-Final Design Report-

SAE Baja 2023-2024

NORTHERN ARIZONA UNIVERSITY

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DISCLAIMER

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EXECUTIVE SUMMARY

The following report details the progress of the 2023-2024 Society of Automotive Engineers Baja capstone team at Northern Arizona University from August 28th to November 28th. This capstone team is adhering to the engineering rules and practices established by the SAE Baja organization, which hosts a nationwide collegiate design series tasking students with the design and construction of a single seat, all-terrain vehicle. The successful performance of this year's team will establish NAU as a competitive engineering school and strengthen its own internal Baja program via technical documentation and underclassman involvement. The team consists of 13 members who have been separated into 4 sub-teams: front end, rear end, drivetrain, and frame. These sub-teams are responsible for the optimization of their own region of the car as well as the geometric integration with the designs of all other sub-teams. Each sub-team began the design process by establishing general customer requirements and technical engineering requirements derived from the SAE Baja rulebook as well as other well-established automotive resources. These requirements were further understood and reinforced with the completion of a team-wide literature review within the automotive space and the SAE Baja community. Sub-teams then completed iterative mathematical modeling cycles to address certain engineering requirements and specific design questions within the scope of their region of design.

Following the initial design conceptualization stage, each sub-team identified several relevant sub-systems within their design that offered the potential for design variation. These variants were assessed against each other via standard mathematical calculation and analysis with respect to technical specifications. A decision matrix was generated by each sub-team to concisely illustrate the results of this concept selection process and organize the efforts of the team's engineers moving forward. At this point, all sub-teams had a well-defined design path with clear expectations of the performance of their specific region as well as the behavior of the car once packaging and design integration occurs.

After this, the team further optimized their designs and analyzed their efficacy within the scope of FMEA to identify critical potential failures and mitigations within the design to address these potential failures. The designs that resulted from this analysis stage were reinforced via further engineering calculations and featured detailed test plans to verify their performance on the car during the manufacturing stage. Fully refined Bills of Materials was generated for each subsystem along with an expansive schedule for ME486C that will allow the team to allocate the appropriate number of resources to ensure manufacturing of the vehicle is completed efficiently and with great quality.

As of November 28th, the team has a nearly finalized CAD assembly of the whole vehicle completed and has begun work on the second prototyping stage as well as final design optimization to meet the December 1st Final CAD and BOM deadline.

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1 Background

The purpose of this chapter is to provide a high-level overview of the SAE Baja ’24 capstone project. A general project description will be presented, followed by the main academic and competition deliverables that the team will be responsible for completing throughout the design cycle. Finally, details concerning project success will be discussed along with relevant metrics to assess design performance at the conclusion of the year.

1.1 Project Description

The SAE Baja competition is a collegiate design series hosted by the Society of Automotive Engineers in which students are tasked with designing and building a single-seat, all-terrain vehicle. This year’s team consists of 13 members overall, with smaller sub-teams of 3-4 members being created. These 4 sub-teams are each responsible for designing specific regions of the car:

- Front End: front suspension & geometry, front brakes, steering
- Rear End: rear suspension & geometry
- Drivetrain: engine, 4WD power transmission, rear brakes
- Frame: frame construction & validation, ergonomics, safety

The team currently has 7 sponsors that have pledged various means of support to the team:

- W.L. Gore: Cash grant with a value of $4750
- Monster Energy: Variable supply of energy drinks depending on availability
- Industrial Metal Supply (IMS): 4130 steel tubing for primary frame members and control arms
- Nova Kinetics: Tig welding and carbon lay-up resources
- Vroom: Free material and laser cutting
- TMS: Donation of Titanium stock and hardware
- Cognito: Monetary donation

These companies are providing invaluable support to the SAE Baja capstone team, but the team will need to secure additional funding to facilitate a successful project. Team budget liaisons have begun the formation of fundraising platforms to accept donations from personal and corporate accounts intending to fund the team directly. In addition, companies including Copper State, Mother Road, Findlay Toyota, HAAS, ETM, Wilwood, and Bass Pro Shops will be contacted for sponsorship in the form of hardware, raw materials, or cash donations by the semester’s end.

Excluding sponsorships, the team has developed a budget that aligns with the expectations of previous SEA Baja capstone teams at NAU. This capstone team requires a larger budget compared to other capstone teams due to the sheer complexity of the tasks being completed. With only a small amount of the budget being covered by initial sponsorships, the team will need to put in sizeable amounts of fundraising effort to ensure that the project stays supported. The sponsorships will be useful in raising money and resources, as will other means of cash flow such as familial/corporate donations and student out-reach events. The full budget can be seen listed below with a breakdown provided by individual sub-teams in Table 1 as well as Section 5.2

Table 1: SAE Baja '24 Team Budget

	Category	Relevant Items	Approximated Cost	Sub-Total
Front	Vehicle Expenses	Brake System Control Arms Rod Ends/Ball Joints Knuckle Material	$2,649.00	$4,674.00
	Spare Parts	Hardware Welding Supplies Bushings	$500.00	
	Competition Expenses	Registration Travel	$1,125.00	
	Contingency (5%)	Unpredicted Expenses	$400.00	
Rear	Vehicle Expenses	Suspension System Drive System	$1,260.00	$2,840.00
	Spare Parts	Camber Links Rod Ends CV Axles	$320.00	
	Competition Expenses	Registration Travel	$1,125.00	
	Contingency (5%)	Unpredicted Expenses	$135.00	
Drive	Vehicle Expenses	Motor Front Gearbox Rear Gearbox ECVT 4WD	$6,359.00	$8,284.00
	Spare Parts	Hardware Gears	$500.00	
	Competition Expenses	Registration Travel	$1,125.00	
	Contingency (5%)	Unpredicted Expenses	$300.00	
Frame	Vehicle Expenses	Frame Materials Paneling Safety Equipment Hardware	$1,041.00	$2,466.00
	Spare Parts	Hardware Tab Materials Tubing	$200.00	
	Competition Expenses	Registration Travel	$1,125.00	
	Contingency (5%)	Unpredicted Expenses	$100.00	
			Total	**$18,264.00**

Aside from the flashy appeal of building a fully functioning race car, this project is important for a variety of more logical reasons. First, successful performance will put NAU on the map as a strong engineering school that can compete with some of the larger and more established SAE Baja programs around the nation. This capstone project will also help strengthen internal Baja knowledge at NAU through rigorous calculation, documentation, and involvement of underclassmen. Lastly, the industry connections made via the team's budget liaisons will establish long-lasting connections with companies that will assist NAU's SAE Baja program for years to come by providing financial and material support.

1.2 Deliverables

This capstone is being carried out in accordance with NAU course requirements as well as SAE Baja design competition requirements. Both sets of deliverables have different due dates and, as such, will be presented separately.

The deliverables that are associated with NAU's capstone requirements (ME476C/ME486C) are mainly based around the technical documentation of the design process from the infant stages of the project. Two initial presentations will be completed that are intended to inform the other students in the capstone program about the design competition, establish foundational understanding for the design's goals, and introduce basic concepts that will be relevant to the vehicle's design process. Following that, the sub-teams will be responsible for presenting prototypes of their systems that represent a low-fidelity model of their region of the vehicle. These prototypes will feature both a physical representation and a virtual representation that addresses any questions the team may be looking to solve in that stage of the design process with regards to fabrication, CAM tool paths, etc. A tentative schedule was provided by Dr. Willy to summarize all these ME476C capstone requirements (Table 2).

Table 2: ME476C Tentative Schedule

Week	Week Starts	Agenda	Individual Assignments	Team Assignments
1	28-Aug	Lecture: Introduction to Capstone	**HW00 & HW01**	
2	4-Sep	Staff/Team Meetings*	**HW02**	**Team Charter**
3	11-Sep	Staff/Team Meetings		
4	18-Sep	**Presentation 1**	**Peer Eval 1**	
5	25-Sep	Staff/Team Meetings		
6	2-Oct	Staff/Team Meetings	**HW03**	
7	9-Oct	**Presentation 2**	**Peer Eval 2**	
8	16-Oct	Staff/Team Meetings		**Report #1**
9	23-Oct	Staff/Team Meetings		**Website Check #1**
10	30-Oct	Staff/Team Meetings		**Analytical Analysis Memo**
11	6-Nov	**Presentation 3 and 1st Prototype****	**Peer Eval 3**	
12	13-Nov	Staff/Team Meetings		
13	20-Nov	Staff/Team Meetings***		**Report #2**
14	27-Nov	Staff/Team Meetings	**HW04**	**Final CAD/BOM**
15	4-Dec	**2nd Prototype Demo**		**Project Management**
Finals	11-Dec		**Final Peer Eval**	**Website check #2**

The deliverables that are associated with SAE's Baja competition deliverables must be completed as well if the team wishes to be able to compete in the 2024 competition in Gorman, California. The first competition deliverable is a request for proposal (RFP) in which the team must justify the novelty of the design, its feasibility in terms of design/fabrication, and marketing/sales plans. Next, the frame sub-team must supply documentation surrounding the construction of the vehicle's frame with material invoice(s), material certification(s), and calculations for all primary members of the frame. The vehicle will need to be analyzed for proper cost reduction practices in its design, as well as preparing a full cost prototyping report the discuss the material and fabrication cost of the vehicle during its manufacturing as well as within the scope of Lean/Six Sigma manufacturing principles. Finally, the team must prepare an extensive design review briefing (DRB) that will be presented to a board of judges during the competition. This DRB will contain information about all 4 sub-team's designs on the car as well as thorough justification for said designs. Like the ME476C deliverables, a schedule for SAE Baja competition deliverables is provided below in Table 3.

Table 3: SAE Baja Competition Deliverables

Deadline	Competition	Due Date**
Presentation - Business RPF * [R]	National	12/20/2023 11:59:59 PM
Tech - Roll Cage Documentation Package * [R]	National	2/1/2024 11:59:59 PM
Cost - Written Cost Reduction Report * [R]	National	3/4/2024 11:59:59 PM
Presentation - Presenter Form * [R]	National	3/11/2024 11:59:59 PM
Online Cost Prototype * [R]	National	3/18/2024 11:59:59 PM
Design - Design Review Briefing * [R]	Baja SAE California	3/25/2024 11:59:59 PM
Design - Design Review Briefing * [R]	Baja SAE Williamsport	4/8/2024 11:59:59 PM
Design - Design Review Briefing * [R]	Baja SAE Michigan	8/12/2024 11:59:59 PM

1.3 Success Metrics

For this project to be considered a success, the team's vehicle will be assessed against a variety of metrics at the end of the semester and academic year. The most basic requirements for success are the construction of a Baja car that will pass technical inspection at competition and full completion of all capstone/competition deliverables presented in Section 1.2

Setting design success aside, another important area to assess is the cohesivity and productivity of the team's 13 members throughout the semester and academic year. All member's final designs will be required to be formulated using numerical testing and results rather than opinions or blind assumptions. In addition, the manufacturing of the car's components will be kept in-house as much as possible to reduce costs and minimize lead times. Lastly, this capstone team will not meet the demanding success metrics of this project without a well-posed schedule. The generation and observance of a schedule, in the form of a Gantt chart, will ensure the efforts of this group are always directed in an efficient manner and that all deliverables will be executed on time. A simple example of a work breakdown structure (WBS) that the team might follow is presented below in Table 4, with a completed Gantt chart in Appendix A: Project Management. If the team can adhere to these operational baselines, the project will be deemed a success with regards to internal team performance.

Table 4: WBS Example

Milestone description	Responsible Sub-Team	Assigned To	Progress	Start	Days
Frame Completion in CAD	Milestone			10/31/2023	1
Presentation 1	All Team	N/A	**100%**	9/12/2023	7
Major Sub-System Decisions	All Team	Team Leads	**100%**	9/15/2023	7
Wire Frame	Frame	Cooper Lead	**80%**	9/12/2023	15
Define Front Suspension Points & Begin CAD	Front	Bryce Lead	**100%**	9/20/2023	7
Define Rear Suspension Points & Begin CAD	Rear	Seth	**75%**	9/20/2023	7
Define Drivetrain Points & Begin CAD	Drivetrain	Henry Lead	**100%**	9/20/2023	7
Measure Hailey & Design Rollcage	Frame	Cooper Lead	**100%**	9/18/2023	10
Concept Generation & Selection	All Team	Team Leads	**100%**	9/26/2023	11
Presentation 2	All Team	N/A	**100%**	10/3/2023	7
Packaging Integration (Wheelbase, car length, etc.)	All Team	Cooper & Henry Lead	**90%**	10/3/2023	15
Report 1 & Webiste 1	All Team	Seth	**30%**	10/20/2023	8
Finalize Frame (footbox, lower rear triangle, rollcage)	Frame	Cooper Lead	**60%**	10/11/2023	22

2 Requirements

This chapter provides an in-depth view of the different types of requirements for the project. These requirements are either set by the SAE as rules for the competition structure or given by the client, in this case, Prof, David Willy and our capstone team. The requirements are separated into two groups, those being customer requirements and engineering requirements. Customer requirements relate to general performance metrics and may or may not be numerically quantifiable. Engineering requirements are the specific, quantifiable requirements, that govern any design decisions for the project. Finally, the house of quality (QFD) provides a comparison between the types of requirements and provides a value of importance to each.

2.1 Customer Requirements (CRs)

2.1.1 Front End

The front end of the vehicle only has a single strict requirement established in the rulebook. As such, many of these CRs are inferred based on desirable vehicle attributes and from extensive benchmarking research (see Section 3.1.1).

- *Vehicle must comply with the dimensions of the SAE Baja course
- Vehicle must have adequate ground clearance
- Vehicle must have adequate traction across all terrains
- Vehicle must be capable of safe operation over rough land terrain
- Vehicle must have agile maneuverability
- Front suspension components must be robust in design (i.e. control arms, hubs, knuckles, tie rods, etc.)

These CRs must be satisfied to design a successful vehicle that will perform well in an SAE Baja competition. Metrics that will evaluate satisfactory design performance are presented in Section 2.2.1

2.1.2 Rear End

The rear end has the least strict requirements when it comes to maintaining a safe and durable vehicle that complies with the SAE rules. The customer requirements come from research from successful teams and conversations with other teams make sure our design is optimized. Some of the areas the team is focusing on in these discussions are:

- Tunability
- Serviceability
- Reliability
- Ease of manufacturing
- Low cost
- Maximum traction
- Maneuverability

These CRs will help keep the team focused on what areas need to be at the forefront of the discussion when talking about the design.

2.1.3 Drivetrain

The drivetrain is composed of several subsystems that will be capable of handling the challenging obstacle courses. To have a fully functioning drivetrain, there are customer requirements that will guide the team in designing a drivetrain. The goal of the drivetrain is to be the most efficient and robust, which is a critical part in designing a competitive SAE Baja vehicle. A list of CR's is stated below:

- High top speed
- Maximum efficiency
- High torque
- High service life
- Low weight
- High transmission range

These CR's will help drive the team in designing a successful drivetrain and equally compete with the other top SAE Baja vehicles.

2.1.4 Frame

The frame is the basic platform for which all other subsystems are integrated onto. For that reason, correct and intentional frame design are vital to the team's success. Attributes which make a frame successful, and therefore competitive, are straightforward. Many of these CR's reflect these attributes and have been carefully considered through extensive frame research of both competing schools' frame designs and previous successful NAU BAJA teams' frame designs. The CR's are as follows:

- Frame must satisfy SAE BAJA Rules
- Frame must be designed for manufacturing
- Frame must be rigid
- Frame must be lightweight
- Frame must be maneuverable
- Frame must be aesthetically pleasing
- Frame must be durable
- Vehicle must be fast
- Frame must be stable
- Frame must be cost-effective

2.2 *Engineering Requirements (ERs)*

2.2.1 Front End

Referring to the qualitative front end customer requirements presented in Section 2.1.1 2.1.1

- Decrease Vehicle Width
  - Max Vehicle Width = 64”
- Increase Ride Height
  - Front Ride Height Minimum = 10”
- Increase Tire Traction
  - Scrub Radius = ~0 degrees
- Increase Capability in Rough Terrain
  - Wheel Travel = ~12” total (3:1 bump to droop)
- Increase Turn-In Angle
  - Pro-Ackerman = 40-100%
- Increase Crash Durability
  - Max Survivable Collision Speed = 40 mph

Engineering requirements allow the front end to guide their geometric design and better communicate design desires with other sub-teams during integration phases. Many of these engineering requirements must serve dual purposes: meeting the engineering requirement and satisfying SAE BAJA rules and regulations.

2.2.2 Rear End

Working with the customer requirements from section 2.1.2 , the team established some quantitative measurements that would be able to highlight the customer needs. These will be critical to the teams’ goals moving throughout this project. These technical requirements are listed below:

- Decrease weight (lbs.)
  - Rear suspension system under 50 lbs.
- Increase strength (psi)
- Increase rearward axle path (in.)
  - 1 in. or rearward movement
- Increase linkage radii (in.)
  - 22 in. camber links
- Increase ground clearance (in.)
  - 11 in. of ground clearance
- Vehicle width (in.)
  - Maximum vehicle width of 64 in.

- Decrease CV axle angle (degrees)
    - 180 degrees

These gave the team a better understanding of some of the areas to have in the back of the mind when coming up with designs. As before these mainly came from the SAE rulebook, research from successful schools, and team discussions when integration with other sub-teams present.

2.2.3 Drivetrain

Regarding the customer requirements from section 2.1.3, the engineering requirements will influence different aspects of the drivetrain design. It is crucial that abiding by these engineering requirements will reflect the performance of the drivetrain. The list of engineering requirements is stated below:

- 40mph top speed
- 80% drivetrain efficiency
- 400lb-ft of torque to the wheels
- 1000-hour service life
- Total drivetrain weight (without engine) 60lbs
- 1:4.5 total transmission range

Using these constraints will help the team better understand the design of the drivetrain and the important aspects of meeting the SAE Baja rules and adhering to the customer requirements as well.

2.2.4 Frame

Directly corresponding to the frame customer requirements of 2.1.4, the engineering requirements quantify these qualitative requirements. Many of these engineering requirements are also driven by the SAE BAJA Rules in addition to customer requirements.

- Decrease Weight
    - Minimize number of primary and secondary members
        - Primary: ~ 30
        - Secondary: ~ 36
- Decrease Body Length
    - Maximum Wheelbase = 64 inches
- Decrease Body Width
    - Maximum Body Width = 64 inches
- Decrease Cost
    - Cost of 4130 CD Steel
        - Primary 1.25” OD x 0.065” thickness
        - Secondary: 1.00” OD x 0.035” thickness
- Increase Strength of Frame

- Optimize Yield Strength

2.3 *House of Quality (HoQ)*

2.3.1 Front End

The CRs and ERs that the front end will be working with throughout the design cycle have a variety of interaction effects and, as such, must be analyzed relative to each other as well as the design success of the car overall. The front end QFD (see Table 5) helped to quantify these interaction effects and allowed the ERs to be ranked in order of relative importance. The optimal targeted design by the front end team this year was also assessed against NAU's Baja car from last year as well as against two ultra-competitive universities that are known for their Baja program (ETS and Cornell). This benchmarking process is covered in more detail in Section 3.1.1

Table 5: Front End QFD

System QFD

Project:	SAE Baja '24
Date:	09/18/2023

#		1	2	3	4	5	6
1	Decrease Vehicle Width						
2	Increase Ride Height						
3	Increase Tire Traction	-3					
4	Increase Capability in Rough Terrain	3	9	6			
5	Increase Turn-In Angle				3		
6	Increase Crash Durability	6	-3		6		

Legend	
A	NAU #74
B	Baja ETS
C	Cornell Racing

#	Customer Needs	Customer Weights	Decrease Vehicle Width	Increase Ride Height	Increase Tire Traction	Increase Capability in Rough Terrain	Increase Turn-In Angle	Increase Crash Durability	1 Poor	2	3 Acceptable	4	5 Excellent
1	Comply with track dimensions	4	9									A	BC
2	Adequate ground clearance	2		9	6	9		3			A	C	B
3	Adequate traction	3	3	3	9	6	3	3			A		BC
4	Safe operation over rough terrain	3	6	6	3	9		9				ABC	
5	Agile manuverability	4	6	3	6	3	9					A	BC
11	Robust design	3		3		3		9			BC	A	
	Technical Requirement Units		Inches	Inches	Degrees (Scrub Rad)	Inches (Wheel Travel)	Degrees	mph					
	Technical Requirement Targets		64	10	0	12	40-100	40					
	Absolute Technical Importance		87	66	72	84	45	69					
	Relative Technical Importance		1	5	3	2	6	4					

The benchmarking process revealed that most of the top universities focus more on high speed, low safety designs that push the limits of the materials used during construction. NAU has traditionally gone for more robust vehicles that sacrifice other elements of performance for strength and durability. These two competing ideologies will be kept in mind throughout the front end's decision-making processes with regards to vehicle design.

The QFD revealed that several of the CRs and ERs work towards the same goal (ratings of 6 and 9) while other tend to work weakly or inversely with each other (ratings of 3 or -3). These interaction effects, along with the relative importance of customer needs, were quantified and summated at the bottom of the QFD to deliver a relative technical importance to each ER. These technical rankings are seen below:

1) Decrease Vehicle Width
2) Increase Capability in Rough Terrain
3) Increase Tire Traction
4) Increase Crash Durability
5) Increase Ride Height
6) Increase Turn-In Angle

The relative technical importance of each ER, in conjunction with the metrics established in Section 2.2.1

2.3.2 Rear End

After the customer requirements and engineering requirements were solidified, the team correlated these to each other. Table 6 helps to illustrate how the requirements interact with each other. The team conducted a benchmark of the teams involved in the competition from high-ranking universities. This benchmarking is covered in more detail in more detail in Section 3.1.2.

Table 6: Rear End QFD

System QFD

Project:	Lumberjack Motorsports SAE Baja Rear Suspension
Date:	9/19/23
	Input areas are in yellow

Decrease weight							
Increase strength	-3						
increase rearward axle path							
Increase linkage radii	-1						
Increase ground clearance	-6	3	1				
Vehicle width	1	3		6	2		
Decrease CV axle angle		3	-2	1			

Legend	
A	CWRUM
B	RIOT Baja
C	TS BAJA

		Technical Requirements							Customer Opinion Survey				
Customer Requirements	Customer Weights	Decrease weight	Increase strength	increase rearward axle path	Increase linkage radii	Increase ground clearance	Vehicle width	Decrease CV axle angle	1 Poor	2	3 Acceptable	4	5 Excellent
Tunability	2	2	3	8	7	3	2	7	C	A			B
Servicability	2	2	6						A		BC		
Reliability	5	3	9	3	2			7			A	B	C
Ease of manufacturing	3	6	7	1	1		1	1		C	A		B
Low cost	5	9	9	3	3		1	3			A	C	B
Maximum Traction	2	7		8	8		3	1				B	AC
Maneuverability	4	5	1	8	6	5	7	1				AB	C
Technical Requirement Units		lb	Psi	in	in	in	in	degrees					
Technical Requirement Units		<50	NA	<1	22	>8	<64	180					
Absolute Technical Importance		120	133	97	82	26	46	73					
Relative Technical Importance		2	1	3	4	7	6	5					

The outcome of this table was to establish common design focus amongst the team. The customer needs were weighted and then ranked for correlation against the engineering requirements the team came up with after design research and discussion. The engineering requirements were then ranked based on importance by using the absolute technical importance, the rankings are below:

1) Increase strength
2) Decrease weight
3) Increase rearward acle path
4) Increase linkage radii
5) Decrease CV axle angle
6) Vehicle width
7) Increase ground clearance

These rankings will benefit the teams' design decisions moving forward by allowing the team to know what is most important. This also allows for the discussion of sacrifices the team will have to make.

2.3.3 Drivetrain

Based on the customer requirements and engineering requirements gathered, the team can then weigh each design requirement based on its importance. The importance was based on benchmarking different designs from top ranking teams. The benchmarking is discussed in detail in Section 3.1.3.

Table 7: Drivetrain QFD

		1	2	3	4	5	6	7
1	top speed							
2	drivetrain efficiency							
3	torque to the wheels	6						
4	service life							
5	total system weight (w/out engine)		3					
6	total transmission range		6					
7	Meets HROE Guard specifications					-9		

Legend
A Cornell 2023
B NAU 2021 #21
C NAU 2023 #74

			Technical Requirements							Customer Opinion Survey				
	Customer Needs	Customer Weights	top speed	drivetrain efficiency	torque to the wheels	service life	total system weight (w/out engine)	total transmission range	Meets HROE Guard specifications	1 Poor	2	3 Acceptable	4	5 Good
1	fast	5	9	6	6	3	6	6	1	C			B	A
2	High effeciency	3	9	9	6	6	3	6	3	C		B		A
3	fast acceleration	5	3	6	9	3	9	9	1	C		B		A
4	durable	1	1	6	1	9	1	4	9			AC		B
5	can crawl and go fast	4	9	6	9	3	3	9	1	C			B	A
6	NEEDS TO BE SAFE	5	1	1	1	6	1	1	9			C	B	A
7	Aesthetically Pleasing	3	9	1	9	3	9	9	3		C		A	B
8														
9														
10														
11														
	Technical Requirement Units		MPH	Unitles	Lbf/Ft	hours	lbs	Unitless	N/A					
	Technical Requirement Targets		40	80	400	1000	60	01-04.5	N/A					
	Absolute Technical Importance		156	125	162	108	129	165	86					
	Relative Technical Importance		4	6	3	7	5	2	1					

Based on the results, the team has determined that there are certain characteristics that the team must follow to have an effective drivetrain. The customer requirements and engineering requirements were evaluated immensely so that it would reduce any design flaws later in the design process. The top engineering requirements the team found most important are listed below:

1) Top Speed
2) Drivetrain Efficiency
3) Torque to the Wheels
4) Service Life
5) Total System Weight
6) Total Transmission Range
7) Meets HROE Guard Specifications

The rankings will help the team in making informed decisions, reducing the risk of design flaws, and ultimately meet the customer requirements based on the expected outcomes. Also, keeping in mind that efficiency and design quality must be met.

2.3.4 Frame

The customer and engineering requirements of the frame sub-team can have varying interactions, ranging from highly correlating to actively opposing each other. The use of a house of quality is helpful in deciding which requirements should be prioritized in the design process. Another benefit of the QFD is the comparison to other current designs. For the benchmarking process, three high scoring, consistently competitive teams were selected, those being ETS Baja, SAE Beaver Racing, and Cornell Baja Racing. The results of the QFD can be shown below in Table 8.

