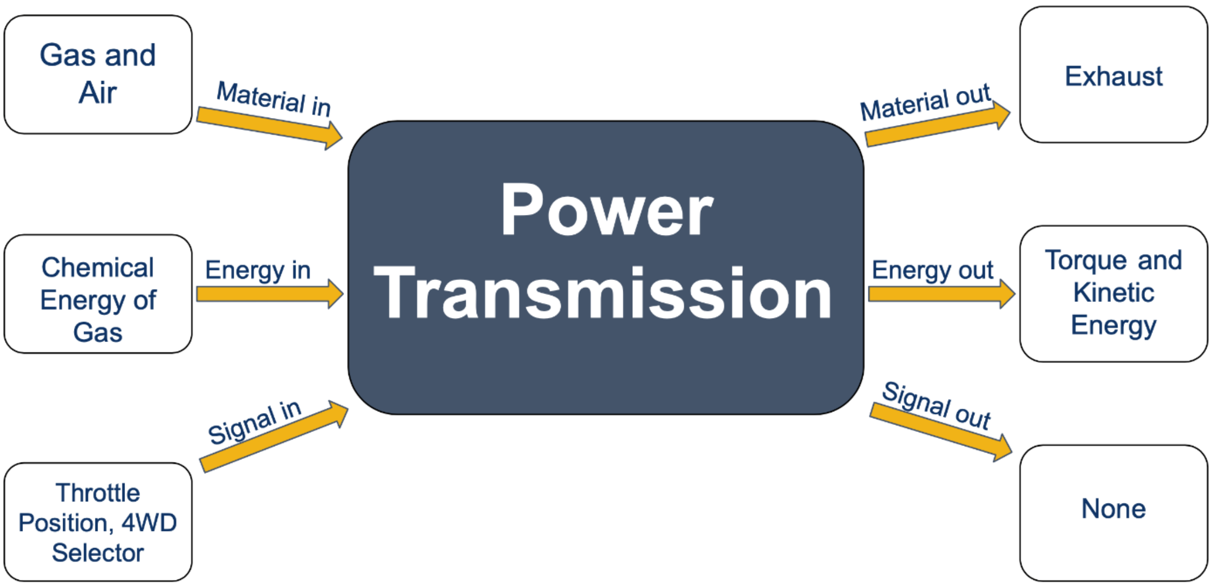


Figure 13: Drivetrain Black Box Model

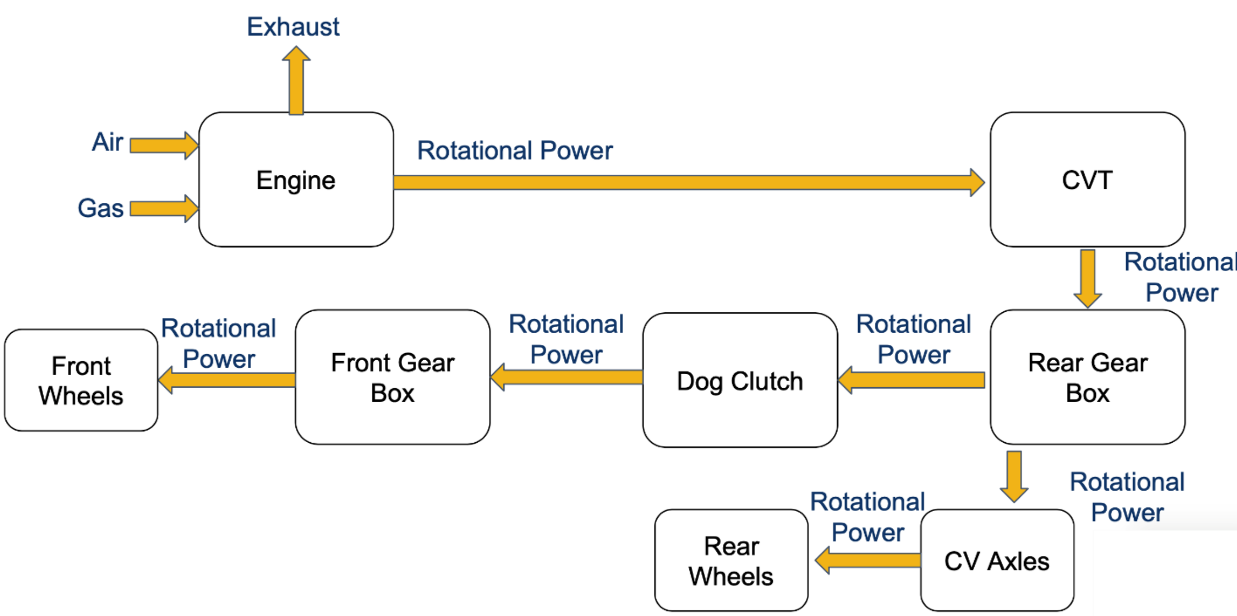
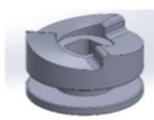
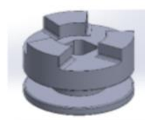
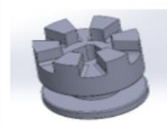


