NAU SAE BAJA 2024-2025 – Drivetrain Team

Design Report 2

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Fall 2024-Spring 2025

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

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DISCLAIMER

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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.

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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. 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.

– *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 design include the 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.

– *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

– *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 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.

– *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.

– *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.

– *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	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
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

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

			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					

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

– *Benchmarking*

3..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 purchasing a mechanically actuated Gaged CVT.

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

3..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.

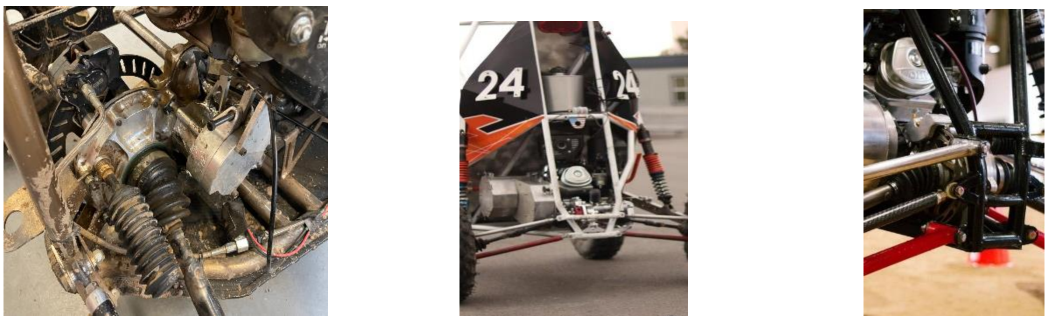


Figure 2: 2024 NAU Baja (left), 2024 RIT Baja (middle), 2024 Cornell (left)

3..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..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

– *Literature Review*

3..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.
- Single Row Cylindrical Roller Bearings – SKF [8]
  - This website contains the catalog ratings for cylindrical roller bearings for SKF bearings.
- Deep-Groove Ball Bearings Data Sheet – SKF [9]
  - This website contains the catalog ratings for deep groove ball bearings for SKF bearings.
- Single Row Tapered Roller Bearings – SKF [10]
  - This website contains the catalog ratings for single row tapered roller bearings for SKF bearings.

3..2 Matthew Dale

- Ball & Roller Bearing Design: Theory, Design, and Application [11]
  - Bearing design and fit to ensure solid fit.
- Non-Destructive Material Testing [12]
  - How to take a stress test to determine fatigue limit.
- Design And Analysis of Wheel Hub of Baja ATV in Ansys. [13]
  - Determining optimum wheel hub size and shape.
- Design and Weight optimization of wheel assembly components using FEA for BAJA [14]
  - Further hub development and optimization.
- Simulation and Optimization of Wheel Hub and Upright of Vehicle: A Review [15]
  - Force visualization and stress testing, along with further part development.
- Ansys Innovation Space [16]
  - 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 [17]
  - Material selection and further part development.
- ASM Material Data Sheet [18]
  - Material selection and material engineering data specs.
- Engineers Edge [19]

    - Press fit resources and equations.
- Ames Web [20]
    - Shrink fit resources and equations.

3..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 [21]
    - 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 [22]
    - 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 [23]
    - 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 [24]
    - 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 [25]
    - 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 [26]
    - 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.
- SAE AISI 4140 Steel Properties [27]
    - Listed tables and properties of my chosen material for the front and rear CV cups, shafts, brake mounts, and press fit's locations.
- ANSYS Software Tutorial. [28]
    - Tutorial for ANSYS for mathematical modeling and consequent design development and refinement.

- Command pro CH440 [29]
    - Information on the power and torque outputs of our engine that helps me calculate loads for our sprag bearings and CV cups and designing around these parameters.

3..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) [30]
    - 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 [31]
    - 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 [32]
    - 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 [33]
    - 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 [34]
    - 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.
- SKF – Roller Bearings [8]

- This website contains the data sheet that was used to select roller bearings for the input shaft of the front gearbox.

- SKF – Deep Groove Ball Bearings [9]
  - This website was used to select the ball bearings for the output shaft of the front gearbox.

3..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? [35]
  - 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 [36]
  - 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 [37]
  - 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 [22]
  - 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 [38]
  - 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 [39]
  - 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
- Industrial Roller Chains and Sprockets [40]
  - Catalog in order to find standard sizes for sprockets and chains, as well as approximate pricing for each
- How to Understand Tension Force Existing in Slack Side of Belt Drive [41]
  - Provides equations for calculating tight and slack side tension forces in a belt/chain drive, as well as definitions of variables used in their calculation

- Conveyor Chain- Designer Guide [42]
    - Gives ideas for how proper design, loadings, and lubrication should work in a functioning roller chain
- Subtended Angle Definition [43]
    - Provides further detail in how force calculations are made for a roller chain, especially that in which the sprockets are of differing sizes

3..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 [44]
    - 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 [45]
    - 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 [46]
    - 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 [47]
    - 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 [48]
    - This video provides a MATLAB simulation on the performance of a CVT transmission.
- Statically Indeterminate Beams: Method of Superposition [49]
    - This video provides an explanation of how to apply the method of superposition to solve redundant reactions like the one on the rear gearbox input shaft.
- ANSI Standard Roller Chains [50]
    - This website provides information on standard chain sizes which proved useful for deciding on the ideal chain number for the 4WD system of the vehicle.

– *Mathematical Modeling*

3..1 Dylan Carley

The vehicle must use a restricted Kohler CH440 engine, producing a maximum torque output of 18.5 ft-lb at 2400 rpm with a maximum rpm value of 3600. The first stage of power delivery is through the continuously variable transmission (CVT), which has a high gear ratio of 0.85:1 and a low gear ratio of 3.5:1. After the CVT, the second stage of power delivery is through the rear reduction gearbox which has a gear ratio of 6.98:1 and is responsible for increasing the torque to the desired level. The gearbox will have a total of six bearings within the casing and one external bearing; three of them are deep-grove ball bearings, two of them are tapered roller bearings, and two of them are cylindrical roller bearings. The analysis will validate the selection process for each bearing and prove that each bearing will be sufficient for the rear reduction gearbox. In Figure [] the bearings are shown in orange, shafts in red, and gears in pink.

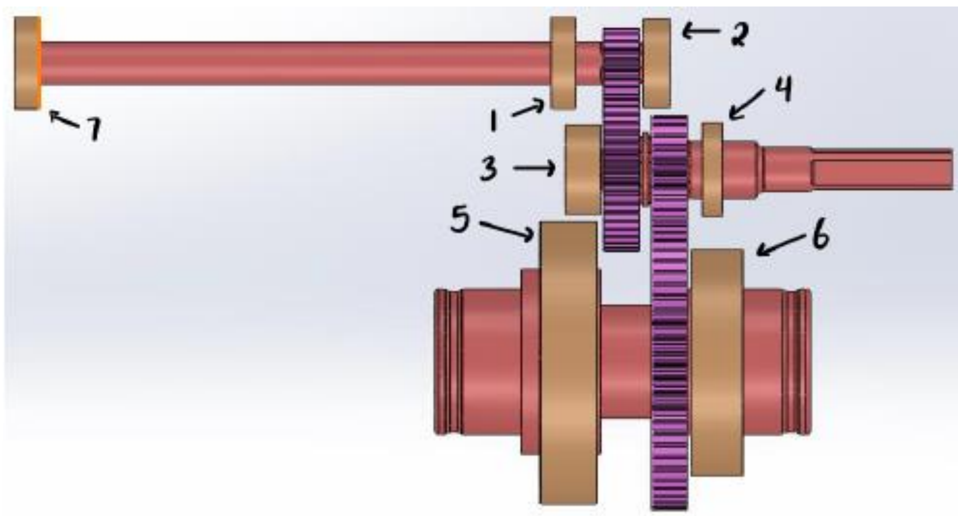


Figure 6: Top View Layout of Bearings

Gear Ratio (m_G):

$$m_G = \frac{N_G}{N_P} \tag{1}$$

N_G = Number of Teeth of big gear

N_P = Number of Teeth of little gear

Desired Life in Revolutions (L_D):

$$L_D = 60 * L_d * n_D \tag{2}$$

L_d = Desired life in hours = 100 hours

n_D = Desired speed of bearing in rpm

Application Factor (a_f):

Table 5: Application Factor

Table 11–5 Load-Application Factors

Type of Application	Load Factor
Precision gearing	1.0–1.1
Commercial gearing	1.1–1.3
Applications with poor bearing seals	1.2
Machinery with no impact	1.0–1.2
Machinery with light impact	1.2–1.5
Machinery with moderate impact	1.5–3.0

a_f = 1.0 (chosen for machinery with no impact)

a Value (a):

a = 3 (for Ball Bearings)

a = 10/3 (for roller bearings – cylindrical or tapered roller)

a = both values are used

Weibull Parameters:

$$x_0 = 0.020$$

$$\theta = 4.459$$

$$b = 1.483$$

Desired Bearing Speed (L_d):

$$L_{d_{12}} = \frac{Max\ engine\ rpm}{Low\ Gear\ Ratio\ of\ CVT} \tag{3}$$

$$L_{d_{34}} = \frac{L_{d_{12}}}{First\ Gear\ Reduction} \tag{4}$$

$$L_{d_{56}} = \frac{L_{d_{34}}}{Second\ Gear\ Reduction} \tag{5}$$

Max engine rpm = 3600

Low Gear Ratio of CVT = 3.5

First Gear Reduction = 2.44

Second Gear Reduction = 2.86

Rating Life Multiple (x_D):

$$x_D = \frac{L_D}{L_{10}} \tag{6}$$

L_D = Desired life in revolutions

L_{10} = Rated life in revolutions = 10^6

Max Torque from CVT ($T_{max_{CVT}}$):

$$T_{max_{CVT}} = Max\ Engine\ Torque * Low\ Gear\ Ratio * 12 \tag{7}$$

$Max\ Engine\ Torque$ = 18.5

$Low\ Gear\ Ratio$ = 3.5

Max Torque after First Gear Reduction ($T_{max_{1st\ red.}}$):

$$T_{max_{1st\ red.}} = T_{max_{CVT}} * First\ Gear\ Reduction \tag{8}$$

$T_{max_{CVT}}$ = 777 lb-in

First Gear Reduction = 2.44

Transmitted Load on Gears (W_t):

$$W_t = \frac{2T}{d} \tag{9}$$

T = Max torque from CVT

d = pitch diameter of gear/pinion

Tangential Force on Gears ($F^t{}_{12}$):

$$F^t{}_{12} = W_t \tag{10}$$

W_t = Transmitted load from previous

Radial Force on Gears ($F^r{}_{12}$):

$$F^r{}_{12} = F^t{}_{12} \tan \emptyset \tag{11}$$

$F^t{}_{12}$ = Tangential load from previous

$\emptyset$ = Pressure angle = 20 deg.

Total Force on Gears (F_{12}):

$$F_{12} = \frac{F^t{}_{12}}{\cos \emptyset} \tag{12}$$

$F^t{}_{12}$ = Tangential load from previous

$\emptyset$ = Pressure angle = 20 deg.

The sum of Moments on Shafts:

$$\sum M_{location} = 0 \tag{13}$$

The sum of Forces in the y-Direction on shafts:

$$\sum F_y = 0 \tag{14}$$

Total Radial Load (F_d):

$$F_d = \sqrt{F_y{}^2 + F_z{}^2} \tag{15}$$

F_y = Radial load in y-dir.

F_z = Radial load in z-dir.

Equivalent Radial Load (F_e):

$$F_e = 0.4F_D + KF_a \tag{16}$$

F_D = Radial load

$K = 1.5$

F_a = Axial load

Induced Load (F_i):

$$F_i = \frac{.47 * F_D}{K} \tag{17}$$

F_D = Radial load

$K = 1.5$

Equivalent Load (F_{eA}/F_{eB}):

$$IF\ F_{iA} \leq (F_{iB} + F_{ae})\ \begin{cases} F_{eA} = 0.4F_{rA} + K(F_{iB} + F_{ae}) \\ F_{eB} = F_{rB} \end{cases} \tag{18}$$

F_{iA} = Inducted load on bearing A

F_{iB} = Inducted load on bearing B

F_{ae} = Equivalent axial load

K = 1.5

F_{rA} = Radial load on bearing A

F_{rB} = Radial load on bearing B

Catalog Rating (C_{10}):

$$C_{10} = a_f F_d \left[\frac{x_D}{x_0 + (\theta - x_0)\left[ln\left(\frac{1}{R_D}\right)\right]^{1/b}} \right]^{1/a} \tag{19}$$

a_f = Application Factor

F_d = Total radial load/Equivalent load

x_D = Life rating multiple

x_0 = 0.020

θ = 4.459

b = 1.483

R_D = Desired reliability

a = 3 or 10/3

Conversion Factor from lbs to kN:

$$kN = lbf * .00445 \tag{20}$$

Reliability (R):

$$R = exp\left[-\left[\frac{x_D\left(\frac{a_f F_D}{C_{10}}\right)^a - x_0}{(\theta - x_0)} \right]^b \right] \tag{21}$$

a_f = Application Factor

F_d = Total radial load/Equivalent load

x_D = Life rating multiple

Table 6: Final bearings with reliability

Bearing #	Desired Speed (rpm)	Life Rating Multiple	Radial Load (lbf)	Catalog Rating (kN)	Chosen Catalog Rating (kN)	Dimensions (IDxODxT)	Reliability
1	4235.3	25.41	413.13	5.42	7.25	19.05 x 41.275 x 11.113	0.973
2	4235.3	25.41	1085.08	12.77	20	17 x 40 x 12	0.979
3	1732.6	10.4	2303.3	20.73	27.5	17 x 40 x 16	0.96
4	1732.6	10.4	483.75	4.71	7.02	25 x 42 x 9	0.984
5	606.42	3.64	1888.1	12.4	286	82.55 x 124.41 x 25.4	0.9999
6	606.42	3.64	1354.5	8.89	103	65 x 100 x 23	0.9999
7	4235.3	25.41	148.09	1.94	7.25	19.05 x 41.275 x 11.113	0.9999

Table 7: Bearing type with item #

Bearing #	Type of Bearing	Sealed or Not	Item #
1	Deep Groove Ball Bearing	Sealed	R12-2RS
2	Cylindrical Roller Bearing	Not	NJ 203 - ECP - SKF
3	Cylindrical Roller Bearing	Not	NJ2203 - ECP - SKF
4	Deep Groove Ball Bearing	Sealed	61905 - 2RS - SKF
5	Tapered Roller Bearing	Not	27687 - 27620 - SKF
6	Tapered Roller Bearing	Not Sealed	32013 - X - Q - SKF
7	Deep Groove Ball Bearing	Sealed	R12-2RS

From the original selections made before completing the calculations, most bearings were changed to accommodate the values that the results showed. The bearings that did not change were 5 and 6, tapered roller bearings with large catalog rating values. The analysis bearing 7 was added on the end of the input shaft to help stabilize the forces acting from the CVT. Once bearing seven was added, the radial loads on bearings 1 and 2 were reduced significantly. Bearings 1 and 7 are the same because both bearings are on a .75-inch shaft. Bearing 2 had to be increased in size to support the radial load exerted by the gears and CVT; it also changed from a deep-groove ball bearing to a cylindrical roller bearing, which can support more radial load with a smaller profile. Bearing 3 experiences the highest radial force since two radial loads act on the shaft from two gears, and the 4WD forces act on the opposite shaft. Bearing 3 also had to be changed from a deep-groove ball bearing to a cylindrical roller bearing for the same reasons as bearing 2. Bearing 4 decreased in size since the radial forces were lower than initially expected. The total cost for all the bearings inside the casing and the one external comes to $342.12.

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.

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

Max Impact Force = 1348 N (as calculated by suspension team)

Youngs Modulus for 6061-T6 Aluminum = 69 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.

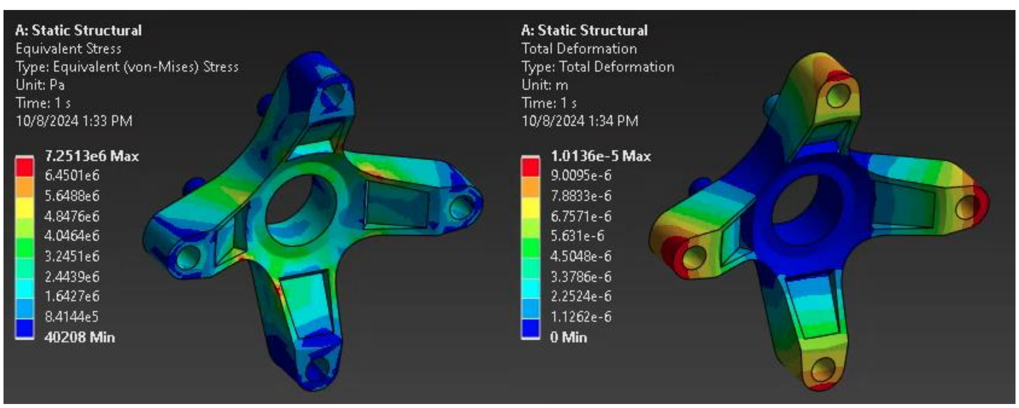


Figure 7: Static Structural ANSYS Test of Initial Hub Design

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.

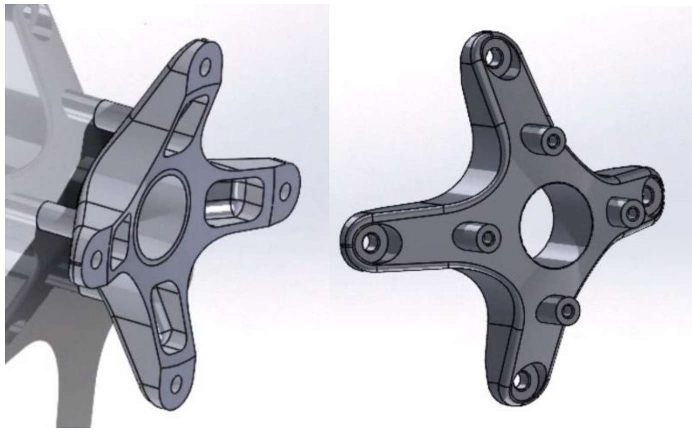


Figure 8: Developed Hub Design

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.

A major part of ensuring a successful hub is how the female spline is going to be attached to the hub. This was a major failure with last year's baja team, and nailing it this year is to be expected. The plan is to buy a wheel hub that matches the axle we are running, and machine that hub down to just short of the spline. This part will be attached to the wheel hub and used to transmit power to the wheels. The attachment between the hub and female spline must be strong enough to transmit 415 lb ft of torque for the duration of competition.

The first assumption to be made is the materials. We know the hub is aluminum 6061 T6, however the exact type of steel used for the spline is unknown, as well as what sort of heat treatment (if any) was used to harden the part. However, it is likely that a heat treatment was used on the splining itself, and the rest of the part has mild to no heat treatment.

Another assumption to be made is the interference between the spline and the hub. Using the machinists handbook, I used a Class FN 4 fit. This fit is a heavy drive fit that is suitable for parts that can be highly stressed. Under FN 4, the recommended interference is between 1.5 and 3.1 thousandths of an inch for a size range of 1.19 to 1.58 inches. Seeing how the inner diameter of the hub is 1.5 inches the interference should be 3 thousandths of an inch.

And finally, I assumed the coefficient friction between steel and aluminum to be 0.4, although the coefficient depends on a bunch of different factors such as surface finish, and we are also planning on using loctite to help lubricate the press, as well as help secure the fit.

The way we are going to achieve a 3 thousandth interference is obviously going to be forcing the parts into each other with a press, but in order to make things go smoother we will heat the aluminum hub to expand it, and cool the steel spline to shrink it. The steel spline will be left in a freezer overnight to get as cold as possible, while the hub will be heated to just under 200 degrees fahrenheit to avoid change in structural properties.

The change in material size can be calculated as:

$$\epsilon_x = \epsilon_y = \epsilon_z = \alpha(\Delta T)$$

Equation 1

With the hub heated to 199 degrees fahrenheit, it should expand by 0.00169 inches, and with the spline cooled to 0 degrees, it should shrink by 0.000432 inches. We can then use another equation to find the pressure of the fit after the parts are heated and cooled respectively, with a new interference of 0.000879 inches.

$$P = \frac{\delta}{\frac{d}{E_0}\left(\frac{d_o^2 + d^2}{d_o^2 - d^2} + \nu_0\right) + \frac{d}{E_i}\left(\frac{d^2 + d_i^2}{d^2 - d_i^2} - \nu_i\right)}$$

Equation 2

With the equation solved, there is a pressure of 6.345 MPa between the two parts. We can then calculate the force needed to press the parts together with this equation:

$$F = \mu P A$$

Equation 3

Solved, we get a force of 10.7 tons needed to press the cooled spline into the heated hub. Then I went back and reused the pressure equation, but this time with an interference of .003 inches for when both parts are back to room temperature to find the force needed to press them in without temperature change, and then max torque with another equation.

$$T = \frac{Fd}{2}$$

Equation 4

The max torque the fit will be able to hold is 627 NM, or 462.45 lb ft once the parts return to room temperature.

Seeing how the max torque expected to be transmitted to the wheels is 415, this should be strong enough for competition, and has a factor of safety of 1.11. The force needed to press the parts together is quite high, although should be doable with the resources available in the machine shop.

3..3 Ethan Niemeyer

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

Material Used - 4140 HT Steel:

Steel	Tensile Strength, Mpa (ksi)	Yield Strength, Mpa (ksi)	Hardness (HB)	Young modulus (psi)	Bulk modulus(pa)	Shear Modulus(pa)	Conditions
AIS 4140	1020 (148)	655 (95)	302	1.02989E+06	5.6351E+09	2.752E+09	Normalized at 870 °C (1600 °F)

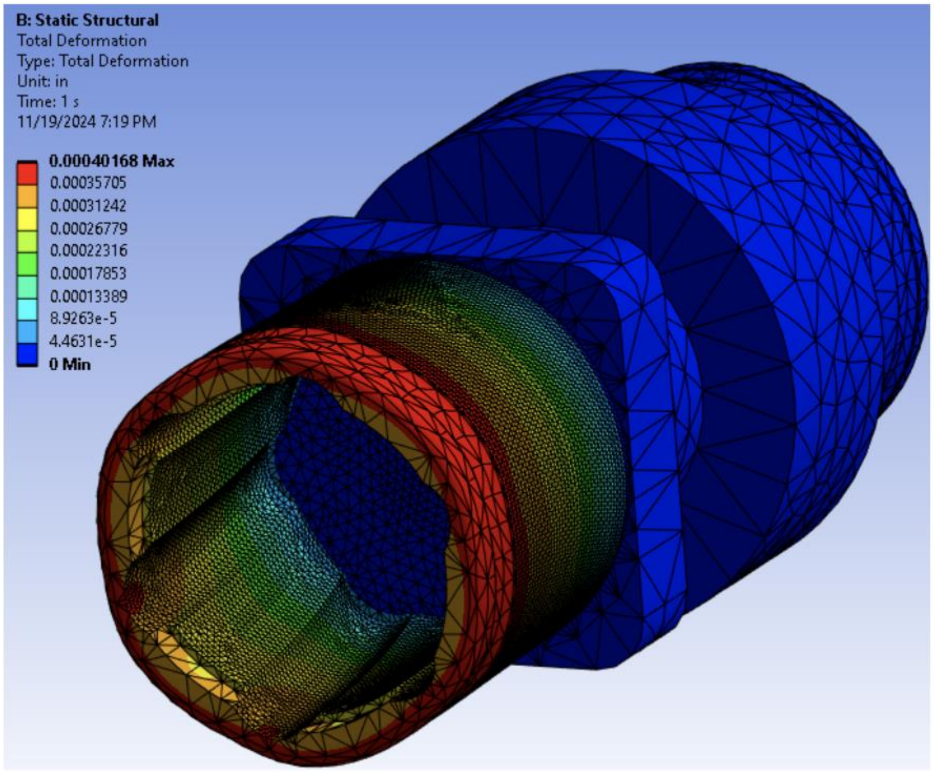
Continued:

Cycles	Alternating Stress (psi)	Poisson's ratio
1E+07	50763	0.29

In ANSYS I applied 503.57 ft-lb (calculated induced load) across the inside and outside of the CV cup, acting as the outer race of a bearing (cut out circles inside cv cup) in order to simulate the cup in a maximum torque scenario. I applied fixed mounts at the square press fit, and bearing axial and radial loads at each bearing location; on the brake disc mount and the opposite side CV Cup. The results from this will inform me of how good my thickness calculation from before are, as I am using that calculated thickness. This is also the thinnest or weakest part of the CV cup and that is why this part of it is being analyzed. The whole component is one piece made from the same 4140 HT steel material, so if this area of the component survives the experienced forces, the rest of the CV cup will also. Finally, plug all assumptions into ANSYS and upload the STP file of the CV Cup-Shaft-Cup, and then perform the analyses.

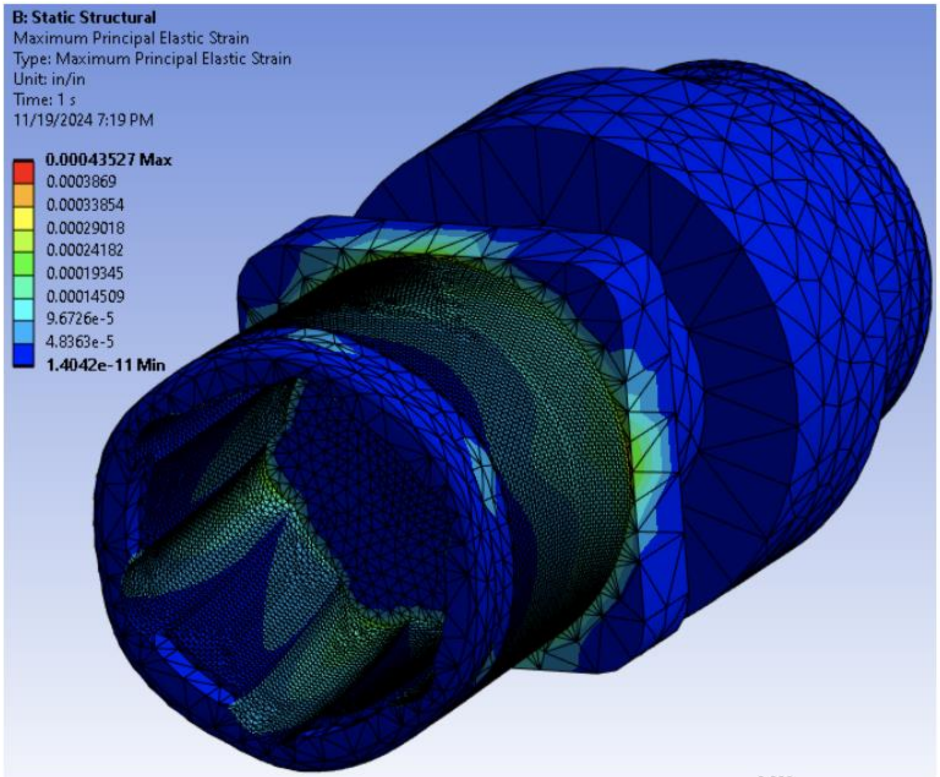
Analyses performed: Total Deformation, Maximum principal strain, and Von-Mises stress.

Results



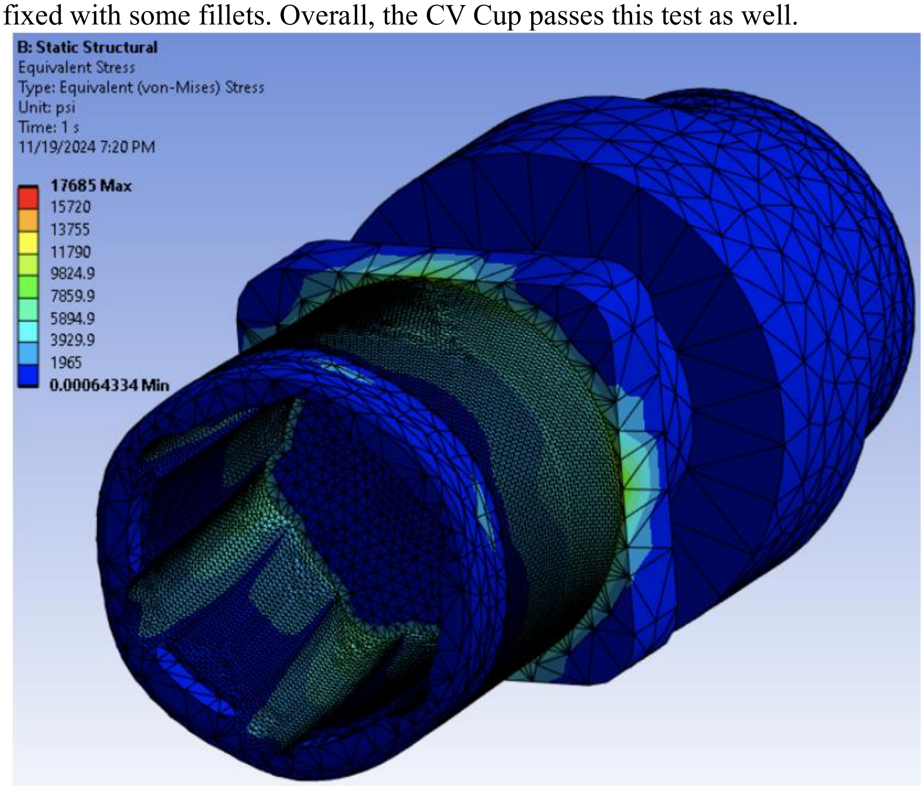
Total Deformation

As you can see, the deformation is worst toward the end of the CV Cup where it is highlighted in red. If you refer to the respective chart and values, you can see that at worst the deformation is 0.00040168 inches, which is negligible at best. Therefore, with our torques and loads we experience at their peak values, the CV Cup will completely hold up in this regard. Even if our expected values change and increase in value for any reason, the CV Cup can easily be thickened accordingly.