Table 8: Frame Team QFD

System QFD

Project: **Baja 24 Frame**
Date: 9/14/23

Decrease weight						
Decrease length of body	6					
Decrease width of body	3	-9				
Decrease Cost	-9	3	3			
increase aerodynamics	-3	-6	3	-3		
Increase strength of frame	6	-3	-9	-6		

Legend
A ETS Baja
B SAE Beaver racing
C Cornell Baja Racing

Customer Needs	Customer Weights	Decrease weight	Decrease length of body	Decrease width of body	Decrease Cost	increase aerodynamics	Increase strength of frame	1 Poor	2	3 Acceptable	4	5 Excellent
		Technical Requirements						**Customer Opinion Survey**				
Rigid	3	1	6	3	3	1	9				ABC	
Easy to manufacture	3	3	3	1	3	6	3			B	AC	
Maneuverable	2	3	9	9	1	1	3					ABC
Aesthetics	1	3	1	3	3	9	1			C	B	A
Durable	2	3	1	3	3	3	9			AC	B	
Satisfy SAE Baja Frame Guidlines	4	3	1	6	3	1	6					ABC
Stable	3	1	3	9	1	3	6				C	AB
Fast	3	6	3	3	9	9	3				BC	A
Lightweight	4	9	6	3	9	3	6				ABC	
Affordable	3	9	6	3	9	6	6			ABC		
Technical Requirement Units		lbs	in	in	$	lbf	psi					
Technical Requirement Targets			64	64								
Absolute Technical Importance		123	112	120	134	108	154					
Relative Technical Importance		3	5	4	2	6	1					

Through benchmarking, some of the top schools in the country can be shown to focus on satisfying the SAE guideline, keeping the vehicle maneuverable, and keeping the frame lightweight and rigid. Of these common needs, the team decided that the most important customer needs were to satisfy the SAE competition rules and keep the frame as lightweight as possible while maintaining a strong structure.

Through the evaluation of the QFD, the team was able to compare the engineering requirements to the customer needs, identifying which ERs are the most important to a successful design. This ranking was made by comparing the weight and correlation of each ER to each CR, with a higher combined score relating to a more important engineering requirement. The final ranking of ERs for the frame is as follows:

1) Increased Strength of Frame
2) Decreased Cost
3) Decreased Weight
4) Decrease Width of Body
5) Decreased Length of Body
6) Increased Aerodynamics

This ranking of ERs allows for the frame team to make design choices that may require favoring one over another, leading to the most optimal design.

3 Research Within Your Design Space

To remain competitive at this year's SAE Baja competition, the team must be aware of all current design requirements and metrics for a highly functional vehicle. The section below details the process of design benchmarking, literature review, and mathematical modeling within each sub-team's design space.

3.1 Benchmarking

3.1.1 Front End

During the benchmarking phase, the front-end team decided to focus their research on high-performing suspension and steering geometry. The three systems under analysis are as follows: scrub radius, front shock placement, and steering design.

3.1.1.1 Scrub Radius

The first system under analysis is the scrub radius of the front knuckle/wheel assembly. The scrub radius is defined as the distance between the tire's centerline axis and the axis created by the control arm mounts on the knuckle when these two axes intersect on the ground. The two most common design cases are a positive scrub radius and a zero-scrub radius (see Figure 1).

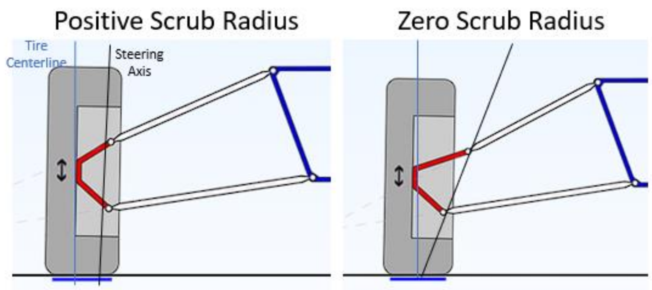


Figure 1: Scrub Radius Definition

An event that is all too common in a Baja competition is a sudden, hard application of the brakes. When this occurs, a force is sent backwards through the contact patch of the tire that is in line with the tire's centerline axis. In the case of a positive scrub radius, this braking force doesn't act in line with the steering axis of the knuckle, causing the generation of a torque. This torque causes the wheels to angle in (also called "toe in"), leading to instability and lack of control for the driver. A way to mitigate this steering influence is by zeroing out the scrub radius; by doing so, all forces generated on the tire will act through the steering axis. Since all forces are kept in line with each other, they don't generate a torque and, thus, no influence on the toe of the vehicle's front wheels is created. For a simple visualization of this effect, see Figure 2.

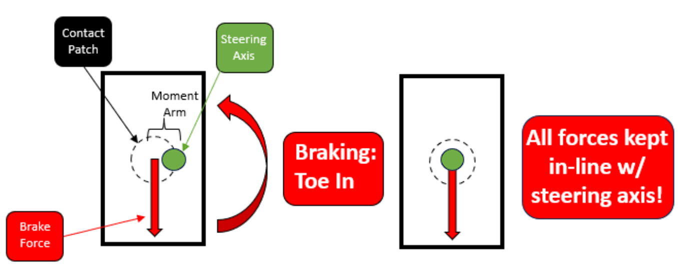


Figure 2: Scrub Radius Impact

For this reason, almost all top teams choose to design for a zero or near-zero scrub radius in their front end assembly. This minimizes the influence on steering and toe characteristics under hard braking (and acceleration) events and leads to better control for the driver. Some high-level teams that adhere to this philosophy are featured below in Figure 3.

Figure 3: Zero Scrub Radius Popularity Amongst Top Teams [101]

3.1.1.2 Front Shock Placement

The front shock mounting position both on the control arm and on the frame are critical for determining suspension characteristics of the vehicle as well as modifying the vehicles center of gravity. In general, there are three separate mounting styles for the front shock: Upper control arm to upper front brace member, upper control arm to lower front brace member, and lower control arm to lower front brace member. Figure 4 shows both of the two frame mounting options highlighted in blue and red.

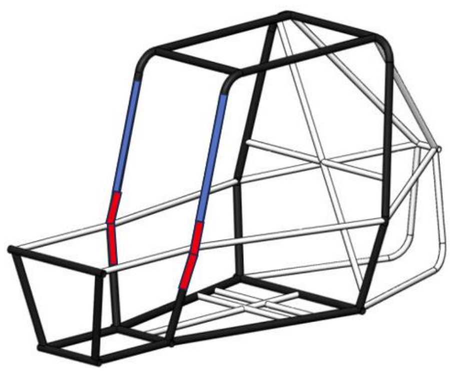


Figure 4: SAE Baja Chassis with Upper Front Brace in Blue and Lower Front Brace in Red

Benchmarking against top performers of the past 6 years we can determine some of the top performing suspension layouts. At the Oregon competition in 2023, ETS won the event overall using a shock mounted to the lower front brace and the upper control arm. At the SAE Ohio competition, The 2023 ETS vehicle can be seen in Figure 5 with the lower front shock placement visible. CWRU took first place also using a front shock mounted between the lower front brace member and the upper control arm. The third benchmark is the 2017 Oshkosh winner UM Ann Arbor who took first place with a shock mounted to the junction between the side impact member and the front brace member.

Figure 5: 2023 ETS SAE Baja Vehicle Utilizing a Lower Front Brace and Upper Control Arm Mount

After completing our benchmarking, we have chosen to move forward mounting the front shock to the upper control arm and the lower front brace member. Utilizing this suspension layout helps to decrease the center of gravity of the vehicle, increases suspension performance by decreasing the ratio between shock travel and wheel travel, and provides more adjustment options once mounted by increasing or decreasing the frame mounting tab lengths.

3.1.1.3 Steering Design

Steering design directly effects the maneuverability of the vehicle. With a maneuverability subcategory at competition, it is vital to optimize the front end so that it performs to competition standards. The three types of steering that the team could move forward with is that of Ackermann, parallel, and Reverse Ackermann steering. Ackermann describes that the inside wheel terns proportionally more than the outside wheel. Reverse Ackermann describes the opposite with the outside wheel turning in more than the inside wheel. Parallel steering describes an identical steering angle from both wheels. A diagram of all three forms of steering is shown below.

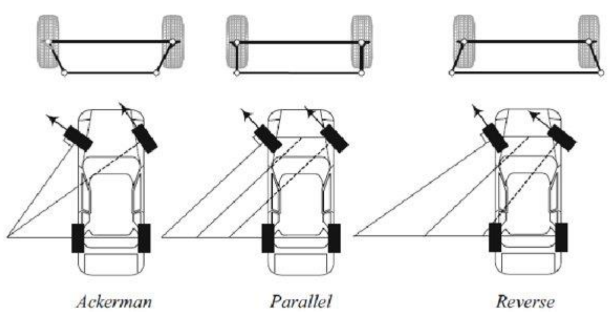


Figure 6 : Three Types of Steering Design

Benchmarking the top three performing teams for maneuverability, the team can analyze what is a typical steering design used in SAE Baja. It was discovered that all three best performing teams in maneuverability at Baja Oregon last year used Ackermann steering. These three teams, ETS Baja, Cornell, and Beaver Racing not only performed first through third in maneuverability but also placed first, third, and second in the overall standings respectively. The figure below displays the use of Ackermann steering on Cornell's

Oregon car. Take notice how the inside wheel is turned noticeably inward more than the outside wheel.

Figure 7 : Cornell's use of Ackermann Steering

Cornell uses a 50% Ackermann ratio.

Ackermann is most useful at very low speeds and tight turns because this is when there is the least wheel slip and load transfer side to side. The performance of Ackermann perfectly mirrors the demands of SAE Baja competitions. Benchmarking against the top teams for the last half a decade in maneuverability, the team has decided to move forward with Ackermann steering. This will help the maneuverability of the vehicle once built and will optimize performance of the car's steering during competition.

3.1.2 Rear End

During the benchmarking and research phase the Rear End team wanted to research different suspension geometries and designs of top performing teams in the competition.

3.1.2.1 Rear Trailing Link System

When looking at the different designs, one that stood out was the trailing arm suspension. This design allows for rearward axle travel. Which allows a little "give" as the wheel is hitting bumps. Three colleges were analyzed, these schools were Rochester Institute of Technology, Louisiana State University, and Johns Hopkins.

Figure 8: Examples of Rear Trailing Link Suspension Systems

Pictured in the left is RIT's vehicle, this team consistently places amongst the very top of the competition. An interesting design aspect they achieve is that the rear camber links allow for negative camber gain to take place as the shock compresses. The takeaway from analyzing LSU's vehicle is that lowering the trailing arm placement increases ride height however it decreases ride if the team is not careful with placement of camber links. Pictured on the right is Johns Hopkins' design, this design utilized the benefits of having a sway bar. The team thought this was interesting however could be hurting the vehicle in a few of the competitions, such as the rock crawl event.

3.1.2.2 *Rear Double A-Arm System*

Analyzing the previous year's BAJA team standings at 3 locations, we noticed a trend of top teams typically utilizing a double a-arm rear suspension. The three schools that were looked at were ETS, Oregon State, and Cornell. ETS used a "Cheater Caster Link", as shown in the 1^{st} photo in Figure 5, to allow for adjustment of caster in the rear wheels. One flaw that the double a-arm setup has is that it lacks adjustability of caster angle. This design modification mitigates that problem. ETS placed 3^{rd} in suspension and 5^{th} overall, proving this design to be worthy of consideration. Oregon State used angled frame mounts, like ETS, but without the additional link. This design, as shown in the 2^{nd} photo in Figure 9, allows for camber gain during suspension compression, allowing for the vehicle to experience increased traction under hard loading corners. This theory will be expanded on in section 3.2.3.3. This design earned Oregon 3^{rd} in maneuverability and 2^{nd} overall. Cornell University utilized vertical frame mounts, as shown in the 3^{rd} photo in Figure 9, which earned the team 2^{nd} in maneuverability and 3^{rd} overall. This design doesn't allow for negative camber gain throughout the travel, rather it results in positive camber gain which makes for lazier cornering capability. The team did, however, mount their upper arm closer to their lower, which, if done correctly, can replicate the camber gain theory.

Figure 9: Examples of Rear Double A-Arm Suspension Systems

3.1.2.3 *Rear Single H-Arm System*

The rear single H-arm suspension system is not the most popular at the SAE Baja competition but when executed properly it can be a super effective system. This suspension system only uses one control arm with a u-joint drive shaft to maintain the structure and location of the wheel. This gives the car increased ground clearance while also dropping the overall weight. Some schools that have been successful with this design are Northern Arizona, Michigan, and Case Western. Looking at these team's previous cars gave us a good idea on how to make the system work properly and what would need to be considered if we chose to move forward with this design.

Figure 10: Examples of Rear Single H-Arm Suspension System

3.1.3 Drivetrain

3.1.3.1 Total Gear Ratio Change

Having a transmission that has a wide range of possible gear ratios is important as it allows for high torque at low speed and a high-top speed. The two options are using a gaged cvt as we have in the past or designing a new geometry and selecting a new belt. The gaged cvt has a range of .9 to 3.9, while a custom cvt could have a range as large as .5 to 4.2, unlocking much more low-end torque and top speed. Having such a large range is unnecessary however, as we are limited on the max usable torque by tire friction and limited on top speed by aerodynamics. We have settled on a custom cvt transmission that's range is .5 to 3.8, as it provides more than necessary torque and more than necessary top speed, but just marginally, so that it is as small as possible, as an ultra-wide range transmission is heavier.

3.1.3.2 Rear End Optimization

For the rear gearbox, there are three different options for how to transfer power from the eCVT to the rear wheels. These options are chain drive, belt drive, and a gearbox. Belt drives are not common in the competition, if they are ever used, due to their various downsides compared to the very similar chain drive option. Gearboxes are very widely used in the Baja competition, particularly by top teams at the competition. Chain drive can also commonly be found, but these designs are significantly less compact than gearboxes, which results in less optimal rear suspension geometry thus reducing the capability of the vehicle. Chain drives also require a different set of maintenance steps, which can be more taxing than a gearbox which is sealed in an oil bath and should rarely need to be altered. The team decided to move forward with the gearbox design because we are more than capable of designing and manufacturing an effective gearbox that will outperform a chain drive power transmission. This rear gearbox will have integrated CV cups and an integrate braking system onto the housing to further cut down on the space that the gearbox will take up in the rear end which optimizes rear end geometry for suspension.

3.1.3.3 Front End Optimization

For the front gearbox, there are two types of axles joints the gearbox can transmit power to its wheels, constant velocity (CV) joint or universal (U) joint. The difference is that U joint axles would be easier to manufacture since it contains a bearing that is pressed between two yokes. The downside of a U joint is its limited range of motion and its significantly more susceptible to binding. With CV joints, it can withstand higher loads and has a larger range of motion due to its construction of using ball bearings. Figure 11 shows a comparison of a CV joint and U joint. The team is determined to run CV joints because every competitor in SAE Baja is using CV joints due to its high-performance capabilities.

Figure 11: CV Joint and U Joint Comparison

3.1.4 Frame

Due to the intense focus on driver safety during competition, SAE has outlined a strict set of rules that pertain specifically to the frame and roll cage envelope of the driver. These restrictions severely limit the possibility for unique design within the sub team. There are some allowances, one of which is the bracing style, which was focused on for the frame team benchmarking.

3.1.4.1 Bracing Orientation

Figure 12: Front (left) vs Rear (right) Frame Bracing

The main two frame styles allowed in competition are front braced and rear braced roll cages, shown in Figure 12. The orientation refers to the location of the secondary bracing material and where it is in reference to the driver. Throughout the benchmarking process, it was found that while front brace designs used to be very popular, they have been much less frequently used by winning teams in the past few years. This shift is likely caused the weight advantages attributed to rear braced frames due to the switch to a larger fuel tank specification by SAE.

3.2 *Literature Review*

For this capstone project, students are required to develop a large depth of knowledge in their specific sub-team's concentration. As such, all members of the team have completed a thorough literature review of textbooks, papers, and online sources that will be relevant throughout the duration of the design cycle.

3.2.1 Front End

3.2.1.1 Abraham Plis

- Suspension Geometry and Computation [3]:
  - Chapter 12 of this textbook contains information and equations relating to the geometric design of double A-arm suspension systems. Items such as relative control arm angles, side-view swing arm angles (SVSA), and instantaneous center (IC) calculations are discussed. Since the front end will feature geometry based around these concepts, the information presented here will be invaluable.
- The Automotive Chassis: Engineering Principles [4]:
  - Chapter 1 of this textbook presents different types of suspension and drive arrangements that generalized automotive designs use. The largest takeaway from this reference is the calculation of anti-dive geometry in the vehicle's front end. When implemented correctly, anti-dive helps reduce the inclination of the Baja vehicle to pitch forward under hard braking. This improves driver control and feedback, making it an important reference for consideration by the front end team.
- Analysis of Steering Knuckle of All Terrain Vehicles (ATV) Using Finite Element Analysis [5]:
  - This research paper details the design methodology and validation of an SAE Baja steering knuckle. There is useful information on designing a knuckle against specific expected forces and how to perform a stress/deformation analysis via FEA to validate the design of the knuckle itself. This will be extremely helpful in the early design stages of the front steering knuckle for this year's car.
- Design and Development of Front Suspension System for an Off-Road Vehicle [6]:
  - This research paper contains information on performing design calculations and finite element analysis (FEA) on a control arm. There is information on advanced suspension attributes such as natural frequency, ride rate, and motion ratio that will be helpful during the front end's suspension deep dive. There is also a thorough discussion of anti-dive calculation and design that is backed up with graphical results performed in Lotus *Shark*, the same software this year's team will be using.
- Design Review of Suspension Assembly of a Baja ATV [7]:
  - This research paper presents a detailed design review of an SAE Baja vehicle from a university in India. The authors walk through their design methodology and their expected outcomes at the beginning of the design cycle and document their hiccups and workarounds throughout the year. The most crucial section of information pertains to the usage of Lotus *Shark*; this paper does an excellent job at providing a rational for certain decisions inside of the software that will be of great use to the front-end team.
- Suspension and Steering Geometry (Front) | Double Wishbone | Anti-Ackerman | SAE Baja | SolidWorks [8]:

- This online resource is a YouTube tutorial that guides the viewer through the modeling of a control arm in SolidWorks. Between this reference and the SolidWorks *Weldments* feature, the construction of the A-arms utilized in the front end should be a quick and efficient process.

- Steering Knuckle | SolidWorks | 3D – Modeling | Baja ATV [9]:
    - This online resource is another YouTube tutorial that gives the viewer recommendations for the design and modeling of an SAE Baja steering knuckle in SolidWorks. Since the geometry of the knuckle can be quite tricky, this reference will help guide the team during the modeling phase and ensure the knuckle is designed for manufacturability.
- Using Structural Bolts for Structural Bolting [109]:
    - This online resource helped the front end team determine the correct specifications to assess the integrity of bolted members of the front end assembly.
- Finding Shear Strength from Tensile Strength [110]:
    - This online resource helped transfer the generic tensile yield strength of a grade 8 steel bolt to a usable yield strength in shear-based loading for bolt strength calculations.
- Bolt Shear Strength – Bearing, Tear Out, and Shear Load Capacity Calculations [111]:
    - This resource provided the equations necessary to evaluate the performance of various diameter bolts for use in the control arm pivots in the front end. Items such as dynamic loading coefficients and double shear area were discussed that directly applied to the team's loading scenario.

3.2.1.2 Bryce Fennell

- Optimal Design of Suspension System of Four-Wheel Drive Baja Racing [34]
    - This online paper offered information on how to develop a suspension system for offroad racing use utilizing Ackerman steering, bump and droop suspension travel, and double A control arm geometry. This resource offered information on numbers to shoot for when considering Ackerman percentage, toe angle, and caster angle.
- Fine-Tuning of the Suspension System of Baja ATV [33]
    - This published paper details suspension tuning goals with figures for tuning the suspension of any offroad vehicle. This paper offered methods for the final stages of suspension tuning after the main suspension design is settled on. After the team determined double A arms will be used, this paper detailed how the control arm mounts for both the vehicle and knuckle change the suspension dynamics of the system as a whole.
- Redesigning the Cooper Union SAE Mini-Baja Front Suspension and Steering [35]
    - This graduate paper shared information gathered from the Cooper Union SAE baja vehicle and discussed design decisions the team made such as double A arm control arm geometry, how they calculated and optimized for steering angle, and how to calculate suspension travel desired for their vehicle. This information was critical for helping the front team determine the positioning and travel of the steering rack to develop Ackermann steering with a 7-foot turning radius.
- Baja SAE, SAE International, 2023 [32]

  - Detailed competition rules from the Society of Automotive Engineers (SAE). This rulebook did not feature rules directly restricting the front suspension designs; however, the rules pertaining to vehicle width and recommended ride heights do change suspension designs and were considered while designing the front end of the vehicle.

- Tutorial on Lotus Suspension Software [31]
  - This YouTube tutorial detailed how to input your vehicles hardpoints into the Lotus Shark software for suspension simulation and optimization. The tutorial also details how to recover data from the program from graphs and tables, how to interpret this data, and how to iterate upon this design.
- Lotus Shark Suspension | Tutorial [30]
  - This was a basic YouTube tutorial of the Lotus Shark suspension simulation software. This tutorial covered how to add your vehicle into the software and ensure the data being recovered from the simulations are accurate. This software also covered steering dynamics within the program and how to input an accurate steering rack to ensure manufacturability during the construction phase.
- Suspension Geometry and Computation [36]
  - Chapter 7 discusses the effects of camber angle and scrub radius on vehicle handling dynamics as the wheel moves throughout its travel. The textbook discusses an optimal suspension design with zero scrub radius and a very slight camber angle which remain constant as the wheel moves throughout the suspension travel.
  - Chapter 12 discusses the Double A arm suspension design including the benefits and drawbacks. The book discusses how to integrate the double A arm design into your vehicle with steering, bump, and droop dynamics and calculations included.
- Road and Offroad Vehicle Dynamics [58]
  - Pg. 379-442 discuss the characteristics of suspension and what each measurement discusses. Two particularly important characteristics for our purposes are toe angle throughout suspension travel and tire scrub radius. Understanding each suspension characteristic helps the front suspension team to optimize the front end of the vehicle.

3.2.1.3 Evan Kamp

- Vehicle Dynamics: Theory and Application [23]
  - Chapter 7, titled Steering Dynamics includes Calculations for Turning Radius and Viable Steering Angles for basic CV axle designs. Using these calculations, a preliminary estimate of turning radius can be made by picking realistic steering angles and an estimate of the vehicle’s center of gravity.
- The Science of Vehicle Dynamics Handling, Braking, and Ride of Road and Race Cars [77]
  - Chapter 5, titled The Kinematics of Cornering explains the kinematic performance of steering. This changes how the car performs under steering and greatly affects the steering column and how comfortable the car is for the driver. Steering comfort was an issue brought up from last year and is something we hope to rectify.
- Analysis of Ackermann Steering Geometry [74]
  - This paper describes the benefits and drawbacks of Ackermann steering geometry. Written and published by the Society of Automotive Engineers, it talks about Ackermann’s

application within SAE Baja. This journal was the main source referenced when deciding to move forward with Ackermann steering.

- Steering System for SAE Baja [46]
    - This journal compared the use of Ackermann steering and Parallel steering. Comparing turning radius and performance calculations based on identical geometry, Ackermann vastly outmatches the use of Parallel in the application of SAE Baja. This journal also outlined Ackermann angles and percentages that the team would later benchmark as goals for its own design.
- Design and Optimization of Steering Assembly for Baja ATV Vehicle [12]
    - This journal outlined knuckle design accounting for Ackermann steering. This resource was helpful when later using Lotus Shark and changing knuckle geometry. This resource will be used once again when accounting for knuckle machining.
- Tech Explained: Ackermann Steering Geometry [59]
    - This online resource outlined viable Ackermann angles as a function of slip angle and lateral force. It is important to account for these two variables when picking Ackermann steering angles in order to maximize performance from the system.
- Baja Virtual Presentation Series [79]
    - Day 8's presentation on steering calculations was very helpful when making preliminary steering calculations. This helped the team be able to make preliminary radius calculations in order to best optimize the geometry of the front end.
- Shigley's Mechanical Engineering Design [22]
    - Machinery's Handbook [25]. Explanation on how the machining can be done for components of the front suspension. This is vital for the construction of components such as the knuckle.
- McMaster-Carr [21]
    - This was used to find the hardware that was able to be used when constructing the front end. This is important when choosing hardware for construction. This was also used for solidworks models that could be used within the assembly.
- Materials selection in mechanical design [18]
    - Chapters 5 and 6 gave helpful incites in using aluminum in a gearbox. This was ultimately passed up for steel for the design of the rack within the rack and pinion system.

3.2.2 Rear End

3.2.2.1 Joey Barta

- W. F. Milliken and D. L. Milliken, Race Car Vehicle Dynamics [48]
    - This SAE written textbook is highly regarded as the "bible" of suspension engineering. It was recommended by previous members of the NAU BAJA and Formula teams, a member of Cornell's BAJA team, as well as online forums. The authors developed many of the vehicle dynamics theories in the book.
- R. G. Budynas, Shigley's Mechanical Engineering Design [22]
    - This textbook provides useful theories and formulas for failure prevention as well as design

for mechanical elements.

- J. C. Dixon, Tires, Suspension and Handling [49]
  - Also SAE certified, this textbook provides detailed coverage of the theory and practice of vehicle cornering and handling. The book includes classical equations to back the theories.
- Suspension Types – SUSPROG [50]
  - This online resource illustrates potential rear suspensions with downloadable excel files pertaining to each. It serves as a useful source for the early stages of suspension design.
- J. Isaac-Lowry, "Suspension Design: Types of Suspensions," [51]
  - This online resource provides a short list of applicable designs to reference in the early stages of design.
- SLASIM: Suspension Analysis Program [52]
  - This online resource references software through MATLAB that analyzes functionality of suspension kinematics.
- Setup Suspension 101 [79]
  - This online resource expands on preload, compression, rebound, ride-height, and crossover spacers tuning.
- Suspension Geometry Calculator [65]
  - This online software provides an intuitive, simple suspension geometry calculator to play around with basic geometry before diving into Lotus Shark.

3.2.2.2 Seth DeLuca

- Vehicle Suspension System Technology and Design Chapter 4 [61]
  - Analysis and Design of Suspension Mechanisms looks thoroughly at the different parameters the suspension should be considering such as camber, toe angle and roll axis.
- Geometric Design of Independent Suspension Linkages [62]
  - This is another good resource to refer to during design of this suspension system. This resource includes information regarding joint and link types.
- Fine-Tuning of the Suspension System [63]
  - Includes information on optimizing the suspension based on weight and driver preferences. This will be helpful when the team is finding ways to better the suspension system.
- Design Analysis of 3 Link Trailing Arm [64]
  - This analysis discusses the advantages and disadvantages of a trailing arm with camber links. The main benefit of this suspension geometry is a better control of camber through travel of the suspension.
- Design Analysis of H-arm with Camber Link [65]
  - This article highlights the design and manufacturing phase of a H-arm with a camber link. This article would be beneficial if the group chose this geometry, and reference this system in design phases.

- Racing Aspirations suspension Geometry [66]
    - This 2D software allows for a quick analysis of camber links and camber angle.
- Spring rate and wheel rate calculator [67]
    - Calculates spring and wheel rate when given parameters based on a simple geometry, weight, spring angle, and ride height.