Figure 14: Drivetrain Functional Flow Diagram

The functional model provides a more thorough breakdown of all the aspects of the drivetrain and shows all the components needed to drive the vehicle. Knowing all the components needed to drive the vehicle is crucial, and further refinement of all these parts will help us be successful at competition.

4.2 Concept Generation

Table 6: Drivetrain Morphological Matrix

Concept	Design Variants		
CVT Actuating Mechanism	 Cams + Rollers	 Ramps + Rollers	 Electronic
Axles	 CV (Cup alone)	 CV (Cup-Shaft-Cup)	 U-Joints
Gears	 Spur Gears	 Helical Gears	 Bevel Gears
Hubs	 Spline	 Hex	 Press Fit
Dog Clutch	 3-tooth Curvic	 3-tooth Square	 6-tooth Square

Each sub-team had to decide within their space on a general sub-system design to refine for this application from the options shown in Figure #. Factors such as integration, ease of manufacturing, and cost were considered when making these decisions. More information on the decisions made for each respective sub-system is given in section 4.3.

4.3 Selection Criteria

4.3.1 CVT Actuating Mechanism

The first decision that had to be made when designing a CVT was to determine the type of actuating mechanism that will cause the primary clutch of the CVT to engage the belt and change gear ratios. Our top three choices were a cam and roller assembly that is utilized in Polaris vehicles, a ramp and roller assembly that Gaged uses in their systems, and an electronically controlled CVT. After determining the efficiency of a mechanically actuated CVT and analyzing the forces and stresses of that system, it was determined that a cam and roller assembly was the best route to take when deciding on an actuating mechanism.

4.3.2 Axles

Table 7: Axle Selection Table

	Pros	Cons
CV Axle	-Cheaper -Constant Velocity (smooth ride) -Plunging	-Complex geometry in the joints -Hard to replace
Universal Axle	-Easier to replace	-Susceptible to Binding -Expensive -Rough ride -Zero plunging -Acts as a suspension member

After considering this table, our team has chosen to move forward with CV axles due to their overall effectiveness and performance when compared to universal joint axles. CV Axles also allow plunging, which will be necessary with our trailing arm suspension setup. Our team has also decided to move forward with the integrated cup-shaft-cup CV axle design because it will increase the strength of the drivetrain subsystem, while reducing components, hardware and weight.

4.3.3 Gears

Table 8: Gear Selection Criteria

	Pros	Cons
Spur Gears	- Easy to Design - Easy to Manufacture - Better efficiency	- Noisy compared to Helical
Helical Gears	- Less noise than spur gears	- Hard to Design - Hard to Manufacture - Less efficient than Spur Gears
Bevel Gears	- Easy to Desing - Transmit power at a 90-degree angle	- Changing direction of power isn't a useful novelty for this project

The main decision that needed to be made with gear selection was the type of gears we were going to use. The three options that we were choosing from were spur, helical, and bevel gears. Spur gears are widely used by high performance teams in competition due to their simple design and ease of manufacturing. Helical gears are very efficient and produce less noise than spur gears, however they are difficult to manufacture and install. Bevel gears transmit power from non-parallel shafts, usually at a 90-degree shaft angle. This gear type is not ideal in the Baja vehicle due to the mounting of the engine and the limited space available in the front and rear. The team decided to use spur gears in both the front and rear gearboxes due to the simple design, manufacturability, and lower cost of the gears, as we are going to design and then have them manufactured out of house.