Maximum Principal Elastic Strain

The maximum principal elastic strain is shown in the above figure and really doesn't have any critical areas with our experienced loads. The worst is experienced at the sharp cracks of the square press fit. This can easily be fixed with some fillets. Overall, the CV Cup passes this test as well.



Equivalent Stress

This figure shows that the equivalent stress is also worst at the cracks of the square press fit. This will also be fixed via fillets and is a very small value to begin with anyways. After that is done, crack propagation will be kept to a minimum, and the part will be as strong as it can be, without being made from an extremely expensive material. Overall, the CV Cup passes all three tests, and this study will be saved to be further refined in the future if needed.

Sprag Bearing loads:

-Restricted Kohler CH440:
Max-torque = 18.5 ft-lb @ 2400 rpm
-Final drive gear ratio will be 3.9 * 6.98 = 27.22 with CVT and front gearbox
- 18.5 ft-lb torque * 27.22
=503.57 ft-lb
*.9 for mechanical loss = 453.23 ft-lb
Sprag rating must be higher than this value.

3..4 Rowan Jones

<u>Front Gear Box</u>

Gear Dimensions

Table 8: 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	59	4.91667	3.29167	0.625
Pinion	20	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 1:1.09.

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

The most recent iteration of the front gearbox has larger dimensions to accommodate filets and other structural members on the outside of the casing. Both shafts in the front gearbox were completely redesigned to have higher safety factors, changing the geometry of the casing in turn.

The casing material will be 6061-T6 aluminum due to a desirable strength to weight ratio. It is relatively light weight compared to steel and has enough strength to withstand the forces exerted by the bearings. The shafts will be made of 4140 steel and the gears will be made of 4340 HT steel. These components will be experiencing the most amount of force in the gearbox, so steel was the most logical choice.

To calculate the loads experienced by the bearings and the gears, a variety of equations were used. The main governing equations for the gears are:

$$T = T_{engine} Ratio_{CVT} \frac{d_g}{d_p} \tag{1}$$

Where T is the torque experience by the input shaft, T_{engine} is the output torque of the engine, $Ratio_{CVT}$ is the ratio from eh CVT, and d_g and d_p are the pitch diameters of the gear and pinion respectively. The transmitted load is calculated by:

$$W_t = \frac{2T}{d_p} \tag{2}$$

Where W_t is the transmitted load of the pinion form the shaft, T is the torque from the reduction, and d_p is the pitch diameter of the pinion. The radial load is defined by:

$$W_r = W_t \tan(\varphi) \tag{3}$$

Where W_r is the radial load, W_t is the transmitted load, and φ is the pressure angle. These loads will be used in the ANSYS software later to demonstrate what the forces the gears will experience.

The output values of these equations with the appropriate values are:

W_t = 2750 lbf

W_r = 1000 lbf

T = 2292 lbf-in

These values will be used in the finite element analysis of the gears in the next section.

To calculate the bearing reaction forces, moment diagrams were created based on the torque coming from the sprocket. For simplicity and cleanliness, only the output values of these moment diagrams are displayed below.

For the bearings on the input shaft:

Bearing A:

y-direction: 2143 lbf

z-direction: 1050 lbf

Bearing B:
y-direction: 1529 lbf
z-direction: 1501 lbf
For the bearings on the output shaft:
Bearing A:
y-direction: 1245 lbf
z-direction: 987 lbf
Bearing B:
y-direction: 1245 lbf
z-direction: 987 lbf
Bearings were chosen based on the loads listed above as well as the geometrical parameters of the casing. The chosen bearings will be implemented into Prototype 2.

Finite Element Analysis on the Front Gearbox Components

For the setup of the finite element analysis (FEA) on the gear and pinion of the front gearbox, the material properties needed to be identified. The material of the gears in 4340 HT steel, but since the heat-treated version of this steel does not exist in the ANSYS explicit materials list, the 4340 steel that did exist was modified to reflect the properties of the heat-treated version. Below is the breakdown of the different properties used in this FEA.

Properties of Outline Row 3: STEEL 4340

	A	B	C	D	E
1	Property	Value	Unit		
2	Material Field Variables	Table			
3	Density	0.28288	lb in^-3		
4	Isotropic Elasticity				
10	S-N Curve	Tabular			
12	Tensile Yield Strength	1.859E+05	psi		
13	Tensile Ultimate Strength	1.25E+05	psi		
14	Specific Heat Constant Pressure, C_p	477	J kg^-1 ...		
15	Johnson Cook Strength				
24	Bulk Modulus	2.3061E+07	psi		
25	Shear Modulus	1.1864E+07	psi		

Engineering Data for 4340 steel

Table of Properties Row 10: S-N Curve

	A	B
1	Cycles	Alternating Stress (psi)
2	1E+08	84709
*		

S-N Curve Data for 4340 steel

Table of Properties Row 4: Isotropic Elasticity

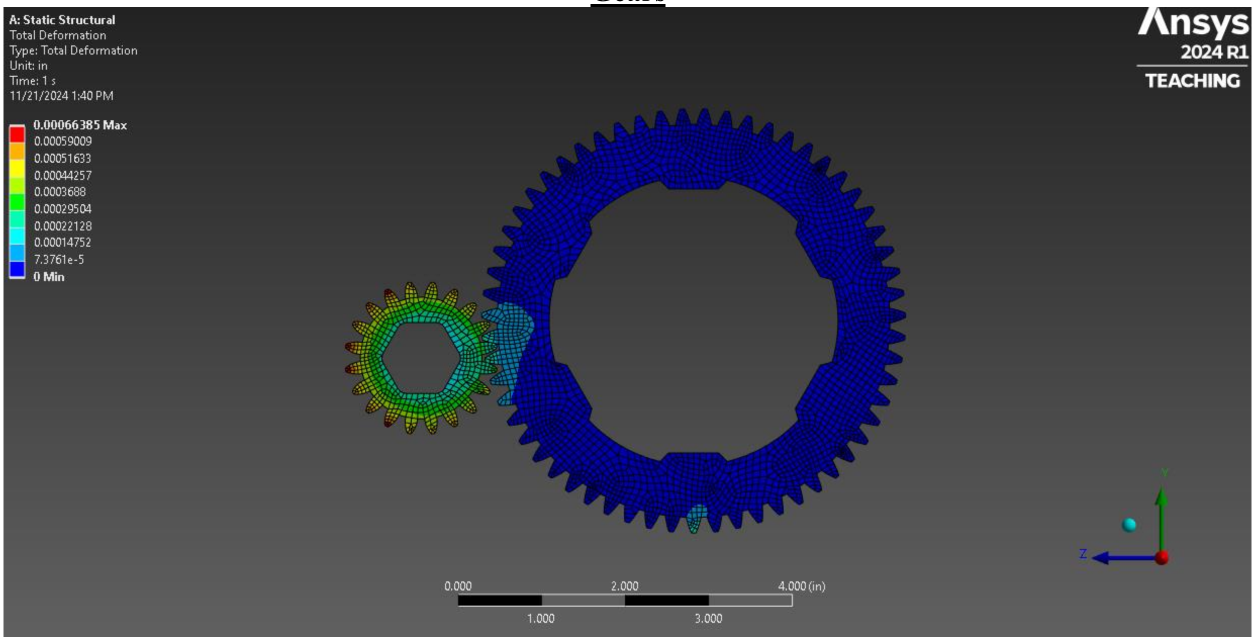
	A	B	C	D	E
1	Temperature (F)	Young's Modulus (psi)	Poisson's Ratio	Bulk Modulus (psi)	Shear Modulus (psi)
2		3E+07	0.3	2.5E+07	1.1538E+07
*					

Isotropic Elasticity definition for 4340 steel

The following process was used to set up the forces being experienced and the supports on the gears:

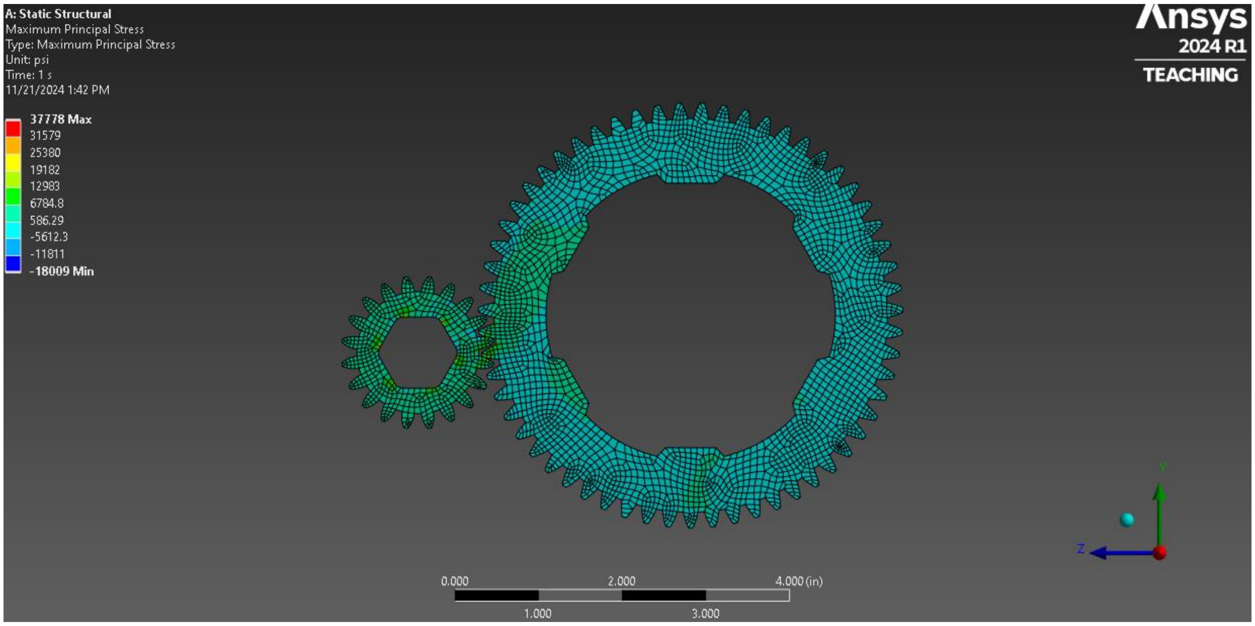
1) Apply fixed support on flat edges of the gear
2) Apply frictionless support to flat edges of the pinion
3) Apply a moment (2500 lbf-in) to all of the exterior faces (teeth and sides, not internal)
4) Apply radial load to gear (1000 lbf)
5) Generate finest mesh possible

Gears



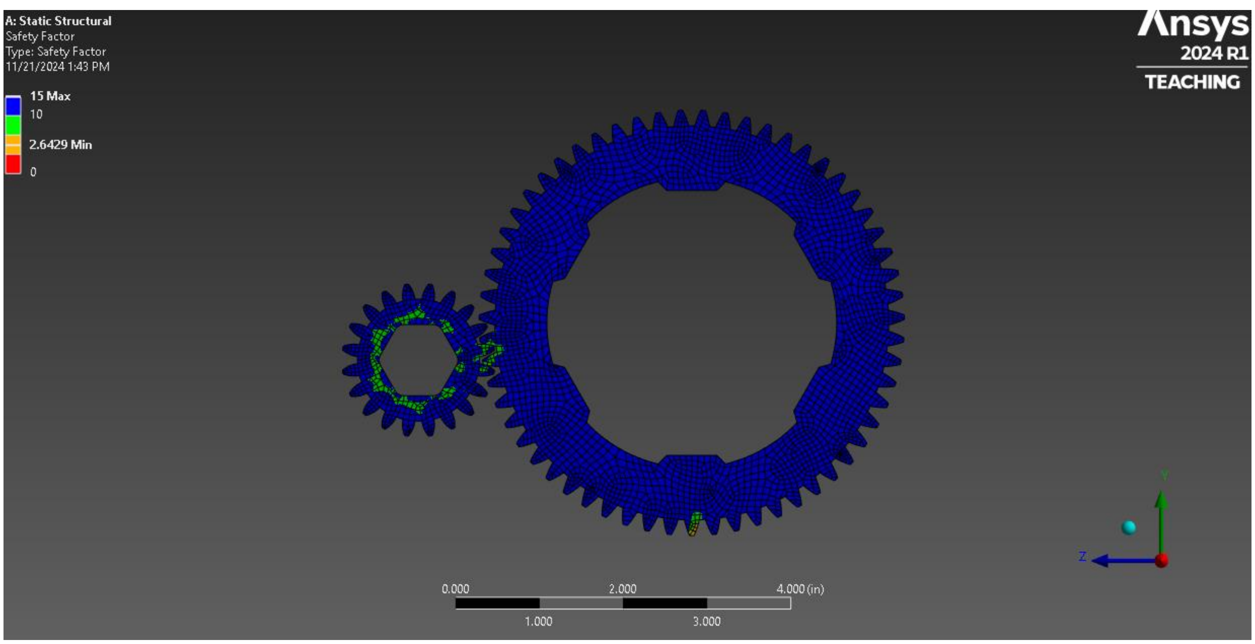
ANSYS Total Deformation

The maximum deformation of 6.5E-04 inches occurred at the tips of the teeth of the pinion, which is expected as the moment is applied on all exterior faces of the pinion. This amount of deformation is minimal and will not compromise the integrity of the component.



ANSYS Maximum Principal Stress

The maximum principal stress of 37,778 psi occurred at the contact point of the teeth of the pinion and gear. As stated in Presentation 1, the allowable contact stress was 65,982 psi, which is a little under 20 kpsi over the maximum principal stress. This is acceptable and will allow for a high safety factor for the gears.



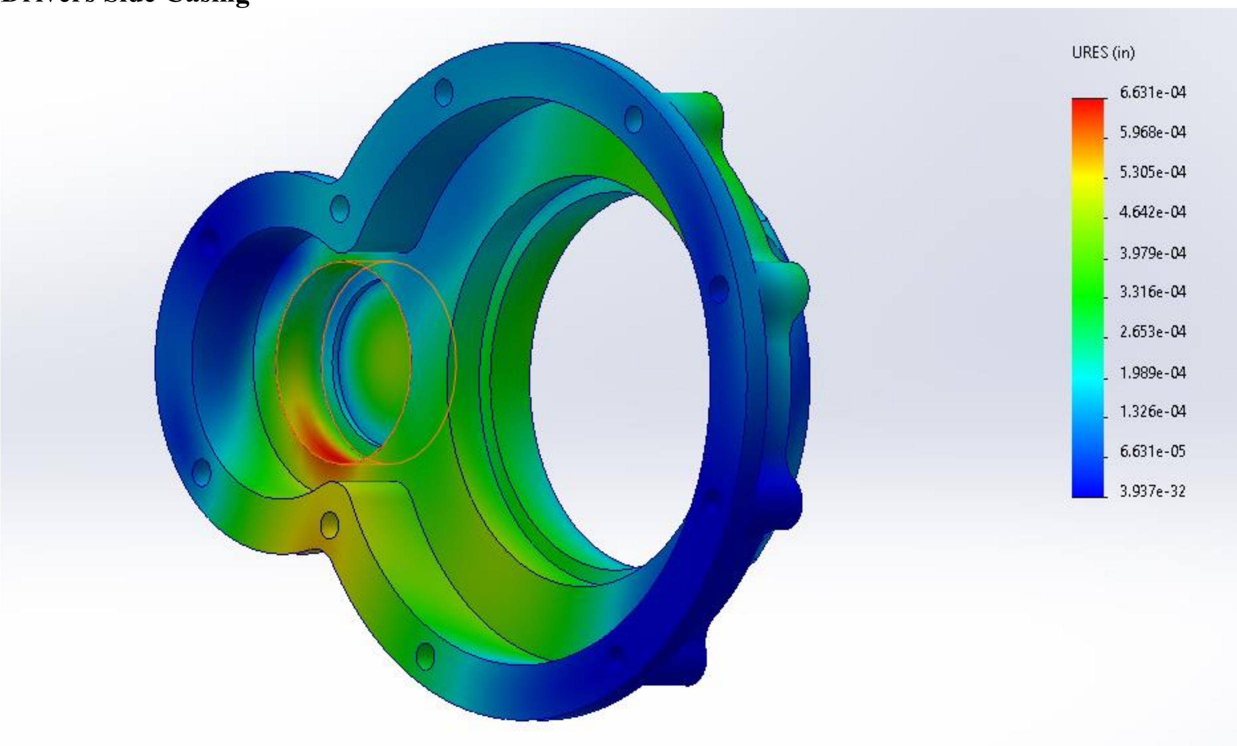
ANSYS Factor of Safety

The lowest factor of safety of 2.6 occurred at the contact point of the teeth meshing and at the radial load applied. This is an acceptable factor of safety for the gears, especially considering this simulation is the worst-case scenario for the gears.

The next section will focus on the casing of the gearbox and how the bearing forces will affect the casing. The setup process for this FEA, which was performed using SOLIDWORKS Simulation, is listed below.

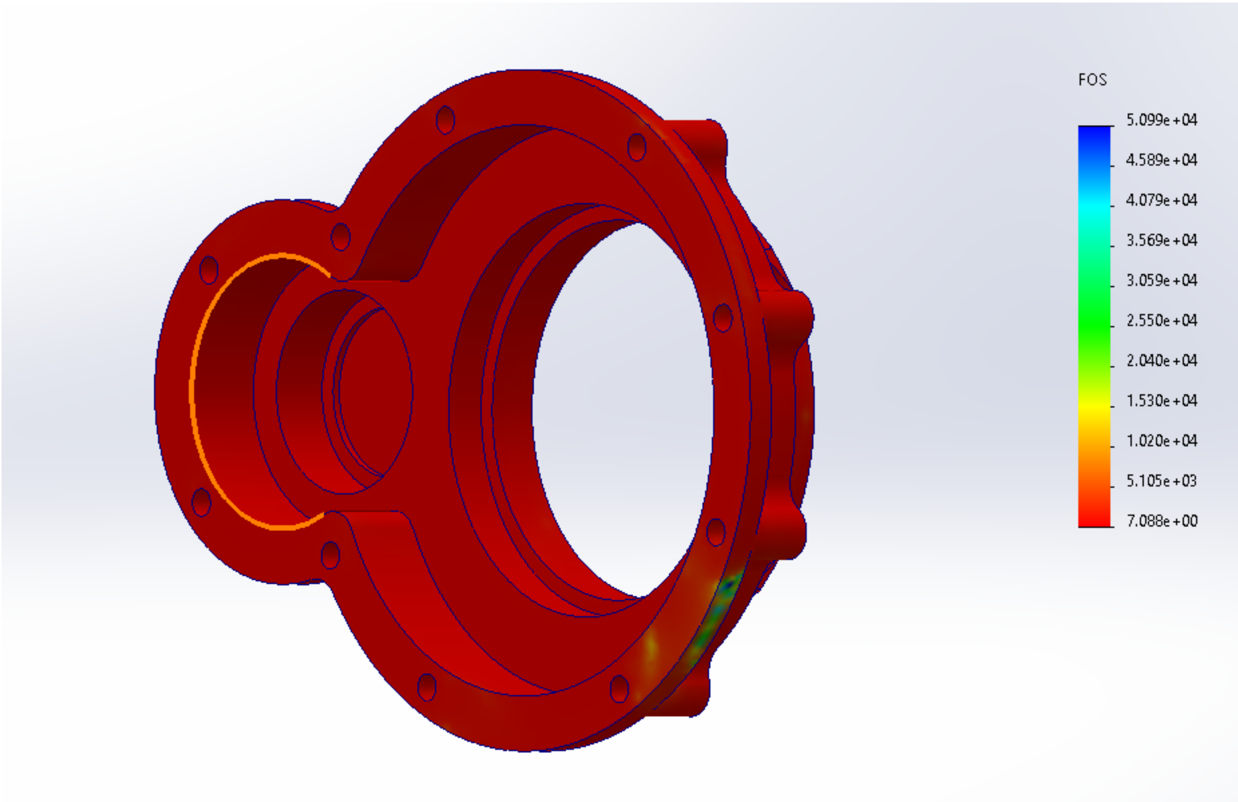
1) Apply material properties of 6061-T6 aluminum to casing
2) Apply fixed supports to the mounting holes of the casing
3) Apply forces normal to both bearing retention locations
   a. 1300 lbf for the output shaft bearing forces
   b. 2500 lbf for the input shaft bearing forces
4) Generate finest mesh possible

Drivers Side Casing



Displacement (Deformation) of the Drivers Side Casing

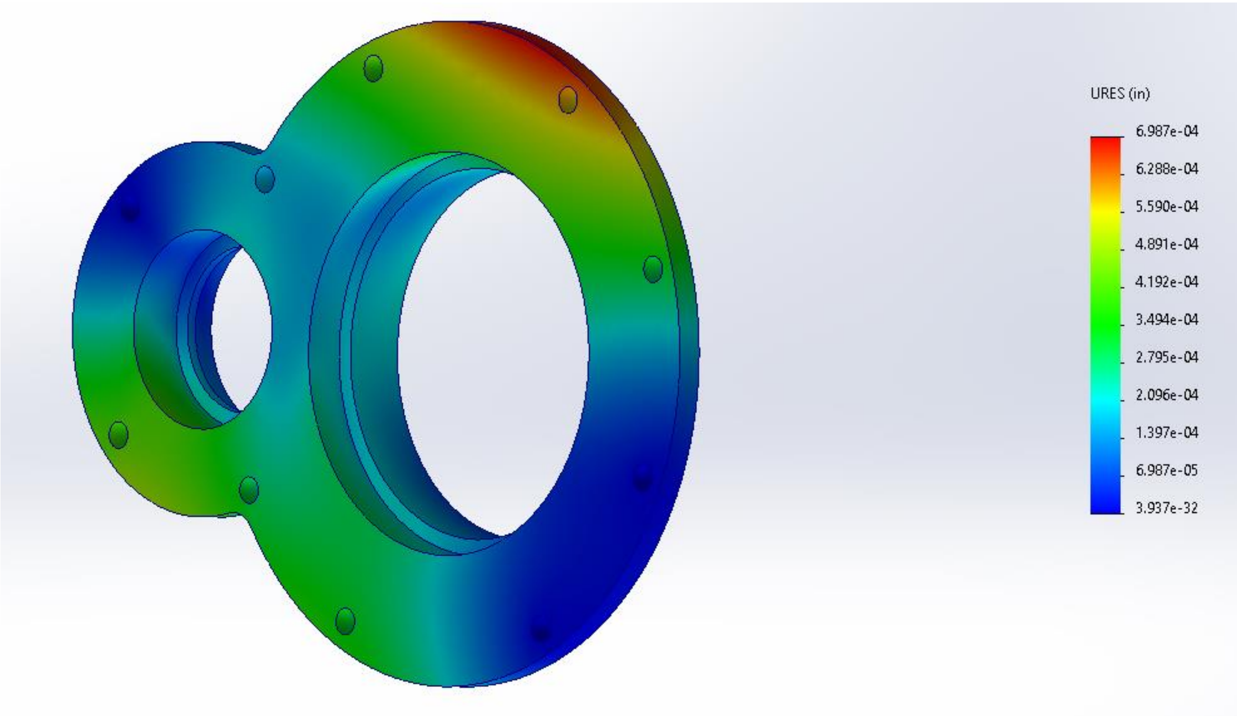
The maximum displacement of 6.631E-04 inches occurred at the bearing retention locations, which is very low and acceptable for the stresses that will occur.



Factor of Safety for the Drivers Side Casing

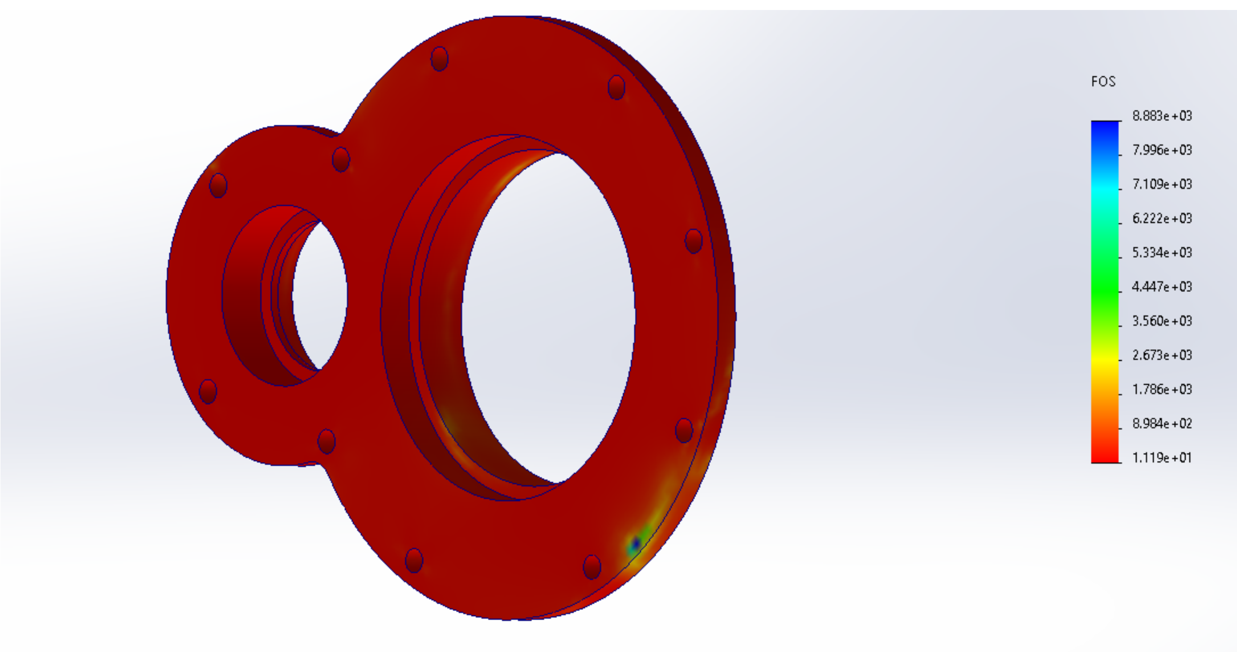
The lowest factor of safety of 7 occurred around most of the casing, which is expected and acceptable as the casing was designed to withstand these forces.

Passengers Side Casing



Displacement (Deformation) of the Passengers Side Casing

The maximum displacement of 6.987E-04 inches occurred on the flange of the casing, which is expected and acceptable given this is a very low displacement.



Factor of Safety for the Passengers Side Casing

The lowest factor of safety of 10 occurred around the casing, which is more than acceptable for this design.

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:

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

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

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

Where γ is the pitch angle. From these values, the chordal speed variation can be derived as $\Delta V/V = 1.7\%$. 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,

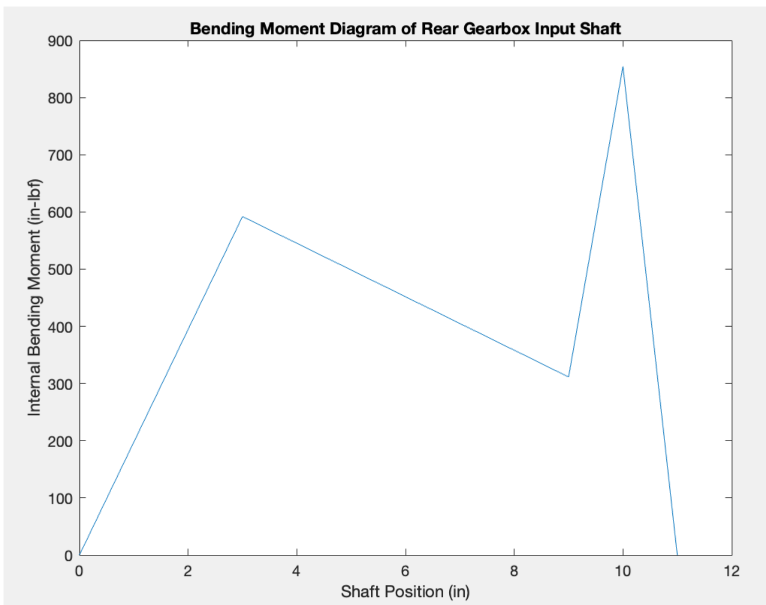
$$\sigma_{curved} = \frac{F}{A_{tooth}} = \frac{3000\ lbf}{0.26\ in\text{^}2} = 11538.46\ psi$$

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.

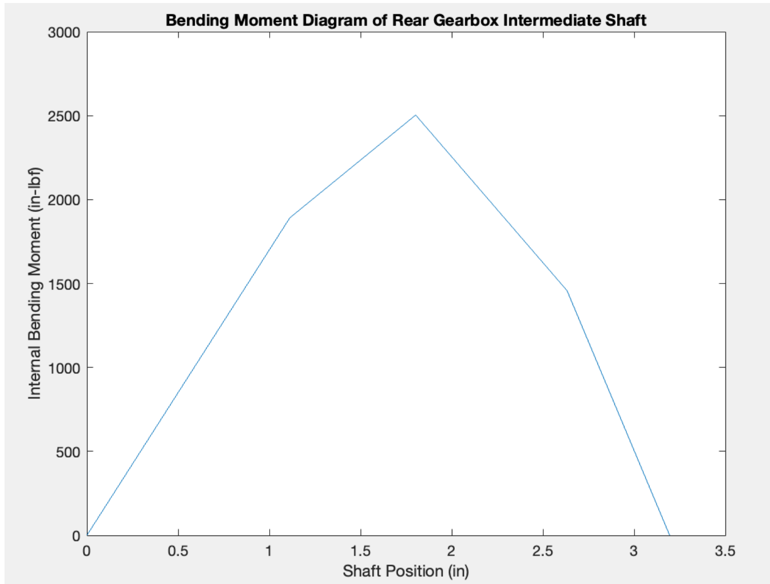
3..6 Seth Scheiwiller

Engineering calculations were conducted on four out of six shafts total in the drivetrain of the SAE Baja vehicle. These shafts include the pinion and intermediate shaft of the rear reduction box, intermediate chain drive shaft, and the pinion shaft of the front gearbox. Ensuring that each shaft critical location is greater than one will guarantee infinite life and prevent fatigue and static failure.