3.2.2.3 Lars Jensen

- Performance Vehicle Dynamics: Engineering and Applications [67]
    - Chapter 7 – Suspension Kinematics, Chapter 8 – Dynamic Modelling of Vehicle Suspension. These two chapters were a great read and helped me learn more about ideal suspension characteristics for vehicles and how to model your designs.
- The Multibody Systems Approach to Vehicle Dynamics [68]
    - Chapter 4 – Modelling and Analysis of Suspensions Systems. This chapter helped my learn more about what kind of analysis should be performed on suspension systems in order to create the best final product.
- Suspension Design and testing of an All-Terrain Vehicle using multi-body dynamics Approach [69]
    - This source dealt with the flow of design calculations for suspension parameters and was directly applied to the initial design decisions for the rear suspension system.
- Optimal Design of Suspension System of Four-wheel Drive Baja Racing [70]
    - This reading looked at geometric design of rear suspension and offered another option for designing a model that could be used to optimize the suspension system.
- Design and Optimization of Rear Wheel Assembly for All-Terrain Vehicle [71]
    - FEA analysis of rear knuckle and hub was something I had very limited experience and this source helped me expand my knowledge on the topic and apply my learnings to the project.
- Float 3 EVOL RC2 Factory Series Owner's Manual [72]
    - Shock service and tuning is an important consideration for this project and this manual will be helpful for setting up the selected shocks to best suit the driver.
- A Square C & D "BAJA ATV Videos" Playlist [73]
    - SolidWorks modeling of suspension systems and knuckles is covered in this video series and directly applies the development of a high performing SAE Baja car.
- Introduction to SolidWorks Finite Element Analysis [74]
    - This video was very helpful for me to learn the basics of FEA modeling and what kind of fixtures I would need to analyze the rear suspension trailing link.
- Interpretation of the results obtained by Finite Element Analysis (FEA) in SolidWorks [75]
    - This web resource covers the results that come from an FEA analysis in SolidWorks and how they can be used in the design process.

- Fundamental of vehicle dynamics [75]
    - This book provided more information that was helpful for generating target values in the Lotus Shark software during the rear suspension design process.

3.2.3 Drivetrain

3.2.3.1 Henry Van Zuyle

- Shigley's Mechanical Engineering Design [22]
    - Chapter 17, Flexible Mechanical Elements
    - Machinery's Handbook [25]
- Chapter, Gearing
- US Patent US20180172150A1, Electromechanically actuated continuously variable transmission system and method of controlling thereof [37]
    - ETS ECVT patent
- An Experimentally-Validated V-Belt Model for Axial Force and Efficiency in a Continuously Variable Transmission [38]
    - Factors that effect CVT efficiency
- Modeling and Tuning of CVT Systems for SAE® Baja Vehicles [40]
- Shaft Splines & Serrations [42]
    - Spline strength and geometry
- Altair Motion View: CVT Model [43]
    - Helped me develop my CVT design software

3.2.3.2 Ryan Fitzpatrick

- Shigley's Mechanical Engineering Design [22]
    - Chapter 6, Fatigue Failure Resulting from Variable Loading : used for failure analysis as well as shaft diameter calculations.
    - Chapter 7, Shafts and Shaft Components : used during shaft analysis and design for manufacture and assembly of components.
    - Chapters 13 & 14, Gears – General & Spur and Helical Gears : used during general gear design phase to determine sizing and other design aspects.
    - Chapter 18, Power Transmission Study : an extra chapter in the book that looks at the design of a two-stage gearbox. This chapter is helpful in guiding my general steps in the design process and ensuring that I am considering all aspects of the design properly.
- Machinery's Handbook [24]
    - Chapter 12, Gearing : This is a secondary source for gear calculations other than Shigley's.
- Methodology for Designing a Gearbox and its Analysis – IJERT [56]

- General gearbox design process and aspects to consider during its design.

- Design and Analysis of Gearbox for SAE Baja Competition – IJERT [55]
  - Gearbox Design for SAE. This is a specific article that outlines the calculations necessary to properly design the gearbox, but it is specific to the particular gearbox in the article which is not the same as the gearbox the team is moving forward with.
- Lightweight Design of Gearbox Housing of Baja Racing Car Based on Topology Optimization – Journal of Physics [81]
  - Gearbox housing design for optimization using topology methods. This will be helpful in the future when my design focus shifts to the housing design.
- Gear Design by AGMA Theory – The Engineering Blog [82]
  - AGMA theory source that includes lube factor. This is important because the Shigley's equations do not account for lube which severely impacts the life cycle calculation of the gears.
- A Look at Belt, Chain and Gear Drive Technology – Power Transmission Engineering [83]
  - Power Transmission Options Discussion. This article helped in the selection of power transmission types.
- Chain Sprocket Calculator [84]
  - Used to calculate the chain drive option which is discussed below in the calculations portion of the report.
- MatWeb Online materials Information Resource [125]
  - Used to compare material properties of steels and aluminums for use in the manufacture of the gearbox materials including shafts, gears, and gearbox housing case.
- McMaster Carr Ball Bearings Catalog [126]
  - Used to research and compare ball bearing options for the three shafts in the rear gearbox for life and load ratings as well as pricing identification.
- SKF Ball Bearing Catalog [127]
  - Used to research and compare ball bearing options for the three shafts in the rear gearbox for life and load ratings as well as pricing identification. This source was used more extensively for life and load ratings that McMaster Carr because its catalog is referenced directly in Shigley's.

3.2.3.3 Donovan Parker

- Shigley's Mechanical Engineering Design [22]
  - Chapter 3, 6, 7, 11, 16
- Machinery's Handbook [24]
  - Machine Elements, Polygon Shafts
- Machine Elements in Mechanical Design [29]

    - Chapter 7, Section 5
- McMaster-Carr [21]
    - Power Transmission
- Design of a Drivetrain for SAE Baja Racing Off-Road Vehicle – IJAEMS [112]
    - Powertrain
- Design Analysis and Fabrication of the Powertrain System for All-Terrain Vehicle – IJERT [111]
    - Calculations
- Belts/ Other Drives – Baja SAE Forums [109]
- SAE Baja '24 Rule Book [110]
    - Belt, Gear, and Chain Drives
- Belts and Chains Play to Their Strengths – Power Transmission [124]
- Belt and Chain CVT: Dynamics and Control – Mechanisms and Machine Theory [123]

3.2.3.4 Jarett Berger

- Shigley's Mechanical Engineering Design [22]
    - Specifically, chapter 14, discusses how to design spur and helical gears using gear design equations. In addition, chapter 18, discusses power transmission that can used in designing the 4WD system.
- Machinery's Handbook [24]
    - This textbook provides thorough explanations and equations for gears, splines, and cams. There are also figures and tables that present gear ratios and example equations that are needed to conduct these calculations.
- Spur Gear Designing and Weight Optimization [45]
    - This paper discusses the approach in designing a spur gear. It includes step by step equations needed to calculate each part of the spur design. In addition, it also focuses on how to save weight through material choices and comparison of weight to strength ratio.
- Design, Analysis, and Simulation of a Four-Wheel-Drive Transmission for an All-Terrain Vehicle [54]
    - This detailed paper analyzes how a 4WD system works and how to design it so that it can successfully operate. It overlooks the rear and front gearboxes and specifically, different types of power transmissions used.
- Design and Analysis of Gearbox for SAE Baja Competition [55]
    - This source analyzes gearbox design and presents a thorough example of each step for designing a gearbox. Since the current SAE Baja regulations are limited, this makes designing the drivetrain more open ended.
- Methodology for Designing a Gearbox and its Analysis [56]
    - This online source shows steps in designing a gearbox, which includes tables and equations. It provides nomenclature for each step in designing a gearbox.

- Design and Analysis of Gearbox with Integrated CV Joints [57]
  - This online source analyzes how to integrate CV joints with output shaft. It discusses splines and how it mates within the gearbox. Additionally, this source provides images of different designs of CV joint integration, which can influence the CV joint integration for the teams SAE Baja vehicle.
- Introduction to SolidWorks Finite Element Analysis [115]
  - This video explained how to use the FEA feature in SolidWorks and how it can be used to analyze different forces acting on different components for the front gearbox.
- Analysis of a Cross Groove Constant Velocity Joint Mechanism Designed for High Performance Racing Conditions [121]
  - This journal article analyzed how a Constant Velocity Joint (CVJ) functions. It provides detailed pictures and equations needed to determine plunge geometry.
- Numerical Analysis Based on a Multi-Body Simulation for a Plunging Type Constant Velocity Joint [122]
  - This source goes into depth about the geometrical features of a Constant Velocity Joint and analyzes contact forces and stiffness of the inner side of the cup. It presents thorough pictures illustrating vectors and equations needed to calculate the kinematics of the individual ball bearings, which can be used for the CV cup analysis.

3.2.4 Frame

3.2.4.1 Gabriel Rabanal

- Materials selection in mechanical design [18]
  - Chapters 5 and 6 of this textbook lay out the process for selecting specific materials for a job. This is helpful in some smaller aspects of the frame design like the skid plate, where multiple different materials can be chosen and must be compared.
- The Automotive Chassis: Engineering Principles [14]
  - While the entire textbook is applicable to the project, chapter 6 focuses on the loading effects on the chassis of the vehicle as well as braking behaviors with different designs. This applies to the frame team as we are the ones designing to accommodate the other sub teams and ensuring that all systems work in a cohesive environment.
- A novel approach for design and analysis of an all-terrain vehicle roll cage [15]
  - This paper contains an in-depth analysis of a rear braced frame design, very similar to the one settled on for the team design. Using the results of this FEA model, the team can evaluate strong and weak points of the tested design and adjust our design accordingly.
- Computational analysis for improved design of an SAE Baja frame structure [16]
  - This paper is another analysis of a rear braced frame, albeit one of a significantly different design. The paper uses a different analysis procedure and draws comparisons to industry vehicle design for the analysis. Combined with the previous reference, the team can look for the strengths of both designs to find optimal solutions to design concepts.
- Design and FE analysis of chassis for solar powered vehicle [17]

- This paper has an in-depth comparison of two different steel types that were considered in our material selection phase. The FEA modeling compares AISI 4130 and AISI 1018 steel types, helping the team decide which material will be better suited to the project frame.

- Mini Baja Vehicle Design Optimization [19]
    - This paper shows FAE modeling for a front braced frame analyzed for failure points using different materials and thicknesses. While the front braced frame design differs from our chosen orientation, the comparison of material thicknesses and general failure locations helps us optimize the frame as much as possible.
- SolidWorks BAJA SAE Tutorials - How to Model a Frame (Revised) [20]
    - This YouTube video from the SolidWorks page is a direct tutorial on how to create a SAE Baja style wireframe in SolidWorks. This is helpful for creating a CAD model that is easy to implement changes to and maintain the integrity of the file as adjustments are made.
- Chassis Build || UBC Baja Build Series Episode: 3 [112]
    - This source is a YouTube video that follows UBC Baja in their 2018 year jigging the Baja frame. The source is helpful in showing a unique manner of jigging the frame to allow for easier construction and changes to be made if necessary.
- Jigging We Just Tried [113]
    - This source is a forum from 2014 where members from different teams discussed the best ways to jig their frames. This forum was helpful to the team in deciding which method we would use for jigging the frame and ensuring proper welding angles.
- SOLIDWORKS - Flatten Pipes using Insert Bends [114]
    - This source demonstrates another method for coping tubes derived from a Solidworks part. This is the method that SAE Formula used for coping, which was one option that we looked at for frame manufacturing.

3.2.4.2 Cooper Williams

- Shigley's Mechanical Engineering Design [22]
    - Chapter 9: Welding, Bonding, and the Design of Permanent Joints
    - This chapter discusses the different factors to consider when selecting a permanent bonding method. Although there are SAE rules about the standards of our TIG welding, examining the factors that could improve the strength of our design could drive some of our design.
- The Automotive Chassis: Engineering Principles [14]
    - Ch 6.1: Vehicle and body center of gravity
    - This entire book is a well of valuable knowledge; however, this chapter will be incredibly important when considering how sub-system integration affects the steering and handling of the vehicle.
- Design and Analysis of Chassis for SAE BAJA Vehicle [19]

- This article will help drive some of the design of the frame. By showing how different forces act on the frame, the angles and lengths of some members can be adjusted, within reason, to minimize the torsion, torque, and ultimately deflection of the members.

- Mathematical Model for Prediction and Optimization of Weld Bead Geometry in All-Position Automatic Welding of Pipes [25]

    - Similar to Shigley's Chapter 9, this article provides specific factors that can optimize TIG weld strength. This article is much more specific and provides the mathematical modeling done to support the equations derived in the article. Although this information may not contribute considerably to manufacturing our frame, it will help when considering how to perform some of the more challenging and meticulous welds.

- Design, analysis and optimization of all-terrain vehicle chassis ensuring structural rigidity [21]

    - This article provides research supporting why rear brace frames are structurally superior to other frame designs, which is why they have been so dominant in the offroad industry currently.

- Design Judging Discussion [28]

    - In this forum, a previous judge reveals some of the thought process of judging a BAJA vehicle at competition. Although slightly dated, this forum will allow our frame team to design towards placing highly which directly correlates to the judges' opinions.

- Getting Started with Weldments in SOLIDWORKS [27]

    - This video is a great starting point for starting to use weldments in SolidWorks. This frame design is weldment intensive, so utilizing this powerful tool is essential to successfully designing a frame for manufacturing.

- MatWeb - The Online Materials Information Resource [117]
    - This website provided material information, specifically modulus of elasticity, for 4130 CD steel. This property was used in the additional calculations that I performed for deflection of SIMs.

- Axiom CNC: Creating a Project from Start to Finish! [118]
    - This video helped me learn many of the basic functions of the Axiom CNC router. One of our sponsors, NovaKinetics, has one of these routers and has confirmed that we can use it to make our jig.

- Axiom CNC Machine Training [119]

    - This video provided a great overview of how the Axiom CNC router system works and the many functions it might have. It was posted by Axiom themselves which was reassuring. It reviewed settings, setup, operation, and basic knowledge that one might need to router a part, such as a jig.

3.2.4.3 Antonio Sagaral

- Shigley's Mechanical engineering Design [22]
  - Ch 2 – 22 discusses formulas and techniques for properly selecting materials. Information is included from material properties to the application in which the material is to be used.
- Fundamentals of Machine Component Design [29]
  - Ch 11 focuses on welding and different bonding techniques. All of which could be incorporated into the frame design and improve certain joints in the frame.
- Design Analysis and Optimization of a BAJA-SAE Frame [39]
  - This paper has useful information about FEA analysis as well as material selection.
- Design and Construction of a Space-frame Chassis [41]
  - In this paper, more FEA techniques are discussed including how to handle different suspension forces in the model.
- DESIGN AND STRUCTURAL ANALYSIS OF BAJA FRAME WITH CONVENTIONAL AND COMPOSITE MATERIALS [44]
  - This paper covers different ways to analyze frame impacts and has various equations to be used in the force analysis.
- [Front Impact Test & Meshing] BAJA SAE Roll Cage/Frame Design in ANSYS Workbench Static Structural [53]
  - This online source has more information on impact testing. This includes different impact testing points as well as strategies to cover a more complete impact analysis
- Baja SAE Frame Investigations [47]
  - This online source outlines the pros and cons of different bracing techniques i.e., front vs. rear braced frames.

3.3 Mathematical Modeling

3.3.1 Front End

3.3.1.1 Abraham Plis – Steering Knuckle

Throughout the mathematical modeling of the steering knuckle, several equations, tools, and examples were utilized to direct the analysis towards the most optimal path. The relevant governing equation that applied was the hole bearing stress equation that allows for bolted connections to be designed against tear-out and deformation (see Figure 13):

$$\sigma_b = \frac{P}{A_b} = \frac{P}{td}$$

Equation 1: Hole Bearing, Single Shear [10]

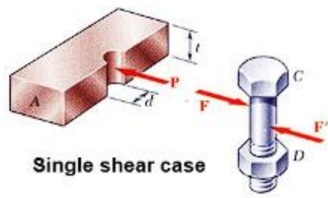


Figure 13: Hole Bearing Diagram

The applicable tool that will help guide the mathematical modeling of the steering knuckle is the team's suspension software Lotus *Shark* (Figure 14). This will help the front end define and optimize the geometry of the knuckle as well as the overall suspension system. The appeal of *Shark* is the versatility of the program, the generation/experimentation of geometric hardpoints, and the real-time feedback of geometric alterations [7].

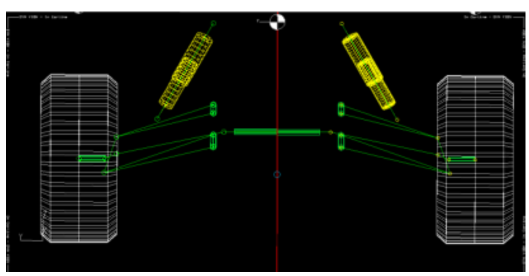


Figure 14: Lotus Shark Software Preview

Lastly, there is a great online example that can be followed to correctly perform FEA on a steering knuckle to optimize its design characteristics [9]. This example will be critical to help the team dial in the geometry of the knuckle and optimization of its weight/manufacturability while not compromising its structural integrity (Figure 15).

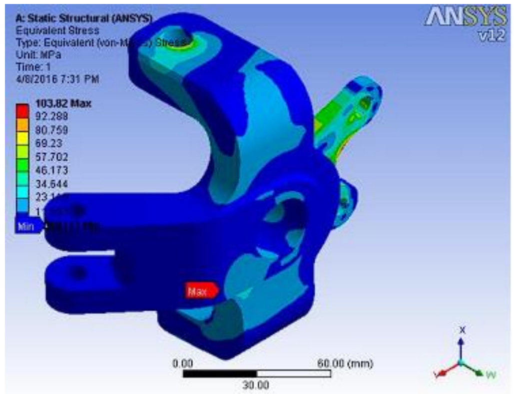


Figure 15: Steering Knuckle Example

The design-oriented motivation for this mathematical model was to see how much bearing stress the central knuckle hole will see from the outboard CV spline during a jump. A simple diagram is presented below in Figure 16 to aid the reader with visualization.

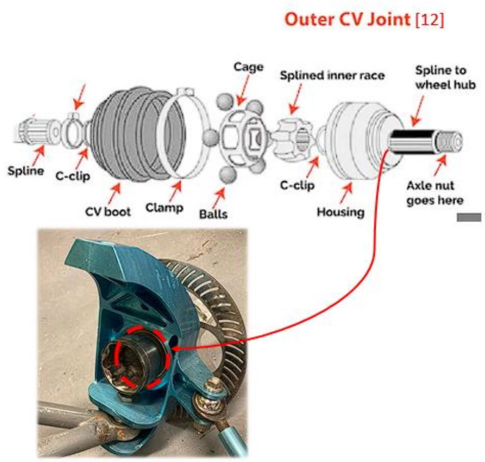


Figure 16: Steering Knuckle Modeling Diagram

A worst-case scenario was assumed in which the total weight of the car (500 lbs. with driver) was delivered to a single wheel from a 6-foot drop with a 3G deceleration occurring in roughly a tenth of a second. The diameter of the spline is 1" and the thickness of the contact surface between the spline and knuckle is roughly 2". To calculate the impact force generated, the following equation can be used [11]:

$$F = \frac{m * \sqrt{g * h}}{t}$$

Equation 2: Impact Force

Combining Equation 1 and Equation 2, a bearing stress of 1080 psi can be calculated to be seen by the knuckle during this event. This stress can be validated by comparing to the yield strength of billet aluminum, which is roughly 26,100 psi [13]. This means the observed bearing stress takes up 4% of the knuckle's yield strength, allowing the team to design with the intention of skeletonizing the knuckle around the inner contact surfaces. This will ensure the knuckle is as lightweight as possible while not compromising its strength.

3.3.1.2 Bryce Fennell – Control Arm

While designing the upper A control arm for the front of the SAE vehicle, weight, and stiffness will be a top priority. Ensuring the arm is as light as possible while remaining strong enough to handle the impacts of racing without an induced failure are critical. There are 3 critical locations on the upper control arms being: Chassis mounting location, shock mounting location, and the steering knuckle mounting location. These three locations can be found in Figure 17 and will be the only areas with a force input.

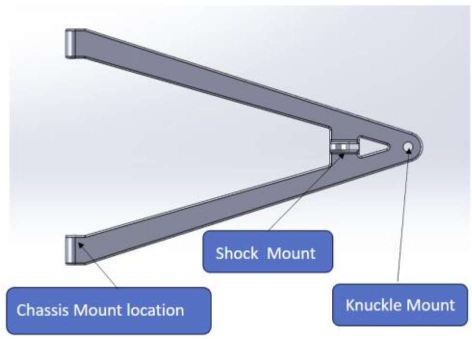


Figure 17: Labeled Upper Front A Control Arm

I performed bending moment and shear force calculations using Equation 3 to determine the position of the maximum shear force and bending moment on the upper control arm. By understanding these maximum positions added strength can be integrated into the design at this location and removed from less critical portions of the control arm.

$$Shear\ Stress\ (Tao) = \sum \frac{Force}{Cross\ Sectional\ Area}$$

Equation 3: Instantaneous Shear Stress in a Member

$$Bending\ Moment\ (M) = \int Shear\ Force\ Diagram\ (SFD)$$

Equation 4: Equation for Graphing Bending Moment in a Member

Using a length of 17 inches with a maximum impact force calculated using a 550lb vehicle falling from 3ft off the ground onto a single front wheel a maximum bending moment was calculated to be 4050 lbf*in occurring 13.56 inches outboard from the chassis mounting location. Figure 18 details the graphs indicating both the maximum shear and bending of the control arm under the maximum realized load.

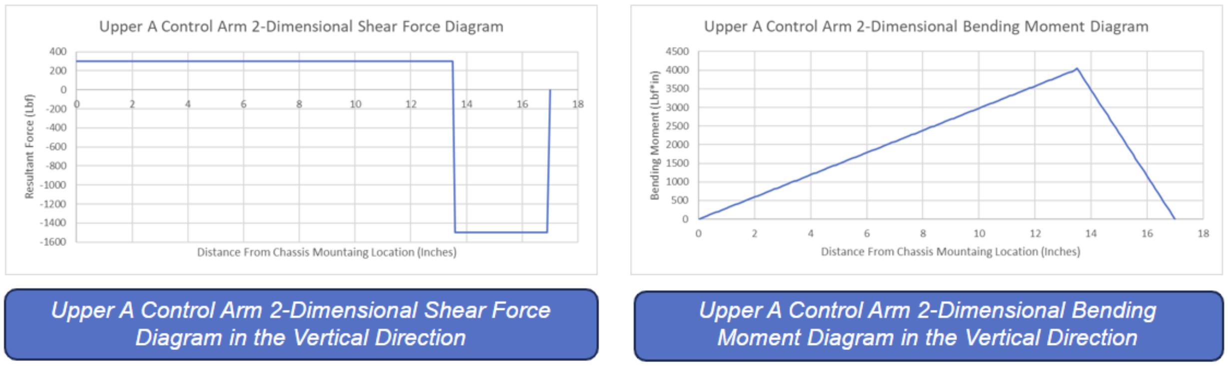


Figure 18: Left: Shear Force Diagram of Upper A Control Arm,

Right: Bending Moment Diagram of Upper A Control Arm

Knowing the location of maximum bending moment in the upper control arm, we will add material and stiffening members to ensure the control arm meets the impact requirements stated above. Additionally, in locations inboard of the maximum bending moment location, material can be removed to reduce the overall weight of the control arm without negatively impacting the safety of the member.

3.3.1.3 Evan Kamp – Ackermann Steering

After deciding that the team wanted to use Ackermann steering geometry, Mathematical modeling was done to make preliminary predictions for the turning radius of the vehicle. Ackermann steering describes that the steering angle of the inside wheel is proportionally steeper than that of the outside wheel. This allows for the inside wheel to effectively lead the car through the range of the turn. The figure below shows a vehicle displaying Ackermann steering with inside angle δ_i being greater than that of δ_o.

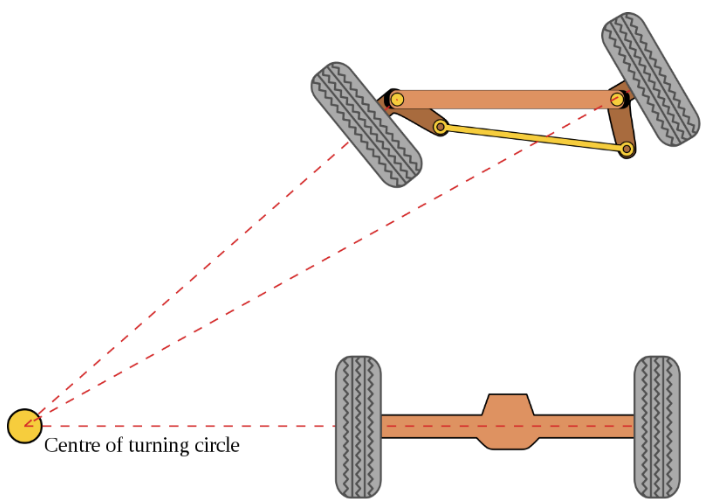


Figure 19 : Ackermann Steering

Ackermann steering calculations are used under the assumption that the slip angles of the steering system are close to or are at 0º during a slow turn. To meet this condition viable steering angles must be decided by using Figure 20 below.

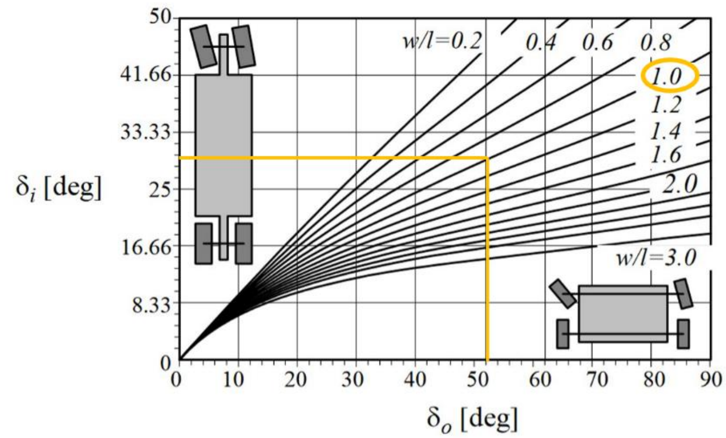


Figure 20 : Viable Steering Angles using Ackermann Steering

With the team trying to use a 1/1 width length ratio, and deducting that the maximum possible turning angle of the teams CV axles, Figure 20 is used in order to find the cooresponding outside angle δ_o when δ_i is at 50º; it also displays the effect of width and length on viable steering angles. This chart is calculated as a function of lateral force and slip angle. After deciding steering angles, turning radius can be calculated.

$$R = \sqrt{a_2^2 + l^2 \cot^2 \delta}.$$

Equation 5: Steering Radius

$$\cot \delta = (\cot \delta_o + \cot \delta_i)\frac{1}{2}$$

Equation 6: Steering Angle

$$\%Ackermann = \frac{\delta_i - \delta_o}{\delta_i} * 100\%$$

Equation 7: Ackerman Angle

Table 9: Steering Calcs

Preliminary Measurements	
Wheel Center Length (l)	64in
Wheel Center Width (w)	64in
Inner wheel angle (δi)	50°
Outer wheel angle (δo)	30°
δavg	40°
Rear wheel to center of gravity (a2)	32in
Results	
Percent Ackerman Used	40%
Hypothetical Turning Radius (R)	6.89ft

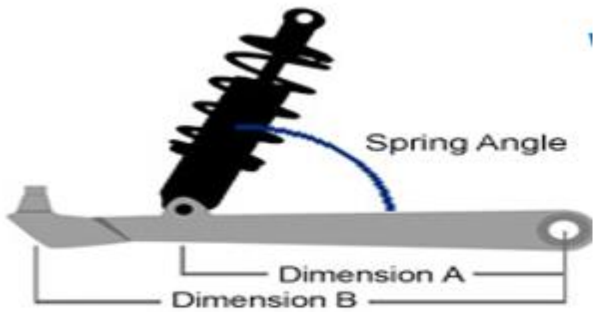
Understanding viable Ackermann angles as a function of slip angle and lateral force allows the team to construct a good steering car when developing steering geometry with the knuckle. In addition, insight to turning radius gives the team confidence that Ackermann steering is a good design and is viable within competition.