4.3.4 Hubs

Table 9: Hub fit Concept Selection

	Pros	Cons
Spline	-Serviceability	-Machinability -Proprietary Standards
Hex	-Serviceability	-Machinability -Axel Redesign
Press Fit	-Machinability	-Serviceability

The main factor to consider when designing the drive fit type for the hubs is the serviceability. Being able to slide the wheel hub on and off as the team needs is crucial for general ease of serviceability and impacts the suspension, brakes, and drivetrain. With that being considered, the press fit is out of the question, even though it would be easy to machine. Now, the team is left with spline and hex fit. Since the axles we are using are already splined, the team will most likely go with splining. However, the axels we have right now are splined to a proprietary standard, and with the fitment needing to be as secure as possible to drive the wheels, the spline standard cannot be guessed. The current plan is to take a matching hub to the axles, machine it down to get the receiving end and press fit that into the hub. Given the over complication of a spline fit as it stands~~,~~; a hex fit will always be in consideration until finalizing development.

4.3.5 Dog Clutch

Table 10: Dog Clutch Selection Table

	Pros	Cons
3 Tooth Square	Easier to manufacture, less intensive design process	Inefficient, will see a higher stress and impart higher shock on the system
3 Tooth Curved	Most efficient and stress reducing design	Tougher to design and manufacture
6 Tooth Square	Similar to 3 tooth, easier design process	Harder to manufacture than its benefits justify, least efficient design due to complexity

The factors guiding the selection process for the dog clutch were stress considerations on the part, as well as general designability and manufacturability. The first idea to be discarded was the 6 square tooth design. It would function similarly to the 3-tooth design, but would be harder to manufacture, with no discernable benefits to support it. Between the 3-tooth square and curved tooth design, the curved design would be harder to design and create, compared to the very simple square tooth design. However, the benefits that the curved design would come with, namely the better stress distribution, efficiency, and shiftability outweigh the possible downsides, which is what the team decided to come to the decision that the curved teeth model was the option to go with.

4.4 Concept Selection

The tables from the selection criteria section above are summarized in the table below. Each selection was made from pre-established calculations made during the initial design process and can be backed up conceptually or mathematically. Along with this, preliminary CAD assemblies with the selections below can be found in Appendix A.

Table 11: Drivetrain Concept Variant Selections

	Variants					
Subsystem	1	Results	2	Results	3	Results
Axle Types	CV (Cup-Shaft-Cup)	✓	CV (Cup alone)	X	Universal -Joint	X
Gear Types	Bevel Gear	X	Helical Gear	X	Spur Gear	✓
Clutches	3-Tooth Square	X	3-Tooth Curvic	✓	6-Tooth Square	X
CVT	Cams	✓	Ramps	X	ECVT	X
Hub	Spline	✓	Hex	X	Press Fit	X

5 CONCLUSIONS

The SAE Baja event is a collegiate design, fabrication, and business competition with the goal of competing with other universities on the design and development of a competitive race car. The goal is to have the custom vehicle perform in five dynamic events at an annual competition with those events being suspension and traction, maneuverability, hill climb, acceleration, and a four-hour endurance race. This competition encourages innovative designs while maintaining safety through a rigorous technical inspection that is split into engine inspection, frame inspection, general inspection, and a brake test. This project has been divided into three sub-teams to ensure successful completion of a high-performance custom vehicle. The sub-teams are chassis and ergonomics, suspension, steering, and brakes, and drivetrain. This report is concerned with the progress of the drivetrain sub-team which includes the concept generation and evaluation that has been done to determine major sub-system decisions, the literature review that has been conducted to make those decisions, functional models, mathematical modeling, and our first design iterations in CAD. The design goals for the drivetrain sub-team are to design a competitive drivetrain that can maintain a top speed of 33mph, optimize weight to be as low as possible, successfully climb a hill with a grade of 30-45 degrees, incorporate a selectable 4WD system, and outperform other universities.