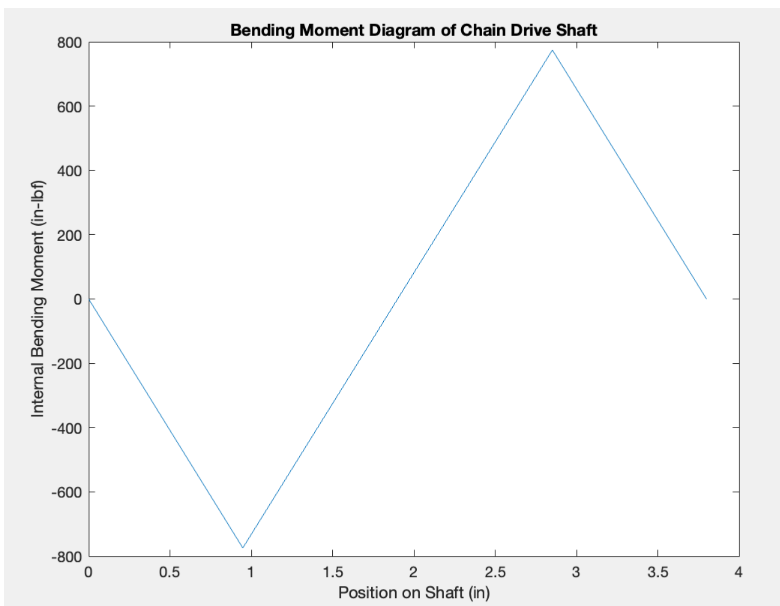
The first step is to calculate the forces acting on each shaft. These numbers were provided by those working on the gears and the bearings. Once those numbers were obtained, the next step was to calculate the bending moment acting at each critical location on each shaft. Because the stress concentrations are highest at points with shoulder fillets on the shafts, it is important to calculate the bending moments and torques acting at each of those points. After those numbers have been found, the next and final step is to calculate the fatigue and yielding FOS at each of those shoulder fillets.



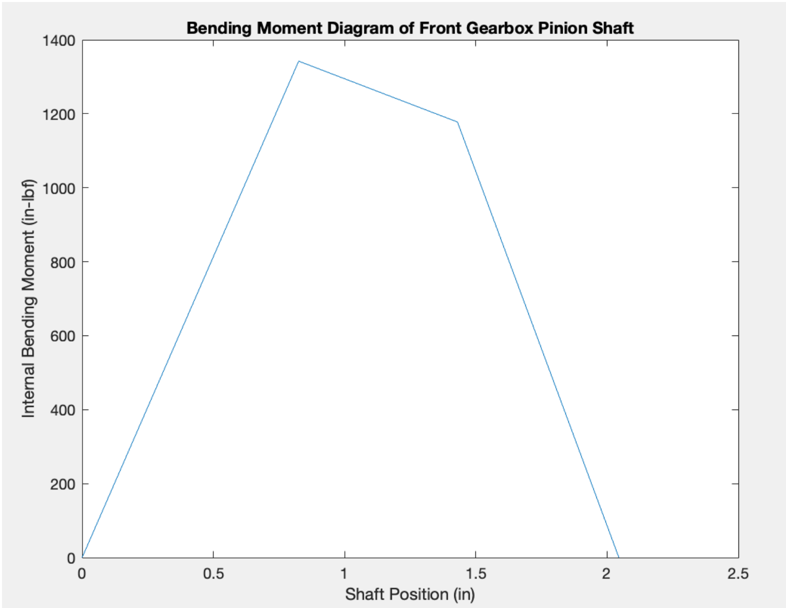
Bending moment diagram of rear gearbox input shaft.



Bending moment diagram of rear gearbox intermediate shaft.



Bending moment diagram of chain drive shaft.



Bending moment of front gearbox pinion shaft.

Because there is a long list of equations used to solve the factors of safety, only the main equations will be displayed in this report. The equations used can be found throughout chapters 6 and 7 and *Shigley's Mechanical Engineering Design* [1].

$$\sigma'_a = (\sigma_a^2 + 3\tau_a^2)^{1/2} = \left[\left(\frac{32K_fM_a}{\pi d^3}\right)^2 + 3\left(\frac{16K_{fs}T_a}{\pi d^3}\right)^2\right]^{1/2}$$

$$\sigma'_m = (\sigma_m^2 + 3\tau_m^2)^{1/2} = \left[\left(\frac{32K_fM_m}{\pi d^3}\right)^2 + 3\left(\frac{16K_{fs}T_m}{\pi d^3}\right)^2\right]^{1/2}$$

Fluctuating von Mises stress equations for alternating and mean loads.

$$n_f = \left(\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{ut}}\right)^{-1}$$

Goodman's theory for fatigue factor of safety.

$$n_y = \frac{S_y}{\sigma'_{\max}}$$

Yielding factor of safety.

Although only the Goodman criterion is displayed in this report, the fatigue factors of safety were calculated for each theory discussed in Shigley's [1]. This includes Morrow, Gerber, Soderberg, ASME-Elliptic, Smith-Watson-Topper, and Walker.

A MATLAB script was generated for each shaft that did the majority of the calculation to allow for ease of iteration and trial and error to determine the most ideal shaft diameters and component placement. These scripts were utilized to get each shaft critical location as minimal as possible without failure to decrease weight. Because each script is approximately 100-400 lines long, the scripts will not be discussed in length for the sake of brevity. Using these MATLAB scripts, the minimum factor of safety for each shaft was calculated and is compiled into the table below.

Resulting FOS For Each Shaft

Shaft	**Yielding FOS**	**Fatigue FOS**	**Resulting Theory**
Rear Gearbox Input Shaft	2.03	1.16	N/A
Rear Gearbox Intermediate Shaft	1.45	1.03	Smith-Watson-Topper
Chain Drive Shaft	1.04	1.03	N/A
Front Gearbox Pinion Shaft	1.44	1.07	Soderberg

The reason for "N/A" in the resulting theory column is because the minimum factor of safety takes place at a region where there is no torque, so all the theories converge to the same number.

4 Design Concepts

– *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.

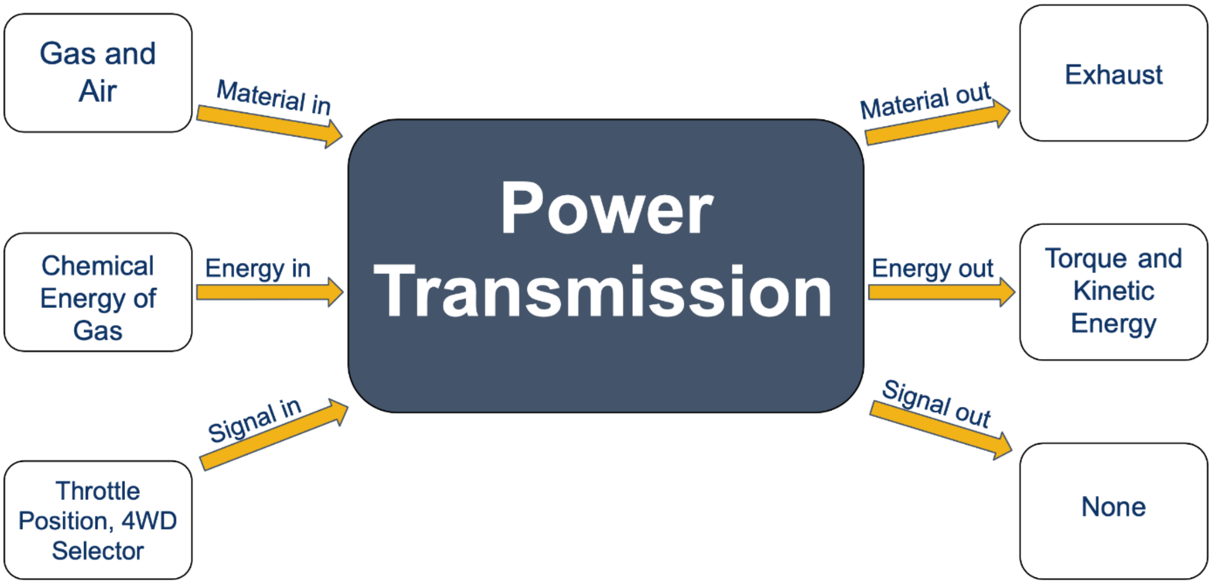


Figure 12: Drivetrain Black Box Model

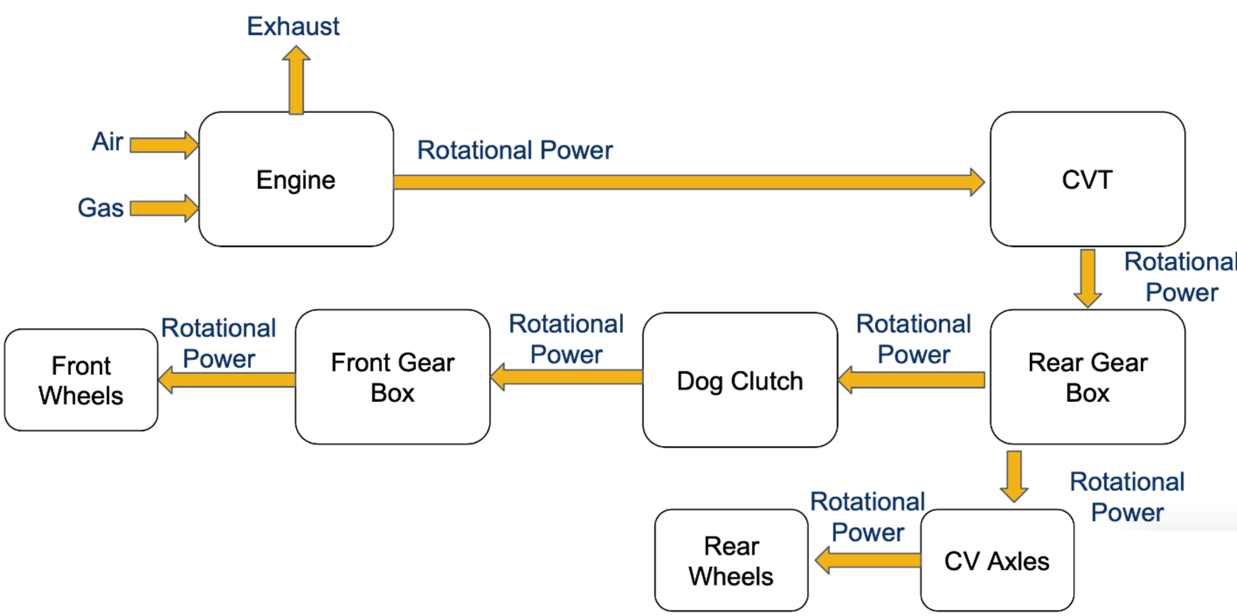
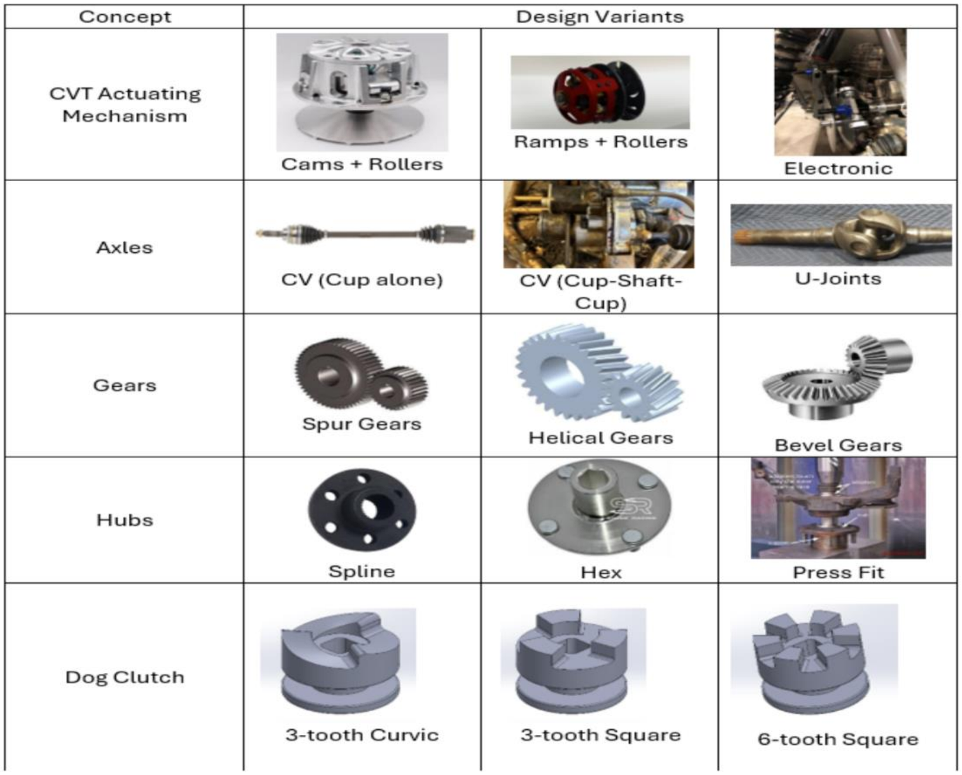


Figure 13: Drivetrain Functional Flow Diagram

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.

– *Concept Generation*

Table 9: 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.

– *Selection Criteria*

4.3.1 CVT Actuating Mechanism

The first decision that had to be made when choosing 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 ramp and roller assembly was the best route to take when deciding on an actuating mechanism.

4.3.2 Axles

Table 10: 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 11: 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 12: 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 13: 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.

– *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 14: 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 Approved Replacement for Section 5: SAE Deliverable

– *Executive Summary for our Proposal*

Executive Summary:
Lumberjack Defense Systems is pleased to submit our proposal to SAE Systems for the development and production of a new ground vehicle platform tailored for covert operations in remote and harsh environments worldwide. Our company has a proven track record in delivering cutting edge, mission critical defense technology. We understand the unique challenges that SAE Systems' clients face, and we are committed to providing a solution that not only meets but exceeds the operational, logistical, and technological demands outlined in the Request for Proposal.

Project Overview
The purpose of this project is to design and produce a highly reliable, self-sufficient, and efficient ground vehicle capable of operating in remote areas with extreme environments. The vehicle will be air-deliverable and must support sustained operations with multiple fuel sources, high fuel efficiency, and the ability to carry a single operator.

Additionally, the vehicle should have features that enhance its ability to facilitate air-ground delivery operations, making it suitable for use in diverse terrains such as jungles, deserts, and mountainous regions. The client's requirements emphasize long-term dependability without the need for resupply, with a special focus on self-sufficiency in isolated locations.

Our solution will focus on:

Self-sufficiency and Reliability:
The vehicle will incorporate advanced fuel systems with multiple fuel source capabilities, ensuring high efficiency and operational longevity even in the most isolated environments. We will design a robust system that guarantees the vehicle's reliability without the need for resupply or infrastructure.

Air Delivery Capability:
The vehicle will be optimized for rapid air transport, ensuring ease of deployment into the field. Its design will accommodate the specific needs of air-delivery systems while maintaining operational readiness immediately upon arrival.

Operator-Focused Design:
A single-operator layout will be engineered to ensure maximum control, comfort, and safety, while

providing intuitive interfaces and accessibility even in extreme field conditions.

Additional Optional Features:

Communication Systems:
A robust communication suite will be included to ensure real-time coordination with air and ground forces.

Autonomous Features:
Optional autonomous driving and navigation systems that can improve mission efficiency, especially in hazardous or highly volatile environments.

Modular Design:
Optional modular components allow for mission-specific customization of the vehicle for different operational needs.

Why Choose Lumberjack Defense Systems?
At Lumberjack Defense Systems, we differentiate ourselves through our commitment to quality, innovation, and operational excellence. Several key factors set us apart from other companies:

Experience in Extreme Environments:
We have extensive experience in developing defense solutions that are specifically designed to perform in harsh and remote locations. Our vehicles have been tested and deployed in a variety of challenging environments, from deserts to dense jungles and high-altitude terrains, ensuring that our designs can withstand the elements.

Cutting-Edge Technology and Engineering:
We utilize the latest in fuel-efficient technologies, vehicle design, and autonomy systems to ensure that our solutions are both high-performing and sustainable. Our expertise in developing multi-fuel systems, autonomous capabilities, and air-deployable solutions ensures that we can meet and exceed the operational demands of SAE Systems' clients.

Proven Track Record in Defense Contracts:
With a history of delivering successful defense solutions, we have built a reputation for excellence in engineering, production, and testing. Our clients trust us for our commitment to quality, timely delivery, and thorough post-production support, ensuring that every vehicle operates at peak efficiency.

Customization and Scalability:
We understand that each mission may require unique specifications, and we offer the flexibility to adapt our solutions to meet evolving needs. Our vehicles are designed with modularity in mind, allowing for custom configurations that can be tailored for specific missions or environments. We also have the capacity to scale production in line with the client's needs as outlined in the RFP.

Comprehensive Support and Lifecycle Management:
Our relationship with clients extends well beyond the delivery of vehicles. We offer full lifecycle management, including training, maintenance, and ongoing support. Our team will work closely with SAE Systems to ensure that all vehicles perform optimally in the field and that all personnel are fully trained in how to operate and maintain the platforms.

Summary:
With years of experience working with defense and military agencies, Lumberjack Defense Systems has developed a deep understanding of the unique challenges faced by our clients. Our team is composed of engineers, designers, and field specialists who have honed their skills through real-world deployments

and technological innovation. We leverage cutting-edge technologies and maintain strict quality control to ensure that every vehicle we produce meets the highest standards of reliability and performance.

6 Design Validation and Initial Prototyping

– *Failure Modes and Effects Analysis (FMEA)*

Part # and Functions	Potential Failure Mode	Potential Effect(s) of Failure	Potential Causes and Mechanisms of Failure	RPN	Recommended Action
Rear CV Cup-Shaft-Cup	Impact deformation	Rear CV Cup failing/ineffectiveness, CV Axle shaft failing/ineffetcivness, Damage to rear gearbox	Car bottoming out, Faulty suspension setup	9	Design suspension effectively
Front CV Cup	Impact deformation	Front CV Cup failing/ineffectiveness, CV Axle shaft failing/ineffetcivness, Damage to front gearbox	Car bottoming out, Faulty suspension setup	15	Design suspension effectively
Axle Shaft	High-Cycle Fatigue	Failure of CV axle shaft	Weakly designed axle shaft; small diameter or weak material	20	Perform FEA and do engineering calcs to ensure shafts are strong enough
Hub Spline	Slipping	Failure of transmission of power to wheels through hubs	Faulty press fit	140	Study up on press fits, ensure parts are manufacture to correct tolerance
Front gearbox Output Shaft	Temperature induced deformation	Deformation leading to damage of contained bearings	To much friction between gears, faulty gearbox design	20	Ensure center to centers on gears are correct, incorporate bearings effectively
Front gearbox Input Shaft	Contact and Cyclic Fatigue and Temperature induced deformation	Shaft shearing, Bearings overheating	Too much contact stress at the fillets in the shaft, friction between the gears too high	36	Ensure shaft calculations incorporate real world stresses, heat treat shaft to increase strength
Input Gear	Contact Fatigue	Teeth Shearing at contact point, Gearbox becomes not functional, no 4WD	Applied torque is too much, causing shearing of the gears	60	Make sure center to centers are correct, use ANSYS, and heat treat the entire gear to provide thew maximum strength
Output Gear	Contact Fatigue	Teeth Shearing at contact point, Gearbox becomes not functional, no 4WD	Applied torque is too much, causing shearing of the gears	60	Make sure center to centers are correct, use ANSYS, and heat treat the entire gear to provide thew maximum strength
Sprag Bearings	Cyclic Fatigue	Torque is not effectively transferred from the output gear to the CV cups	Impact loading, stresses get too high	20	Use largest bearings possible to minimize failure
Ball Bearings	Cyclic Fatigue	more friction occurs, no longer operational	Impact loading, stresses get too high	20	Use largest bearings possible to minimize failure
Roller Chain	Cyclic Fatigue	Chain slips/disconnects, loss of 4WD functionality	Chain incorrectly matched to the sprocket	20	Ensure selected chain and sprockets are compatible
Chain Drive Sprockets	Contact Fatigue	Sprockets shear due to tangential loading, power cannot be transmitted to front gearbox	Tangential loading, stress too high, sprockets incorrectly selected	80	Make sure sprockets are properly aligned, sprocket can adequately handle tension forces from the chain along with a factor of safety
Chain Drive Intermediate Shaft	Contact and Cyclic Fatigue	Shaft shears at contact points with the sprockets, loss of 4WD	Shaft has incorrect geometry/material properties to properly handle stress	40	Design shaft with correct forces and stresses applied
Jaw Clutches	Contact Stress	Jaw teeth begin shearing due to contact with each other	Teeth exert too much force on each other	40	Design jaws to withstand high forces
Front Hub	Braking torque, impact deformation	Brake failure, wheel off center, wheel disconnection, failure to drive	Higher impact than calculated	60	Increase factor of safety part is designed for, perform ANSYS and engineering calculations to ensure part is strong enough
Rear Hub	Impact deformation	Wheel off center, wheel disconnection, failure to drive	Higher impact than calculated	40	Increase factor of safety part is designed for, perform ANSYS and engineering calculations to ensure part is strong enough
Input Gear	Contact Fatigue	Teeth Shearing at contact point, Gearbox becomes not functional, no 4WD	Applied torque is too much, causing shearing of the gears	60	Make sure center to centers are correct, use ANSYS, and heat treat the entire gear to provide thew maximum strength
Intermediate Gear Driven	Contact Fatigue	Teeth Shearing at contact point, Gearbox becomes not functional, no 4WD	Applied torque is too much, causing shearing of the gears	60	Make sure center to centers are correct, use ANSYS, and heat treat the entire gear to provide thew maximum strength
Intermediate Gear Driving	Contact Fatigue	Teeth Shearing at contact point, Gearbox becomes not functional, no 4WD	Applied torque is too much, causing shearing of the gears	60	Make sure center to centers are correct, use ANSYS, and heat treat the entire gear to provide thew maximum strength
Output Gear	Contact Fatigue	Teeth Shearing at contact point, Gearbox becomes not functional, no 4WD	Applied torque is too much, causing shearing of the gears	60	Make sure center to centers are correct, use ANSYS, and heat treat the entire gear to provide thew maximum strength

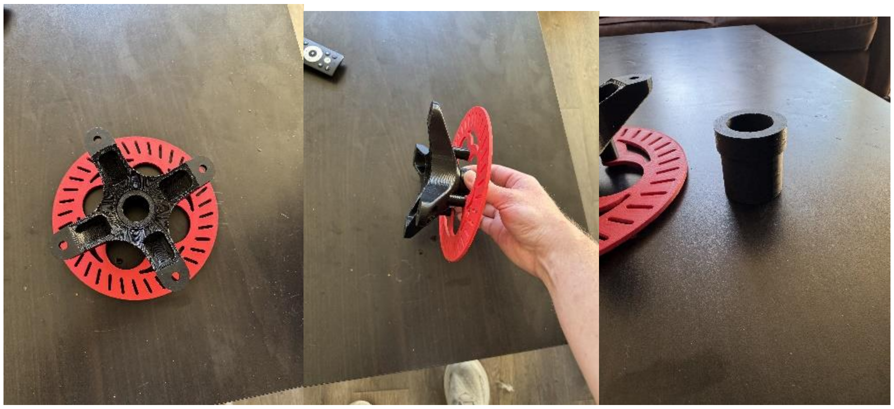
– *Initial Prototyping*

Prototype Pictures:

Rear Gearbox and CV Cup-Shaft-Cup Assembly

Front Gearbox

Hub



CV Cup Questions:

What question is being asked with the prototype?

- How the CV cup fits to the rear gearbox
- How the CV cup fits to the brakes
- How the CV cup meshes with the CV Axle

The methods/manufacturing used for prototype?

- 3D printing
- Press fitting

Answer to questions:

- The CV cup fits great to both the brake disc and rear gearbox, but the table-driven planes of the extrusions must be tweaked to better accommodate symmetry within the design
- The CV cup perfectly meshes with the CV axle

How did the answer inform the design?

- The answer informed that the table-driven features of the CV cup need to be adjusted to better accommodate design needs and symmetry of the final assembly.

Rear Gearbox Questions:

What question is being asked with the prototype?

- How do the gears fit within the casing?
- Are the center-to-centers of the gears, correct?
- What aspects of the design need to change to be able to manufacture the casings?

The methods/manufacturing used for prototype?

- 3D printing

- Press fitting
- Cardboard as spacers

Answer to questions:

- The gears do fit within the casing.
- The center-to-center of the gears look as correct as they can with the level of detail in the 3D print.
- Some of the filets both inside and outside need to change for the casing to be manufactured in house.

How did the answer inform the design?
After the prototype demonstration and completing HW 4, most of the bearings had to be changed to actually be able to support the radial loads that are experienced on the shafts. The actual gears and shafts were correct and most of the design could be manufactured as soon as possible and there would be no problems.

Front Gearbox Questions:

What question is being asked with the prototype?

- Do the gears mesh well and fit in the casing?
- Is the center-to-center correct for the gear and pinion to allow for successful power transfer?
- Do the two sides of the casing match and mesh for assembly?

The methods/manufacturing used for prototype?

- 3D printing
- Press fitting
- Cardboard used to act as spacers

Answer to questions:

- The gears mesh well and do fit within the casing.
- The gear’s center-to-center looks as correct as they can with the tolerances of the 3D print.
- Casing sides mesh and match well, hole sizes do need to change.

How did the answer inform the design?

After prototyping, bearings with higher load capacities were chosen to withstand the loads from the power transmission from the rear. For manufacturability, the filets were changed around the casing to allow for easier tooling. The casing dimensions changed as well to accommodate the new gears that were implanted after the prototype. At this point in the project, the casing is finalized and ready to be manufactured.

Front hub questions:

What question is being asked with the prototype?

- How much space on the axle does the hub take up?
- Does the hub give the wheel enough room for suspension?
- Is there room for the brake rotor?

The methods/manufacturing used for prototype?

- 3d printing
- Screwed the brake rotor in

Answers to questions:

- There is 0.24 inches left on the spline.
- The wheel does have enough room for suspension travel.
- The brake rotor fits.

– *Other Engineering Calculations*

CV Axle Sprag Bearing loads:
-Restricted Kohler CH440:
Max-torque = 18.5 ft-lb @ 2400 rpm
-Final drive gear ratio will be 3.9 * 6.98 = 27.22 with CVT and front gearbox
- 18.5 ft-lb torque * 27.22
=503.57 ft-lb
*.9 for mechanical loss = 453.23 ft-lb
Sprag rating must be higher than this value.

– *Future Testing Potential*

CV Axles:

Some future testing that will be done for the CV Axles will be the 3D printing of the two front CV cups, and the final front driveshaft that is press fit into the final driving gear in the front gearbox. Our selected Sprag Bearings will then be press fit into the drive shaft, and the front cv cups will then be press fit into the sprag bearings, creating a one way driving front CV Cup-Shaft-Cup assembly. This design closely resembles that of the rear CV Cup-Shaft-Cup, except it does not have a brake mount, the press fit is a custom hex gear instead of a square, and instead of being one solid piece, it incorporates two sprag bearings and some press fitting in order for it to be driving in one direction, while it is free moving in the other. This characteristic is needed in the front gear box as it will reduce stress on the chain drive and 4WD dog box.

Rear Bearings:

Once the bearings are here in-person they will be measured for dimensional accuracy and the holes and tolerances on the gearbox casing will be updated so that each bearing has a perfect press fit.

Front Gearbox Bearing Retention:

When the bearings arrive, fitment will be tested with a 3D printed prototype to see how the bearings fit in the casing and on their respective shafts. Press fitting will be thoroughly analyzed to ensure proper bearing placement and retention. The front gearbox is designed so that the bearings should have no place to go but in their specific retention places, so this should not be too worrisome.