3.3.2 Rear End

3.3.2.1 Seth DeLuca – Shock Mounting Angle

In this analysis, the angle of the shock was the main variable that was being analyzed. The shock angle refers to the angle of the shock referenced to the suspension links.

Figure 21: Depicts some of the variables of the suspension system [66].



To conduct this analysis a few assumptions were made. These estimated assumptions were made to keep calculations simple and get an idea of what the shock angle should be. These assumptions and variables are listed below:

- Corner weight of the vehicle in the rear ≈ 150 lbs
- Unsprung corner weight of the vehicle in the rear ≈ 45 lbs
- Dimension A ≈ 16 in
- Dimension B ≈ 16 in
- Shock Ride Height ≈ 2.27 in
- Shock Angle ≈ 60, 70, 80, 90 degrees

A series of formulas were then needed to complete the analysis of this design. These equations are listed below [66]:

$$Sprung\ weight = Corner\ weight - Unsprung\ weight$$

Equation 8: Sprung weight

$$Motion\ Ratio = \left(\frac{Dimension\ A}{Dimension\ B}\right) * \sin\ (Spring\ angle)$$

Equation 9: Motion ratio

$$Static\ load = \frac{Sprung\ weight}{Motion\ ratio}$$

Equation 10: Static Load

$$Spring\ rate = \frac{Static\ load}{Shock\ ride\ height}$$

Equation 11: Spring rate

$$Effective\ wheel\ rate = Spring\ rate * Motion\ ratio^2$$

Equation 12: Effective wheel rate

These equations were then utilized to calculate the spring rate and effective wheel rate for a varying angle of the shock.

Figure 22: Analysis of Effective wheel rate and Spring rate based on varying shock angles.

$$Spring\ rate = \frac{(Corner\ weight - Unsprung\ weight)}{\frac{Dimension\ A}{Dimension\ B} * \sin(Spring\ angle)} \div Shock\ ride\ height$$

90 degrees

$Spring\ rate = 57.76$ lb / in

$Effective\ wheel\ rate = 36.96$ lb / in

80 degrees

$Spring\ rate = 58.65$ lb / in

$Effective\ wheel\ rate = 36.40$ lb / in

70 degrees

$Spring\ rate = 61.46$ lb / in

$Effective\ wheel\ rate = 34.73$ lb / in

60 degrees

$Spring\ rate = 66.69$ lb / in

$Effective\ wheel\ rate = 32.01$ lb / in

These calculations allowed the team to see that having the shock angle closer to 90 degrees will allow the system to be compressed with less force and allow more bump and rebound travel.

3.3.2.2 Lars Jensen – Rearward Axle Travel

An idea from mountain biking is increasing the rearward axle path of the suspension to get the wheel out of the way of obstacles. This calculation applies that idea to the Baja car and allows for different suspension geometries to be tested against each other. Raising the front pivot point of the trailing link away from the bottom of the car increased the rearward axle path giving the sub team direction in the design process. Having the front pivot higher is beneficial to the suspension characteristics and will help the sub team achieve their goal of a suspension system that always maintains maximum traction. A 30 in. trailing link was used for this model and is accurate with what the final suspension geometry is going to look like. Figure 23 shows the layout of the different suspension geometries and Figure 24 shows the resulting rearward travel measurement.

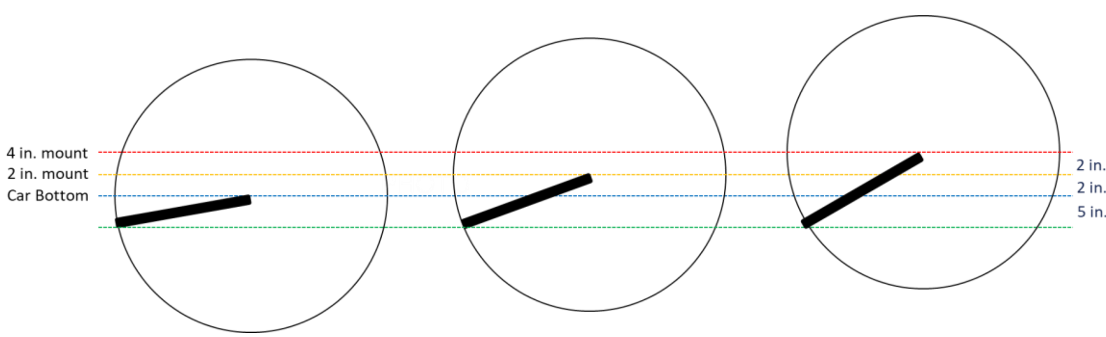


Figure 23: Trailing Link Configurations

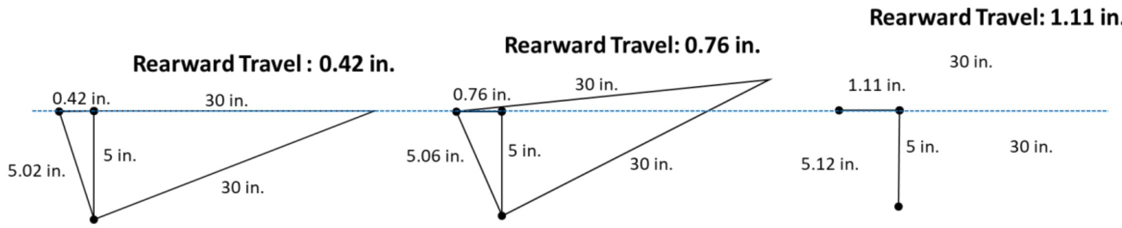


Figure 24: Rearward Axle Path Measurements

3.3.2.3 Joey Barta – Camber Gain

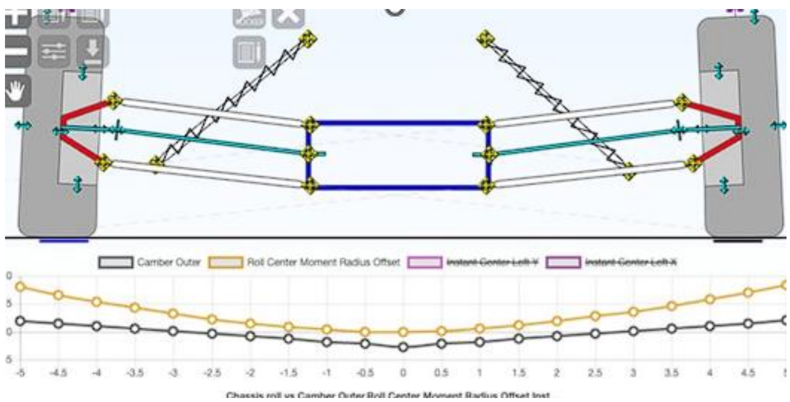


Figure 27: Camber vs. Roll Center

Figure 25 - Initial Rear Suspension Dimensions

Figure 26 - Initial Rear Suspension Dimensions

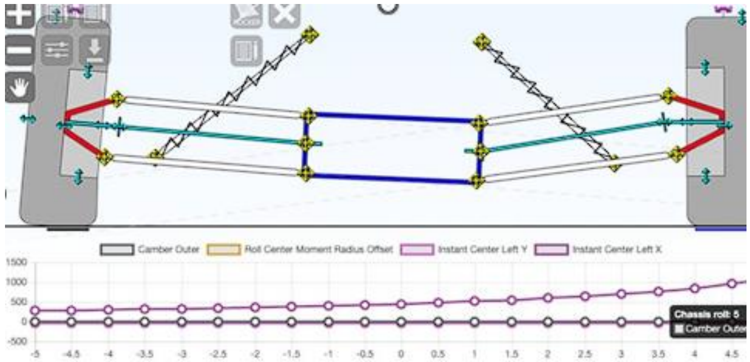


Figure 28: Camber vs. IC

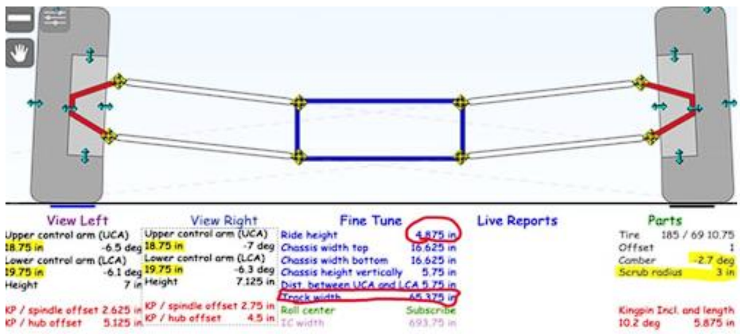


Figure 29: Rear Suspension Initial

Using Racing Aspirations' Suspension Calculator [65], a varying upper camber link length was experimented with to find a suitable ratio of lower link length to upper link length. Although the software used is much more restricted than Lotus Shark, it allowed for us to identify what sort of motion our suspension will go through under hard cornering, and unsprung weight. Our properties were defined by a track width of 65 in. (marginally higher than our realistic length), ride height of 5 in. (about half of our vehicles height), camber of -2.5 degrees, and a ratio of lower to upper link length of 1.05. After this baseline was defined, tests were run through the software that measured instant center with respect to camber and roll center moment radius offset compared with camber. The findings from these measurements confirmed that a shorter upper chamber link with result in negative camber gain from compressive suspension motion.

3.3.3 Drivetrain

3.3.3.1 Henry Van Zuyle

Having a CVT transmission that has the desired range and gear ratios with a selected belt is an important consideration for the performance of our vehicle. Calculating this took some relatively complex systems of equations, Figure 31, that were solved with MATLAB. The use of MATLAB also allowed for quick and easy iterations to dial in the desired gear range and ratios. Using these equations and multiple iterations, all variables were eventually decided on. Using a Gates 19G3450 belt, the center-to-center distance is 9.5", the sheave angle is 12.77 degrees, and the maximum primary side actuation force is 412 lbs., when friction is included. This primary side clamping force was then used to select an appropriate motor and drive screw. With a ½-10 lead screw, and a cast iron nut, the equations in Figure 32 were able to be used to determine a peak torque of 443 oz-in was required. Having that torque then allowed us to select an appropriate motor. Figure 30 Shows the torque curve of the motor that was selected. It is able to output much higher torques than calculated, but to keep temperature down and to allow for things like dirt contamination of the lead screw, a motor that was larger than necessary was decided to be the right choice.

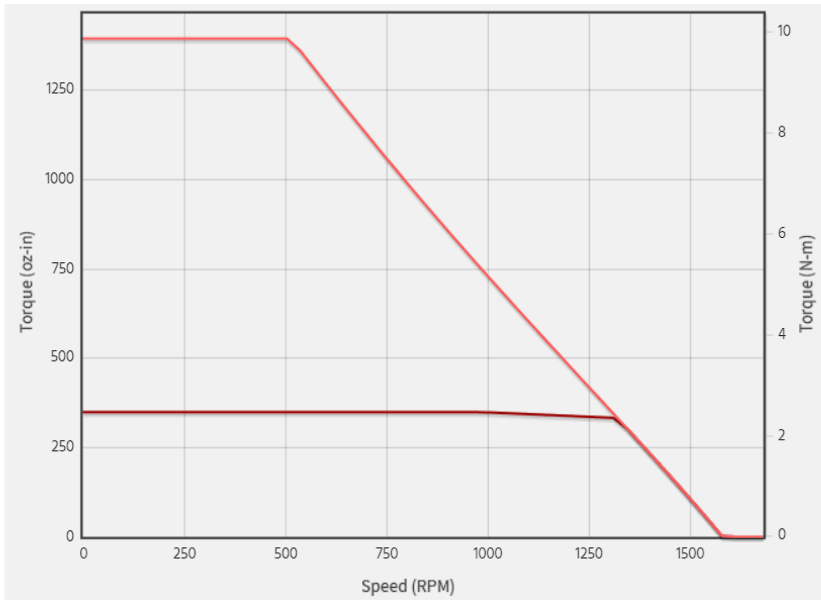


Figure 30: M-3432F-LS-08D Torque Curve

If $\beta \geq \pi$

$$T_0(lbf) = \frac{2\sin\left(\frac{\beta}{2}\right) * \left[2F_{Clamp}\tan\left(\frac{\phi}{2}\right) + \frac{1}{12}M_{Belt} * R^2 * \omega^2\right]}{cos\left(\frac{1}{2}(\beta - \pi)\right) * (e^{\mu_e \beta} + 1)}$$

If $\beta \leq \pi$

$$T_0(lbf) = \frac{2\sin\left(\frac{\beta}{2}\right) * \left[2F_{Clamp}\tan\left(\frac{\phi}{2}\right) + \frac{1}{12}M_{Belt} * R^2 * \omega^2\right]}{cos\left(\frac{1}{2}(\pi - \beta)\right) * (e^{\mu_e \beta} + 1)}$$

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

$$T_1 = T_0 e^{\mu_e \beta}$$

$$T_{1\_Primary}(lbf) = T_{1\_Secondary}$$

Figure 31: CVT Force and Geometry Equations

Torque(raise) = F*Dm/2*(L+u*PI*DM)/(PI*Dm-u*L)

Torque(lower)= F*Dm/2*(L-u*PI*DM)/(PI*Dm+u*L)

Figure 32: Lead Screw Equations

3.3.3.2 Ryan Fitzpatrick

For the rear gearbox, it was important to consider all methods of power transmission. To do this, I did research and basic calculations to determine if power should be transmitted via chain or gear drive. The reason a belt drive was immediately out of contention for the design was that it generally has all of the benefits of a chain drive, while taking up more space and being less efficient than the chain drive system. The main calculations that I was considering in this analysis were overall size of each design, and the efficiency of each design. The gearbox calculations were done according to the methods within the Shigleys textbook for Machine Design [22]. The resulting geometry for the gearbox design can be seen below in Figure 34. The chain drive calculations were done using multiple sources to compare the two designs to each other, as well as an online calculator. By taking the torques and angular velocity on each stage of the transmission, a minimum ANSI Chain Number of #50 was determined to be necessary for the chain drive system. These values for torques, angular velocities, and chain number were input into an online sprocket calculator [84] and the dimensions shown in Figure 33 were determined for a chain drive.

The results and conclusions of these calculations are listed below:

- Gearbox is slightly more efficient (Chain drives have efficiencies "up to about 98%" [83] versus gearboxes efficiencies of "less than 2%" [22]).
- Gearbox takes up 60% less space (see modeling below in Figure 34).
- Gearbox requires less maintenance [83].
- The top teams run gearboxes.
- From these calculations, we have decided to move forward with a gearbox as opposed to a chain drive.

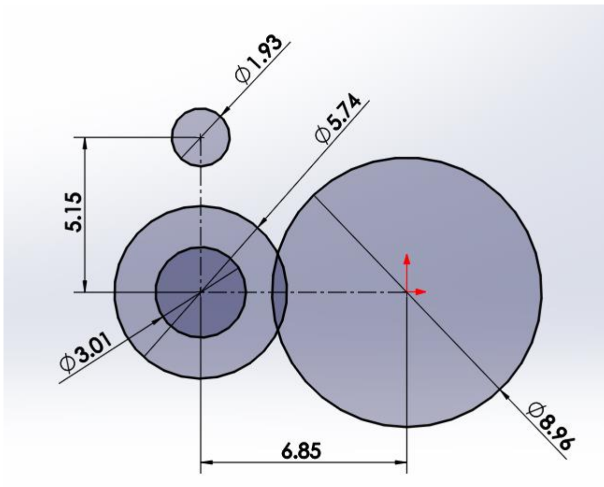


Figure 33: Chain Drive Geometry

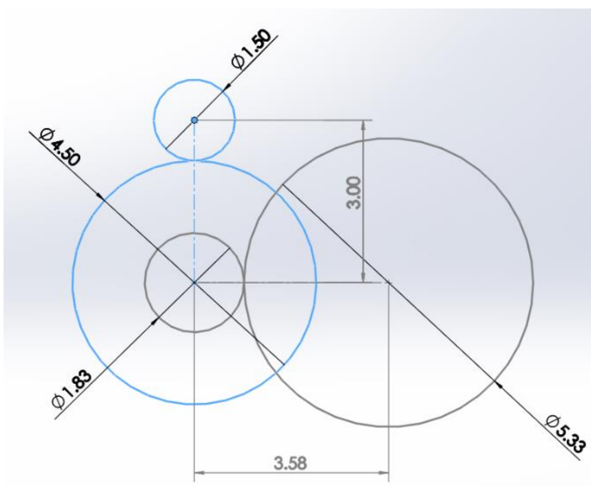


Figure 34: Gearbox Geometry

The next calculation required was to design the gearbox according to the required specifications of the team's design goals and based on the outputs of the eCVT. The goal of this design process was to achieve a reduction of ~9.2 via a two-stage compound gear train while minimizing the space occupied by the gearbox to allow for better rear suspension geometry. The equations listed below in Figure 35 were used in the initial design stage of the gearbox.

Diametral Pitch : $P = \frac{N}{d}$; $P\ (1st\ Stage) = 16\ \frac{teeth}{in}$; $P\ (2nd\ Stage) = 12\ \frac{teeth}{in}$

Train Value : $e = \frac{product\ of\ driving\ tooth\ numbers}{product\ of\ driven\ tooth\ numbers}$; $e_{target} = \frac{1}{9.2}$

Minimum Pinion Teeth : $N_{P,min} = \frac{2k}{(1+2m)sin^2(\Phi)}\left(m + \sqrt{m^2 + (1+2m)sin^2(\Phi)}\right)$

where… k = 1 (for full-depth teeth)

Φ = 20deg (standard pressure angle)

Input RPM = 1200

Output RPM = 120-140

Figure 35: Gearbox Design Equations

Using these equations, I was able to determine the pitch diameters and number of teeth for each gear (listed below in Figure 36), which then allowed me to calculate the overall train value of the two-stage reduction.

Gear	d (diameter, in)	N (number of teeth)
2	1.5	24
3	4.5	72
4	1.83	22
5	5.33	64

Figure 36: Rear Gearbox Pitch Diameters and Teeth Numbers

3.3.3.3 Donovan Parker

The material and parts that the equations below model are what our current chosen belt size, the magnitude of force our pulley clutch side will be experiencing, the strength of the coupling clutches themselves, along with a factor of safety for the clutches and acceleration for proper coupling. We were already aware of what ratio we wanted from front to rear pulley and we already knew how much clearance we would have where the system would be, so our diameters were nicely decided for us and with that information, the first five equations below represent appropriate wrap angle, ideal belt length, and forces from the tensioned belt, in that order. In addition, with the provided information that came from the designs of the pulley, we made the clutch diameter the same for ease of manufacturing and mathematically tested its feasibility and strength. The last five equations represent how much force and stress go into both clutch members during engagement and if they can handle repeated load. The factor of safety calculation confirms that they can.

$$\Phi_d = \pi - 2sin^{-1}\frac{D-d}{2C} = 2.282\text{rads} = 130.757°$$

$$\Phi_D = \pi + 2sin^{-1}\frac{D-d}{2C} = 4.001\text{rads} = 229.243°$$

$$L = \sqrt{4C^2 - (D-d)^2} + \frac{1}{2}(D\Phi_D + d\Phi_d) = 110.92\ in. \ = 9.24\ ft$$

$$F_1 = \frac{T \times 12}{r} = 1048\ lbf\ (Tension\ Side\ Force)$$

$$F_2 = F_1 - \frac{2T}{d} = 0\ lbf\ (Slack\ Side\ Force)$$

$$\Delta r = r_o - r_i\ ,\ \ \Delta d = 2 \times \Delta r$$

$$F = \frac{T}{r_i/12} = 1742.0\text{ lbs}$$

$$\sigma = \frac{F}{A} = 11{,}531\frac{lb}{in^2}$$

$$n = \frac{\pi S \Delta d^2}{4F} = 28.84$$

$$a = \frac{F}{m} = 553.35\frac{ft}{s^2}$$

Where:

Φ_d = Smaller pulley wrap angle

Φ_D = Larger pulley wrap angle

L = Belt length

F_1 = Tension side force

F_2 = Slack side force

T = Torque

F = Force

σ = Normal Stress on Teeth

A = Planar Surface Area on Teeth

a = Acceleration

r = Radius, d = Diameter

m = mass

S = Material Strength

n = Factor of Safety

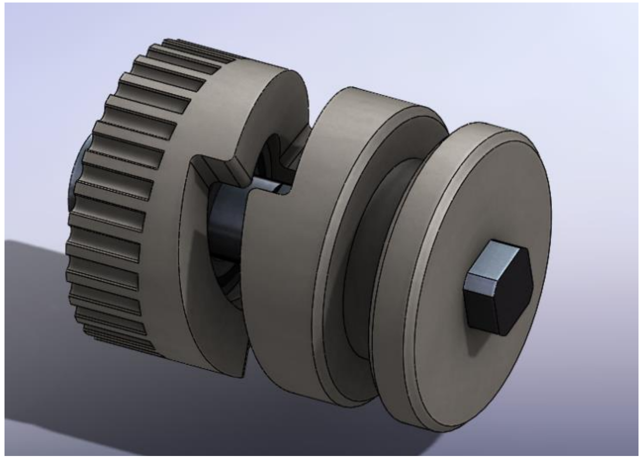


Figure 36: Mathematically Modeled Clutch and Pulley Design

3.3.3.4 Jarett Berger

For the front gearbox, it will be running a stage 1 gear reduction, seen in Figure 37, powered by a flat belt from the rear gearbox. A stage 1 gear reduction is critical since the goal is to have the front tires spin less than the rear tires so that when steering there is more traction. In addition to the front gear box design, the team has decided to integrate the CV joints into the gear box so that it creates a narrower front end and can optimize steering and suspension geometry, which will be discussed in section 5.5.6.1.

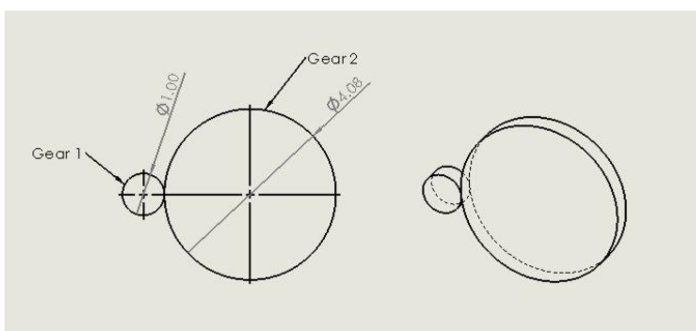


Figure 37: Preliminary Gear Design

The Equation 13 from Shigley's Mechanical Engineering Design [22] was used to find the number of teeth and diametral pitch, which is shown below:

Minimum Teeth for Pinion: $N_P = \frac{2k}{3sin^2\phi}(1 + \sqrt{3sin^2\phi})$

Maximum Teeth for Gear: $N_G = \frac{N_P sin^2(\phi) - 4k^2}{4k - 2N_P * sin^2(\phi)}$

Minimum Teeth for Pinion: $N_P = \frac{2k}{3sin^2\phi}(1 + \sqrt{3sin^2\phi})$

Maximum Teeth for Gear: $N_G = \frac{N_P sin^2(\phi) - 4k^2}{4k - 2N_P * sin^2(\phi)}$

Equation 13: N teeth and Diametral Pitch

: Max. and Min. Number of Teeth

K (Full-Depth) = 1
Pressure angle Φ = 20deg

Determining the number of teeth is crucial since the team's goal is to achieve a gear ratio of 1:4.5. Based on the calculations, the team has determined the correct number of teeth and pitch diameters for gear 1 and gear 2 for the front gear box. The results are illustrated in the table below:

Table 10: Front Gearbox Calculation

	Gear 1	Gear 2
Teeth	12	49
Pitch Dia. (in)	1	4.08333
Dia Pitch (teeth/in)	12	12

3.3.4 Frame

3.3.4.1 Gabriel Rabanal

Up to this point, multiple engineering calculations have been made to assist in design decisions for the frame. The first was a comparison of the bending stresses of different possible frame materials and their respective cross-sectional areas. These calculations were used to evaluate the type of steel that would be used in construction. Equation 14 and Equation 15 are the governing equations for the calculations made.

$$S_b = \frac{S_y I}{c}$$

Equation 14: Bending Stress

$$A_s = \pi(\frac{D_o}{2})^2 - \pi(\frac{D_o}{2} - t_{wall})^2$$

Equation 15: Cross Sectional Area of a Pipe

Calculations were also performed to assist in the choice of fuel tank mounts that were used. The volume of material used, weight of design, and ease of manufacturing were accounted for in the calculations and decision process. For material volume and weight, Equation 16 and Equation 17 were used.

$$Volume = (\pi {r_o}^2 - \pi {r_i}^2) * L$$

Equation 16: Volume of a Tube

$$Weight = \frac{\rho}{V}$$

Equation 17: Weight Equation

3.3.4.2 Cooper Williams

At this point, there are only so many calculations to be done for the frame, and they are going well. In order to relate these cross-sectional area and yield strength calculations to cost-effective material, I used a density calculation using Equation 18.

$$\rho = \frac{m}{V}$$

Equation 18: Density of Steels

Although incredibly rudimentary, this calculation allows our team to directly compare the cross-sectional areas and yield strengths of different materials to their cost. We can also use this comparison to analyze the effects that different materials will have on the overall weight of our car. If we select a steel with a higher yield strength, we can use tubing with a thinner wall. By doing this we minimize the cross-sectional area and therefore the amount of material used for the frame. This directly correlates to the weight of the frame. Essentially, we can generate a weight to strength to cost ratio to analyze the plethora of frame tubing options.