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7 APPENDICES

7.1 Appendix A: Preliminary CAD from Concept Generation

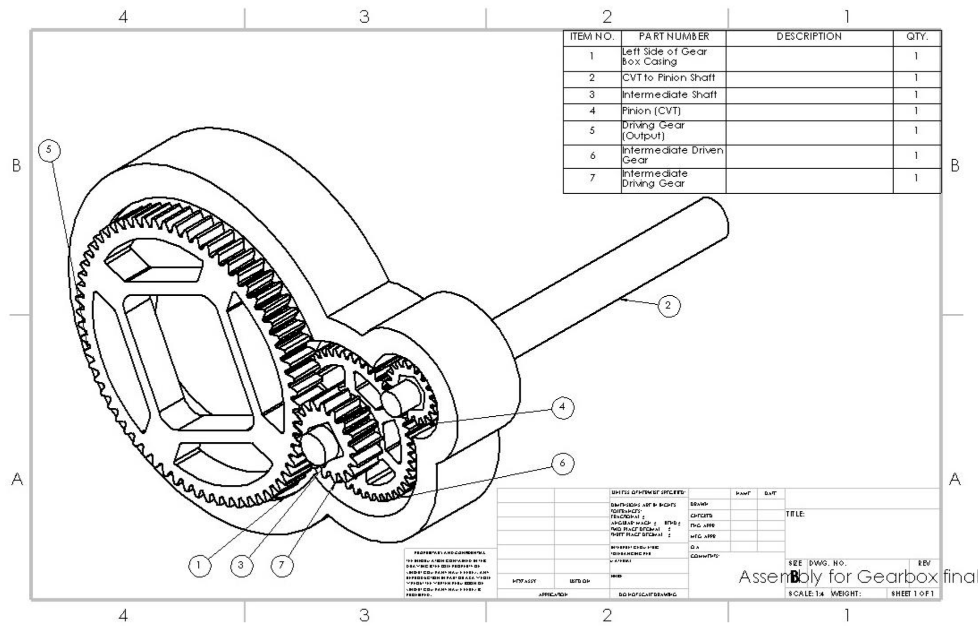


Figure 15: Assembly of Rear Gearbox

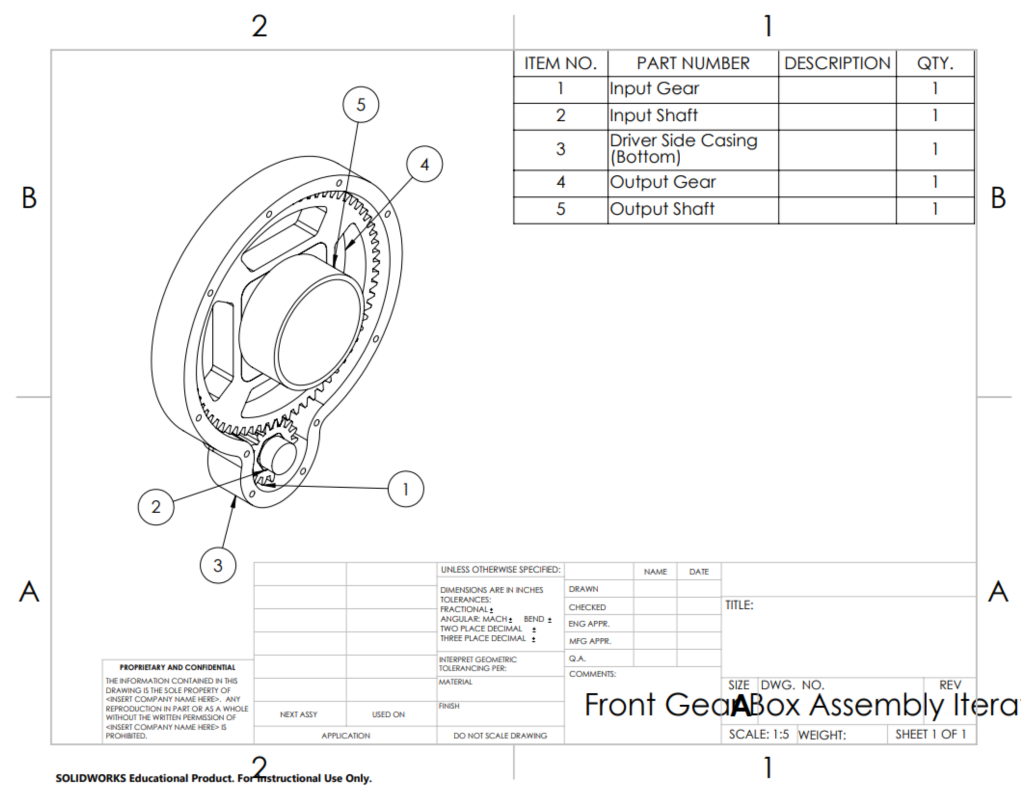
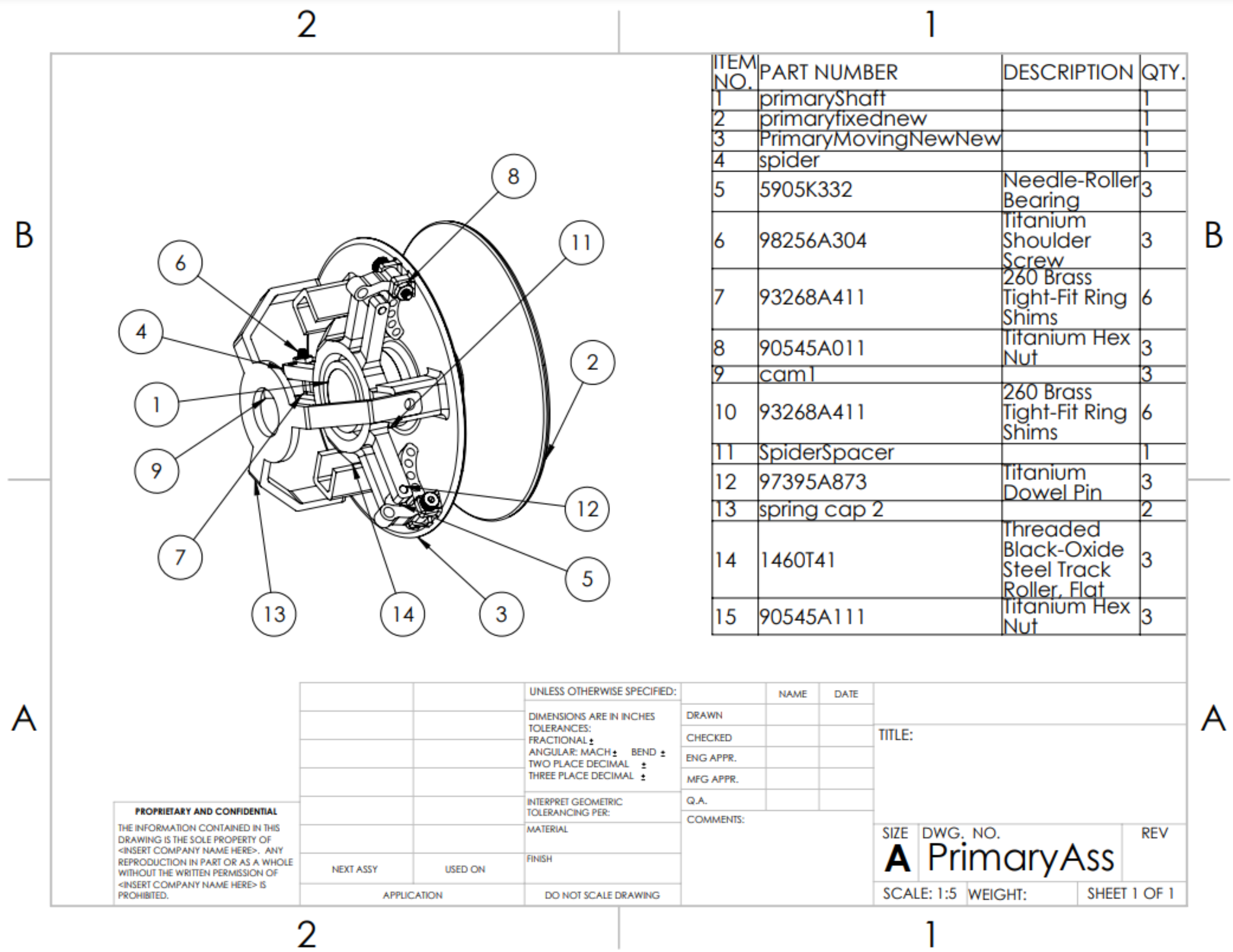


Figure 16: Assembly of Front Gearbox



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	primaryShaft		1
2	primaryfixednew		1
3	PrimaryMovingNewNew		1
4	spider		1
5	5905K332	Needle-Roller Bearing	3
6	98256A304	Titanium Shoulder Screw	3
7	93268A411	260 Brass Tight-Fit Ring Shims	6
8	90545A011	Titanium Hex Nut	3
9	cam1		3
10	93268A411	260 Brass Tight-Fit Ring Shims	6
11	SpiderSpacer		1
12	97395A873	Titanium Dowel Pin	3
13	spring cap 2		2
14	1460T41	Threaded Black-Oxide Steel Track Roller, Flat	3
15	90545A111	Titanium Hex Nut	3

Figure 17: Assembly of Primary Clutch of CVT