7 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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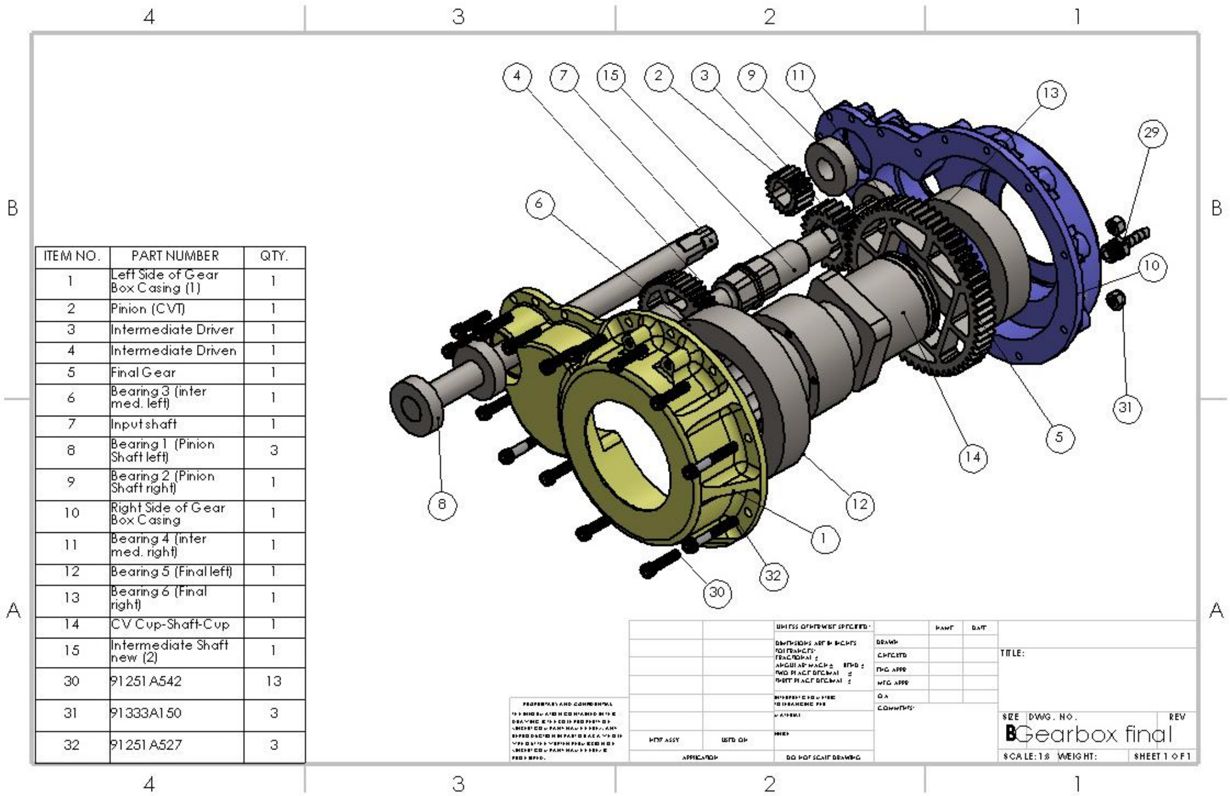
9 APPENDICES

– *Appendix A: Preliminary CAD from Concept Generation*

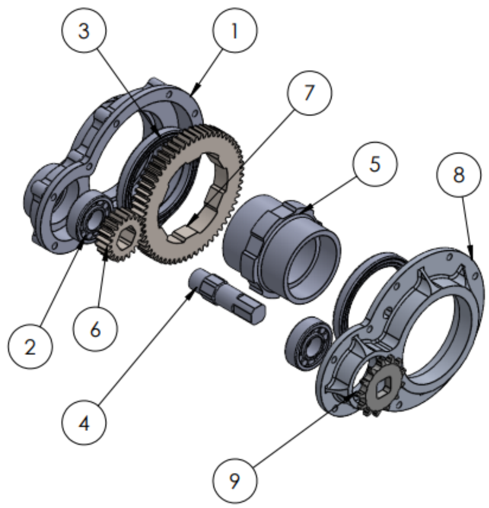


ITEM NO.	PART NUMBER	QTY.
1	Left Side of Gear Box Casing (1)	1
2	Pinion (CVT)	1
3	Intermediate Driver	1
4	Intermediate Driven	1
5	Final Gear	1
6	Bearing 3 (inter med. left)	1
7	Input shaft	1
8	Bearing 1 (Pinion Shaft left)	3
9	Bearing 2 (Pinion Shaft right)	1
10	Right Side of Gear Box Casing	1
11	Bearing 4 (inter med. right)	1
12	Bearing 5 (Final left)	1
13	Bearing 6 (Final right)	1
14	CV Cup-Shaft-Cup	1
15	Intermediate Shaft new (2)	1
30	91251A542	13
31	91333A150	3
32	91251A527	3

Assembly of Rear Gearbox

2 1

B



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	Bottom Casing (DS) v3		1
2	skf_bearing_nj_204_ecp_2		2
3	skf_bearing_61815-2rs1_2		2
4	Input Shaft v5		1
5	Front Reduction Driving Shaft R2		1
6	Input Gear v4		1
7	Output Gear v4		1
8	Top Casing (PS) v3		1
9	2299K19	Flat Sprocket	1

A

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC
TOLERANCING PER:
MATERIAL
FINISH
DO NOT SCALE DRAWING

	NAME	DATE
DRAWN		
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

TITLE:

SIZE A DWG. NO. FGBv4 Full Assembly REV v2

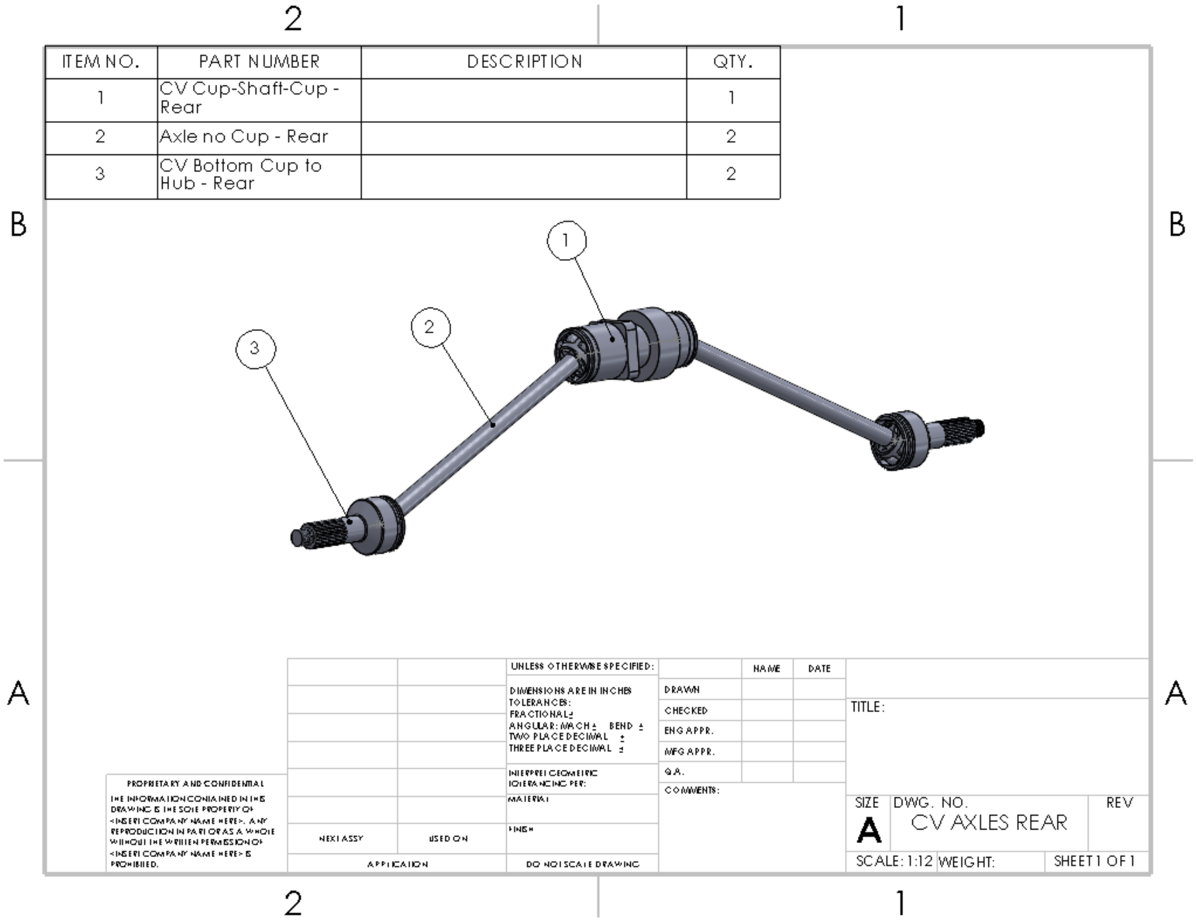
SCALE: 1:5 WEIGHT: SHEET 1 OF 1

NEXT ASSY USED ON
APPLICATION

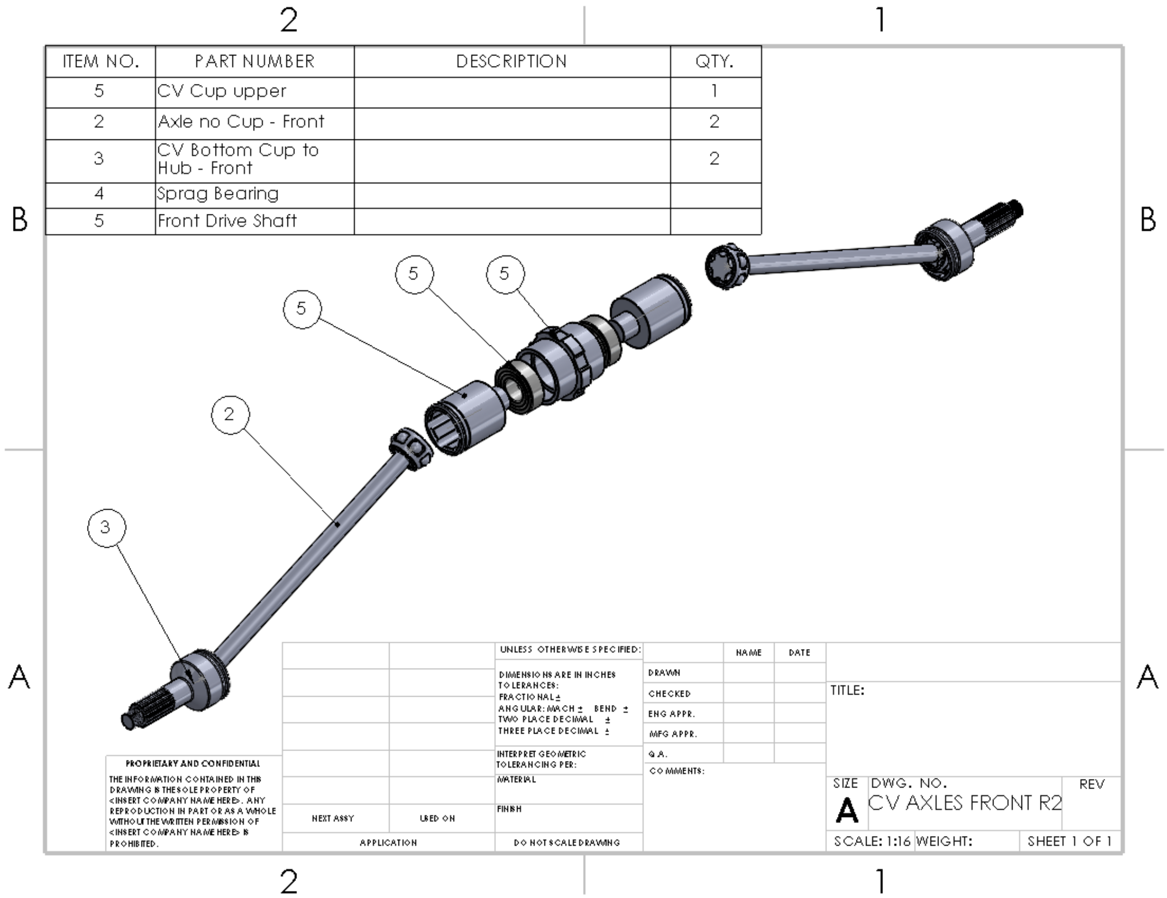
PROPRIETARY AND CONFIDENTIAL
THE INFORMATION CONTAINED IN THIS DRAWING IS THE SOLE PROPERTY OF <INSERT COMPANY NAME HERE>. ANY REPRODUCTION IN PART OR AS A WHOLE WITHOUT THE WRITTEN PERMISSION OF <INSERT COMPANY NAME HERE> IS PROHIBITED.

2 1

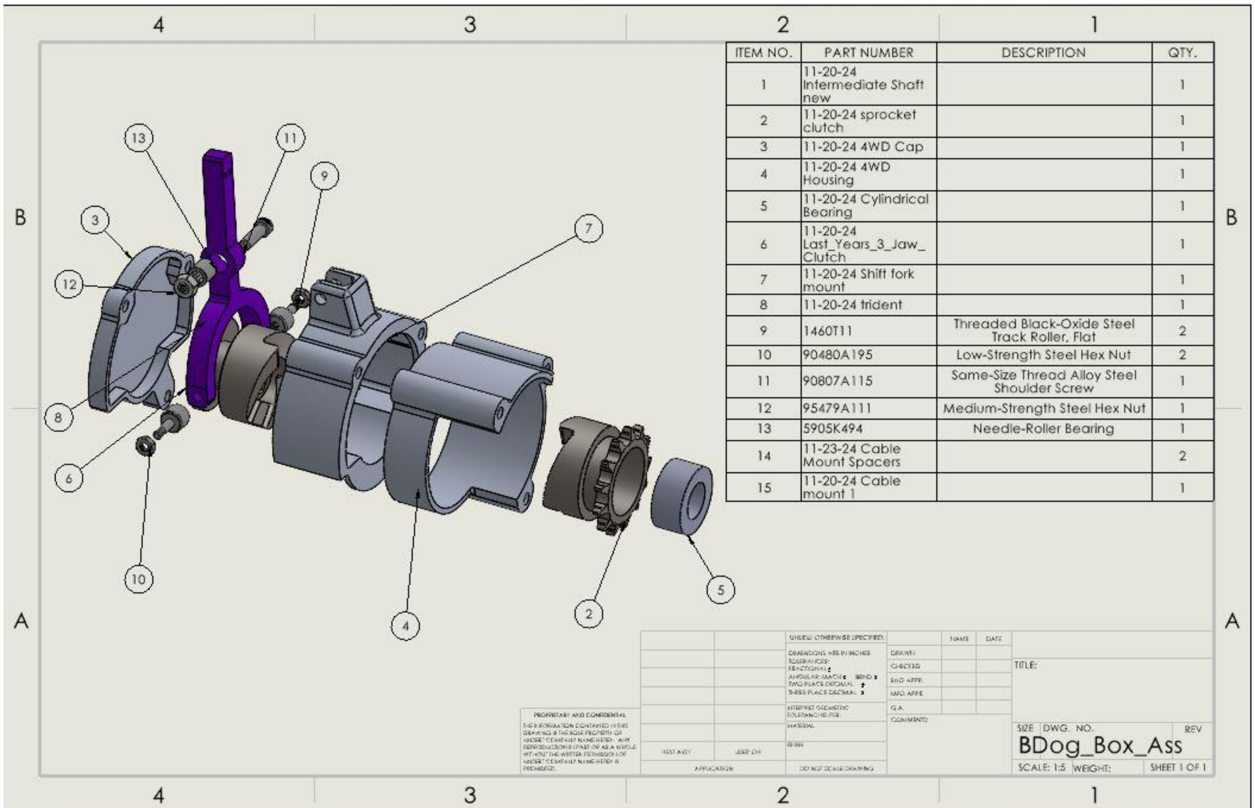
Assembly of Front Gearbox



Assembly of Rear CV Axles

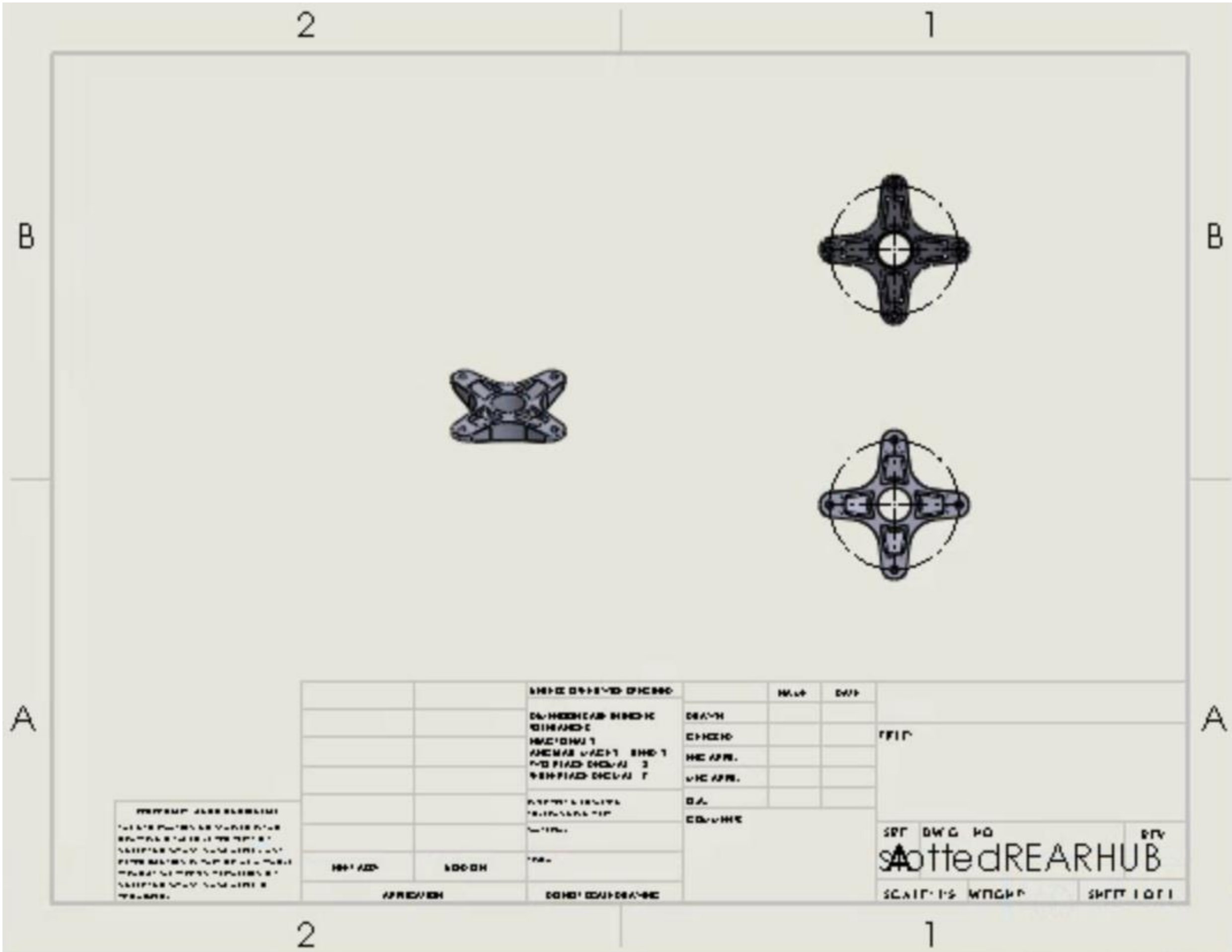


Exploded Assembly of Front CV Axles

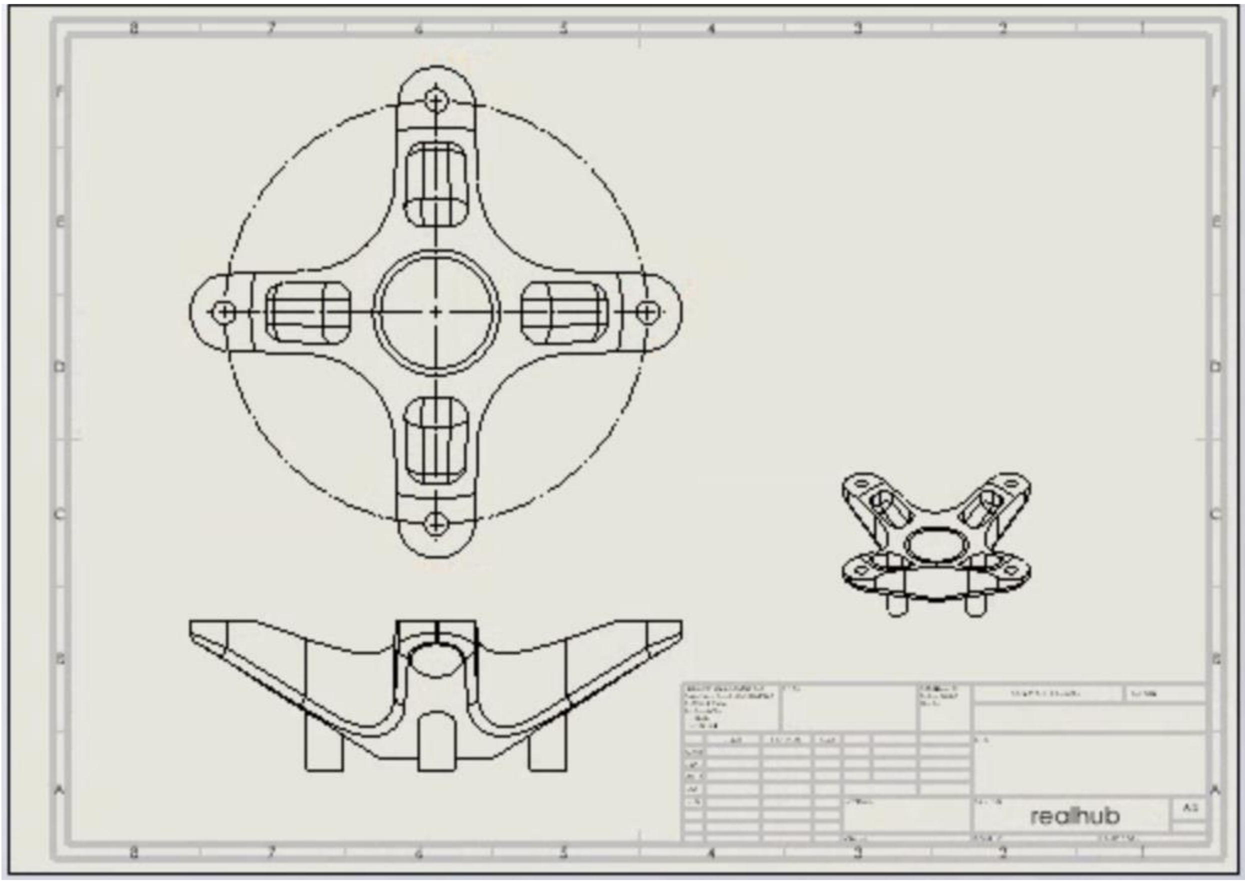


ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	11-20-24 Intermediate Shaft new		1
2	11-20-24 sprocket clutch		1
3	11-20-24 4WD Cap		1
4	11-20-24 4WD Housing		1
5	11-20-24 Cylindrical Bearing		1
6	11-20-24 Last_Years_3_Jaw_Clutch		1
7	11-20-24 Shift fork mount		1
8	11-20-24 trident		1
9	1460T11	Threaded Black-Oxide Steel Track Roller, Flat	2
10	90480A195	Low-Strength Steel Hex Nut	2
11	90807A115	Same-Size Thread Alloy Steel Shoulder Screw	1
12	95479A111	Medium-Strength Steel Hex Nut	1
13	5905K494	Needle-Roller Bearing	1
14	11-23-24 Cable Mount Spacers		2
15	11-20-24 Cable mount 1		1

Exploded view of dog box assembly.



Rear Hub



Front Hub

Design Report for SAE BAJA Fall 2024-Spring 2025
Steering, Brakes, and Suspension

David Polkabla Jr: Steering Lead

Taylor Hewitt: Brakes Lead

Brennan Pongratz: Co-Project Manager

Ryan Key: Suspension Lead

Ryan Latulippe: Suspension team member

Oliver Husmann: Suspension team member

Project Sponsors: W.L. Gore, H&S Field Services, Poba Medical, Harsh Co., Monster Energy, Nova Kinetics, and KC HiLiTES

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

The Suspension, Brakes, and Steering sub-team of the NAU Baja SAE team has made significant strides in designing and developing critical vehicle systems for the SAE Baja competition. The team's primary goal has been to create robust, high-performance components that improve the vehicle's overall performance, reliability, and handling in key events like Hill Climb, Endurance, and Suspension & Traction. The focus areas have included refining suspension geometry, ensuring brake system reliability, and developing a responsive steering mechanism.

The sub-team has completed CAD models for all major components and conducted Finite Element Analysis (FEA) to evaluate their strength and durability under competition conditions. Initial prototypes, such as PVC control arms and 3D-printed parts, were tested to verify design assumptions, yielding valuable insights for further refinement. Additionally, brake rotors and hubs were manufactured and successfully tested for load-bearing capacity, demonstrating their potential under high-stress conditions. The steering system has also been prototyped, and early tests indicate it provides the necessary responsiveness for precise control in tight maneuvers.

While these accomplishments reflect substantial progress, challenges remain, particularly in further optimizing the suspension geometry and steering system to enhance vehicle handling. The team is focused on finalizing designs to meet manufacturability requirements and comply with safety standards. Ongoing testing will continue to validate the performance and reliability of these components under real-world conditions.

In summary, the Suspension, Brakes, and Steering sub-team has established a strong foundation for building a competitive vehicle. Their progress in refining designs and conducting rigorous testing positions them well for continued development as they move closer to delivering a high-performing vehicle for the upcoming SAE Baja competition.

Contents

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. It will also cover design concepts, iterations, and the key criteria that influence design changes.

1.1 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.

1.2 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.

1.3 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 7 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

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.

2.1 Customer Requirements (CRs)

2.1.1 Brakes

To satisfy the customer's brake requirements, we must focus on several key aspects throughout the development process. Safety is a top priority, so we need to ensure all materials and components comply with safety standards and undergo rigorous testing to eliminate potential hazards. Affordability is also critical, which means selecting cost-effective materials and manufacturing methods while still preserving the necessary functionality through value engineering. To avoid hydraulic issues, it's essential to inspect the brake lines thoroughly and use proper fittings to prevent leaks. Additionally, the brake system should be designed to prevent overheating during heavy use. Addressing these factors is crucial for ensuring 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.

2.2 Engineering Requirements (ERs)

2.2.1 Brakes

The engineering requirements for the braking system focus on several critical objectives to enhance performance and safety. First, the braking force must be optimized to ensure the vehicle can stop effectively in all conditions. For example, we calculated that a braking force of 335 pounds is required to stop the vehicle traveling at 40 miles per hour within 4 seconds. In line with SAE guidelines, the brake pedal must be able to withstand a maximum force of 450 pounds. To meet this, the pedal will be constructed from aluminum, which is both durable and lightweight. Safety is a top priority, so the design will include features to protect both the driver and passengers. Additionally, reducing the force needed to operate the brake pedal is essential, with a target maximum pedal force of 55.8 pounds to ensure ease of use. By addressing these factors, the braking system will deliver strong performance while maintaining a high standard of safety.

2.2.2 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.

Any turning radius smaller than 7 feet would fulfill the goal, however due to the maximum rotation of the CV axle being 45^{o}. The minimal steering radius is 7.5 feet

2.2.3 Suspension

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, which the team categorized under 700lbs with the driver. 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 (under 64 inch requirement, 62 inch goal)—which provides a maximum value in the rulebook, vehicle length (48-60 inches), approach and departure angle, and an efficiently designed knuckle, categorized from being lightweight. 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.

2.3 House of Quality (HoQ)

2.3.1 Brakes

The house of quality for the brake system (Figure _) 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.

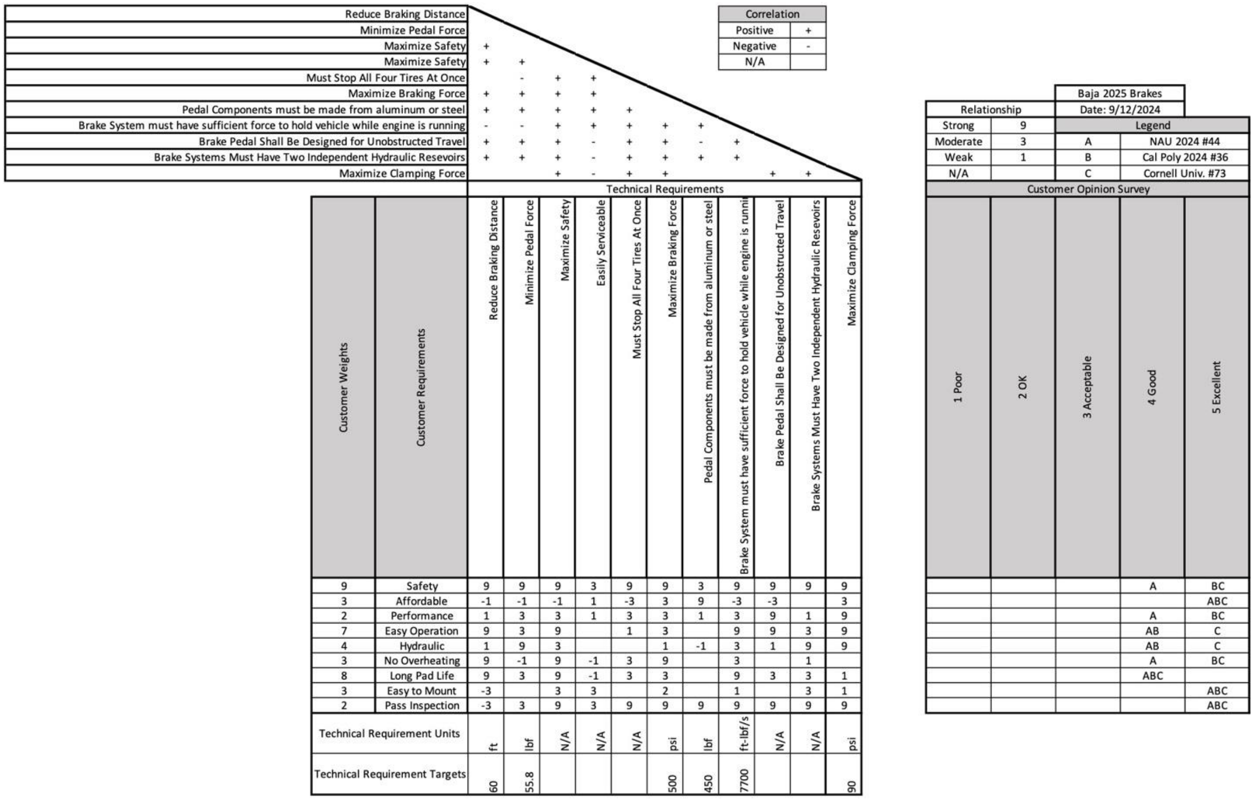


Figure 1: SAE 2024-2025 Brake QFD

2.3.2 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 that 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 are the 2024 NAU #44, Cal Poly #36, and Cornell University #73.

2.3.3 Suspension

Figure 2 presents 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.

Technical Requirements	Light weight	B.8.7	Optimal ride height/ground clearance	B.1.6	Vehicle length/approach angle	Singular known replaceable failure point (bolt)	Efficiently designed knuckle	Optimal camber angles	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 clearance										
B.1.6 - Limitations - Vehicle width (max 64 in)	-		+							
Vehicle length/approach angle			+							
Singular known replaceable failure point (bolt)	+									
Efficiently designed knuckle	+					+				
Optimal camber angles (-1 to -3 degrees)			+							
Optimal caster angle (5 to 10 degrees)			+							
Optimize maximum suspension travel (12 to 16 in)			+		+			+	+	

Baja 2025 Suspension	
Correlation	
Positive	+
Negative	-
N/A	

Relationship		Date: 11/3/2024	
Strong	9	Legend	
Moderate	3	A	2024 Cornell #73
Weak	1	B	2024 SDSU #43
N/A		C	2024 NAU #44

Customer Weights	Customer Requirements	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.	Degrees	Degrees	In.					
Technical Requirement Targets		<50	<6.35	12-16	64	48-60			-1 - -3	5-10	12-16					

Figure 2: Suspension QFD

3 Research Within Your Design Space

3.1 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 2: From Left To Right 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. Cornell excelled in the suspension and endurance categories according to 2024 results. SDSU also did well in the suspension category. NAU’s 2024 car did the best in maneuverability out of all of the categories. Each car is displayed from left to right: 2024 Cornell, 2024 SDSU, 2024 NAU.