3.3.4.3 Antonio Sagaral

When going through the material selection process. There were three main considerations that had to be accounted for and all three came from the SAE BAJA rules. The first one being that the material had to be steel with a carbon content equal to or greater than 18%. In the rules, a baseline material was given for general guidance, and this was 1018 CR steel with a minimum outside diameter of 25 mm and the wall thickness must be at least 1.57 mm. A different material may be used if its bending strength and bending stiffness was higher than the 1018 CR steel at its baseline specifications. To choose the material, Equation 16 and 19 were used.

$$k_b = EI$$

Equation 19: Bending Stiffness

4 Design Concepts

4.1 *Functional Decomposition*

4.1.1 Front End

Figure 38: Front End Black Box

A black box model was used to determine the energy and mass flow throughout the front of the vehicle. Inputs into the model will all be derived from the course features and will be split into foot, hand, and course energy inputs. We will use the black box model to further refine our front-end designs and optimize for energy efficiency within our subsystem.

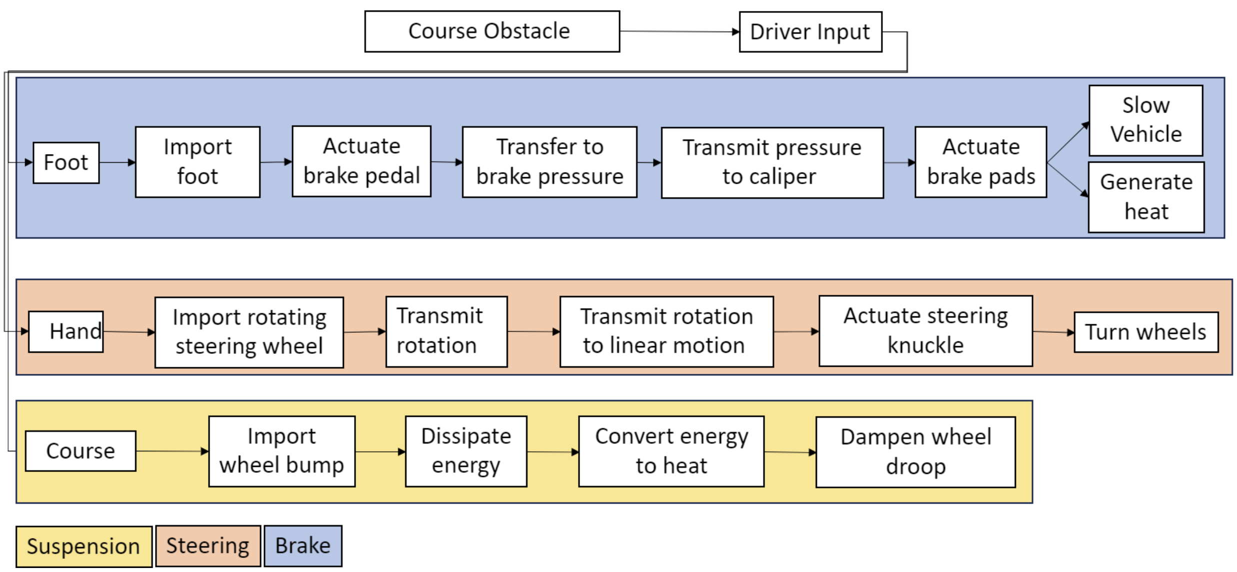


Figure 39: Front End Functional Model

4.1.2 Rear End

Figure 40: Rear Black Box Model

The black box model was used to identify energy coming through and out of the rear suspension to define what the basic functions of the rear suspension are. Notable components of the black box model are the compression and rebound of the shocks signaling chatter in the vehicle and suspension components. This signal will either verify or refute our suspension design and be incremental to the tuning process.

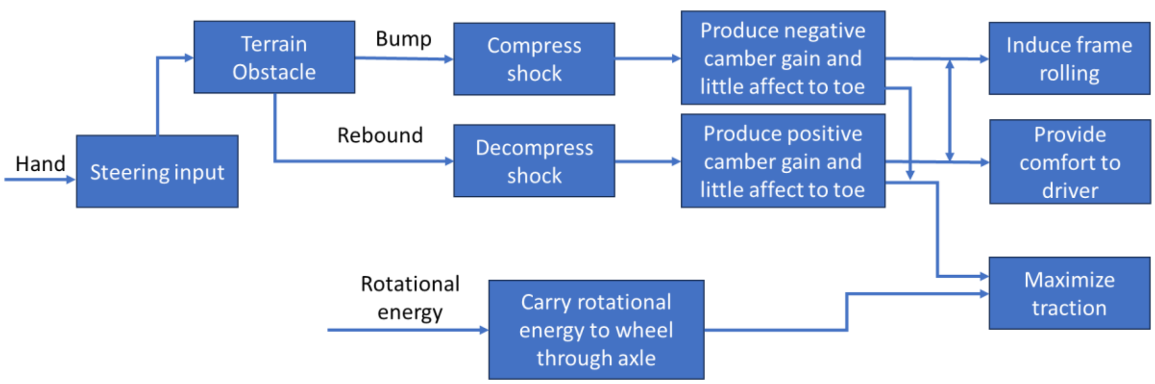


Figure 41: Rear Functional Model

The functional model allowed for the team to visualize all things necessary to design in our project. It provided a visual to identify the inputs and outputs and how energy is being transmitted for the rear suspension. The main aspects of the model are when the shock is being compressed the team would like there to be negative camber gain. This will maximize traction while also allowing a smoother ride for the operator of the vehicle. When the shock is decompressing or extending, the wheel shall undergo positive camber gain. This allows there to be some "give" in the suspension when landing and maintains and maximizes traction. These are the main influences of how rear suspension acts and this will help the team begin concept design.

4.1.3 Drivetrain

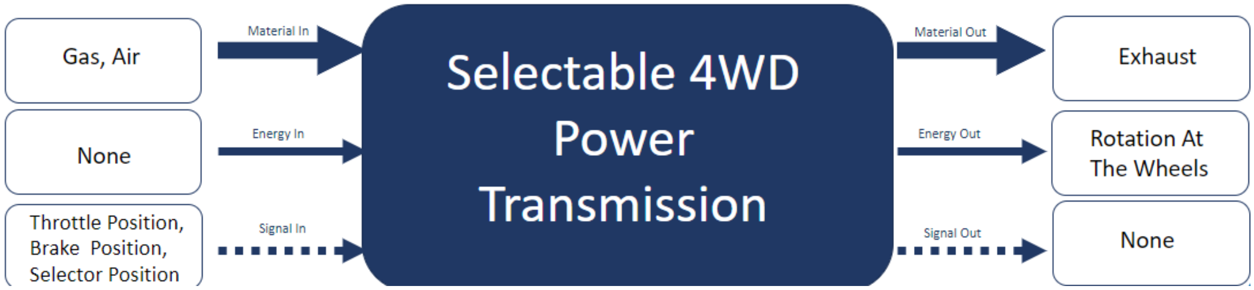


Figure 42: Drivetrain Black Box Model

A black box model was used to define several inputs and outputs for the overall function of the selectable 4WD power transmission. This helped the team in determining the functions needed to meet the expected outcomes. The black box model provides an outline for the functional model where it goes into specific details of how the system operates.

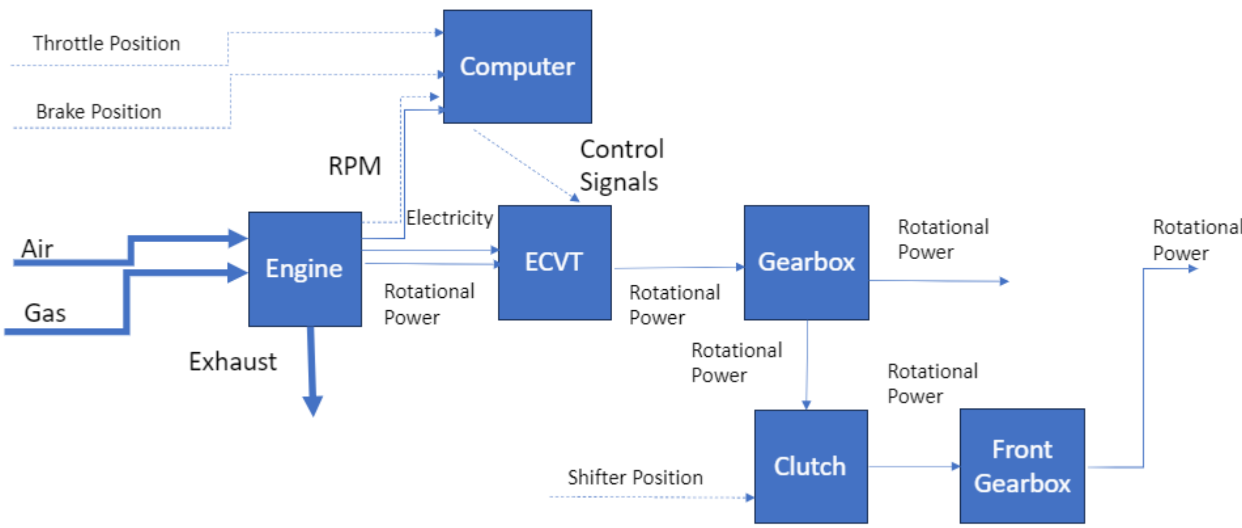


Figure 43: Drivetrain Functional Model

4.1.4 Frame

Figure 44: Frame Black Box Model

The black box model for frame design, shown in Figure 44, was used to identify the purpose of the frame in the simplest terms. Identifying the passage of materials, energy, and signals and how the inputs are affected by the design is important to the function of the vehicle. The biggest impact to the frame from an input standpoint is the load forces induced by the front and rear suspension.

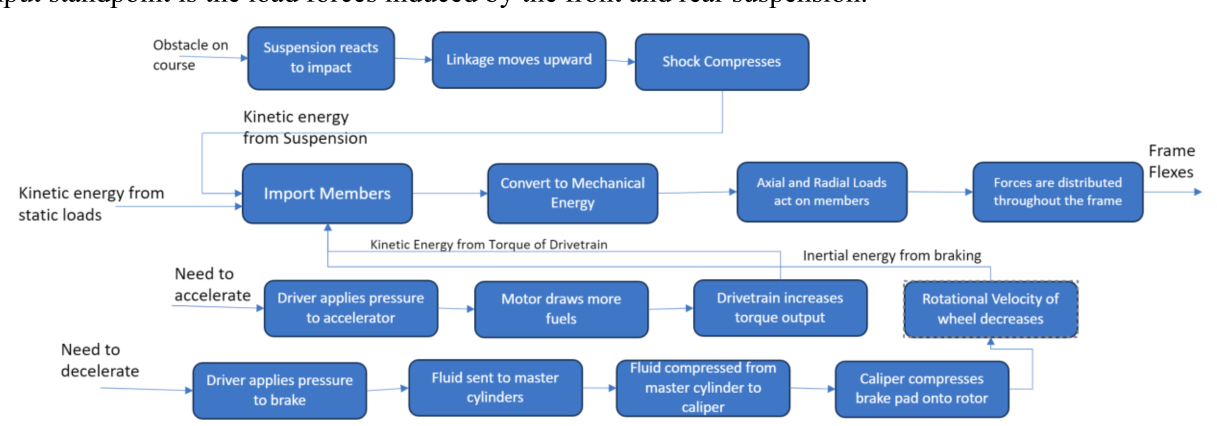


Figure 45: Frame Functional Model

The functional model for the frame sub-system was derived from the frame's role with other sub-systems. Through the other systems inputs, the frame can respond and react accordingly. This reflects the driven nature of the frame design process. Note that the only output from the frame is deflection in the members or flex in the frame. The frame is a structure and so it has very few outputs in proportion to the number of inputs. If the frame welds all meet technical requirements, then this functional model should be accurate to how the frame will function during competition.

4.2 *Concept Generation*

4.2.1 Front End

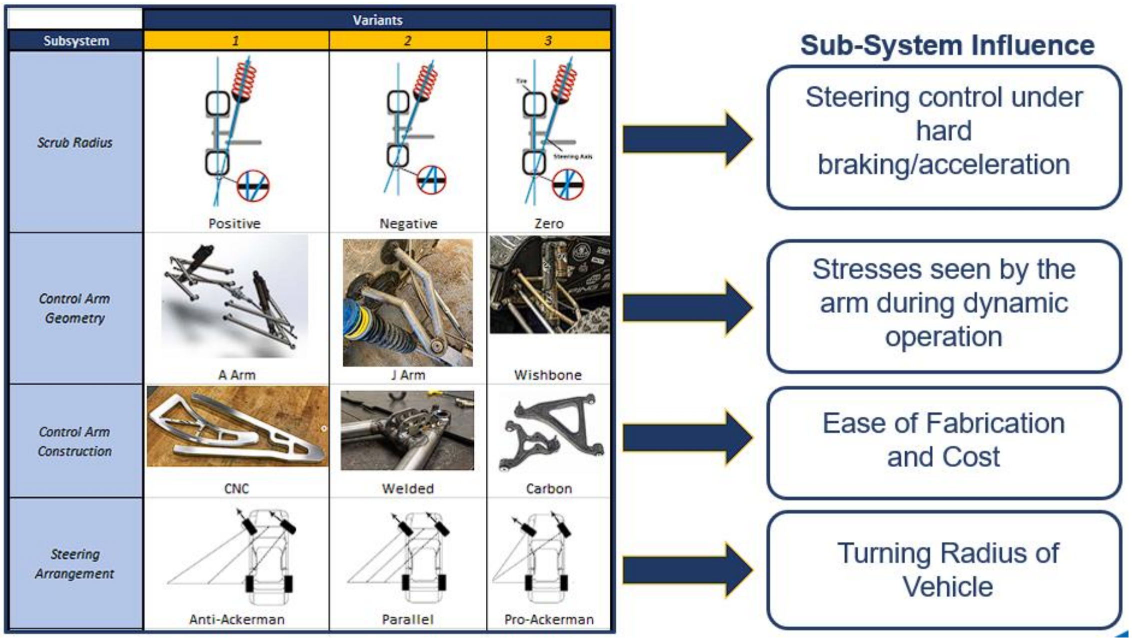


Figure 46: Front End Concept Generation 1

- *Scrub Radius*
    - Positive: Easier to geometrically achieve but worse braking performance
    - Negative: Difficult to geometrically achieve and leads to torque steer
    - Zero: Realistic to achieve and has no negative impact on handling or suspension performance
- *Control Arm Geometry*
    - A Arm: This design is easy to manufacture and is structurally rigid. This design can lead to interference issues with the shock and CV axle.
    - J Arm: This design allows for clearance with the shock to be mounted vertically. This can lead to more difficult manufacturing.
    - Wishbone: Wishbone design can allow for increased clearance with the shock. This is also more difficult to manufacture and can induce unnecessary stress.
- *Control Arm Construction*
    - CNC Aluminum: Is very lightweight and can lead to complex and adjustable design. This construction also is much more expensive and difficult.
    - Welded Construction: Is rigid and adjustable. This is simple to do with the current capabilities of the team in house. Can be heavy if done incorrectly.
    - Carbon: Visually impressive however can be very complex and pricey. Must be outsourced.

- *Steering Geometry*
    - Ackermann: Turns well in low traction, low speed situations. Induced stress on CV axles.
    - Anti-Ackermann: Turns well in high traction situations. Induced stress on CV axles.
    - Parallel: Drives well but is not optimized for performance. Easy to develop steering geometry.

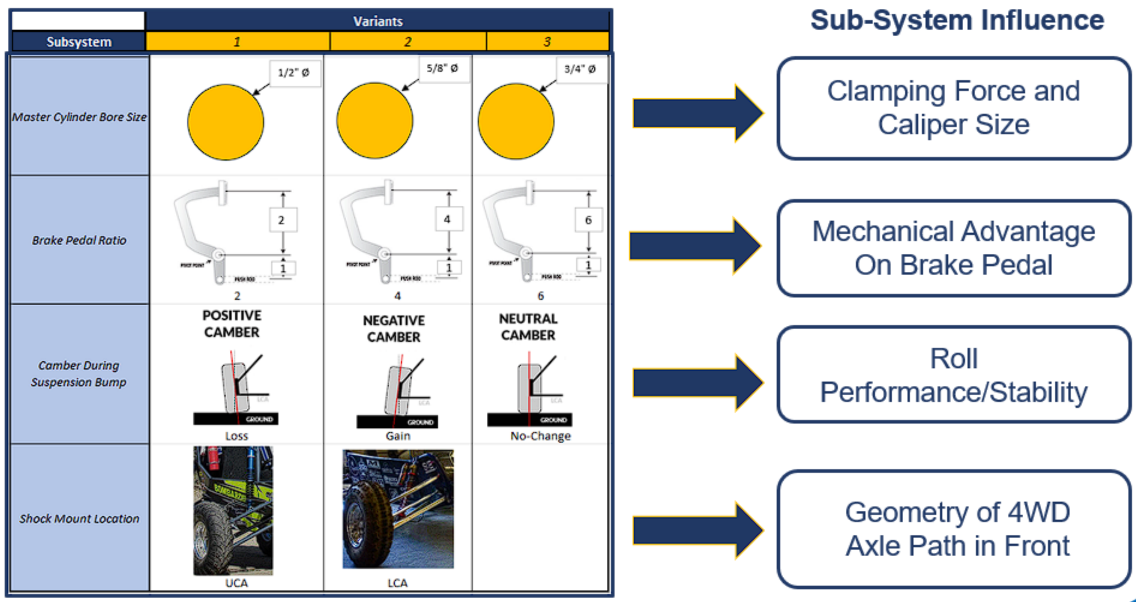


Figure 47: Front End Concept Generation 2

- *Master Cylinder Bore Size*
    - 1/2: Lowest fluid flow. Increased clamping force.
    - 5/8: Medium fluid flow. Medium clamping force.
    - 3/4: Highest fluid flow. Decreased clamping force.
- *Brake Pedal Ratio*
    - 2: Lowest Mechanical Advantage. Decreased Length.
    - 4: Medium Mechanical Advantage. Medium Length.
    - 6: Highest Mechanical Advantage. Increased Length.
- *Camber Gain During Suspension Bump*
    - Loss: Decreased traction and increased risk for vehicle roll.
    - Gain: Increased traction in a turn. Decreased risk for vehicle roll.
    - No-Change: Consistent vehicle dynamics with maximum traction.
- *Shock Mount Location*
    - Upper Control Arm: Increased stress on upper control arm, decreased interference with CV axle.
    - Lower Control Arm: Increased interference with CV axle but with better suspension tuning.

4.2.2 Rear End

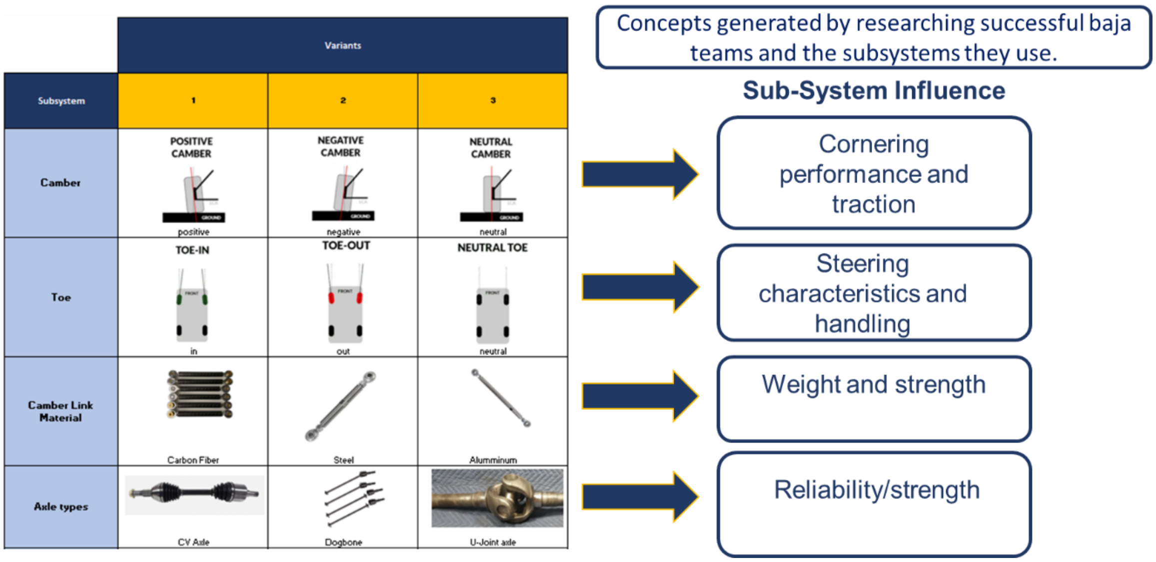


Figure 48: Rear Concept Generation Part 1

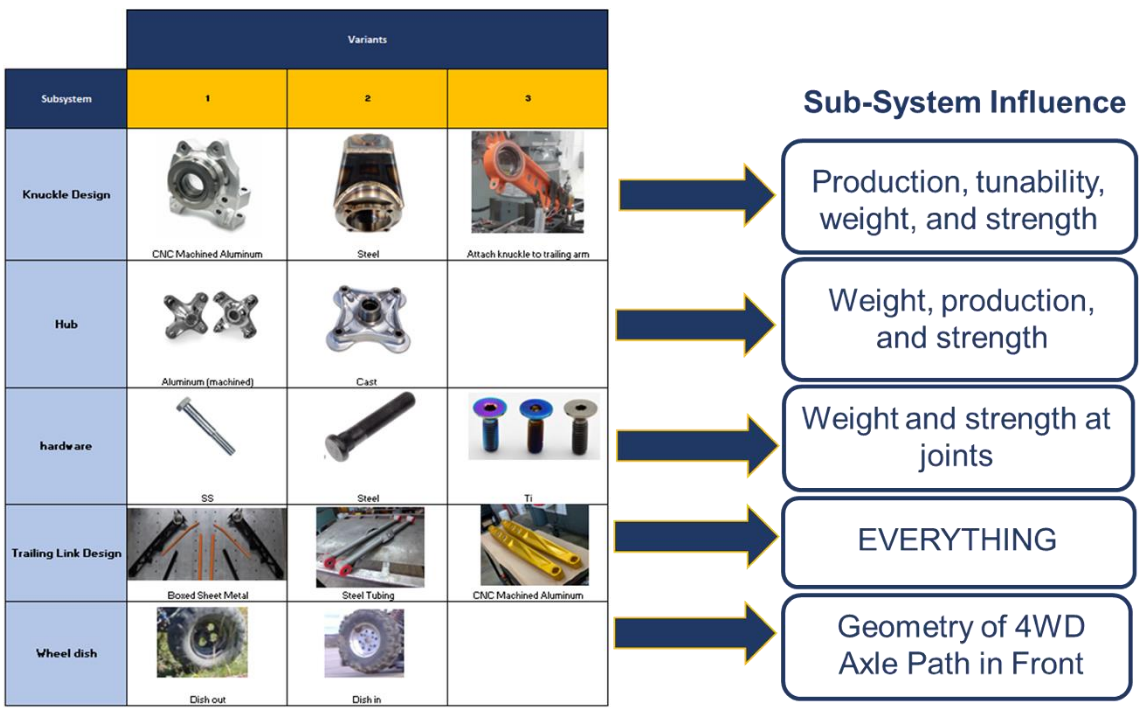


Figure 49: Rear Concept Generation Part 2

The key takeaways from the concept generation figures above are that strength, lightness, reliability, and performance are all critical to the design of the rear suspension. The process of determining which variant of each subsystem is to be used will follow be explained in detail later in the report.

4.2.3 Drivetrain

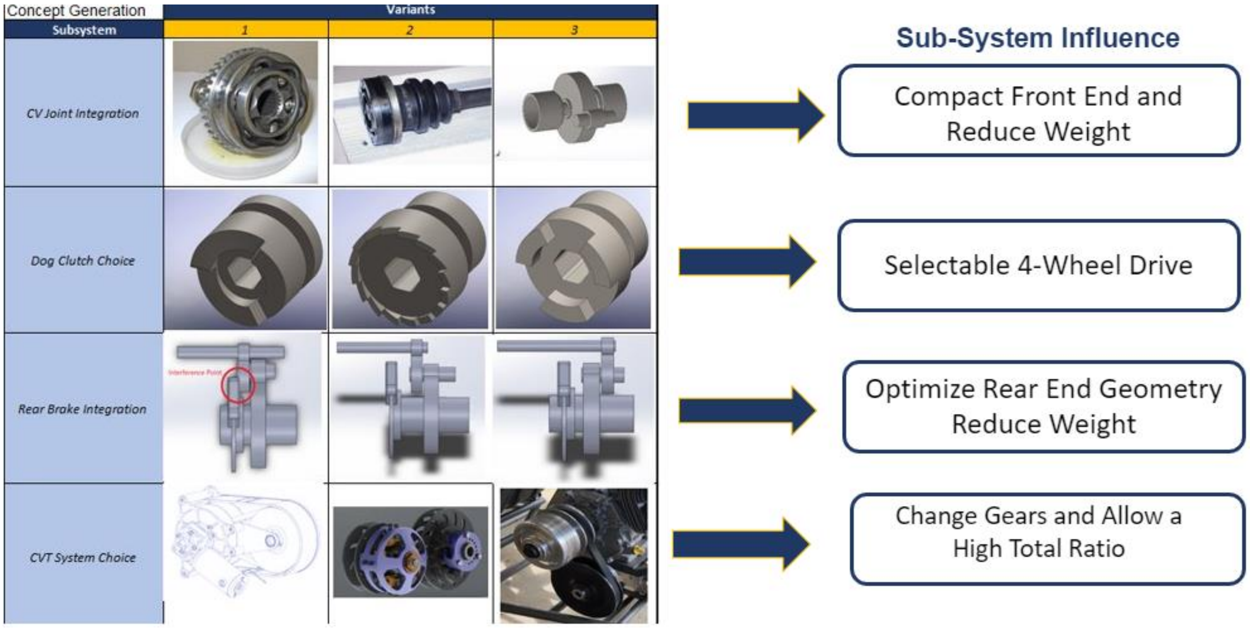


Figure 50: Drivetrain Concept Gen

In this concept generation, there are four different subsystems within the drivetrain that are presented in *Figure 50*. Each subsystem contains a different variant of the system and brief explanation how it influences the overall system of the drivetrain. These variants provide the team different design options and will be tested through simulations and calculations in section 4.3.2 of the report.

4.2.4 Frame

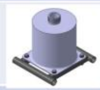
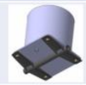
Subsystem	Variant 1	Variant 2	
Frame Type			Frame Integrity and Sub-System Integration
Fuel Tank Mounting			Time, money, effort in manufacturing and integrating
Seat Design			Seat mounting, driver safety
Side Impact Member			Overall frame length and structural integrity
Tube Material	AISI 4130	AISI 1018	Strength of material, weight, cost

Figure 52: Frame Concept Variants

There are few design aspects where the frame team has creative freedoms. Between the SAE BAJA Rules and conforming to other sub-systems, the frame is a largely driven component. However, the frame type, fuel tank mounting, seat mount design, Side Impact Member design, and tube material. In Selection Criteria, members of the Frame Team will determine the optimal variant given a number of factors using calculations and general engineering knowledge.