Figure 3: Cornell University, SDSU, and NAU 2024 SAE Baja Vehicles

3.2 *Literature Review (everyone add sources if needed, template calls for 10+ per student)*

3.2.1 Brakes

3.2.1.1 Taylor Hewitt

1. Shigley's Mechanical Engineering Design (Chapter 16: Brakes) [1]

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.

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

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.

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.

9. Tribological Behavior of Friction Materials of a Disk-Brake Pad Braking System Affected by Structural Changes—A Review [9]

This article explores the tribological behavior of modern brake disk-pad materials, focusing on friction, wear, and how structural changes affect braking system performance. It emphasizes the importance of understanding these factors for designing efficient, reliable braking systems, with attention to friction coefficients, wear resistance, heat transfer, and noise. Test results offer valuable insights for optimizing braking system design through computer modeling and analysis.

10. Braking performance of friction materials: A review of manufacturing process impact and future trends [10]

This paper explores how manufacturing parameters affect brake friction materials' properties and performance, highlighting the role of material heterogeneity. It outlines common manufacturing processes, key properties for modern brake systems, and innovative methods like optimized algorithms and 3D printing to enhance brake material performance.

3.2.2 Steering

3.2.2.1 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 [11]

   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 [12]

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 [13]

   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 [14]

   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 [15]

   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 [16]

   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.

9. Kinematics Design Methodology | Suspension Design Series Ep.1 [17]

   This video is in a series of videos that informs the viewer on how to use the Optimum Kinematics software.

10. DESIGN WITH ME! Actuation Pick-up Points | Suspension Design Series Ep.4 [18]

    This is the fourth video in the series mentioned above, this taught me how to apply the earlier knowledge

3.2.3 Suspension

3.2.3.1 Ryan Key

1. Tune To Win [19]

   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 [20]

   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 [21]

   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 [22]

   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 [23]

   "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 [24]

   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 [25]

   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.

3.2.3.2 Ryan Latulippe

1. Dixon Suspension Geometry and Computation [11]

   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 [26]

   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 [27]

   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 [22]

   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 [28]

   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 [15]

   The article presents information regarding camber and caster angles, which are some parameters which are essential to understand when understanding handling and drivability, especially alignment, as that is one of the biggest issues that messes with drivability.

7. Bump Steer [29]

"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 Video [30]

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.

9. Optimum Kinematics Optimization Module Video [31]

This video is centered around introducing and teaching the optimization module for optimum kinematics. This module is an add-on with the original main program which helps the user visualize their systems and their simulation results on a deeper level.

10. Optimum Kinematics Design Video [32]

This video highlights various techniques for the user in designing a proper suspension system. There are multiple videos within this series that talk about various design tips which will translate in a suspension system with optimal travel with the equipment being used.

3.2.3.3 Oliver Husmann

1. Performance Vehicle Dynamics: Engineering and Applications (Chapters 7 and 8) [33]

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)* [34]

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* [35]

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* [36]

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* [37]

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* [38]

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* [39]

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* [40]

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.

9. Shigley's Mechanical Engineering Design (Chapter 15) [1]

This chapter focuses on the design and analysis of bolted and welded joints, which are critical for ensuring the structural integrity of assemblies. It includes guidelines for calculating preload, bolt stresses, and the effects of combined loading. For SAE Baja, this chapter provides foundational knowledge for designing bolted joints in the suspension system, particularly for components like the knuckle and ball joint connections. The insights from this chapter ensure that bolted joints are optimized for strength while accommodating necessary articulation.

10. Standard Specification for Steel Bolts (ASTM A307) [41]

The ASTM A307 standard defines the strength and quality of steel bolts, ensuring they meet safety requirements. It guides our bolt selection for the SAE Baja suspension, ensuring reliability under off-road loads and impacts.

3.2.3.4 Brennan Pongratz

1. *Shigley's Mechanical Engineering Design [1]*

Chapter 17 discusses flexible mechanical elements such as a v-belt that will be the driving the CVT. It also has information on bearing selection based on its application.

2. *Machinery's Handbook [42]*

This source has an abundance of information on part fabrication but specifically press fit standards and thread standards. The press fit standards are essential to proper bearing performance.

3. *Design and Manufacturing of a Continuously Variable Transmission (CVT) [43]*

This source was used for the ratio between the primary and secondary sides of the CVT and some general calculations for v-belt selection.

4. *Collegiate Design Series Baja SAE Rules [45]*

This paper is an essential reference for the entire team, but we are specifically interested in guards for hazardous releases of energy. Also making sure that pinch points are not exposed to the driver.

5. *Olav Aaen's Clutch Tuning Handbook [45]*

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.

6. *Modeling of a Continuously Variable Transmission [46]*

This source helped us generate a Matlab model of the CVT with parameters that we can implement into our CAD designs.

7. *Fatigue Design Curves and Analysis for Aluminum [47]*

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 including the knuckle.

8. *Design of a Double Wishbone Baja SAE Suspension System [48]*

This source discusses initially designing suspension and ways to optimize. It also discusses CV-axle angles and shock mounting angles.

9. *What is Bump Steer, How it Works, and How to Measure It [49]*

This source discusses what bump steer is and how it can be unwanted. Also what design choices can be made to minimize or maximize it.

10. *What is Frictionless Support? [50]*

This source discusses what a frictionless support is in Ansys. It also talks about when to use it and what it will simulate based on the type of face selected.

3.3 Mathematical Modeling

3.3.1 Brakes

3.3.1.1 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: $W = 29460\frac{lbft}{s^2}\ , and\ d = 88\ ft$

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: Equation 6 is used to determine the brake pedal force needed to stop the vehicle.

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

Equation 6: Brake pedal force [6]

Where BPR = 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.

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: $d_{mc} = 0.813\ in$

3.3.2 Steering

3.3.2.1 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 45°

$$Inner\ Turn\ Radius: R_{in} = \frac{L}{\tan(\theta_{in})} \Rightarrow \frac{60in}{\tan(45^o)} = 60in$$

$$Vehicle\ Turn\ Radius: R = R_{in} + \frac{Trackwidth}{2} \Rightarrow 60in + \frac{62in}{2} = 91in\ or\ 7.58ft$$

$$Outer\ Turn\ Radius: R_{out} = R + \frac{Trackwidth}{2} \Rightarrow 91in + \frac{62in}{2} = 122in\ or\ 10.17ft$$

$$Outer\ Steering\ Angle: \theta_{out} = \tan^{-1}\left(\frac{L}{R_{out}}\right) \Rightarrow \tan^{-1}\left(\frac{60in}{122}\right) = 26.19^{o}$$

Steering Wheel Rotation:

The following equation was used to find the relationship between steering wheel rotation and wheel rotation. This leads to approximately 5.75 inches of rack travel per 360º of rotation.

$$\tan(\theta_0) = \frac{\sin(\beta) * \tan(\theta_i)}{\cos(\beta)}$$

3.3.3 Suspension

3.3.3.1 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.

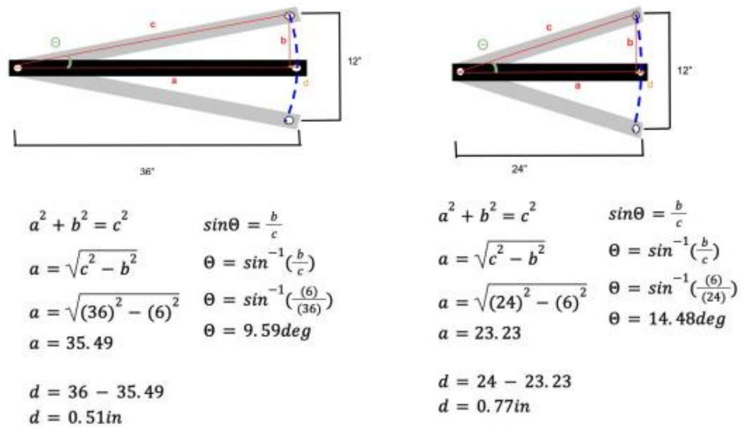


Figure 4: Arm Length Calculations

3.3.3.2.2 Trailing link FEA 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 to soften the vehicle and support it as efficiently as possible. While the shocks can 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 of 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.

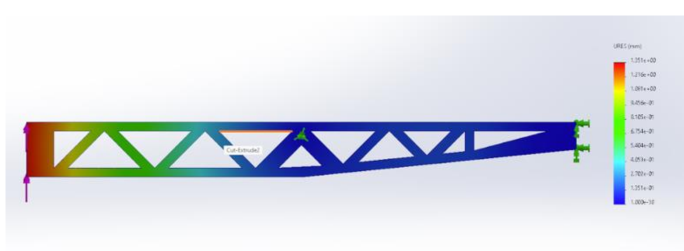


Figure 5: Trailing Arm FEA for Deflection

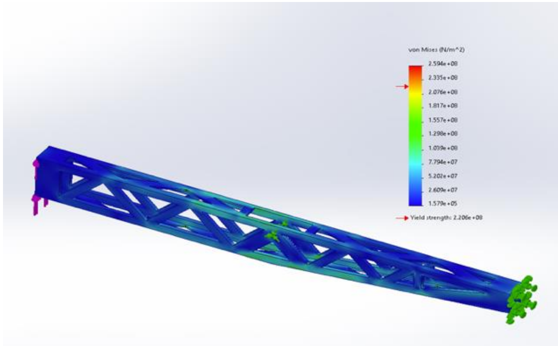


Figure 6: Trailing Arm FEA for Stresses

3.3.3.2.2 Trailing Link Iteration 2 and FEA Simulation

Following more design critiques and SolidWorks design, the team has decided to move away from the boxed and trussed trailing link design and continue forward with a tubed trailing link. This new design retains the strength required while simplifying the fabrication process and integrating weight savings. The new design is pictured below in figure 7.

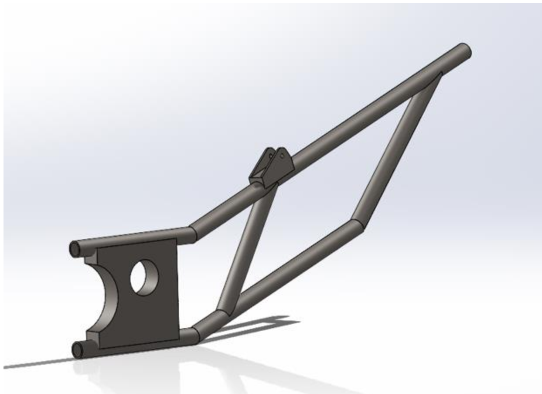


Figure 7: Trailing Link Iteration 2

The new tubular design retains the central mounting location, main pivot location, and camber link supports. The rear camber links will be the same as the previous iteration, as well as the plan for heims, bolts, and other hardware. There is also an added support directly under the shock mount, aiding in rigidity and strength. With the main link geometry to be constructed of bent and coped steel tubing, and the bearing housing to be milled, the fabrication process is streamlined and can be completed in house as opposed to sending parts away to be cut. The bends are specifically designed for strength and ease of cutting coping and welding. Furthermore, below is the FEA simulation results of the new design which showed minimal deflection and shows that there is still room for material removal in the bearing housing component.

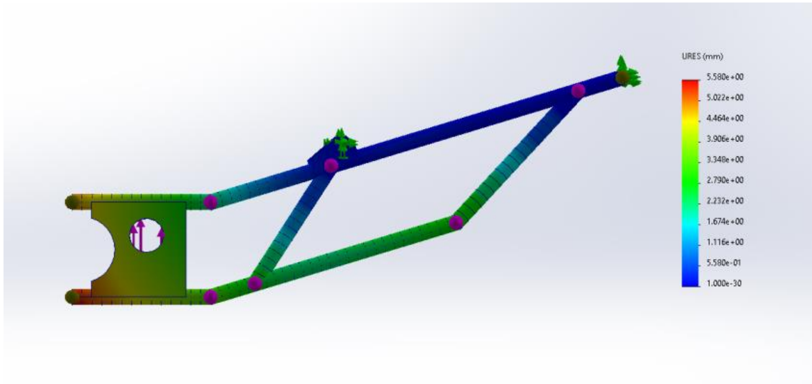


Figure 8: Trailing Link FEA For Deflection

With the CAD model proving the validity of the design, a one-to-one scale prototype was able to be constructed to the same dimensions and including the required hardware and mounting locations. This prototype was able to further validate the design and ensures that there will be no binding or clearance issues with the new design and the accompanying hardware. Below is a photograph of the completed prototype number one.

Figure 9: Trailing Link First Prototype

3.3.3.2 Ryan Latulippe

3.3.3.2.1 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.

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

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

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

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

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

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

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

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

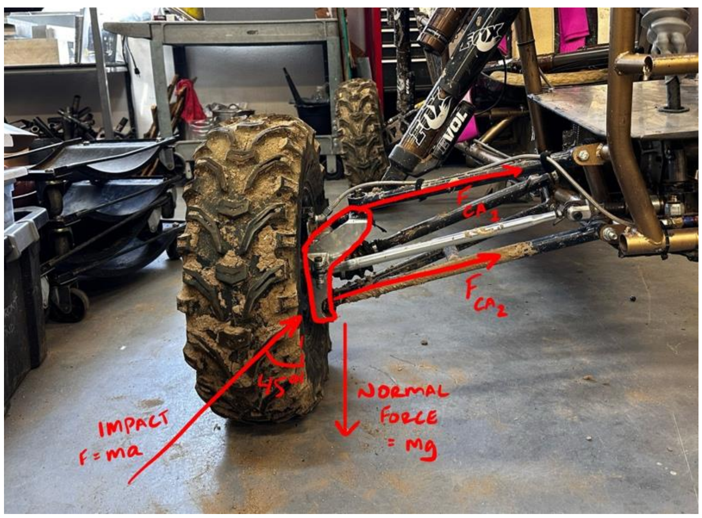


Figure 10: Math for Impact force

3.3.3.2.2 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)

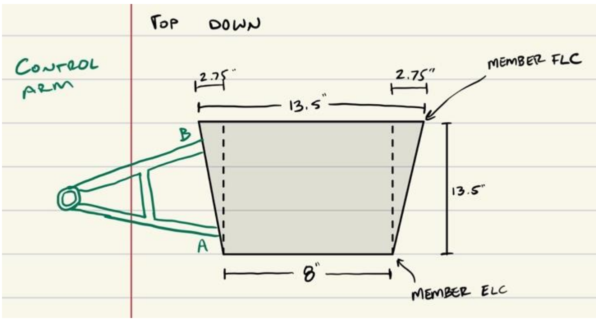


Figure 11: Math for Control Arm

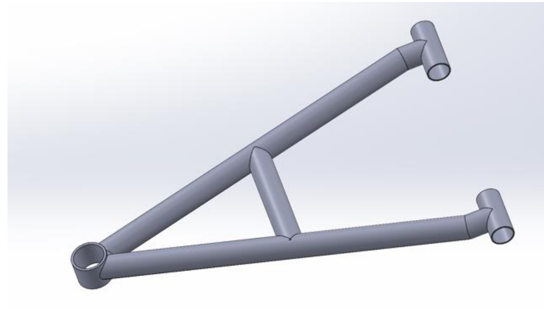


Figure 12: CAD for A-Arm

3.3.3.3 Oliver Husmann

3.3.3.3.1 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.

3.3.3.3.2 Engineering Simulations Using SolidWorks

The SolidWorks simulation was conducted to evaluate the structural performance of the front knuckle variation 1 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.

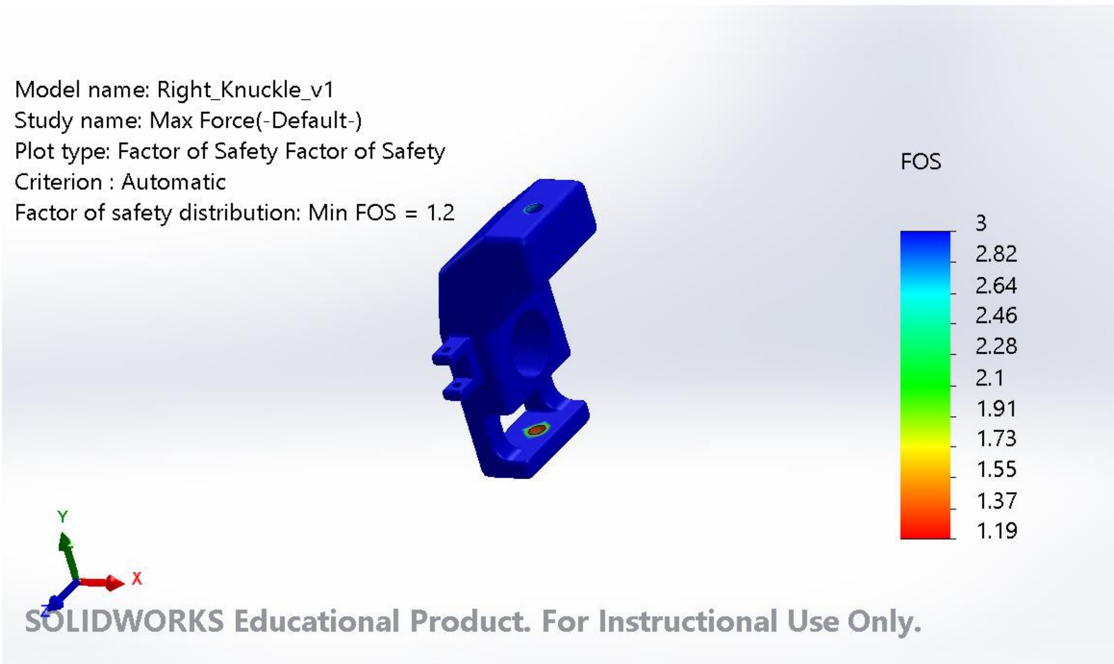


Figure 13: Factor of Safety Visualization for the Front Knuckle Design

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 (seen in section 3.3.3.4 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.

3.3.3.3.3 Factor of Safety for bolts

For the bolts that will be analyzed the minimum shear strength is 84,000 psi

Shear stress Calculations:

For a bolt with a shoulder diameter of 1/4’’ (0.125in)

$$Shear = \frac{2000\ lbf}{\pi(0.125)^2} = 40{,}743\ psi$$

For a bolt with a shoulder diameter of 3/8’’ (0.1875in)

$$Shear = \frac{2000\ lbf}{\pi(0.1875)^2} = 18{,}108\ psi$$

Factor of Safety (FoS):

For the 1/4" shoulder diameter

$$FoS = \frac{Minimum\ Shear\ Strength}{Shear\ Stress} = \frac{84{,}000}{40{,}743} = 2.06$$

For the 3/8’’ shoulder diameter

$$FoS = \frac{Minimum\ Shear\ Strength}{Shear\ Stress} = \frac{84{,}000}{18{,}108} = 4.63$$

The calculated Factors of Safety for both bolt sizes exceed the typical minimum design requirement of 1.5, confirming that the bolts are structurally sufficient for the SAE Baja application. We are opting to use the 1/4" bolt in conjunction with a misalignment spacer to provide increased angular movement of the ball joint due to reduced physical obstruction, which will improve suspension articulation. Although the smaller bolt results in a lower Factor of Safety (2.06) compared to the 3/8” bolt (4.63), it still meets the design requirements.

3.3.3.4 Brennan Pongratz

3.3.3.4.1 Engineering Simulations Using Ansys for Knuckle Second Iteration

The steering knuckle will be one of the most complex parts to machine so we need confidence in the design before that major step is taken. To help understand the strength and durability of the design, Ansys Finite Element Analysis will be used to simulate loading conditions. The areas of the most concern are the upper control arm, tie rod, and lower control arm mounts which attach the knuckle to the rest of the system that can be seen in Figure 11.

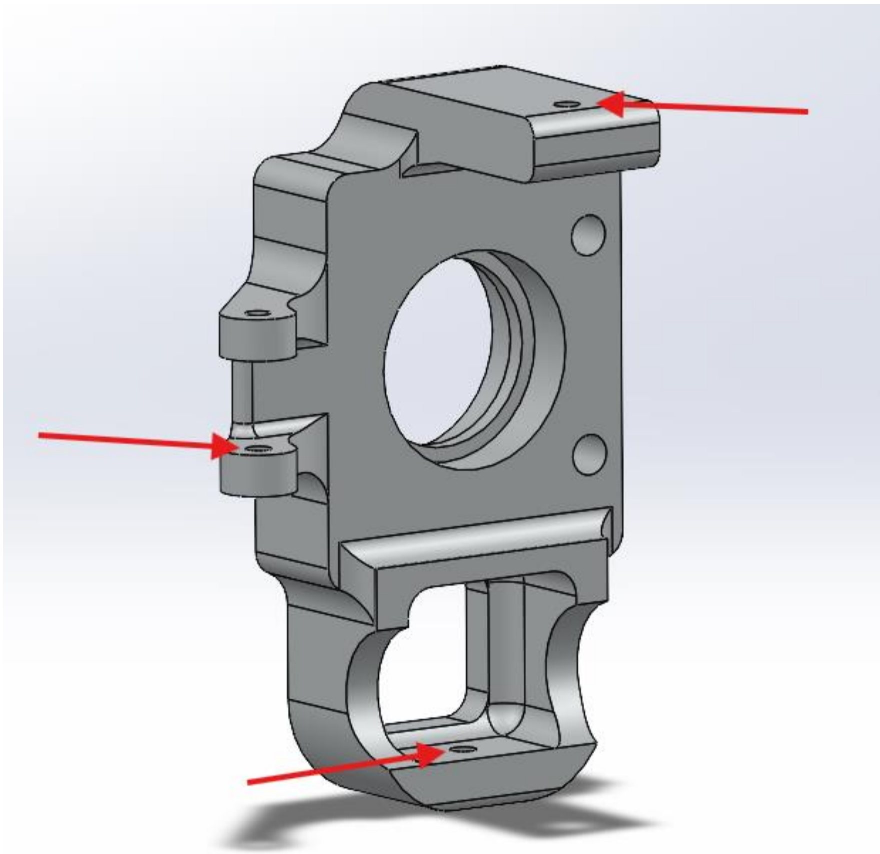


Figure 14: Image of Knuckle Showing Mounts for Suspension and Steering

Frictionless supports are applied to the upper and lower control arm mounts to allow rotation and axial movement but not any radial displacement and a fixed support is applied to the tie rod mount. This simulates a collision with the wheel fully turned. The simulation is shown in Figure 12.

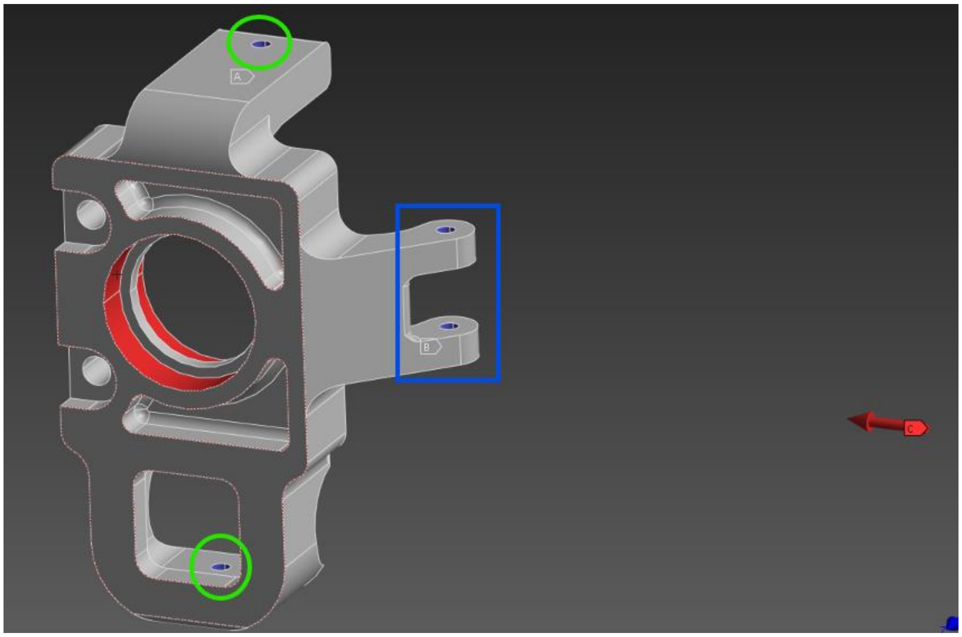


Figure 15: FoS of Knuckle

The red arrow shows the location and direction of the applied load that has a magnitude of 2200lbf. The results desired from this simulation are the stresses throughout the knuckle and the location of the lowest factor of safety. The location of the lowest FoS is shown in Figure 13.

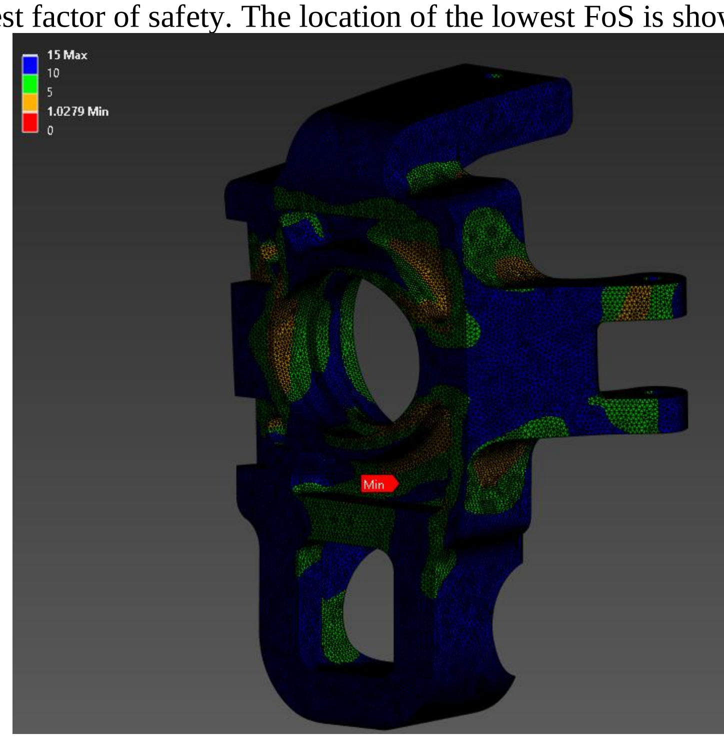


Figure 16: Max Deformation on Knuckle

This simulation results in a FoS greater than 1 which is acceptable for this application. This knuckle design shown is not the final iteration so more analysis will be needed but this informs that mounts with this general design will be strong enough.

4 Design Concepts

4.1 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 17: 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.

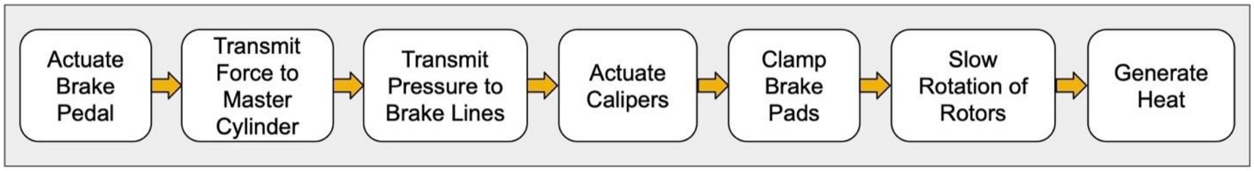


Figure 18: Functional Model for Baking System

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.

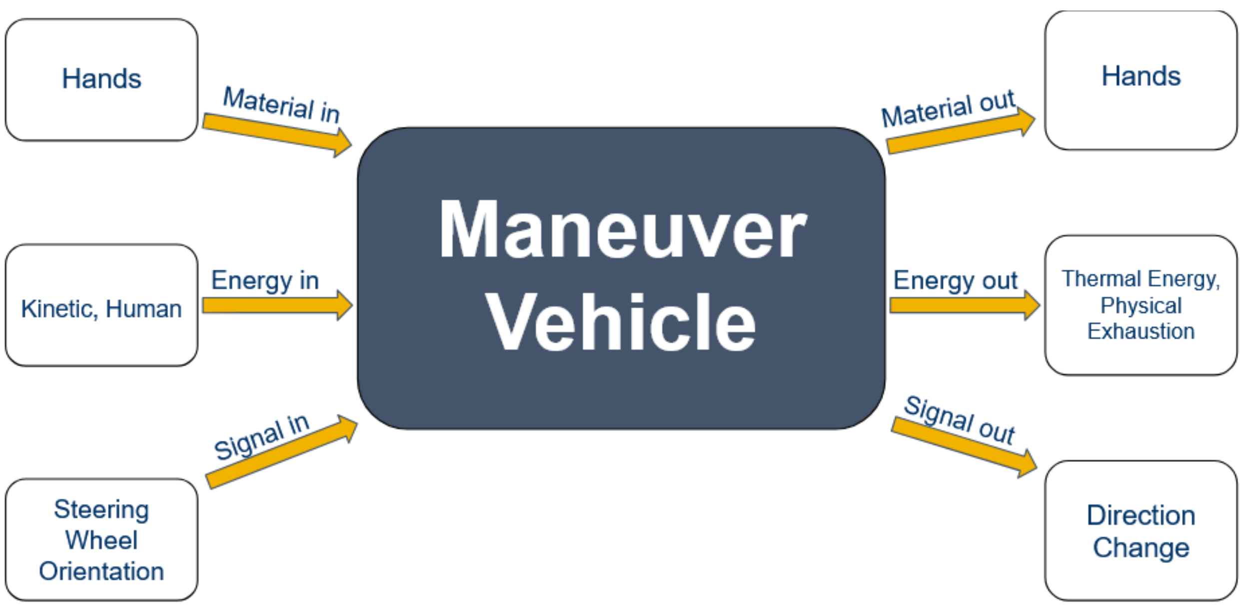


Figure 19: Black Box Model for the Steering System

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.

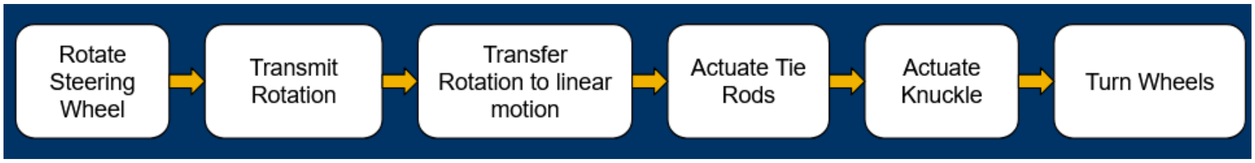


Figure 20: Steering Functional Model

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.

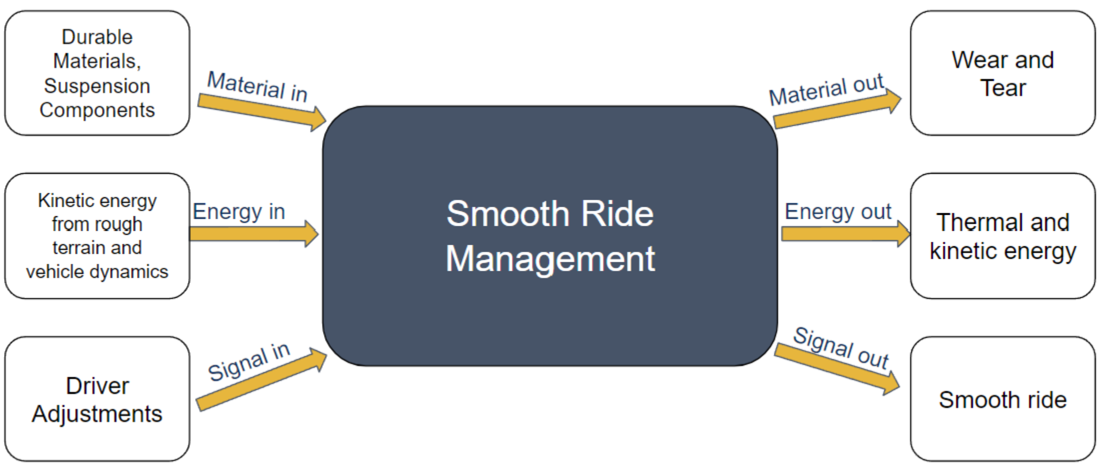


Figure 21: Black Box Model of the Suspension System

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.

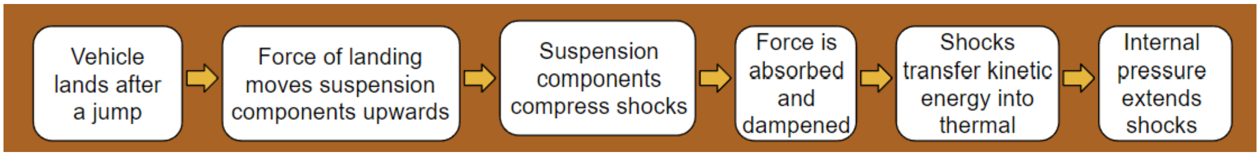


Figure 22: Functional Model of the Suspension System

4.2 Concept Generation

4.2.1 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.

4.2.2 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.

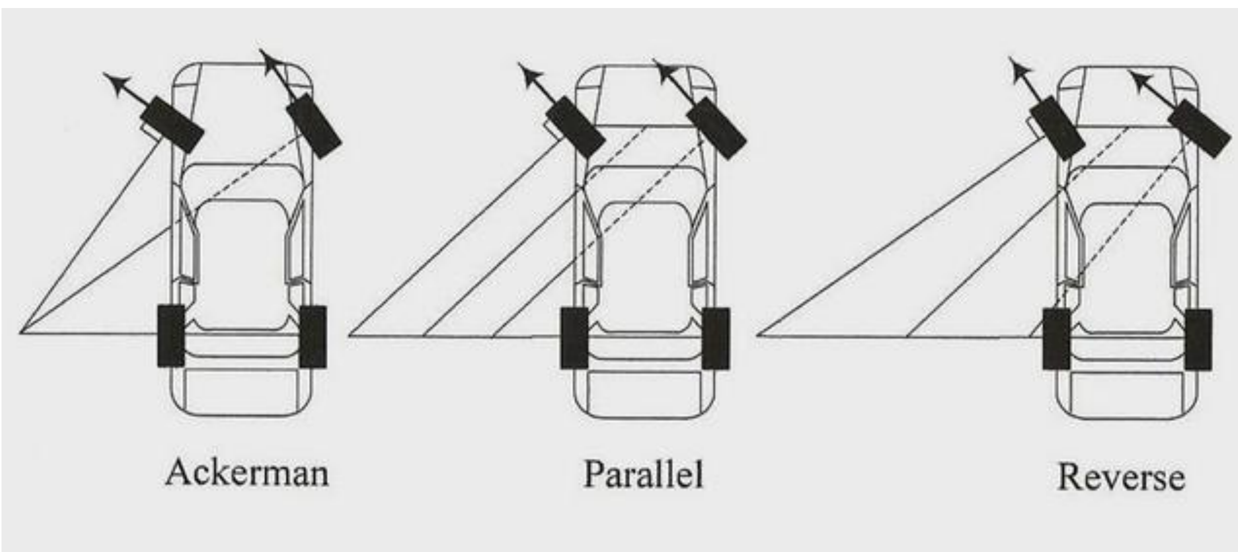


Figure 23: Steering Geometries

4.2.3 Suspension

Shock Mounting Location

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. Considering this analysis and information, the team chose to mount our shocks to the upper control arm on the front double wishbone system.

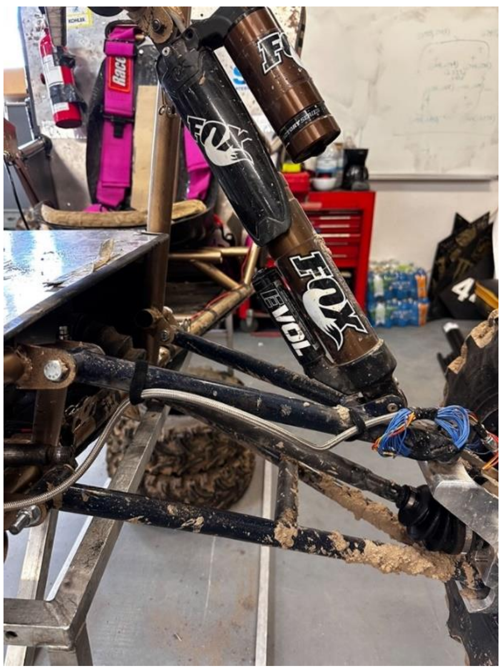


Figure 24: Upper Control Arm Mount

Figure 25: 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).

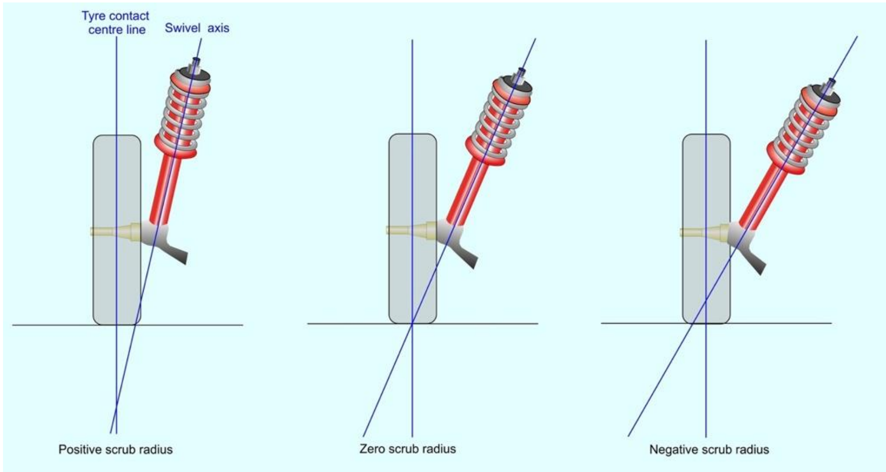


Figure 26: Different Types of Scrub Radius

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.

4.3 Selection Criteria

4.3.1 Brakes

Force Minimization:

The bore diameter of the master cylinder is determined by the braking force exerted on the vehicle. A larger braking force requires a large bore diameter to exert the pressure needed for the calipers to fully clamp on the brake rotors. From equation 14, the minimum bore diameter needed was found to be 0.813 inches. From this, we determine the appropriate bore diameter to be 7/8 inches.

The brake pedal force is determined by the maximum braking force, and the brake pedal ratio which can be seen in equation _. The brake pedal ratio determines the pedal force as the brake force is a number that will not change. From equation 6, we determined the optimal brake pedal ratio to be 6:1 which resulted in a brake pedal force of 55.8 pounds.

4.3.2 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.

4.3.3 Suspension

The suspension system for the SAE Baja vehicle was selected based on critical factors such as shock mounting location, suspension travel, durability, and geometry. An Upper Control Arm (UCA) mount was chosen over the more traditional Lower Control Arm (LCA) mount for the front double-wishbone suspension due to its superior performance in off-road conditions. While LCA mounts often face clearance issues, the UCA mount allows for greater suspension travel and improved shock performance without interference from surrounding components. Suspension travel, a key factor for absorbing impacts and maintaining consistent wheel contact on rough terrain, is critical for stability and control. The UCA mount supports longer suspension travel, enhancing the vehicle's ability to handle jumps, uneven surfaces, and hard landings while maintaining traction and stability. Its geometry also aligns with design goals, enabling optimal caster and camber angles for better handling and reduced tire wear. Additionally, the UCA mount's elevated design minimizes the risk of damage from debris and obstacles, ensuring reliable performance in harsh environments. This combination of benefits makes the UCA mount the ideal choice for the SAE Baja suspension system.

4.4 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**

4.4.1 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.

4.4.2 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.

4.4.3 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 Approved Replacement for Section 5: SAE Deliverable

Plan for Timeline

The implementation plan is structured to ensure timely delivery of a robust vehicle platform that meets SAE Systems' requirements. The timeline includes the following phases:

1. Initial Research and Development (Months 1–4):

- Finalize the conceptual design and create the first prototype.
- Conduct detailed simulations and initial materials testing.
- Deliver a functional prototype to SAE Systems by the end of Month 4.

2. Prototype Testing and Validation (Months 5–8):

- Execute rigorous testing under conditions mimicking extreme environments, including jungles, deserts, and mountainous terrain.
- Validate air delivery capabilities and performance under scenarios where repair is not an option.
- Incorporate feedback to finalize a production-ready design.

3. Initial Production Run (Months 9–12):

- Manufacture 50 units for SAE Systems' internal testing and development.
- Deliver these units by the end of Year 1.

4. Full-Scale Production (Years 2–4):

- Scale production to meet contractual obligations while ensuring quality and cost-efficiency.
- Annually review and refine production processes to optimize efficiency.

Team Members the Client May Work With

The following key personnel will collaborate directly with SAE Systems to ensure smooth project execution:

1. Project Lead:

- **Name:** Seth Scheiwiller
- **Role:** Oversee project execution, maintain alignment with milestones, and act as the primary client liaison.

2. Lead Engineer:

- **Name:** Rowan Jones
- **Role:** Manage design, prototyping, and testing processes to ensure the vehicle meets client specifications.

3. Production Manager:

- **Name:** Brennan Pongratz
- **Role:** Oversee manufacturing layout and production scale-up, ensuring alignment with cost and quality goals.

4. Business and Marketing Lead:

- **Name:** Dylan Carley
- **Role:** Develop procurement strategies, negotiate with suppliers, and ensure materials meet quality standards within budget constraints.

5. Diversity and Sustainability Officer:

- **Name:** Wyatt Walker
- **Role:** Ensure inclusivity, innovation, and sustainable practices in all aspects of the project.

Plan for Content

The implementation plan's content will address critical areas to demonstrate our readiness and capability to meet SAE Systems' requirements. These areas include:

1. Vehicle Design and Testing:

- Comprehensive design details, including compatibility with multiple fuel sources, air delivery capabilities, and features to keep the operator safe. The prototype will be tested in extreme environments to ensure durability.

2. Manufacturing and Cost Projections:

- A detailed production plan outlining scalability, cost-efficiency, and quality assurance measures would be provided along with an analysis of parts sourcing and manufacturing techniques to meet the client's needs while staying within the budget.

3. Marketing and Sustainability:

- A marketing plan to attract employees, distributors, and suppliers will be necessary. Sustainable production and end-of-product considerations will be made to stay ethical throughout the process.

4. Team and Infrastructure Snapshot:

- Overview of our company's capabilities, including location, market share, workforce diversity, and financial performance. Also, an insight into infrastructure to make sure it is ready to handle defense-level production.

6 Design Validation and Initial Prototyping

6.1 Failure Modes and Effects Analysis (FMEA)

6.1.1 Front Suspension

Control Arms

The failure modes and effects analysis (FMEA) for the front control arms of the baja vehicle consist of analyzing the upper control arm (UCA) bending moment, lower control arm (LCA) impact bending, UCA to frame mounts, LCA to frame mounts, UCA to frame shoulder bolts, and LCA to frame shoulder bolts. The failure modes for each component consisted of deformation and impact. The cause of each failure was due to a large impact consistently. The effects of failure were mainly bent members on each respective component, as well as broken welds and parts detaching from the vehicle. Displayed below is a table representation of the information portrayed.

Part	Failure Mode	Effects of Failure	Cause of Failure	Severity (1-10)	Action Taken
UCA Bending moment	Deformation	Bent UCA on impact/max suspension compression	Large impact	9	FEA and logically designed CA shock mounting position
LCA Impact Bending	Deformation	Bent LCA on collision impact	Large impact	9	FEA with a worst case rock impact scenario
UCA to Frame Mount	Impact	Broken welds, detach from vehicle	Large impact	9	Robust frame mount design, FEA
LCA to Frame Mount	Impact	Broken welds, detach from vehicle	Large impact	9	Robust frame mount design, FEA
UCA to Frame S-Bolts	Impact	UCA detach from vehicle	Large impact	4	Strength research and confirmation, 1/4" shoulder bolt
LCA to Frame S-Bolts	Impact	LCA detach from vehicle	Large impact	4	Strength research and confirmation, 1/4" shoulder bolt

Figure 27: Control arm FMEA table

6.1.2 Knuckle

Part	Failure Mode	Effects of Failure	Cause of Failure	Severity (1-10)	Action Taken
UCA Mount	Impact or Fatigue	Detach from Vehicle	Large Impact or Cyclic Loading	9	Over-built mounts
LCA Mounts	Impact or Fatigue	Detach from Vehicle	Large Impact or Cyclic Loading	9	Over-built mounts
Tie Rod Mounts	Impact or Fatigue	Detach from Vehicle	Large Impact or Cyclic Loading	8	Tie Rod Failure
Brake Caliper Mounts	Fatigue	Tangling Brake Lines	Cyclic Loading or Large Braking Force	4	Robust Bolts and Mounts
Bearing	Seizing	Wheels Lock	Exceed rating or Dirt	6	Sealed Bearings
Bearing	Detach	Axle Removes	Large Impact	6	Proper Press Fits and Axle Nut

Figure 28: Knuckle FMEA Table

The most important components are where the knuckle mounts to the frame because separation could severely damage other parts in the system. FEA testing done on these components will be thorough and result in safety factors greater than 1.

6.1.3 Rear Suspension

Part	Failure Mode	Effects of Failure	Cause of Failure	Severity (1-10)	Action Taken
Trailing Link	Bending	Compromised suspention	Bending Moment	9	FEA and strength testing
Camber Links	Bending	Incorrect camber	Impact	7	Stong material, possible high clearance links
Shocks	Bottom/top out	Blown shocks	Impact	7	Proper tunning, limit straps
CV Axle	Plunging/extention	Damage to drivetrain, limited travel	Incorrect geometry	5	Design for minimal plunge through full cycle
Shock Mounts	Cyclic fatigue	Loss of shock support	Weak mounts	5	Robust mount construction
Heim Joints	Over rotation	Limited travel	Binding	3	Correct hiem joint selection
Bolts	Shear failure	Compromised suspention	Weak bolts	3	Correct bolt sizing

Figure 29: FMEA for Rear Suspension

6.1.4 Brakes

Part	Failure Mode	Effects of Failure	Cause of Failure	Severity (1-10)	Action Taken
Rotors	Fatigue	Breaks due to large braking force	Slamming on the brakes	9	Use high strength steel
Calipers	Fatigue	Breaks due to large braking force	Slamming on the brakes	8	Use quality calipers
Front Rotor Bolts	Fatigue	Shears from torque	Driver has to slam on the brakes	8	Use bolts with high yield strength
Rear Rotor Bolts	Fatigue	Shears from torque	Driver has to slam on the brakes	8	Use bolts with high yield strength
Front Caliper Bolt	Fatigue	Shears from torque	Driver has to slam on the brakes	8	Use bolts with high yield strength
Rear Caliper Bolt	Fatigue	Shears from torque	Driver has to slam on the brakes	8	Use bolts with high yield strength

Figure 30: FMEA for Front and Rear Brakes

The biggest priorities in braking system design are the rotors and rotor bolts, as they are critical to performance and safety. To prevent failures, Finite Element Analysis (FEA) testing is used to identify potential weak points and optimize the design. Additionally, selecting the appropriate materials for these components ensures they can withstand the high stresses and wear during operation. This combination of advanced testing and careful material choice helps minimize the risk of failure and ensures the reliability and longevity of the braking system.

6.1.5 Steering

Part	Failure Mode	Effects of Failure	Cause of Failure	Severity (1-10)	Action Taken
Tie Rod Mounts	Fatigue	Breaks due to impact or collision	Large Impact	9	Tie Rod calculations
Rack and Pinion	Fatigue	Breaks due to impact or collision	Large Impact	9	Tie Rod calculations
Rack and Pinion	Interference	Dust or mud gets into rack	Dust from other divers, mud build up	4	Use boots to cover the rack
Tie Rod Mounts	Bending	Tie rod bends or buckles	Large Impact	4	Tie Rod calculations

Figure 31: FMEA for Steering

The primary consideration for the steering system is the placement and integrity of the tie rod mounts on the knuckle. To safeguard the knuckle, the tie rods will be designed with a lower factor of safety, ensuring they are the first components to fail under excessive loads. This intentional failure hierarchy protects the knuckle from potential damage.

6.2 Initial Prototyping

6.2.1 Suspension and Steering

6.2.1A - Front Suspension and Steering

The first initial prototype for front suspension was carried out so we could analyze our current measurements, steering angle, tie rod up travel clearance, and approximate ride height. In order to bring this prototype to life, the team utilized 3D printing to produce the knuckle, control arm to frame mounts, and the control arm to knuckle ball joint cup components. The team also used PVC pipe for the control arm members, and a steel tube which was cut to spec to emulate a tie rod.

For the front suspension portion of the prototype, the team figured out that the knuckle design at the time of the prototype interfered with the actuation of the control arms. The knuckle design has since been changed. The steering angle with the knuckle design at the time of the prototype was not found due to the lower control arm cup causing inaccurate rotation of the knuckle. The estimated ride height is approximately 12-15 inches based on the control arm orientation, retrieved from the prototype demonstration.

The first prototype demonstration informed the team in a variety of ways as mentioned in the previous paragraph. The knuckle design has since been modified to address the clearance issue in the lower control arm cup area. The lower control arm cup also needed to be adjusted to work with the ball joints and knuckle design, all of which has been addressed. The control arm angles from the prototype design also was determined to be too aggressive for optimal travel. This could have resulted in binding with the knuckle and cup area. This has also been resolved since prototype 1.

Figure 32: Front Suspension Prototype

6.2.2 Brakes

6.2.2.1 Front Brake Prototype

To prevent interference between the caliper and the wheel, it's crucial to ensure proper clearance between the brake components and the rotating wheel. The calipers and rotors will be mounted on the knuckle in a way that avoids obstruction during wheel rotation, allowing smooth operation. The optimal caliper position on the knuckle should ensure easy integration with the suspension, provide sufficient clearance for effective operation, and allow the calipers to fully engage with the rotors without interfering with other components.

When the calipers and rotors are properly mounted with enough clearance, there will be no interference with the wheel's rotation. The best position for the caliper is opposite the steering and tie rod mounts, optimizing space and ensuring full engagement with the rotor. This setup also prevents interference with other suspension components, ensuring efficient braking performance.

This design requirement emphasizes precise positioning and clearance management. Proper alignment of the caliper ensures full rotor engagement and avoids contact with the wheel, enhancing both performance and safety. By placing the caliper opposite the tie rod mounts, the design maximizes space and integrates seamlessly with other components, contributing to a more efficient and compact braking system.

Figure 33: Front Brake Prototype

6.2.2.2 Rear Brake Prototype

The larger brake diameter does fit within the available packaging space for the rear gearbox without causing any interference. By carefully designing the brake system, the increased size of the brake components, including the rotor, can be accommodated without compromising the gearbox's packaging or function. This ensures that the braking system and the rear gearbox can coexist in the same space while maintaining optimal performance for both systems.

The caliper will be securely mounted to the rear gearbox using a custom caliper mount. This mount ensures that the caliper can fully clamp onto the rear rotor, providing efficient braking performance. By designing and manufacturing a mount that attaches to the gearbox casing, we can align the caliper correctly with the rotor. This alignment is crucial to ensure that the brake system functions properly, with the caliper engaging the rotor evenly for effective braking.

This requirement informs the design by highlighting the importance of precise integration between the brake system and the rear gearbox. Ensuring that the larger brake diameter fits within the available packaging space drives the need for careful component positioning and clearance management. The custom caliper mount guarantees that the caliper is correctly aligned with the rear rotor, ensuring the braking system functions effectively without interfering with the gearbox. By designing a solution that integrates both systems, we can optimize performance, packaging, and functionality.

Figure 34: Rear Brake Prototype

6.3 Other Engineering Calculations

6.3.1 Suspension Analysis Software (Optimum Kinematics)

Optimum Kinematics is a suspension analysis software that the team has chosen to use to plot points and narrow down how various mounting points will affect how the suspension will travel and cycle. The team has been referencing the frame assembly in SolidWorks and determining our mounting points based on that as well as how we envision everything fitting together. From the points that are determined in SolidWorks, those points can be plotted in optimum to be able to visualize the geometry of control arms and shock placement, along with running simulations to ensure the system cycles as it should.

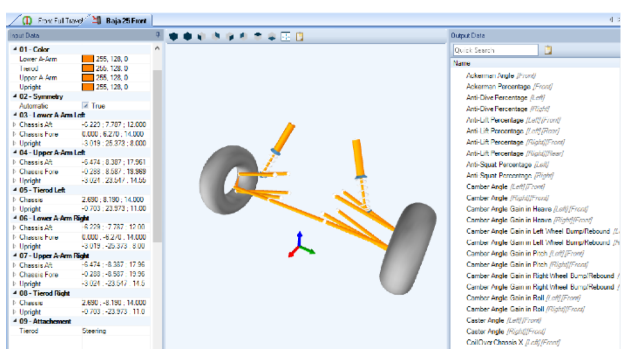


Figure 35: Optimum Kinematics model, input data, and output data

Thus far, Optimum has been able to inform the team of bump steer with the original points, and with that information we were able to adjust based on that bump steer output metric. Through multiple iterations within optimum, we were able to achieve zero bump steer, meeting our initial design criteria for ourselves from the QFD.

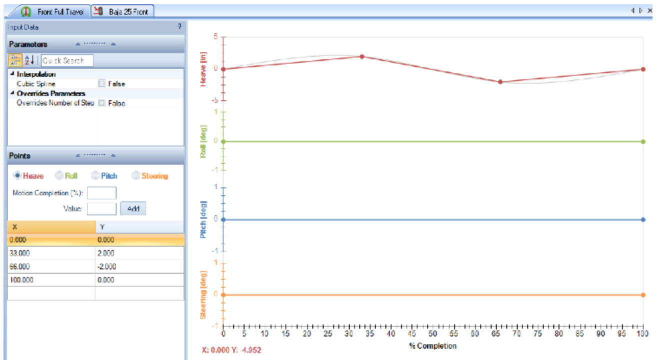


Figure 36: Optimum Kinematics Simulation

In the future, we will continue to utilize optimum kinematics for its simulation and analysis features as we did to analyze bump steer. The steering team has been looking into optimum simulations surrounding Ackerman steering values to meet their goals in the steering sector of the vehicle. Optimum Kinematics has been a useful tool thus far and will continue to assist in getting the team the information and metrics to build a operational and well performing Baja vehicle.

6.3.2 Brake Finite Element Analysis (FEA)

FEA was done for the front and rear rotors to ensure that none of the rotors will fail if the rotors endure heavy braking over short and long spans.

6.3.2.1 Front Brakes

The Finite Element Analysis of the front brake rotor was performed to evaluate its performance under loading conditions and ensure it meets safety and durability standards. The results show a maximum deformation of just 1.139×10^{-6} inches, indicating minimal displacement and confirming the rotor remains well within its elastic limit. The maximum stress on the rotor is 68.3 PSI, which is well below the material's failure threshold, ensuring the rotor's structural integrity under normal braking forces. Additionally, the design includes a safety factor of 15, meaning the rotor can withstand up to 15 times the anticipated maximum load before failure. This generous safety margin ensures the rotor's reliability and long-term performance, even under extreme conditions. In summary, the FEA confirms that the front brake rotors are durable and safe for this year's SAE Baja vehicle.

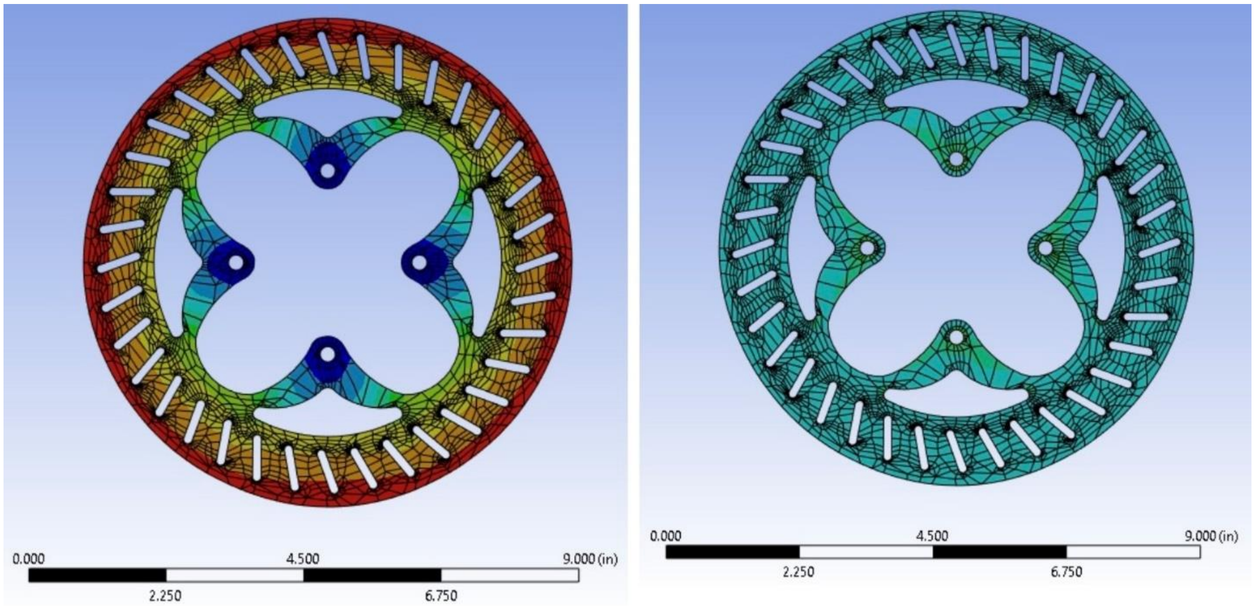


Figure 37: Left to Right Max Deformation and Max Stress of Front Rotor

6.3.2.2 Rear Brakes

The Finite Element Analysis of the rear brake rotor indicates strong performance under typical loading conditions. The rotor experiences a maximum deformation of only 7.7819×10^{-7} inches, suggesting little displacement and confirming it stays within elastic limit. The peak stress is 57.7 PSI, which is well within the material's safe threshold, ensuring the rotor remains structurally sound under braking forces. Additionally, with a safety factor of 15, the rotor can tolerate up to 15 times the anticipated maximum load before failure, providing a significant margin of safety. These findings confirm the rotor's reliability and robustness for demanding rear brake applications in automotive systems.