4.3 Selection Criteria

4.3.1 Front End

4.3.1.1 Abraham Plis

For the front end of the vehicle, suspension geometry and performance are paramount to the success of the car overall. As such, the mathematical justification presented in this sub-section is relevant to these concepts.

The first sub-system under analysis is the scrub radius, previously discussed in Section 3.1.1.1

$$F_{braking} = \frac{Weight\ of\ Car}{2} * (Coeff.Friction\ Tire\ to\ Asphalt)$$

Equation 19: Braking Force Calculation

$$T_{wheel} = F_{braking} * D_{Moment\ Arm}$$

Equation 20: Wheel Torque, Toe Orientation

In conjunction with these equations, some physical characteristics can be assumed about the scenario as well: 550 lb. total car weight, 0.7 coefficient of friction between tire and asphalt, and a 4-inch moment arm (see Figure 53). This scenario produces unwanted torque of 64 lbf*ft about the steering axis of the tire when a positive/negative scrub radius is present. This will compromise the handling characteristics of the car, making these variants undesirable. Instead, a zero-scrub radius is selected as the ideal variant for this sub-system because it mitigates the influence on the steering of the car and leads to better handling for the driver.

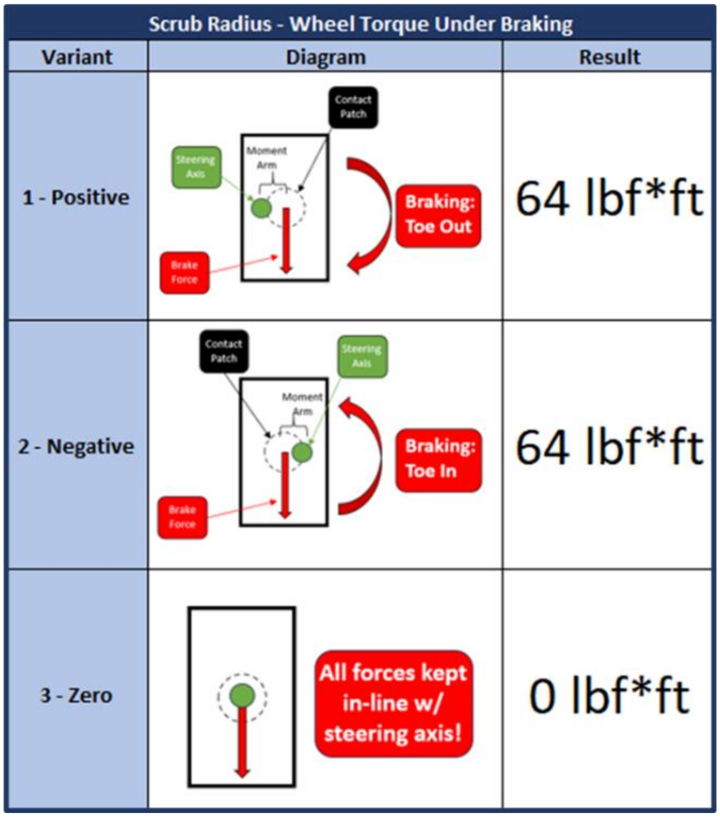


Figure 53: Scrub Radius Justification

The next sub-system under analysis is the behavior of the front suspension camber during a suspension bump event. The term "camber" refers to the angle that the tires make relative to the ground from a front-on perspective (negative camber angles the tires in, vice versa for positive). Another relevant term is "camber gain" which refers to how much negative camber the wheels gain during a compression of the shocks (bump event). When the car takes a fast corner, the weight is shifted to the outside wheel, which places that side of the car in a compression event. To maintain proper tire contact with the ground, the wheels must stay as upright as possible, even when that side of the car is being compressed and tilted over. To promote wheel verticality during a roll-induced compression, camber gain must be designed into the system. Camber gain will ensure that the outside wheel stays vertical and maintains adequate contact with the ground; camber loss and lack of camber gain/loss will lead to undesirable performance during a roll and cause handling issues for the driver. Please see Figure 54 for an annotated visualization of each variant for added reference.

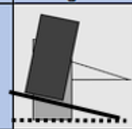
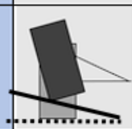
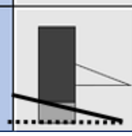
Camber During Bump		
Variant	**Diagram**	**Result**
1 - Gain		Better Roll Performance
2 - Loss		Worse Roll Performance
3 - No Change		Subpar Roll Performance

Figure 54: Camber Performance During Bump

The last sub-system under analysis is the mounting location of the front shocks. Since the car will be running a double A-arm suspension system in the front, the shock can be mounted to either to upper control arm or the lower control arm. With the car being 4WD capable, there will also be a cv axle that is positioned between the two control arms and runs from the inboard portion of the vehicle to the centerline of the wheel. With the two variants being proposed, a graphical illustration will help to visualize the glaring issue with the latter option (see Figure 55).

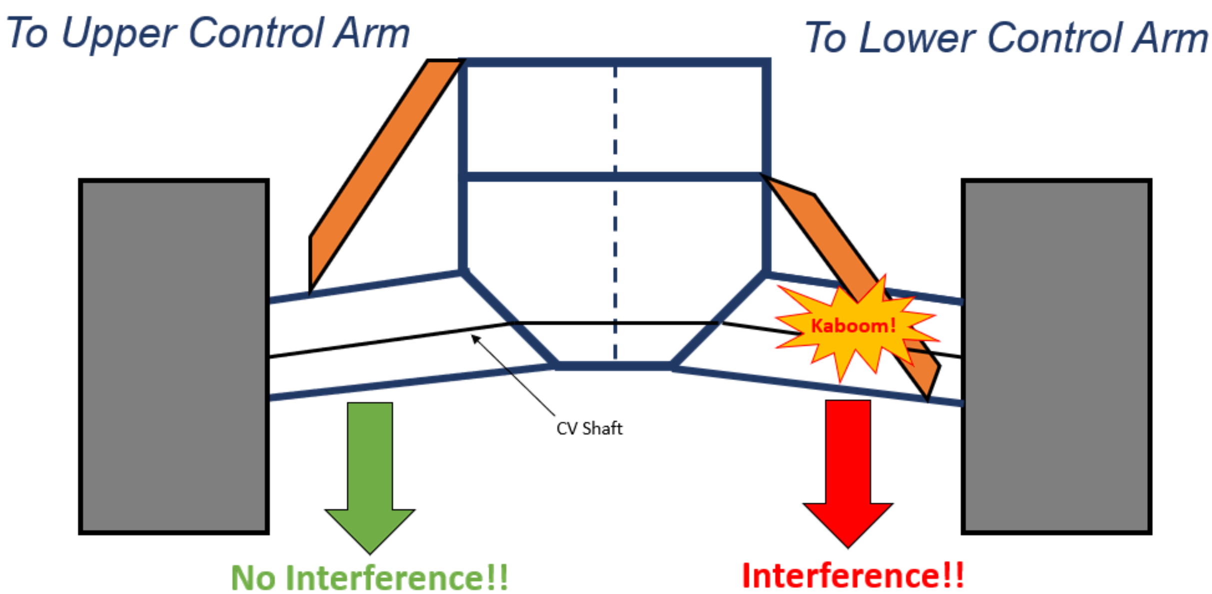


Figure 55: Shock Mount Location

As illustrated, running the shock to the lower control arm will create geometric interference with the cv axle if oriented in a traditional manner. To make this mounting location work, the geometry of the lower control arm will have to be changed in such a manner that unwanted bending forces will be introduced and clearance will be an issue. In contrast, mounting the shock to the upper control arm yields no geometric interference with the cv axle and allows for a traditionally designed, straight-member A-arm. This makes the upper control arm the desirable variant for the shock mounting location sub-system.

4.3.1.2 Bryce Fennell

Analysis was performed on the upper control arm using SolidWorks Finite Element Analysis (FEA) to determine the max stress and the expected deformation of the arm under a maximum force compression. For the theoretical compression a vehicle and driver weight of 550 lbs. was used with a vehicle fall height of 3 feet. This theorized maximum compression event was created as a worst-case scenario to ensure the upper control arms of the vehicle can withstand the force. In the figure below, the left-hand image depicts the anticipated internal developed stress in the upper A control arm while the right hand image depicts the anticipated deformation of the control arm under such load.

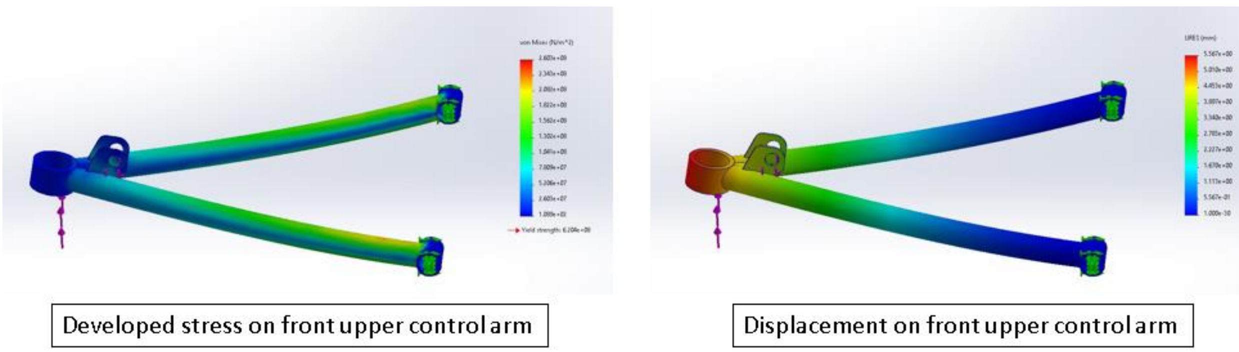


Figure 56: Left: Developed internal Stress in Upper Control Arm. Right: Control Arm displacement under Force

Sections highlighted in green and yellow show areas of increased internal stress for the left image while sections highlighted in red and yellow depict areas of greatest deformation for the right image. After performing an in-depth analysis of the upper A control arm, a factor of safety (FOS) of 2.38 was achieved indicating the current design is strong enough to be used without additional bracing and could be lighter should a thinner walled tube be used.

The next analysis was performed on the brake pedal length and the relationship between the brake pedal center mount, the brake pedal pivot point, and the master cylinder mount. A simple SolidWorks drawing of the anticipated pedal dimensions can be seen in Figure 57 below. After testing the desired amount of brake pedal throw (the distance the pedal actuates to apply full braking force from zero brake force) for the vehicle's operator, a throw of 6 inches was deemed as optimal.

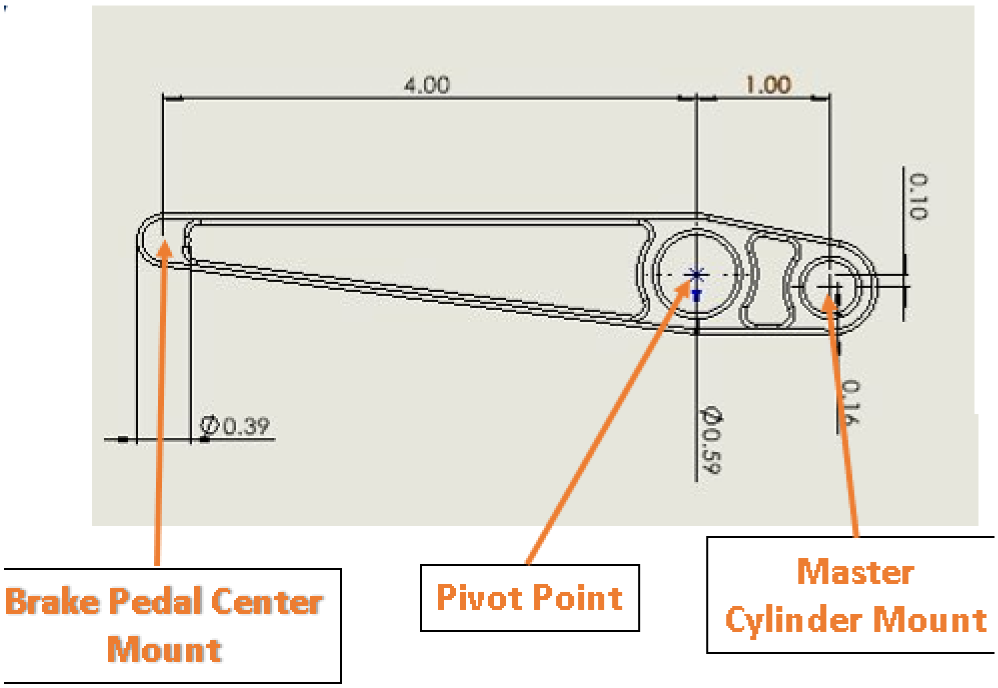


Figure 57: SolidWorks Drawing of Anticipated Pedal Dimensions

With the knowledge of 6-inches of pedal throw and a desired system operating pressure of 140psi, a MATLAB script was written to determine the pedal length versus pedal throw and the pedal length versus system pressure. Outputs from this script can be seen in the figure below:

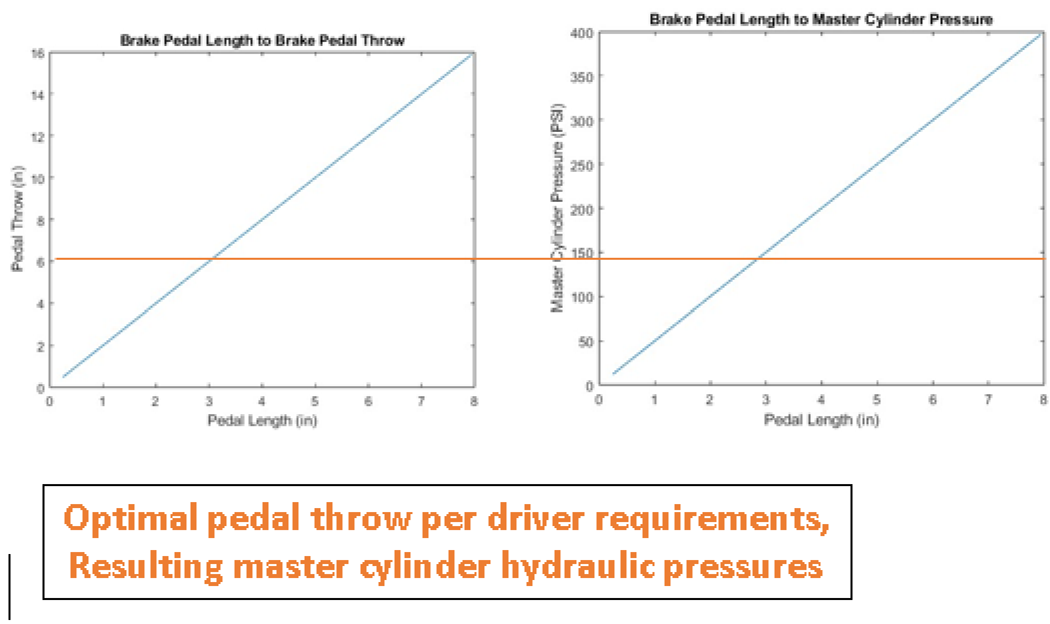


Figure 58: Left: Brake Pedal Length vs Brake Pedal Throw. Right: Brake Pedal Length vs System Pressure

The results from the MATLAB figures above indicate a pedal length of 3 inches from the pivot point results in an optimal system pressure of 142psi with a pedal throw of 6.1 inches. Further optimization work can be done to reduce the pedal throw to below 6 inches while maintaining a system pressure above 140psi, but the current design will be sufficient for production and meets the requirements for both the driver and SAE competition.

The final analysis was performed to choose the correct bore of the master cylinder. As the internal bore of the master cylinder is increased more fluid can flow per distance the piston is compressed; however, the fluid pressure of the system will be reduced as a result of the greater internal diameter. It is important when sizing a brake system to ensure enough fluid is moved through the system to actuate the caliper while also ensuring the system pressure is sufficient to bring the vehicle to a stop quickly. For the SAE competition the braking requirement states the vehicle must be able to lock up all 4 wheels while driving on dry asphalt.

To calculate the brake master cylinder bore size needed a MATLAB script was written which takes differing master cylinder bore sizes as an input and outputs the anticipated torque the brake calipers can exert on the brake rotor disk at the wheels. Engineering assumptions were made to achieve a wheel brake torque such as the use of 6-inch brake rotors, a .4 coefficient of friction between the brake pads and rotor disk, zero loss in pressure throughout the system, and a brake caliper piston area of 1.23 inches. Results from this MATLAB script can be seen in figure below.

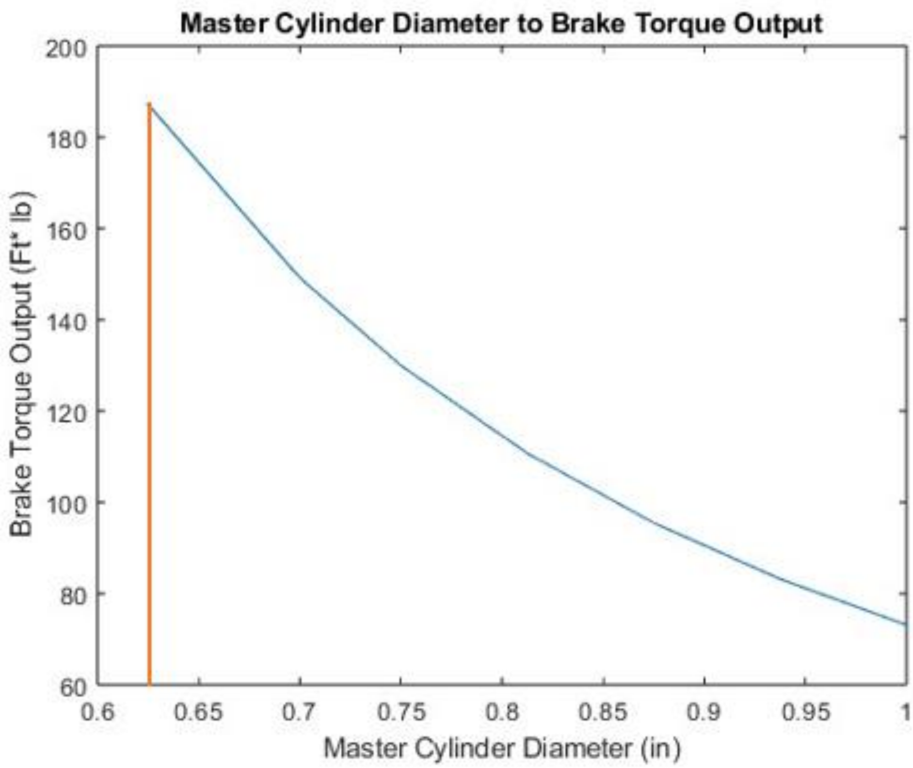


Figure 59: Master Cylinder Bore Size Versus Brake Torque Output at The Wheel

Analyzing the results from the MATLAB script indicate a minimum bore size will result in the greatest system pressure. Sourcing master cylinders from Tilton we will be using a 5/8-inch master cylinder inner bore resulting in 190 ft*lb. of torque at the wheel.

4.3.1.3 Evan Kamp

With weight and strength being vital in the production of a Baja vehicle, the primary concern for each on the front end is the Upper and Lower control arms. Because of this concern the team analyzed both Welded and CNC control arm construction. Each were analyzed based on their Material Cost, design time, manufacturing time, and potential other benefits. The team had also included Carbon in its concept generation, however due to its immense cost and manufacturing difficulty it was not included within this analysis.

Table 11: Control Arm Construction

Type of Control Arm	Material Cost	Design Time	Manufacturing Time	Other Benefits
Welded 	4'x4 control arms 1"OD with 1/16" ID Using 4130 Steel **$40x4 = $160**	1 hour CAD for both the upper and lower control arm. 10 minutes to mirror to passenger side **TOTAL 2hr 10min**	1 hour of jigging with 45 minutes of welding **TOTAL 1hr 45min**	Additional Adjustability with welded
CNC Aluminum 	2'x1'x2" billet for each Control Arm **$600x4 = $2400**	4 hour CAD for both upper and lower control arm. 10 minutes to mirror to passenger side **TOTAL 8hr 10min**	2 hours of Programming 2 hours of Machining **TOTAL 4hr**	If done correctly could be Lightweight

Due to the concern of fundraising for this project, the increased cost to CNC all four control arms is considerable. Manufacturing time is also a big consideration when accounting to produce the control arms. In the end, the benefits of CNC control arms simply do not outweigh the cost and thus the team will be moving forward with welded control arms.

Due to the inherent benefits of using Ackermann steering within SAE Baja, Ackermann Steering was decided to be the decided form of steering for the team. Adjusted steering calculations and radius predictions can be made with the data developed within Lotus Shark software by the team. The figure below displays the percent Ackermann generated by the team's front end geometry.

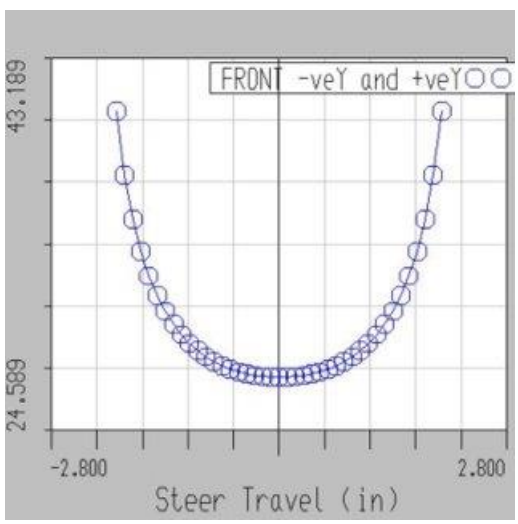


Figure 60: Percent Ackerman

Using this adjusted percent Ackermann, a new steering radius prediction can be made using adjusted geometry for the team. This is shown in the table below.

Table 12: Preliminary Steering Measurements via Shark

Inner wheel angle (δi)	50°
Outer wheel angle (δo)	28.4°
δavg	39.2°
Rear wheel to center of gravity (a2)	32
Percent Ackerman Used	43.189%
Projected Turning Radius (R)	6.93ft

4.3.2 Rear End

4.3.2.1 Seth DeLuca

This subsystem being looked at first is camber angle. Camber is the angle of the tire with the ground (If the wheel is 90 degrees to the ground, then the camber angle is 0 degrees). The camber was analyzed and calculated using Lotus SHARK software which need points the team uploaded from SolidWorks. The system can be seen below:

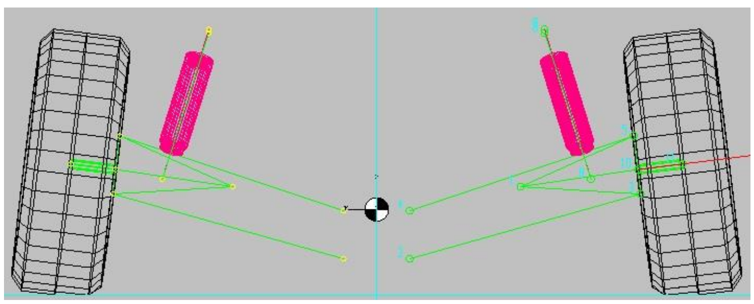


Figure 61: This is the suspension system under full compression.

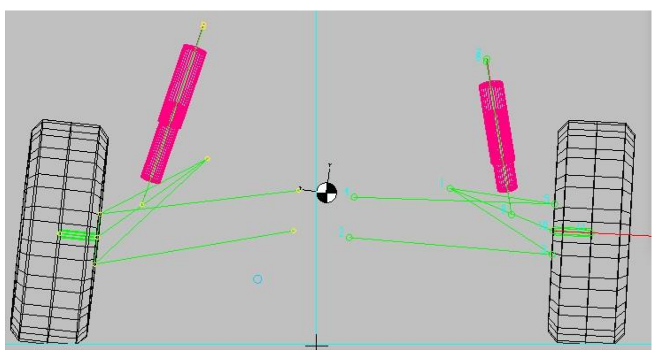


Figure 62: This is the system under full roll or cornering.

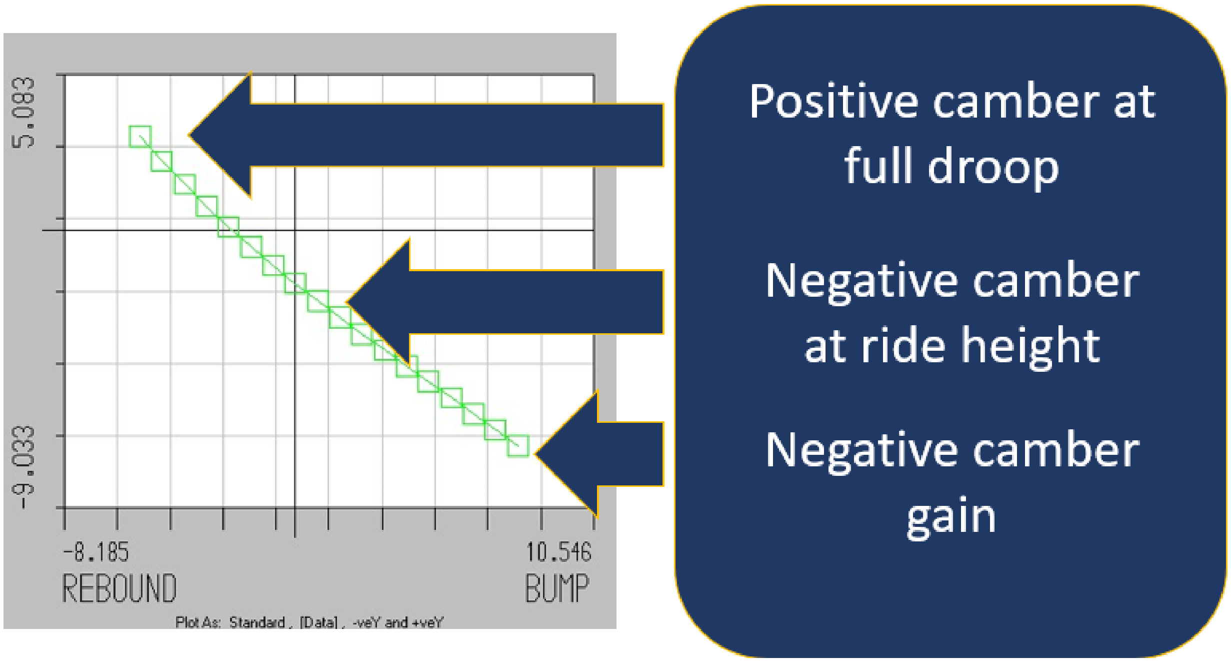


Figure 63: Graph of the camber angle at different points of travel. Listed are main takeaways from the graph.