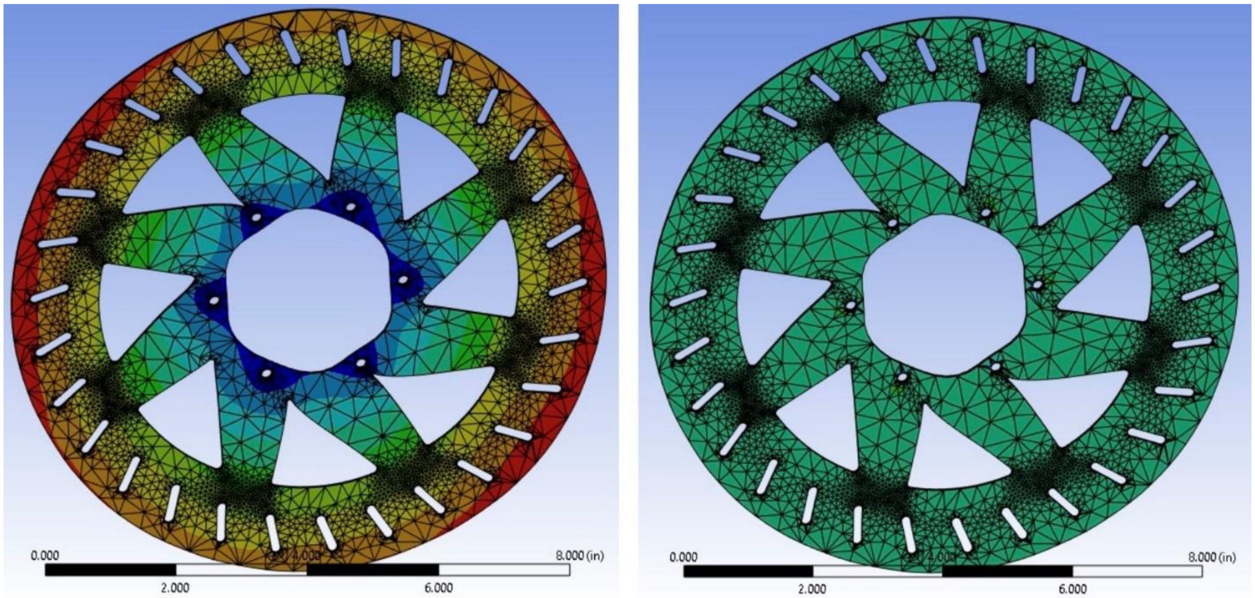


Figure 38: Left to Right Max Deformation and Max Stress of Rear Rotor

6.3.3 LCA and UCA Frame Tabs: Factor of Safety Evaluation

The analysis focused on validating the strength and safety of the Lower Control Arm (LCA) and Upper Control Arm (UCA) tabs under a maximum load of 2,000 lbs. The critical requirement was achieving a minimum Factor of Safety (FoS) of 3 to ensure durability and reliability during extreme conditions. The components were fixed at the points where they are welded to the frame and evaluated using finite element analysis (FEA) in SolidWorks.

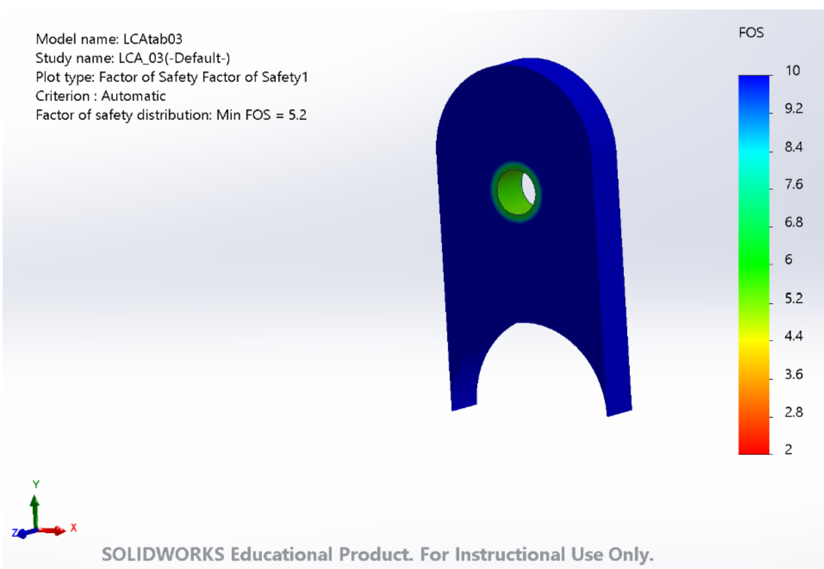


Figure 39: LCA Tab FoS

The results indicate that the LCA tab far exceeds the minimum required safety factor of 3. The stress is concentrated near the mounting hole but remains well within the material's safe limits. This confirms the design's robustness under the maximum load.

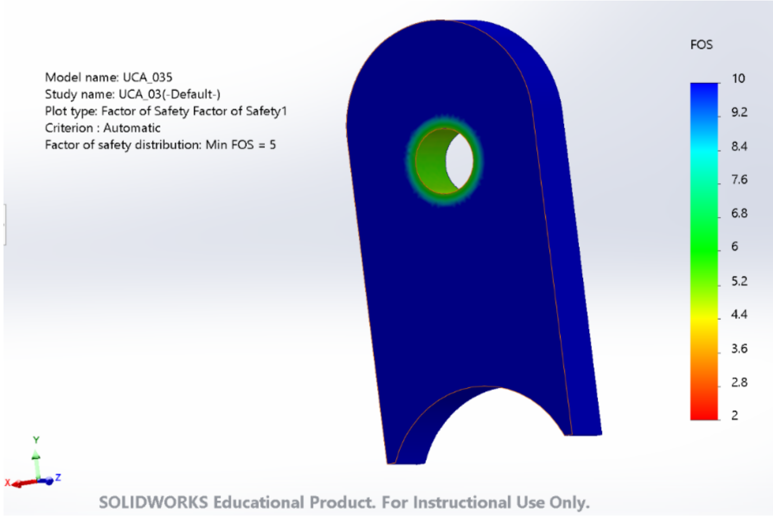


Figure 40: UCA Tab FoS

Like the LCA tab, the UCA tab also exceeds the required FoS of 3. Stress concentrations are localized near the mounting hole, but the overall design ensures structural integrity under the maximum applied load.

The FEA results validate the strength and reliability of the LCA and UCA tabs under the specified loading conditions. Both components exceeded the minimum FoS requirement of 3, demonstrating their suitability for the SAE Baja vehicle. Moving forward, these results confirm that the tabs are ready for manufacturing and integration into the vehicle's suspension system without the need for further design modifications.

6.4 Future Testing Potential

Future testing will focus on validating the suspension, brakes, and steering systems to ensure they perform as expected under real-world conditions. For the suspension system, the vehicle will be driven over rough terrain to evaluate how well it absorbs impacts, maintains stability, and prevents excessive wear or failure in its components. Suspension travel, ground clearance, and alignment will be measured under dynamic loads to verify calculations and software simulations.

The braking system will be tested to confirm it can reliably lock up all four wheels, meeting competition safety requirements. Stopping distance and consistency will also be evaluated under varying conditions to ensure optimal performance. The steering system will undergo testing to verify smooth operation, responsiveness, and alignment, ensuring the vehicle meets the required 7-foot turning radius. Driver feedback will be collected throughout testing to assess comfort, handling, and overall user experience, with adjustments made as needed to refine performance before competition.

7 CONCLUSION

This report summarizes the progress made by the suspension, brakes, and steering sub-team of the NAU Baja SAE team in designing and developing systems to meet the demanding requirements of the SAE Baja competition. The goal was to create durable, high-performing components that enhance the vehicle's performance, reliability, and control in key events like Hill Climb, Endurance, and Suspension & Traction. The team focused on optimizing suspension geometry, ensuring braking reliability, and designing a responsive steering system. Major milestones included completing CAD models, conducting finite element analysis (FEA), and fabricating prototypes for testing. Initial prototypes, including PVC control arms and 3D-printed components, were tested to validate software results, and components such as brake rotors and hubs were manufactured. These efforts provide a solid foundation for refining designs and ensuring manufacturability and safety compliance. The progress achieved this semester moves the project closer to delivering a competitive vehicle, balancing performance, and cost-effectiveness to support a strong result in the competition.

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SAE Baja Chassis Team

Conceptual 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, KC Hilites, Steve Sanghi

Faculty Advisor: David Willy

Instructor: David Willy

DISCLAIMER

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

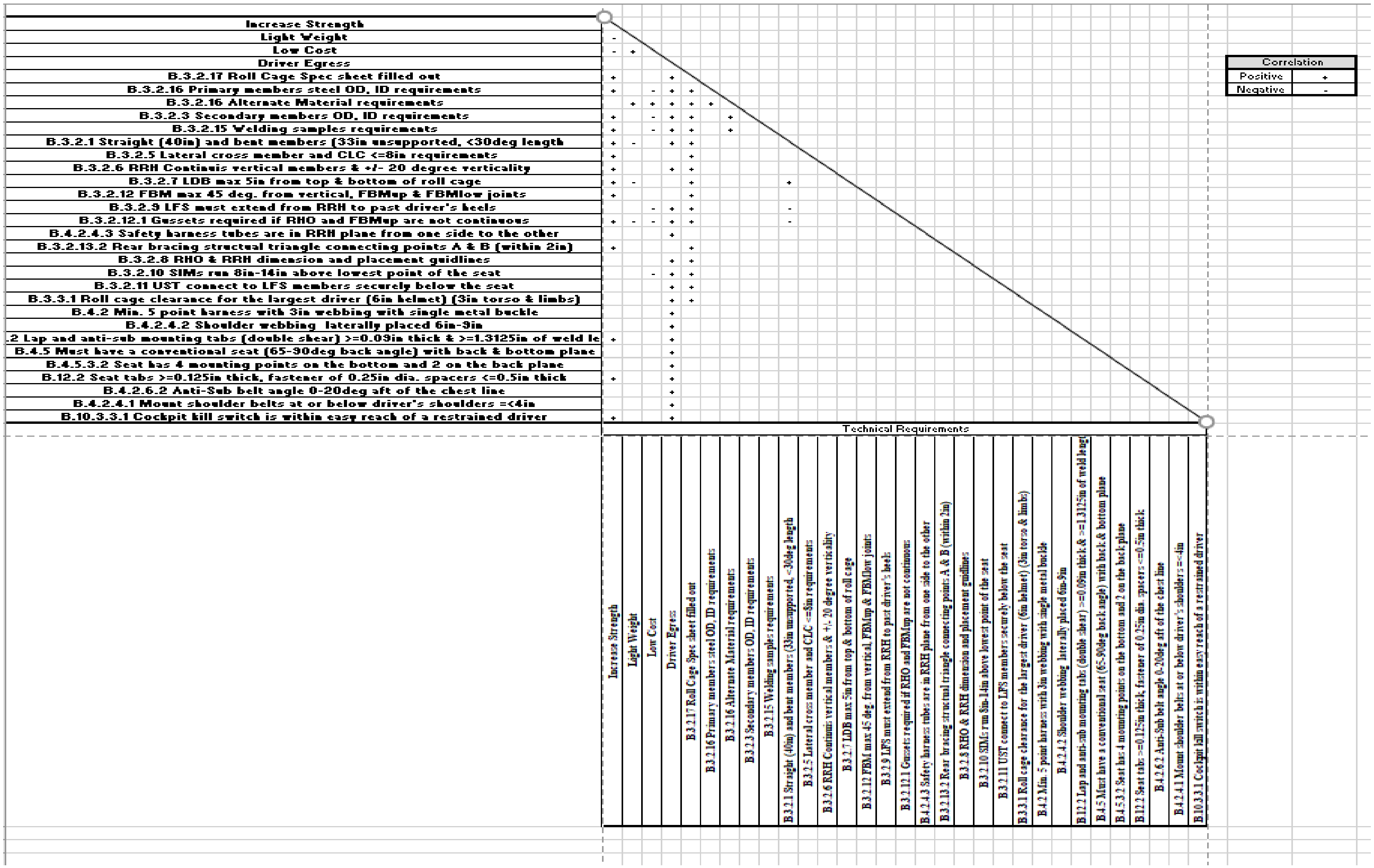
Technical Requirements

Customer 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
2	Performance	9	9	-3	9		9	3	9					1	3	3	9
3	Safety	9	-3	-3	9	9	9	9	9	9	9	9	9	9	9	9	9
5	Durability	9	-3	-1	1		9	9	9	9	9	9	3	9	9	9	9
4	Affordable	-3	3	9	1		9	9	9		3		1	3	1	3	1
8	Comfort	1	1	-3	9								9			3	
7	Ease of Fabrication	-3	3	-1	-3		3	3	3	9	3	3	3	3	3	3	-3
9	Aesthetics	1	1	-3	-1					3	3	3	9	3	9	3	9
1	Pass Techs	9	-1	-1	9	9	9	9	9	9	9	9	9	9	9	9	9
6	Balanced Weight	-3	-1	-3	1		3	3	3		3	3	3		3	1	1
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
Technical Requirement Targets			100	2500	5		1,0.065,0.11		1,0.035		40,33	8		5			

Customer Weights	Customer Requirments	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 structual 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		3	1	1	1		1								3		A		C	B
3	Safety	9	9	9	9	9	9	9	9	9	3	3	9	9	9	9					ABC
5	Durability	3	3	9	9	3				3	1	1	3						A		BC
4	Affordable	1	1	3	1	1		1		1	3	3	1	1			C	B	A		
8	Comfort	9		9	9	9	9	9	9		9		1	9	9		A			B	C
7	Ease of Fabrication	-1	-3	-3	1	-3	-3	1	1	-1	-3	-3	-3		-3	-1		AC	B		
9	Aesthetics		3	9	9		9	3			3				1				A	B	C
1	Pass Techs	9	9	9	9	9	9	9	9	9	9	9	9	9	9	9				A	BC
6	Balanced Weight		9	3	1	3	3												A	B	C
Technical Requirement Units		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					8,14		6,3	3	6,9	0.09, 1.3125			0.125,0.25,0.		4						

Customer Opinion Survey

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

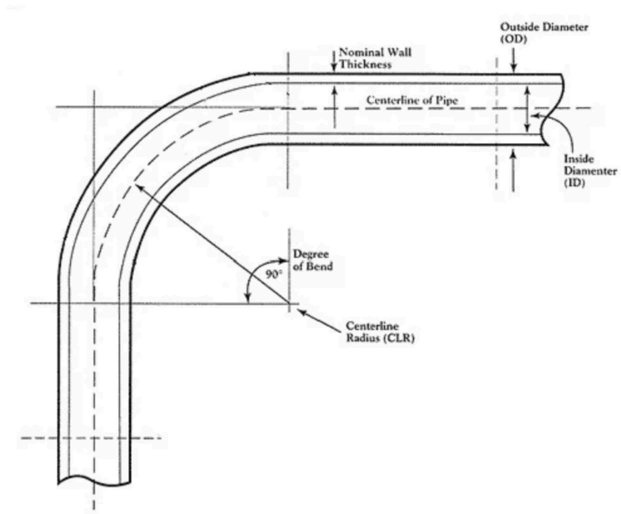
3.3.1 Charles:

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:

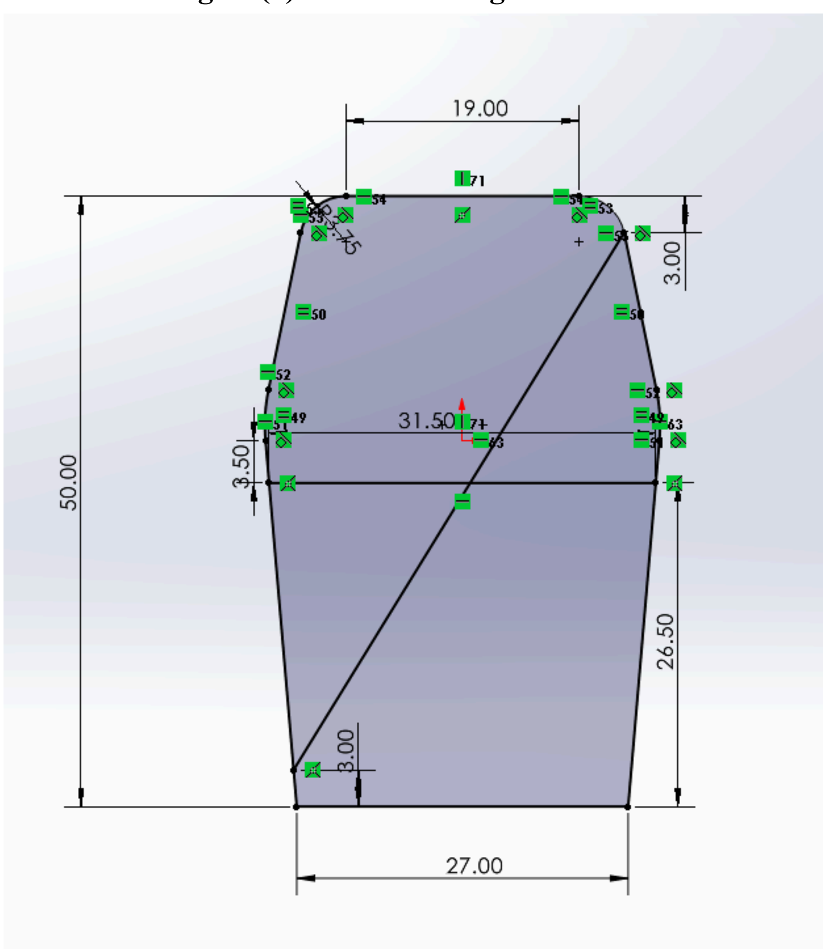
$$rad = \theta \cdot \frac{\pi}{180} \qquad L = rad \cdot \ clr$$

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.

$$0.28 \; * \; 14.7 \; = 4.12\text{”}$$

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:

$$OD = 1.25\ in$$

$$Wall\ Thickness = .065in$$

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

$$Lenght \approx 49ft$$

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

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

Secondary Members:

$$OD = 1\ in$$

$$Wall\ Thickness = .035in$$

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

$$Lenght \approx 36ft$$

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

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

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

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.

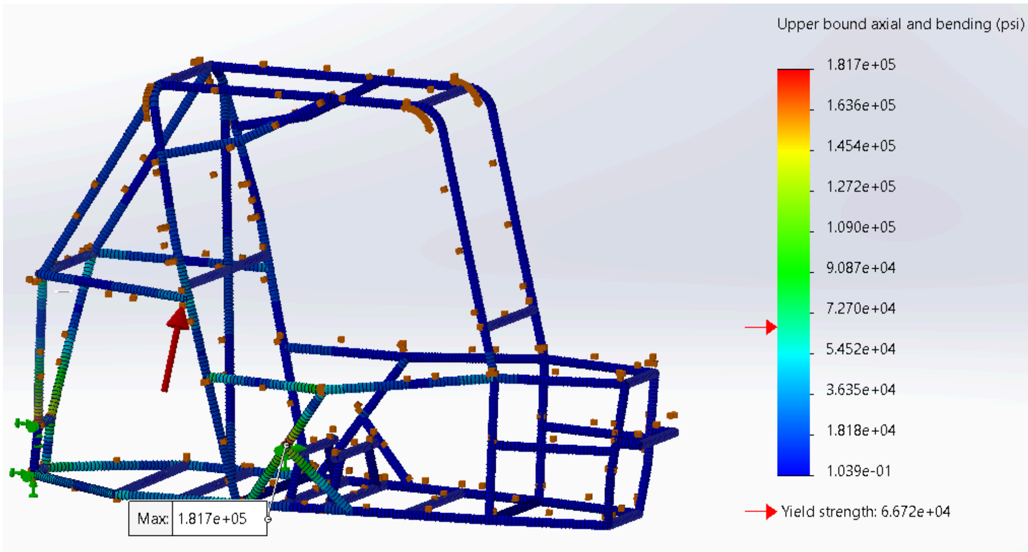


Figure (10): FEA of Stress Analysis, Jumping the Car and Falling 10 ft Then Landing on One Rear Wheel

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.

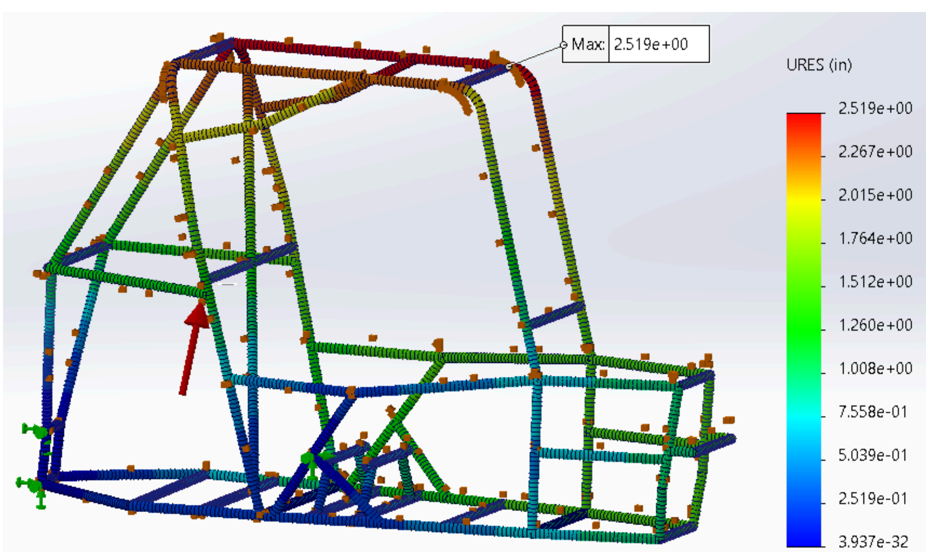


Figure (11): FEA of Displacement Analysis, Jumping the Car and Falling 10 ft then Landing on One Rear Wheel

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.

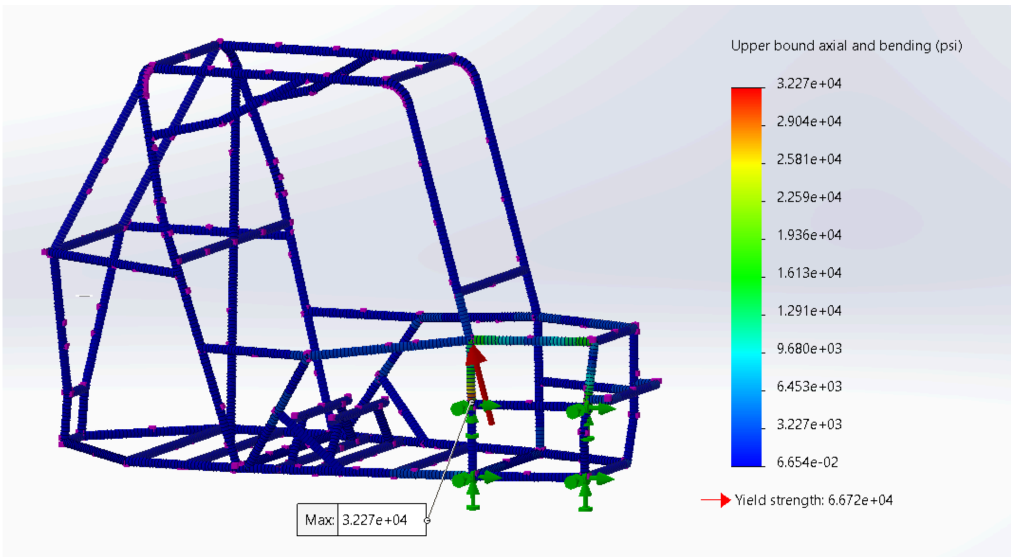


Figure (12): FEA of Stress Analysis, Jumping the Car and Falling 10 ft then Landing on One Front Wheel

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.

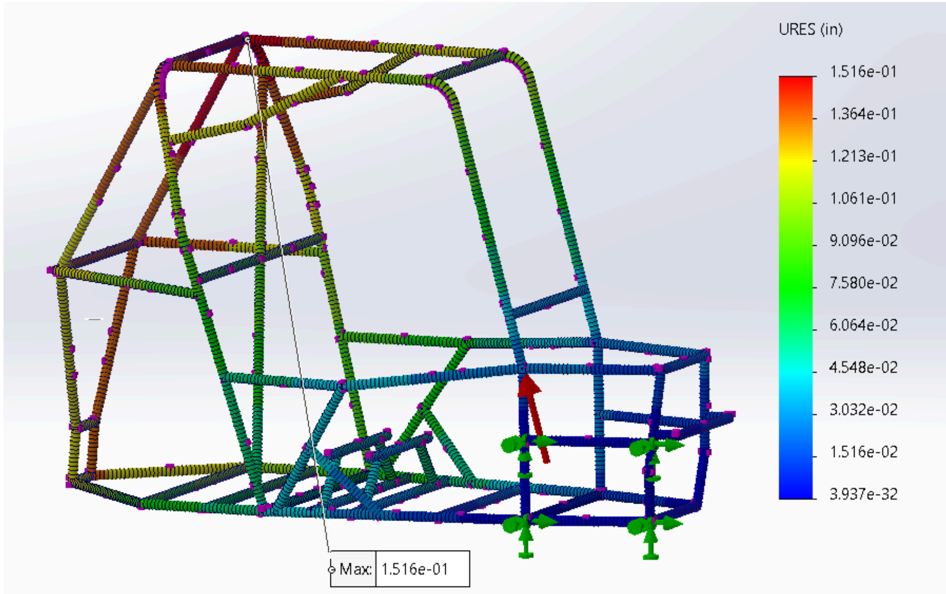


Figure (13): FEA of Displacement Analysis, Jumping the Car and Falling 10 ft then Landing on One Font Wheel

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.

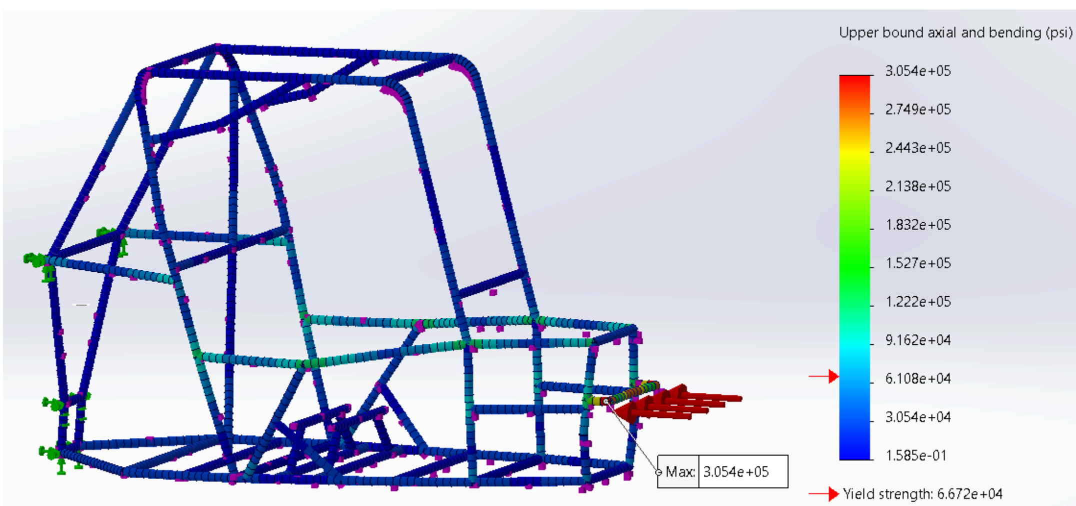


Figure (14): FEA of Stress Analysis: Hitting a Wall Going 25 mph

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.

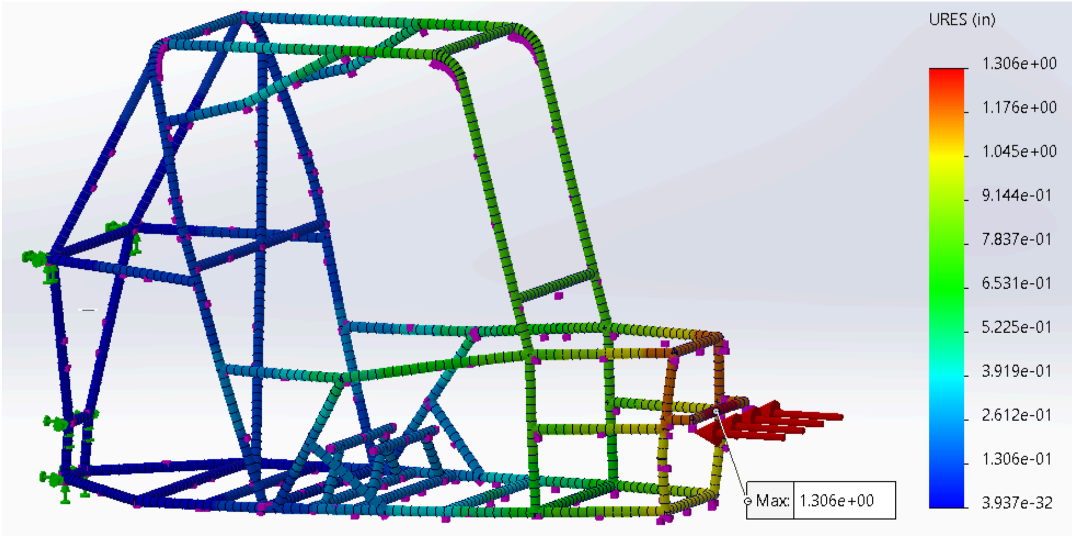


Figure (15): FEA of Displacement Analysis: Hitting a Wall Going 25 mph

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.

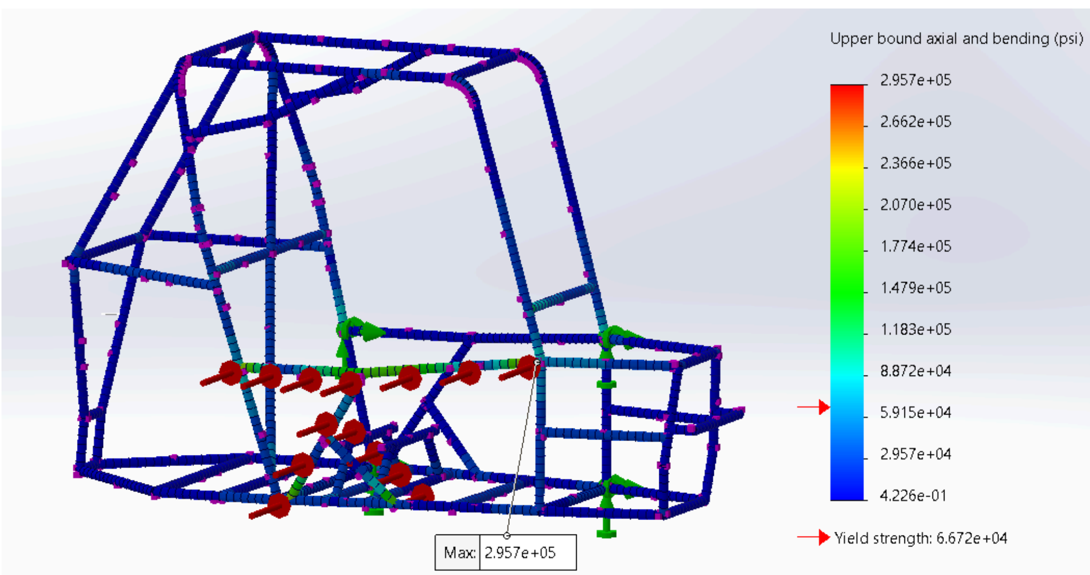


Figure (16): FEA of Stress Analysis: Car Getting T-Boned by Another Driver going 25 mph

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.