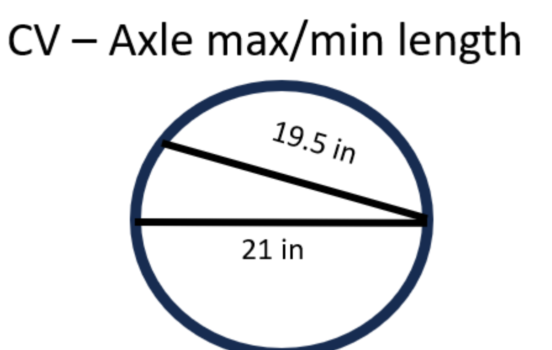
This analysis in SHARK allowed the team to visualize the system designed in SolidWorks. This was an iterative process just because it is hard to make changes to the design in SHARK. The team would decide to make changes to affect the camber being output in SolidWorks, then the team needed to convert this point to Lotus SHARK's coordinate system. The team wanted positive camber at full droop because this allows there to be negative camber at ride height. Negative camber at ride height allows for maximum traction when taking corners and turns. This analysis was critical to the verification that our design will work effectively while keeping prices and weight to a minimum.

The next subsystem looked at was the different axle types available for the suspension. There was some overlap here with Drivetrain, however this analysis shows what would be important for Rear suspension. There are two basic axle types, a universal joint (U-joint) or a constant-velocity axle (CV) [95]. The benefits and weaknesses of both can be summarized in the table below:

Table 13: Pros and cons of different axle types.

Axle type	Pro	Con
Universal Joint (U-Joint)	-Easy to replace -Allows angle change	-Acts as a suspension member -More stress on drivetrain's subsystems -Rougher ride -Spline- Very expensive to buy and can't manufacture at machine shop.
Constant-Velocity axle (CV)	-Allows angle change -Changes length at different points in travel (plunges) -Cheaper	-Hard to replace

The team picked the rear-end suspension system of a trailing arm with two camber links. This means the team needs the axle to induce plunging throughout the wheel travel. This ruled out U-joints from being utilized not to mention the added cost of U-joints. To find the amount of plunge needed to allow for travel of the suspension to not be affected, a simple analysis was conducted below:



Change in length needed for suspension geometry	Change in length of CV Axle	Change in U-joint (Non–spline)
1.5 in	1-3 in	0 in

Figure 64: Analysis of the maximum and minimum length of the axle based on SolidWorks measurements.

This analysis validates the use of CV axles for the teams' design thus far. Moving forward the teams will continue to validate decisions through the lens of calculations and thorough analysis.

A first version of a hub was developed on SolidWorks. This gave an idea of measurements necessary to create a hub. After this was completed a FEA on the hub was done utilizing 6061 Alloy Aluminum, selected due to strength, market availability, and cost. The stress analysis can be seen below:

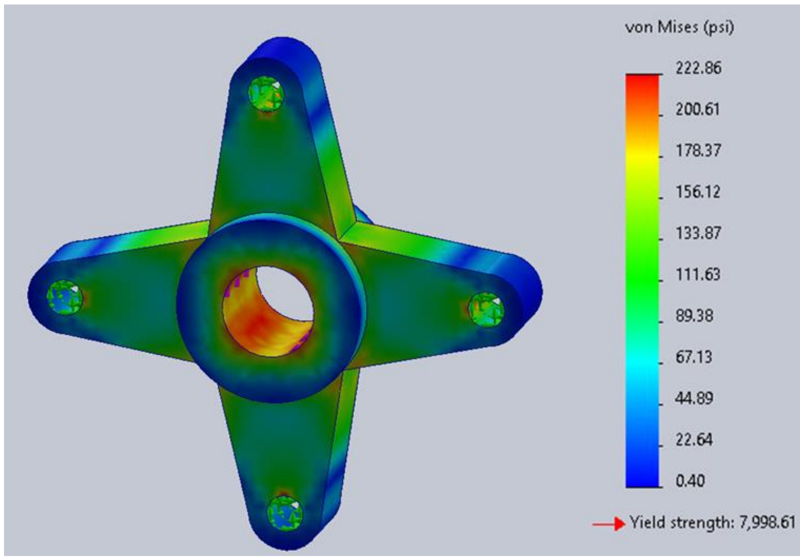


*Figure 65: Stress analysis of the hub based on applying a torque of 1500 lb.*in to the center and fixing the 4 bolt holes.*

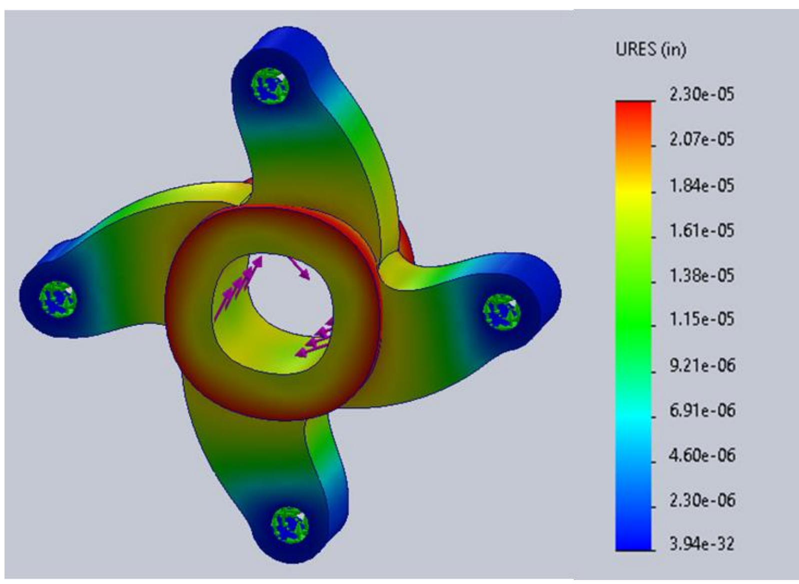


*Figure 66: Displacement analysis of the hub based on applying a torque of 1500 lb.*in to the center and fixing the 4 bolt holes.*

This analysis showed that the stresses above are far less than the materials yield strength. This highlights the opportunity to do some material subtraction which would decrease weight while not impacting strength as long as the team maintains stress values lower than the yield strength.

4.3.2.2 Lars Jensen

After the team developed the necessary hard points for the rear suspension based on mathematical modeling, we were able to run the geometry through SHARK. The final product from shark is shown in Figure 67 and Figure 68 below. This took a few iterations to get right but the sub team is happy with the results. The goal was to keep toe angle between zero and two degrees throughout the suspension cycle. With this design we were able to achieve a toe angle change of around 1.6 degrees which is within the goal of two degrees. The plot in Figure 68 shows the toe angle measurement at different part of the travel between max bump and rebound.

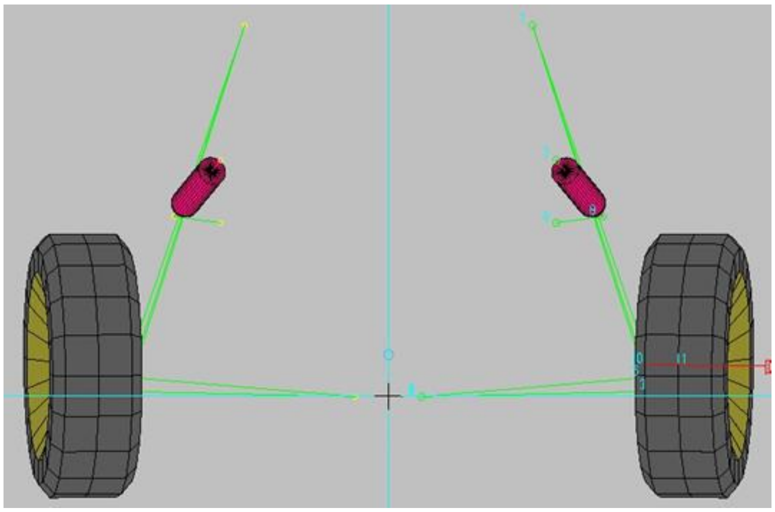


Figure 67: SHARK Top View

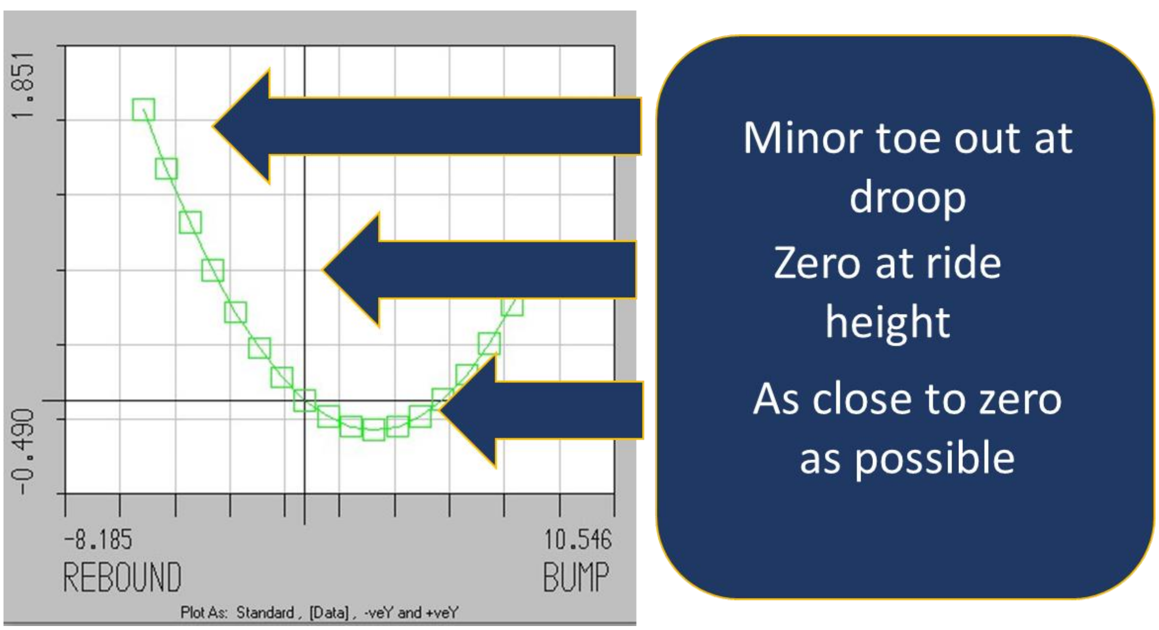


Figure 68: SHARK Toe Analysis

The knuckle is a crucial part of the rear suspension and is responsible for holding everything together. This is why we performed an analysis on the welds holding the camber link tabs to the knuckle. This analysis used measurements from the current design and assumptions for the loads these mounts will experience. Using Equation 21 and Equation 22 it was determined that the welds for these tabs will experience a normal stress of 1,270.59 psi and a shear stress of 35.29 psi. This will be accommodated for in the final design.

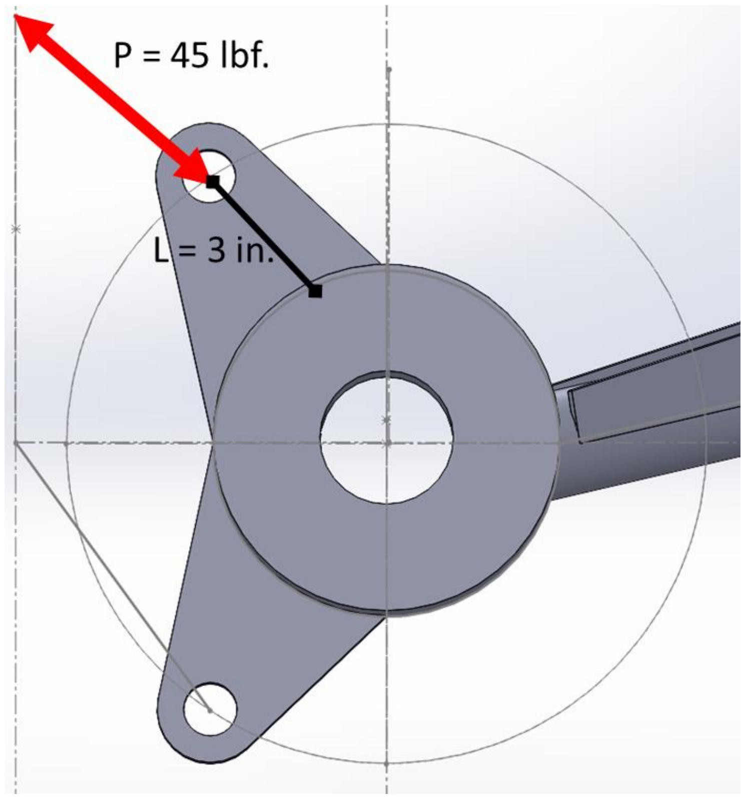


Figure 69: Knuckle Design

- σ=$Normal\ Stress, psi$
- τ=$Shear\ Stress, psi$
- P=$External\ Applied\ Load, lbf$
- L=$Linear\ Distance, in$
- h=$Size\ of\ Weld, in$
- l=$length\ of\ Weld, in$

$$\sigma_b = \frac{6PL}{lh^2} = \mathbf{1,270.59}\ \boldsymbol{psi}$$

Equation 21: Weld Normal Stress

$$\tau = \frac{P}{lh} = \mathbf{35.29}\ \boldsymbol{psi}$$

Equation 22: Weld Shear Stress

The trailing link for this suspension design is very long compared to the vehicle meaning it will experience some serious amounts of force when going through the travel. The goal for this calculation was to see the shear and bending experienced by the trailing link if the baja car landed on one rear wheel and the shock was bottomed out. The diagrams for this calculation can be seen below in Figure 70 and Figure 71. The maximum bending moment that the trailing link will experience is equal to 7,136.87 lbf-in. which will be considered in the final design.

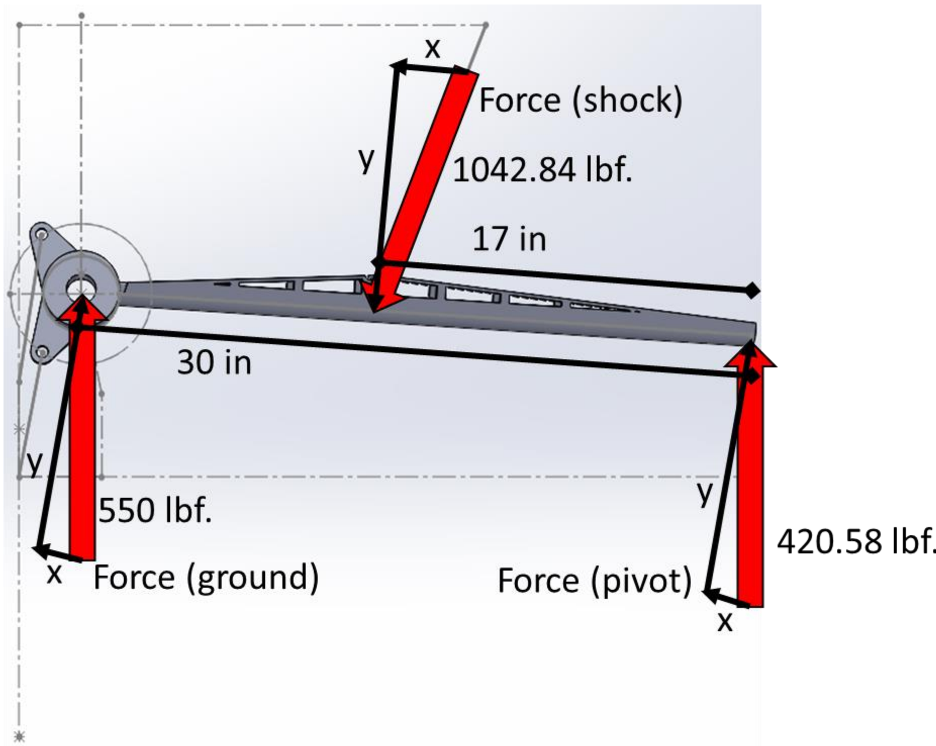


Figure 70: Trailing Link Design

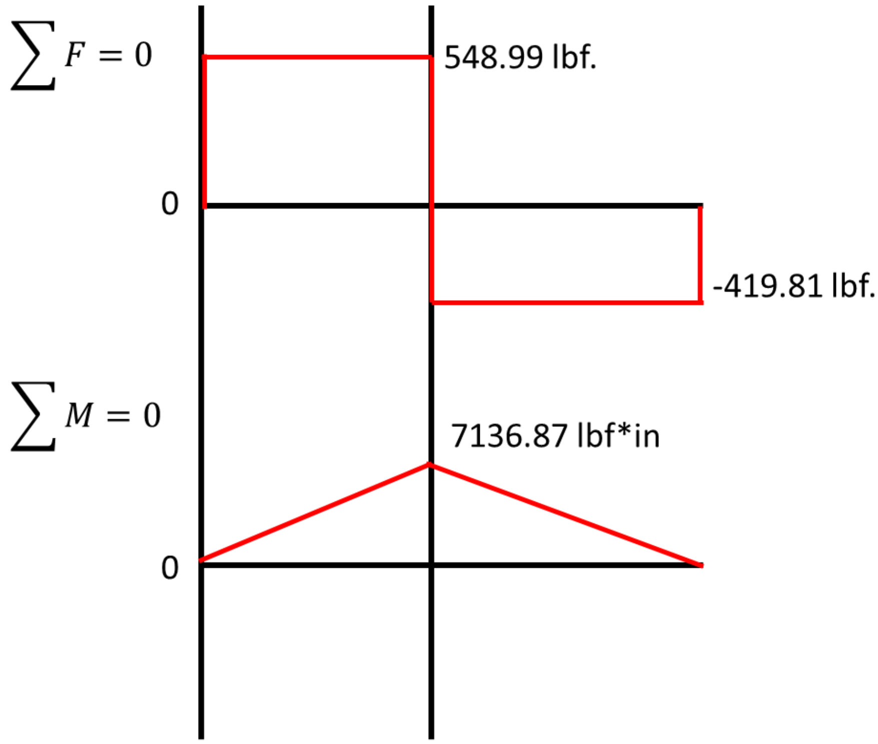
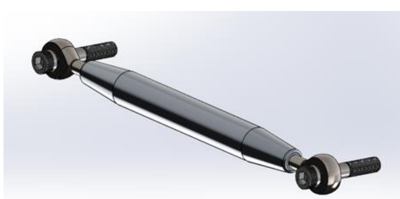
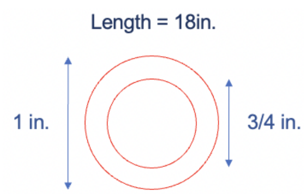


Figure 71: Shear and Moment Diagram

4.3.2.3 Joey Barta





Assumptions
- Uniform Cross-Section
- Analysis under load (compression)
- No affect from ball joints and screw connections

Aluminum Camber Link – tapered tube

$$Q_{max} = \frac{2}{3}(r_o^3 - r_i^3) \qquad \tau_{max} = \frac{4V}{3A}\left(\frac{r_o^2 + r_o\, r_i + r_i^2}{r_o^2 + r_i^2}\right) = \frac{4\times 45lbf}{3\times 0.3436in^2}(1.48in^2) \gg 258.44psi$$

$$S_Y for\ 6063 - T6\ Aluminum = 31{,}118\ psi \qquad FoS = \frac{31{,}118}{258.44} = 120.4$$

By performing a classical calculation on the first iterative design of the rear camber link, a factor of safety of 120.4 was found. This was found by assuming a uniform cross-section, a steady state compression of the suspension at full vehicle weight, no affect from ball joints or screws, and a wall thickness of 1/8 inch. These values are bound to be changed upon further research, which will lower the factor of safety. With this said, the value is still very high and it may be in the interest of weight saving to consider a carbon-fiber upper camber link

Other Possible Options:

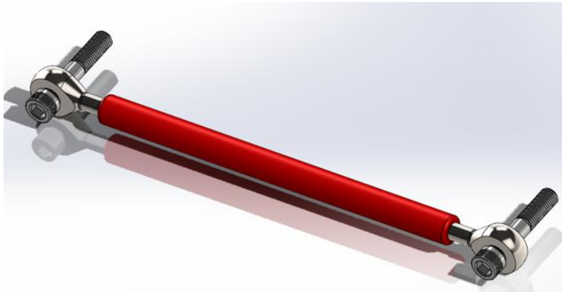


Figure 72: Carbon-Fiber Camber Link

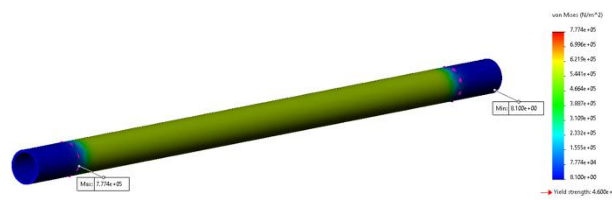


Figure 73: Steel Tube FEA

Carbon Fiber Camber Link – Hollow uniform cross-section tube with 7075 T6 machined aluminum insert

$S_{Ut} for\ Carbon\ Fiber = 650{,}000\ psi$

Note: Carbon Fiber has varying strength in different axis' making it difficult to analyze/compare

Steel Camber Link –uniform cross-section tube

$S_Y for\ 4130\ Steel = 66{,}717\ psi$

Exploring other options for camber link materials, carbon fiber tubing and 4130 steel (same size as that of which is used on the secondary members on the frame). Looking at the carbon fiber model, it was difficult to pinpoint standard and reliable properties for the material, as it's not listed in SolidWorks and different sources have different standards. This is because carbon fiber has varying tensile and compressive strength in different axis' since it is a textile soaked in epoxy. Because of this, in tension, it has stronger properties that aluminum and most steel, and in compression, it can be incredibly brittle and buckle under load.

Going forward, carbon-fiber will be explored as the primary option for upper links where aluminum and steel will be considered for lower links to protect against debris and crashes. Steel and carbon-fiber links will be constructed in similar ways by use of a tube (of specified material) and an insert that will be glued or welded to the tube to create a strong bond for the rod ends to thread into. The aluminum link would be fabricated by a tube with opposite threads tapped on either end. After, excess material will be milled off to lighten the design.

18-8 Stainless Steel

- $S_Y = 31{,}200\ psi$
- $S_{UT} = 73{,}200\ psi$
- $\rho = 0.2890\ lb/in^3$
- **$7.58 per screw**
- **Corrosion Resistant**

Figure 74: Stainless Steel Screw.

Grade 2 Titanium

- $S_Y = 39{,}900 - 59{,}500\ psi$
- $S_{UT} = 49{,}900 psi$
- **40% Lighter than Carbon Steel**
- $\rho = 0.1629\ lb/in^3$
- **Expensive!**
- **$54.54 per screw**

Figure *75:* Titanium Screw.

Black-Oxide Alloy Steel

- $S_{UT} = 90{,}000\ psi$
- **Tensile = 170,000 psi**
- **Mass density = 0.2782 lb/in^3**
- **Cheap!**
- **$1.82 per screw**
- **Many size options**

Figure *76:* Black-Oxide Steel Screw

As shown above, black-oxide steel screws are the most effective and will be used as the screw of choice moving forward. With that said, if the team obtained a sponsorship with a titanium fastener supplier, it would be a great alternative to steel.

4.3.3 Drivetrain

4.3.3.1 Jarett Berger

After determining the correct gear ratio for the front gearbox, the next step is to calculate the bearing reaction forces for both the input and output shafts. To calculate the radial load on the shaft, it was first important to calculate the reaction forces in both x and y directions. Additionally, the equations from Shigley's Mechanical Engineering Design [22] was used determine bearing life and the catalog load rating to help the bearing selection process. These calculations were performed in MATLAB and equation are shown below:

```
H = 11.45; % Horsepower (HP)
N = 3600; % Desired speed (rev/min)
G1 = 17; % Number of teeth
G2 = 65; % Number of teeth
D = 12; % Diametral Pitch (teeth/in)
p = 20; % Pressure Angle
DP1 = G1/D; % Diametral Pitch for Gear 1
DP2 = G2/D; % Diametral Pitch for Gear 2
Ti = (550*H*60)/(N*2*pi)*12; % Input Torque (lb*in)

Rx = Ti/(DP2/2); % Reaction force in x direction
Ry = Rx*tan(p); % Reaction force in y direction
R = sqrt(Rx^2+Ry^2); % Resultant Bearing Reaction Force
disp(R) % Displays Bearing Reaction Force

Fr = 39.3822; % Bearing Reaction Force
a = 1/3; % Bearing load life
LD = 1000; % Desired design life (hours)
LR = 10^6; % Rating life (hours)

L = LD*N*60; % Desing life
C = Fr(L/(LR)^a); % Catalog Load Rating Equation
disp(C) % Displays Catalog Load Rating
```

Figure 77: MATLAB Calculation

$$C_{10} = F_R = F_D \left(\frac{L_D}{L_R}\right)^{1/a} = F_D \left(\frac{\mathscr{L}_D n_D 60}{\mathscr{L}_R n_R 60}\right)^{1/a}$$

Equation 23: Catalog Load Rating

After preforming these calculations, it was determined that a larger diameter bearing would need to withstand the forces acting on the shaft. Additionally, integrating a sprag clutch, a one way bearing that can take higher loads, into the CV joint would not alter the bearing selection process due to the sprag clutch having a large width.

4.3.3.2 Donovan Parker

A lot of research went into creating both the front and rear end pulleys that matched the HTD tooth profile of the belt and the pitch of the belt as well. Below you can see the equation used to generate the number of teeth for the pulley that is 3 inches in diameter and in the figure below that you can see the parametric design table that takes some of the values from the variables in the equation function and apply them to the actual designs. Both tables used are functions in SolidWorks, and the make up of the designs in the last figure below consist of initializing all of the variables and equations we need in the equation function, in which they contribute to completing the very first design. Next to use the parametric design table SolidWorks opens excel and allows you to write in another configuration and make any changes you see fit. Although it cannot be seen each of the cells under the configuration “4.5 in pulley” have equations within them that use the same equations and variables the first configuration used except for the fact that adjust are made to accommodate for the ratio between to the two pulleys, thus generating a new pulley with the right ratio, number of teeth, correct pitch, etc.