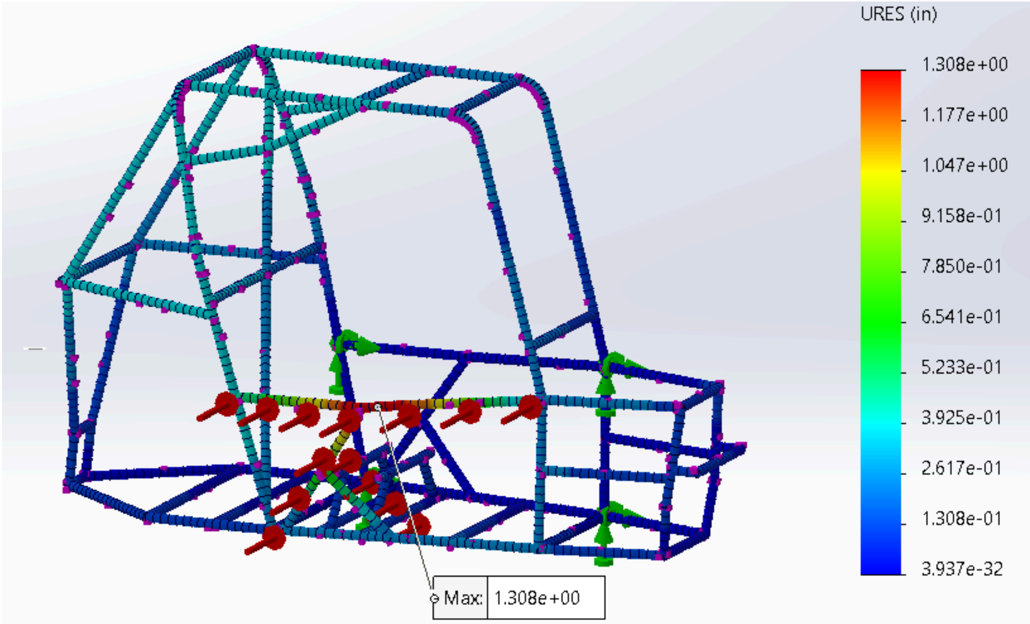


Figure (17): FEA of Displacement Analysis: Car Getting T-Boned by Another Driver going 25 mph

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

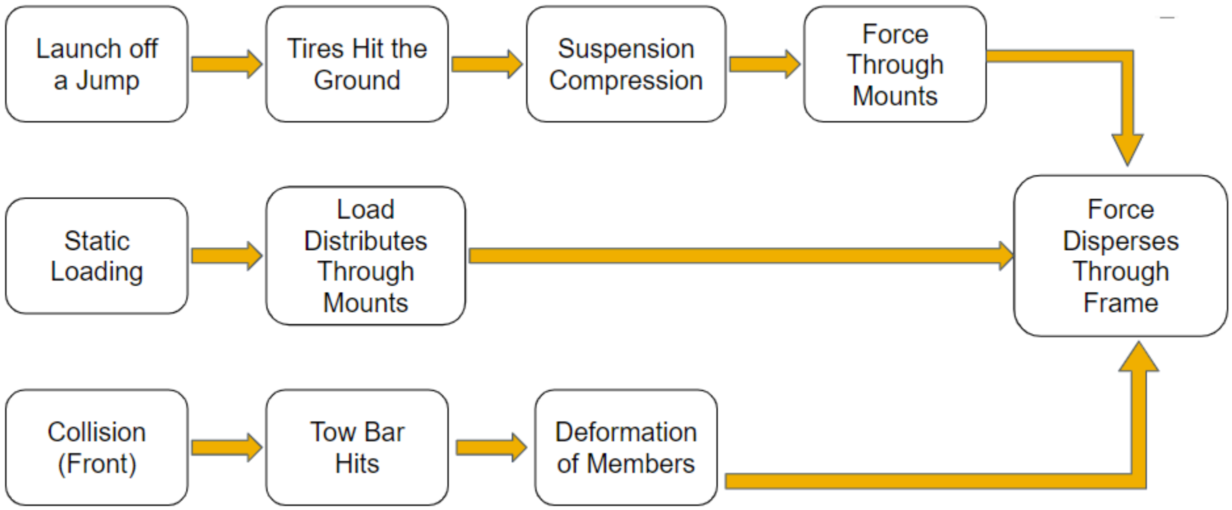


Figure (18): Functional Model

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.

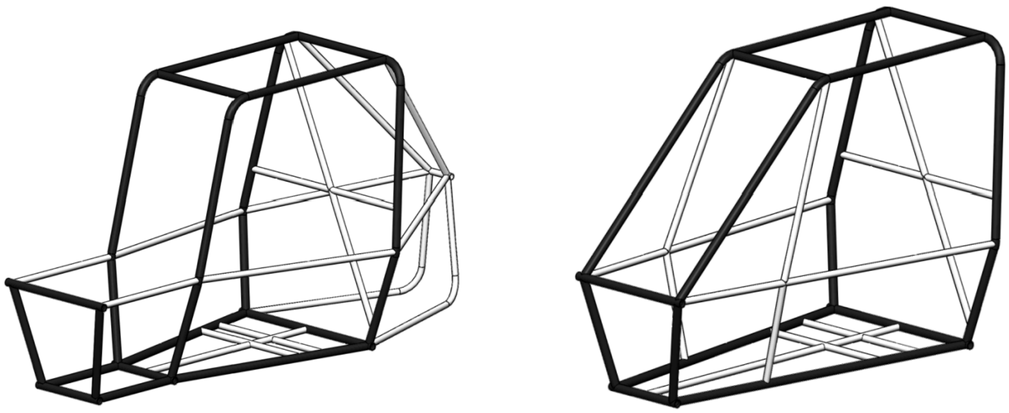


Figure (19): Rear Braced frame (left) Front Braced Frame (right)

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

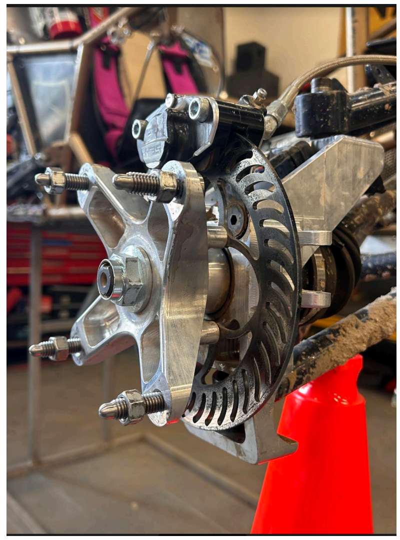


Figure (21): Out-Board Brakes

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

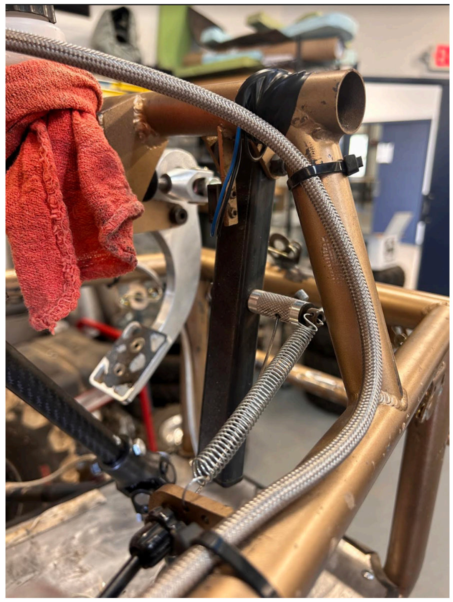


Figure (23): Hanging Pedals

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.

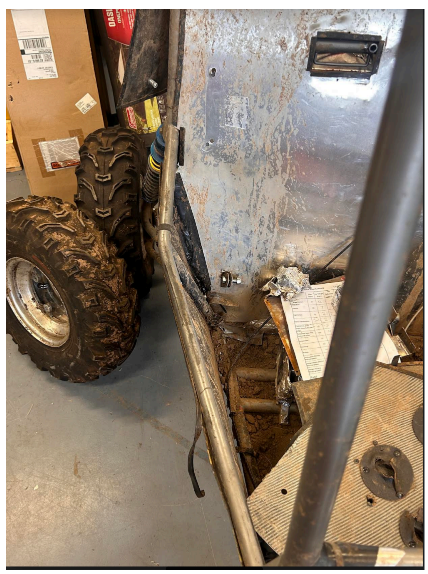


Figure (24): SIM Angled Outward

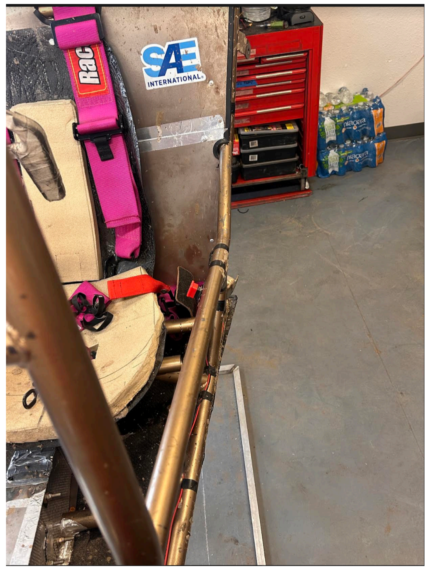


Figure (25): SIM Angled Inward

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

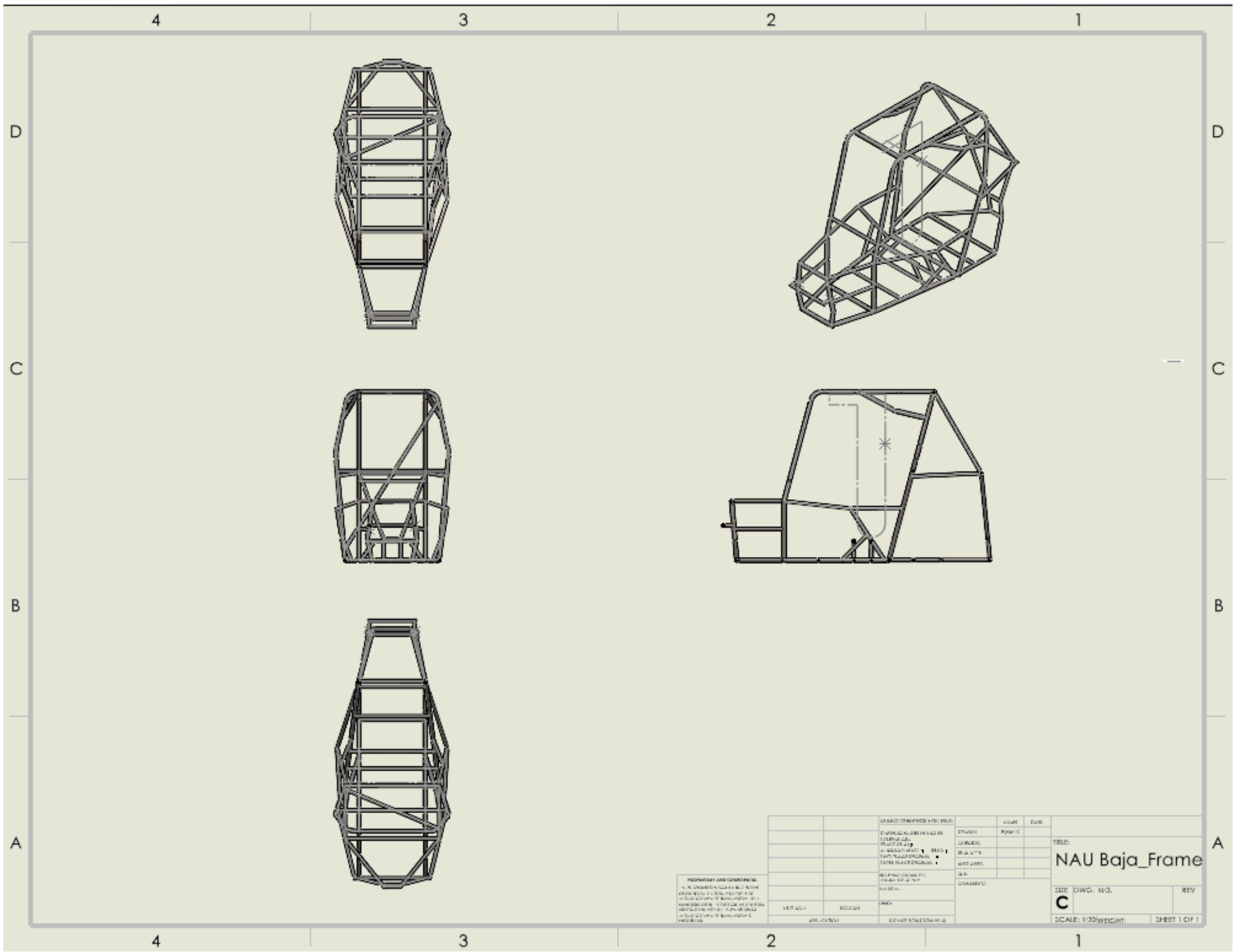


Figure (26): Current State CAD

NUMBER	MEMBER	LENGTH (IN)	BEND (DEG)	QTY
1	RRH	124	78.5 2x, 16.4 2x	1
2	FAB UPPER	27	0	2
3	RLC	21.5	0	1
4	SBT RIGHT	14.5	0	1
5	FAB MID	20.6	0	2
6	RV	26.32	20.5	2
7	RH1	4.5	0	1
8	RH2	5	0	1
9	FAB LOW	32.8	38.69	2
10	ENGINE SUPPORT	27	0	2
11	SIM	35.5	35.5	2
12	SIM SUPPORT 1	17.9	0	2
13	ALC	27	0	1
14	SIM SUPPORT 2	10.85	0	2
15	UST 1	25.3	0	1
16	UST 2	23.67	0	1
17	LFS	44.41	9.5	2
18	UST 3	20.3	0	1
19	SIM FRONT	15	0	2
20	FLC	17	0	1
21	UPPER CA SUPPORT	15.71	0	2
22	FBM FRONT	17.2	20	2
23	ELC	17	0	1
24	TOW BAR SUPPORT	2.5	0	2
25	TOW BAR	18.7	0	1
26	GLC	17	0	1
27	FBM LATERAL	17	0	1
28	SSM	6.5	0	4
29	SS	15	0	2
30	RHO	83.2	75, 15	2
31	CLC	17	0	1
32	SBT LEFT	14.5	0	1
33	LBM	51	0	1
34	RHO GUSSET	11.5	18	0

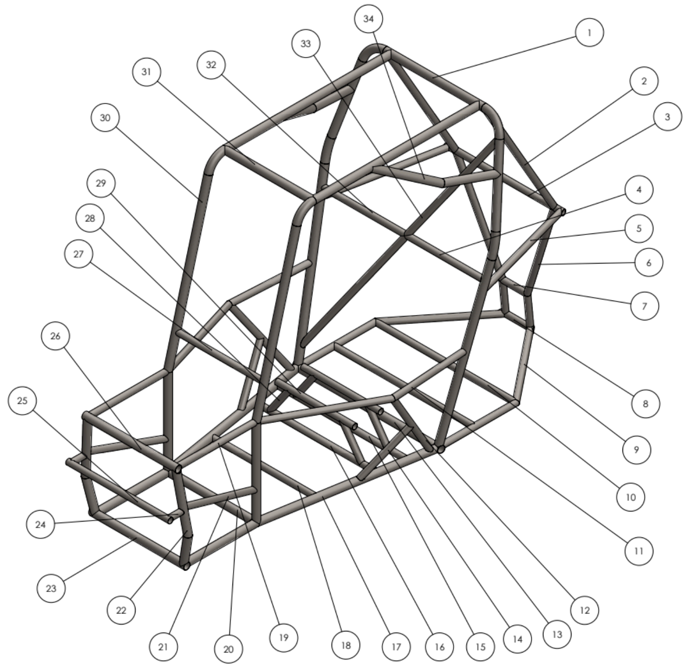


Figure (27) Cut List

5 Business RFP

5.1 Overview

According to SAE requirements, the team is tasked with writing an RFP. This is a business document that announces a project and describes it. This is an approved substitute, per Dr Willy for the team's schedule, budget, and bill of materials. The chassis team was responsible for the cover letter for the document.

5.2 Formal Cover Letter

Dear Society of Automotive Engineering,

Northern Arizona University's 2025 SAE Baja team is excited for the opportunity to participate in the 2025 Arizona SAE Baja competition. The team has eagerly been designing the best car possible to compete with in all of the competition categories. As seniors in the mechanical engineering department we are thrilled to put our newly acquired skills to the test and learn more about automotive design in the process.

As part of the SAE guidelines for the competition the team has been designing everything on the car to adhere to the 2025 rulebook. The goal for NAU is to perform well at competition and to provide a safe experience for the drivers during the various races. This will put our knowledge of engineering principles to the test and ensure that the team makes matured manufacturing decisions. The team has made great strides to further spread the word about SAE Baja by acquiring sponsorships that provide financial support or manufacturing services.

NAU has been competing as SAE Baja competitions for years and has always sparked interest in automotive engineering for the mechanical engineering students. The NAU 2025 Baja team will continue to prepare and fabricate a Baja car that both SAE and NAU will be proud to have at the competition. We look forward to working more closely with SAE to ensure a safe and enjoyable competition coming in the next year.

Sincerely,
NAU SAE Baja 2025 Team

6 Design Validation and Initial Prototyping

6.1 Failure Modes and Effects Analysis (FMEA)

Table(2): FMEA

Part # and Functions	Potential Failure Mode	Potential Effect(s) of Failure	Potential Causes and Mechanisms of Failure	RPN	Recommended Action
Mounting Points	Mounting Points Break	Loss of control	Overloading/Poor Design	54	Reinforce mounting points
Frame Material	Fatigue	Cracking or bending of the frame	Hitting Obstacles & Low-Quality Tube	144	Perform FEA on frame
Shock Absorption	Poor Shock Absorption	Increased stress on frame members	Improper shock loading	64	Test vehicle for the worst-case scenario
Weight Distribution	Imbalanced Weight	Poor handling & possible roll-over	Faulty design /unaccounted weight	210	Test vehicle with every driver & analyze weight distribution on SolidWorks
Frame Bracing	Inadequate Bracing	Increased bending and flexing of the frame	Improper design	56	Perform FEA on frame
Frame Structure	Weld Failure	Structural integrity compromised	Insufficient weld penetration	80	Ensure every welder passes the weld certificate
Frame Members	Tube Failure Under Load	Tubes crumble/Fold	Improper tube sizing or unsupported spans	9	Use FEA to ensure no unsupported spans & Proper tube dimensions are met
Frame Geometry	Integration Interference	Premature wear on components	Design errors/insufficient testing	60	Assemble 3D model of frame & rigorous testing
Joints	Joints Break	Member Separation	Poor joint design, improper welding	30	Keep joint geometry as simple as possible

From Table 2 the FMEA that the chassis team performed identifies several failure scenarios, the reason why that failure could occur, the result that would occur from the failure, and how the team can mitigate the failures to better ensure the frame will not fail under these conditions. The Risk Priority Number (RPN) represents which failure scenarios are most likely to happen and if they do occur, which one would negatively impact the performance of the car the most. From the table above, the failures with the highest RPN values are an imbalanced frame and material fatigue. Both of these failures could result in the frame being inoperable and unsafe for our driver and other competitors. The frame team plans to mitigate these failures by performing FEA analyses to reinforce the frame and ensure the weight is as balanced as possible to prevent any fatigue on the members and possible rollover of the frame.

6.2 Initial Prototyping

Figure (28): PVC Prototype

6.2.1 Prototyping Questions

For the first prototype, the team decided to make a one-to-one scale PVC roll cage and front end. This was to allow the team to see the cockpit and front end as it shows in the CAD of the frame. This was beneficial for the chassis sub-team and the entire Baja team as a whole since this gave other members of the team an idea as to how big the frame would look. This also informed the team on how the driver fit in the frame and how coping angles worked.

6.2.2 Prototyping Process and Results

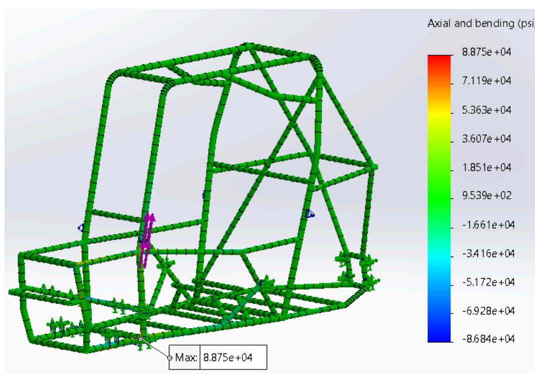
The process was also to reflect the actual fabrication process of the metal frame. The cutting and coping process was still newer to the team and assisted in getting the members familiar with the coping angles and lining up the frame properly. Though bending PVC is very different from bending steel tubes, it gave ideas as to how bends will look on the actual metal tube bender. The prototype was according to the specifications of the frame at the time but the front end was too large and suspension would need better angles for mounts.

6.2.3 Informing the Design

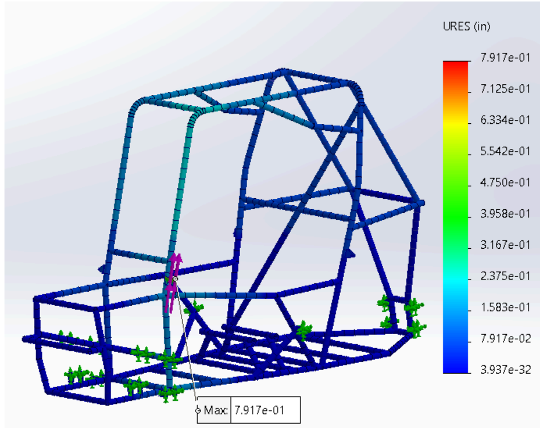
After the PVC prototype was completely assembled the team observed that the front end was very large and that the plan for inboard brakes was changed. As a result, the front end was shrunk down to better accommodate the changes that were made. The rear roll hoop was also changed to allow since the driver space was slightly larger than what was needed. The entire frame was made less wide as a result to make it lighter and easier for other sub-teams to design around.

6.3 Other Engineering Calculations

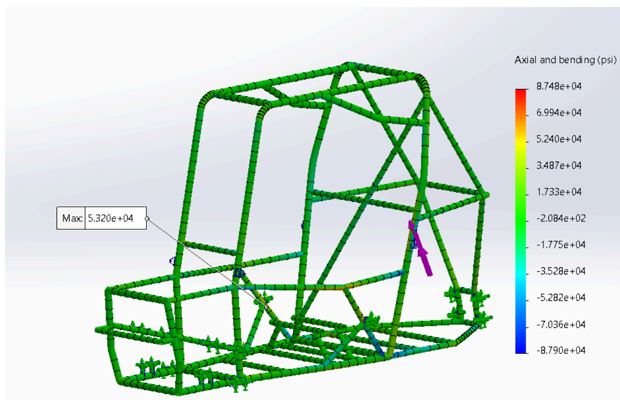
Since the concept selection, the chassis team has performed further refinement of the FEA using Solidworks Simulation and similar scenarios. The results of the FEA are shown in Figures 27-34.



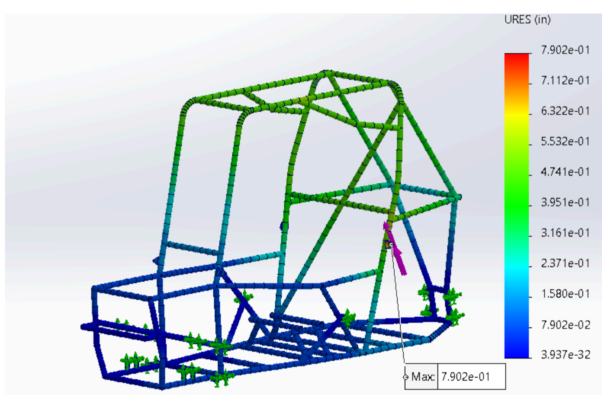
Figure(29): FEA of Stress Analysis, Jumping the Car and Falling 5 ft then Landing on One Front Wheel



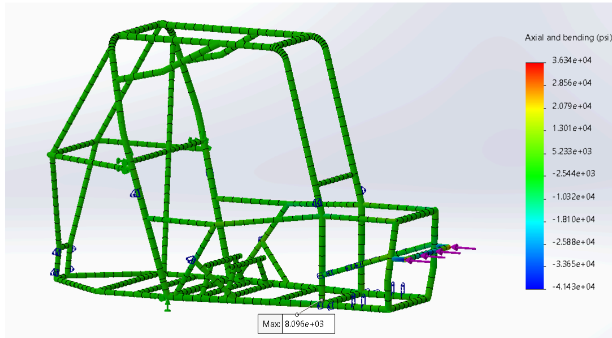
Figure(30): FEA of Displacement Analysis, Jumping the Car and Falling 5 ft then Landing on One Font Wheel



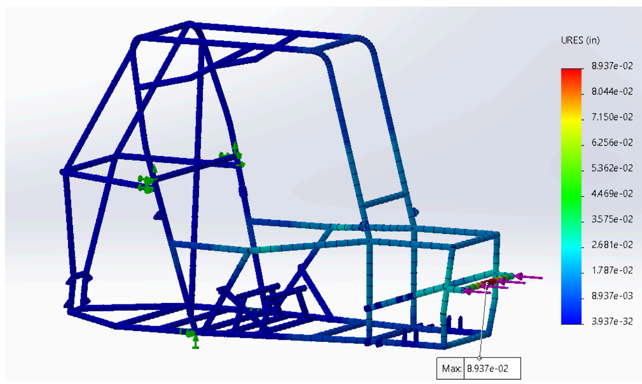
Figure(31): FEA of Stress Analysis, Jumping the Car and Falling 5 ft Then Landing on One Rear Wheel



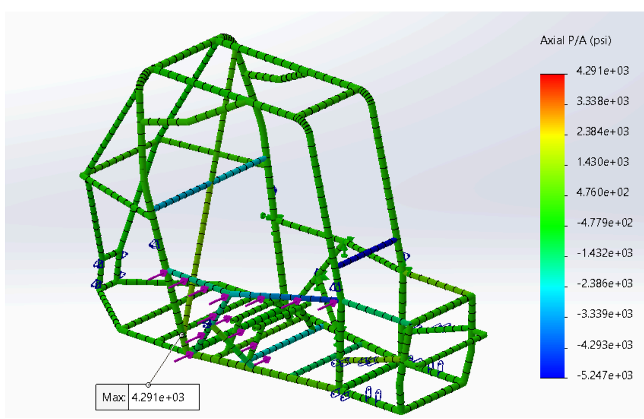
Figure(32): FEA of Displacement Analysis, Jumping the Car and Falling 5 ft then Landing on One Rear Wheel



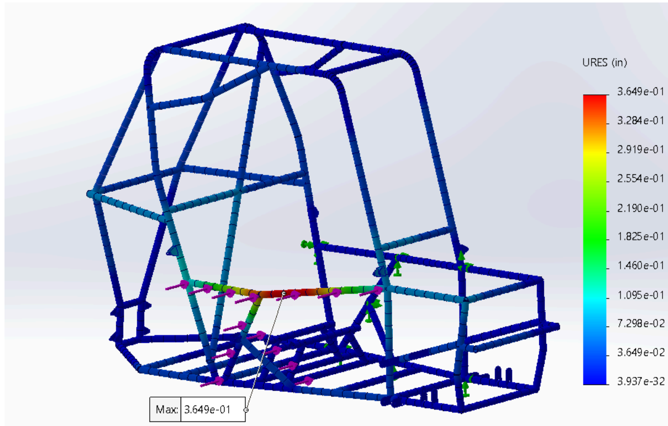
Figure(33): FEA of Stress Analysis: Hitting a Wall Going 20 mph



Figure(34): FEA of Displacement Analysis: Hitting a Wall Going 20 mph



Figure(35): FEA of Stress Analysis: Car Getting T-Boned by Another Driver going 20 mph



Figure(36): FEA of Displacement Analysis: Car Getting T-Boned by Another Driver going 20 mph

From the figures above, the chassis team's engineering calculations consist of FEA simulations in SolidWorks, which has proven extremely beneficial in refining the car's frame. Overall, the chassis team has been most concerned with the tubes crumpling or deforming under stress resulting in driver injury or leaving the car inoperable. To better understand how the frame will perform during competition, the team has come up with four worst-case scenarios that the car could experience during competition. From the FEA results the frame will be plenty strong if these worst-case scenarios occur. Any deformation to the tubes is minimal and would not result in driver injury, additionally the deformations will not impact the other sub-teams ability to operate. Lastly, the stresses that the frame could experience in the scenarios from the figures above are well below the yield strength of the 4130 Chromoly tube, which is 63,000 psi. The main takeaway from these FEA analysis, is that the suspension mounts need to be reinforced to withstand the loads that they will experience. From Figures 27-23, the stress on the suspension mounts would break the mounts off of the tubes they are welded to.

6.4 Future Testing Potential

As a sub team as we are manufacturing the frame we are referencing the rules and the current CAD design to ensure that it is within rulebook guidelines. After the frame is fully welded the team will plan on doing several technical inspections that will simulate the competition inspection to the best of our ability. Once the car is fully assembled with all components the next step would be to test drive in a fashion that will simulate the competition and the various events.This will allow the team to see if anything should be adjusted to improve performance, as well as improving integration amongst other components. Due the costly and labor intensive nature of this project, there will not be much testing done on the prototypes aside from virtual testing. Previous years cars will be used as a benchmark for testing due to the similarity of the cars and the availability as well.

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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8 Appendices

8.1 Appendix A

Table(1): Engineering Requirements

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