1	"p"	= 0.314961in	0.314961in	
2	"PD"	= 3.14961in	3.149610in	
3	"n"	= ("PD" * pi) / "p"	31.415927in	
4	"d"	= 3in	3.000000in	
5	"w"	= 0.7874in	0.787400in	
6	"D1@Sketch1"	= "d"	3in	
7	"D1@Boss-Extrude1"	= "w"	0.7874in	
8	"D1@Sketch3"	= "d"	3in	
9	"D1@Sketch7"	= "d"	3in	

Figure 78: Design Equations in SolidWorks

Design Table for: Official Pulley Design

	D1@Sketch1	D1@Boss-Extrude1	D1@Sketch2	D3@CirPattern1	D1@CirPattern1	$LIBRARY:MATERIAL@Official Pulley Design
3in Clutch Pulley	="d"	="w"	1.5	360	30	SOLIDWORKS Materials:AISI 4130 Steel, annealed at 865C
4.5 in pulley	4.5	0.7874	2.25	360	46	SOLIDWORKS Materials:AISI 4130 Steel, annealed at 865C

Figure 79: Parametric Design Table in SolidWorks

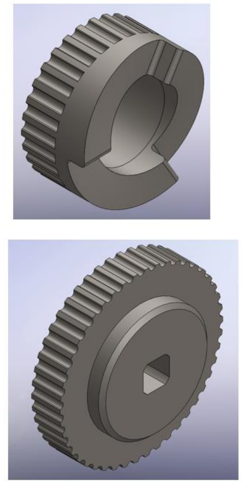


Figure 80: Resulting Pulleys from design equations

4.3.3.3 Ryan Fitzpatrick

The calculation that was performed for the gearbox design alternatives selection was the optimization of the brake integration. The initial gearbox design interfered with the mounting of the brake components to the gearbox, so a solution needed to be found. To fix this problem, two design alternatives were created. The first alternative, Design B in the middle of Figure 81, was to extend the gearbox housing at the cost of increasing the weight of the casing around the gear train. The second alternative Design C to the right in Figure 81, was to flip the first stage of the gearbox to the passenger side of the vehicle which would allow material to be saved from the gearbox housing design but increase the length of the input shafts of both the rear and front gearboxes. This increase in length of the shafts also came at the cost of increased weight of both of these shafts. The calculation to determine which of these designs to move forward with was seeing which of these designs saved more weight compared to the initial gearbox design, Design A to the left in Figure 81. The equations used in the calculations are pictured below in Figure 82. To be able to evaluate the weight of each quickly and efficiently, I wrote a MATLAB script that would calculate the weight savings of Design C compared to Design B. The MATLAB script as well as the results are pictured below in Figure 83. The result of the calculation was that Design C resulted in a net weight reduction of -0.2205 pounds. This means that Design C (flipping stage 1 to passenger side of vehicle) results in a weight GAIN of 0.2205 pounds. Design B eliminates the brake caliper interference from Design A and weighs less than Design C with less design alterations to the initial design.

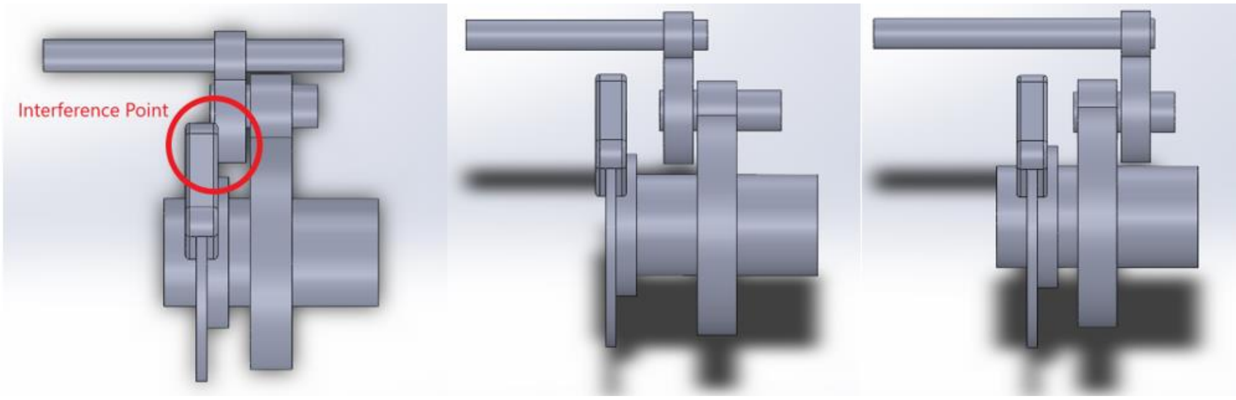


Figure 81: Gearbox Design Alternatives

Equations:

$$Weight = V_{component} * \rho_{material}$$

$$\text{Where...} V = Volume\ (in^3)$$

$$\rho = density\ (\frac{lb}{in^3})$$

$$\rho_{steel,4140} = 0.285\ \frac{lb}{in^3}$$

$$\rho_{aluminium,6061\ T6} = 0.0975\ \frac{lb}{in^3}$$

Figure 82: Gearbox Design Alternatives Equations

```
d_gearbox = gear5_pitch_diameter ; % Depth of gearbox casing at point of
 interest (in)
t_gearbox = 0.125 ; % Thickness of gearbox casing
L_reduction = (0.5*gear4_pitch_diameter + gear5_pitch_diameter) -
 (0.5*gear3_pitch_diameter) ; % Length of casing material reduction (in)
w_reduction = gear_width_23 + 0.1 ; % Width of casing material reduction (in)
V_reduction =
 (2*(L_reduction*w_reduction*t_gearbox))+(w_reduction*d_gearbox*t_gearbox) ; %
 Volume of reduced material (in^3)
density_Al = 0.0975 ; % Density of 6061 T6 Aluminum (lb/in^3)
Weight_reduction_lbs = V_reduction*density_Al % Weight of gearbox casing
 removed (lbs)
density_St = 0.284 ; % Density of 4140 Steel (lb/in^3)
Weight_shaftA_increase_lbs = (1.0667*((pi/4)*0.75^2))*density_St % Weight of
 Shaft A material increase (lbs)
Weight_Frontshaft_increase_lbs = (1.0667*((pi/4)*1^2))*density_St % Weight of
 Front Shaft material increase (lbs)
net_Weight_reduction_lbs = Weight_reduction_lbs - Weight_shaftA_increase_lbs -
 Weight_Frontshaft_increase_lbs % Net weight reduction of Option 3 compared to
 Option 2.


Weight_reduction_lbs =

    0.1512


Weight_shaftA_increase_lbs =

    0.1338


Weight_Frontshaft_increase_lbs =

    0.2379


net_Weight_reduction_lbs =

   -0.2205
```

Figure 83: Gearbox Design Alternatives MATLAB Script

4.3.3.4 Henry Van Zuyle

Selecting between transmissions was one of the most important drive train decisions. The first consideration was choosing a CVT or some other type of transmission. This was an obvious choice as all top competitors use CVTs and they are lighter and more performant than any other type of transmission at this scale. Once we had gotten that choice out of the way, the real decision-making happened. The choices were a custom mechanically controlled CVT, a gaged CVT as we have run in the past, or a custom electronically controlled CVT. The Gaged CVT is by far the cheapest and easiest to implement, but lacks performance, especially in the range department, with tis limited .9:3.9 ratio range. We had concluded that high range was one of the most important parts of this CVT system choice, so having a high range custom CVT was going to be a necessity. We then chose an ECVT over a mechanical CVT due to its increased ability to be tuned, and its higher performance potential.

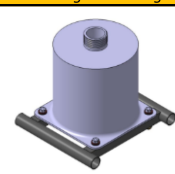
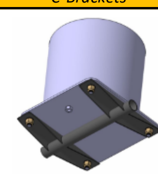
Figure 84: Potential CVT System Choices

4.3.4 Frame

4.3.4.1 Gabriel Rabanal

One decision for frame design that allows decision in design is the mounting type of the fuel tank. In the regulations, teams can choose between using sheet metal tabs mounted to primary or secondary tubing or square tubing mounts supported by primary or secondary material. To choose which design worked best for the frame design, two criteria were evaluated: the weight of each respective design and ease of manufacturing. Using Equation 16: Volume of a and Equation 17: Weight Equation, each design was evaluated to see which would weigh less and involve less material. For ease of manufacture, the number of weld jigs and types of specialty materials were taken into consideration. The results of the calculations are shown below in Table 14.

Table 14: Fuel Tank Mount Selection

	Variants	
Engineering Recs	*Rectangular Tubing*	*C-Brackets*
Figure		
Material Volume (in^3)	6.9492	6.289
Weight (lb)	1.97357	1.78608
# of Weld Jigs	2-3	1-2
Types of alternative materials	1	0

The C-bracket tabs proved to be the best option, having both a lower estimated weight and easier manufacturing process. Because the sheet metal is the same width as the suspension mounting tabs, no alternative materials will need to be sourced for manufacture. Additionally, the repetitive nature of the brackets and fewer mount points directly to the roll cage allow for fewer welds and easier jig operations, leading to an easier manufacturing process.

4.3.4.2 Cooper Williams

Due to the specificity of SAE BAJA Rules, there are very few areas of design where the frame team has creative freedom. One of these areas is the Side Impact Member (SIM). Straight SIM's use fewer feet of material; however, the straight SIM cannot withstand the same forces that a flared SIM can. By applying a few equations from Statics and Mechanics of Materials, this becomes obvious.

$$y_{AB} = \frac{Fbx}{6EIl}(x^2 + b^2 - l^2)$$

Equation 24: Deflection Simply Supported Intermediate Load

$$y_{BC} = \frac{Fa(l-x)}{6EIl}(x^2 + b^2 - 2lx)$$

Equation 25: Remainder Deflection Simply Supported Intermediate Load

$$\Sigma F_1 = R_1 + R_2$$

Equation 26: Resultant Forces: Two Supports

$$\Sigma F_1 = R_1 + R_2 + R_3$$

Equation 27: Resultant Forces: Three Supports

Even just by concept, it is clear that a structure with more supports will withstand a force better than a structure with fewer supports. This became clear when running an finite element analysis of the two variants in SolidWorks as can be seen below.

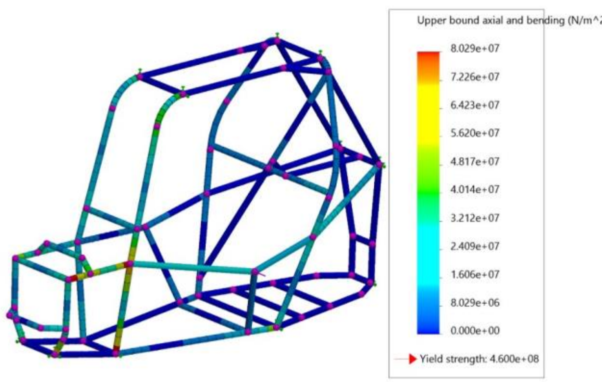


Figure 85: FEA of Flared SIM

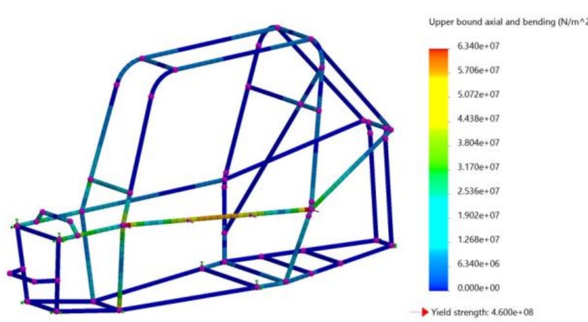


Figure 86: FEA of Straight SIM

Through this analysis it is obvious that along the straight SIM the largest bending is around 5.706e+07 (N/m^2). In the flared SIM, the same for was applied and a largest bending is about 2.409e+07(N/m^2). These pressures make the advantages of the Flared SIM obvious, despite the use of more material. Listed below are some characteristics found to aid in the visualization and quantification of this material difference.

Table 15: SIM Material Quantification

	Variants	
Engineering Recs	*Flared SIM*	*Straight SIM*
Maximum Width (in)	32	28
Total Primary Tubing (ft)	45	48
Total Secondary Tubing (ft)	45	49
Overall Vehicle Length (in)	67	75

4.3.4.3 Antonio Sagaral

The tubes for the seat to be mounted upon had a few requirements in the SAE rulebook. There had to be significant tubing for the driver not to fall through the frame in the event of seat failure. The bottom of the seat also has to be attached in a minimum of 4 places to the frame. Two seat mount variants were created and compared.

Figure 85: Drilled and Sleeved Tube Mounts

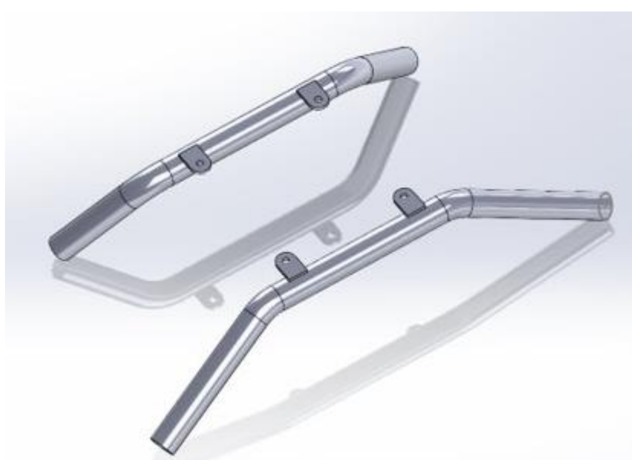


Figure 86: Tab Mounts

The drilled and sleeved tube mount is the better option. This is because it will require less material and less time manufacturing the mounts as the tabs would have to be cut and welded on in the proper orientation and the sleeves can just be placed in the holes and quickly welded.

4.4 Concept Selection

4.4.1 Front End

	Variants					
Subsystem	*1*	*Rating*	*2*	*Rating*	*3*	*Rating*
Scrub Radius	Positive	X	Negative	X	Zero	✓
Control Arm Geometry	A Arm	✓	J Arm	X	Wishbone	X
Control Arm Construction	CNC	X	Welded	✓	Carbon	X
Steering Arrangement	Anti-Ackerman	X	Parallel	X	Ackerman	✓
Master Cylinder Bore	1/2"	X	5/8"	✓	3/4"	X
Brake Pedal Ratio	2	X	4	✓	6	X
Camber During Suspension Bump	Loss	X	Gain	✓	No-Change	X
Shock Mount Location	UCA	✓	LCA	X		

Figure 87: Front End Decision Matrix

The current state of the 2024 SAE Baja front end can be seen in the figure below. Currently all geometry is set with the next steps being knuckle weight optimization and brake rotor integration. The upper A control arms will be welded, the steering tie rods will be carbon tubes with aluminum press fit inserts, and the knuckle and hub will be CNC milled from 6061 aluminum billet. Below the SolidWorks model of the current front end is a drawing of the assembly with a bill of materials attached.

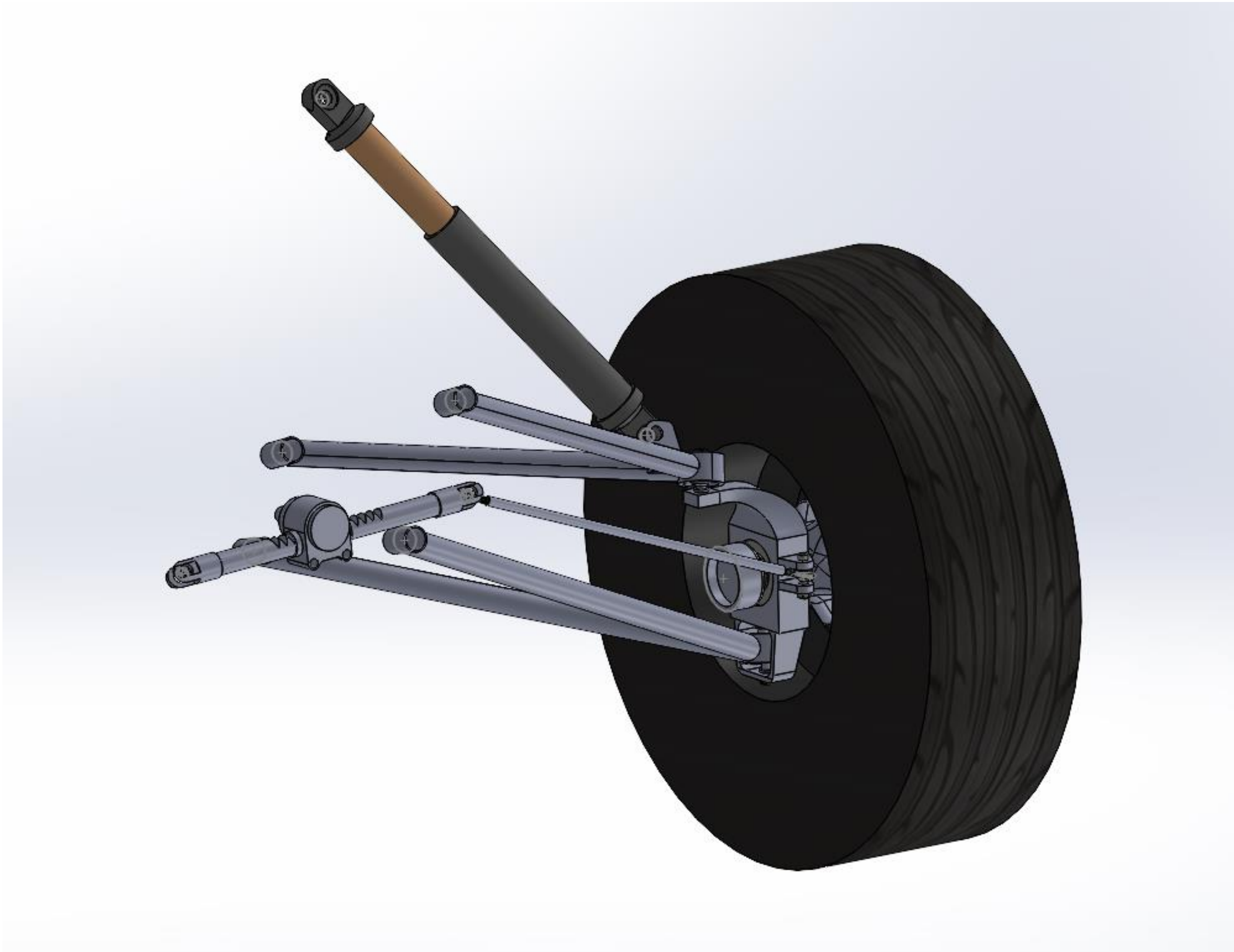
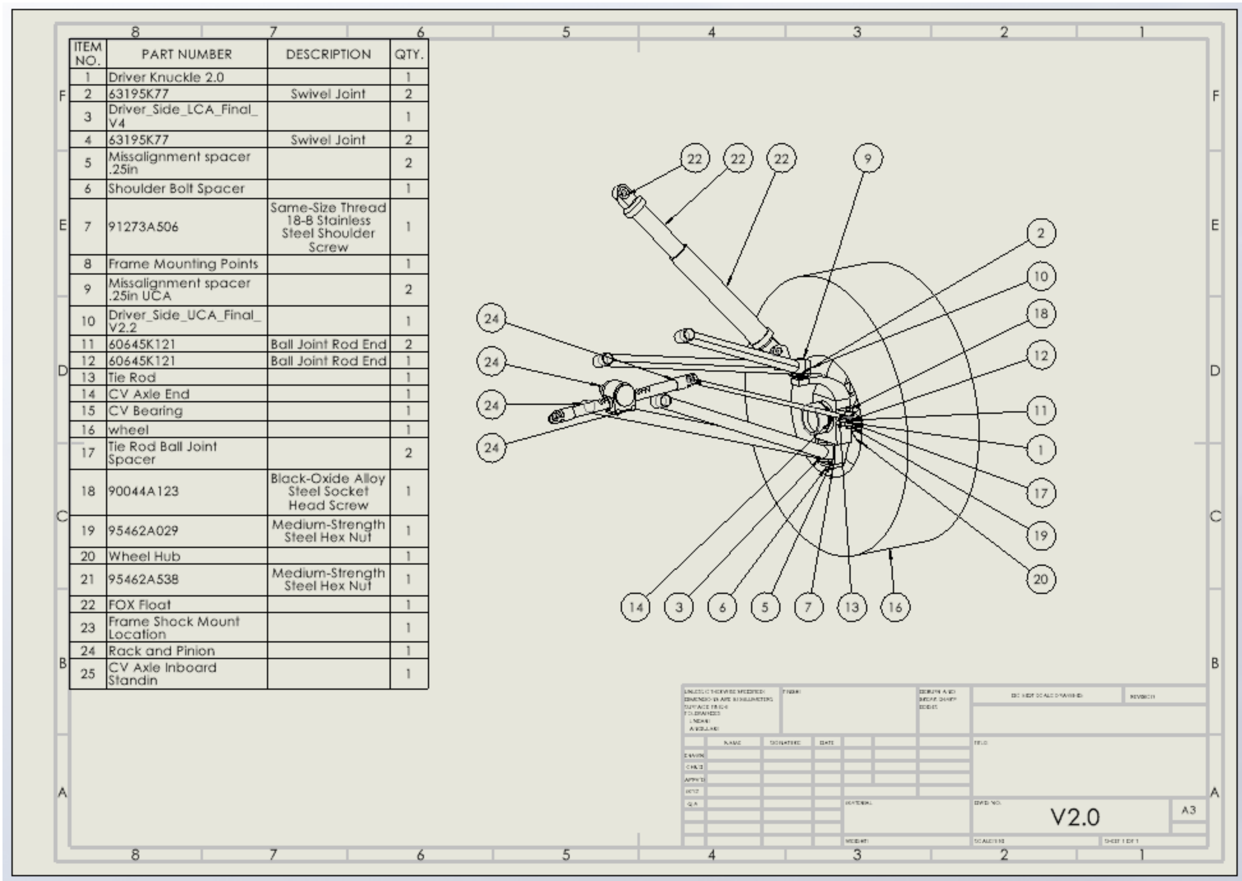


Figure 88: Current SolidWorks Model for Baja Vehicle Front Suspension and Steering



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	Driver Knuckle 2.0		1
2	63195K77	Swivel Joint	2
3	Driver_Side_LCA_Final_V4		1
4	63195K77	Swivel Joint	2
5	Missalignment spacer .25in		2
6	Shoulder Bolt Spacer		1
7	91273A506	Same-Size Thread 18-8 Stainless Steel Shoulder Screw	1
8	Frame Mounting Points		1
9	Missalignment spacer .25in UCA		2
10	Driver_Side_UCA_Final_V2.2		1
11	60645K121	Ball Joint Rod End	2
12	60645K121	Ball Joint Rod End	1
13	Tie Rod		1
14	CV Axle End		1
15	CV Bearing		1
16	wheel		1
17	Tie Rod Ball Joint Spacer		2
18	90044A123	Black-Oxide Alloy Steel Socket Head Screw	1
19	95462A029	Medium-Strength Steel Hex Nut	1
20	Wheel Hub		1
21	95462A538	Medium-Strength Steel Hex Nut	1
22	FOX Float		1
23	Frame Shock Mount Location		1
24	Rack and Pinion		1
25	CV Axle Inboard Standin		1

Figure 89: SolidWorks Drawing of Vehicle Front End Assembly

4.4.2 Rear End

After the team conducted an analysis for each of the sub-systems, decision were made, of which concepts or designs, we wanted to utilize. The team decided to use steel for the knuckle and all the hardware. Then the team is going to attach the trailing arm to the knuckle. The team is also going to use aluminum for the hub. The decisions made by the team may be seen below:

	Variants					
Subsystem	**1**	**Results**	**2**	**Results**	**3**	**Results**
Camber	Positive	✓	**Negative**	✓	Neutral	✓
Toe	In	✗	Out	✗	**Neutral**	✓
Camber Link Material	Carbon Fiber	✗	Steel	✓	**Alumminum**	✓
Axle types	**CV Axle**	✓	Dogbone	✗	U-Joint axle	✗
Knuckle Design	CNC Machined Aluminum	✗	**Steel**	✓	**Attach knuckle to trailing arm**	✓
Hub	**Aluminum (machined)**	✓	Cast	✗	NA	
Hardware	Stainless Steel	✗	**Steel**	✓	Titanium	✗
Trailing Link Design	**Boxed Sheet Metal**	✓	**Steel Tubing**	✓	CNC Machined Aluminum	✗
Wheel dish	**Dish out**	✓	Dish in	✗	NA	

Figure 90: Rear End Decision Matrix

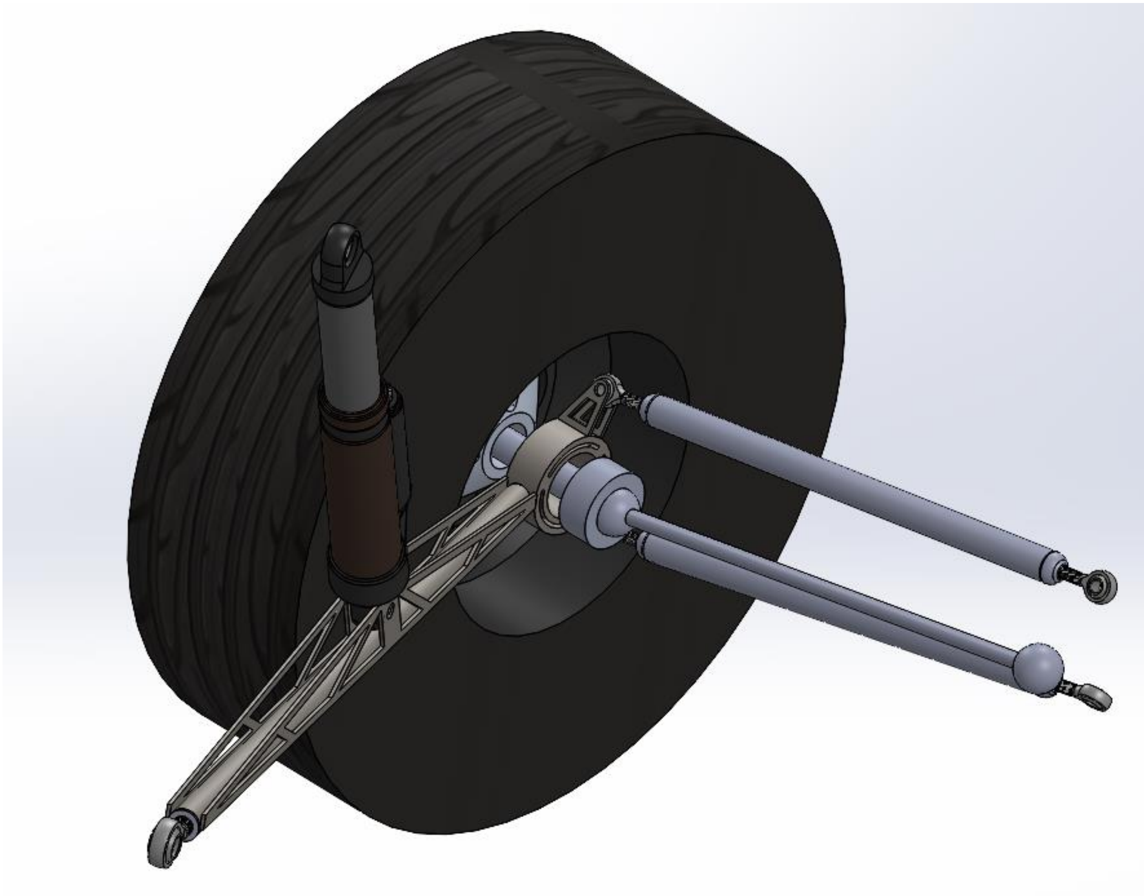
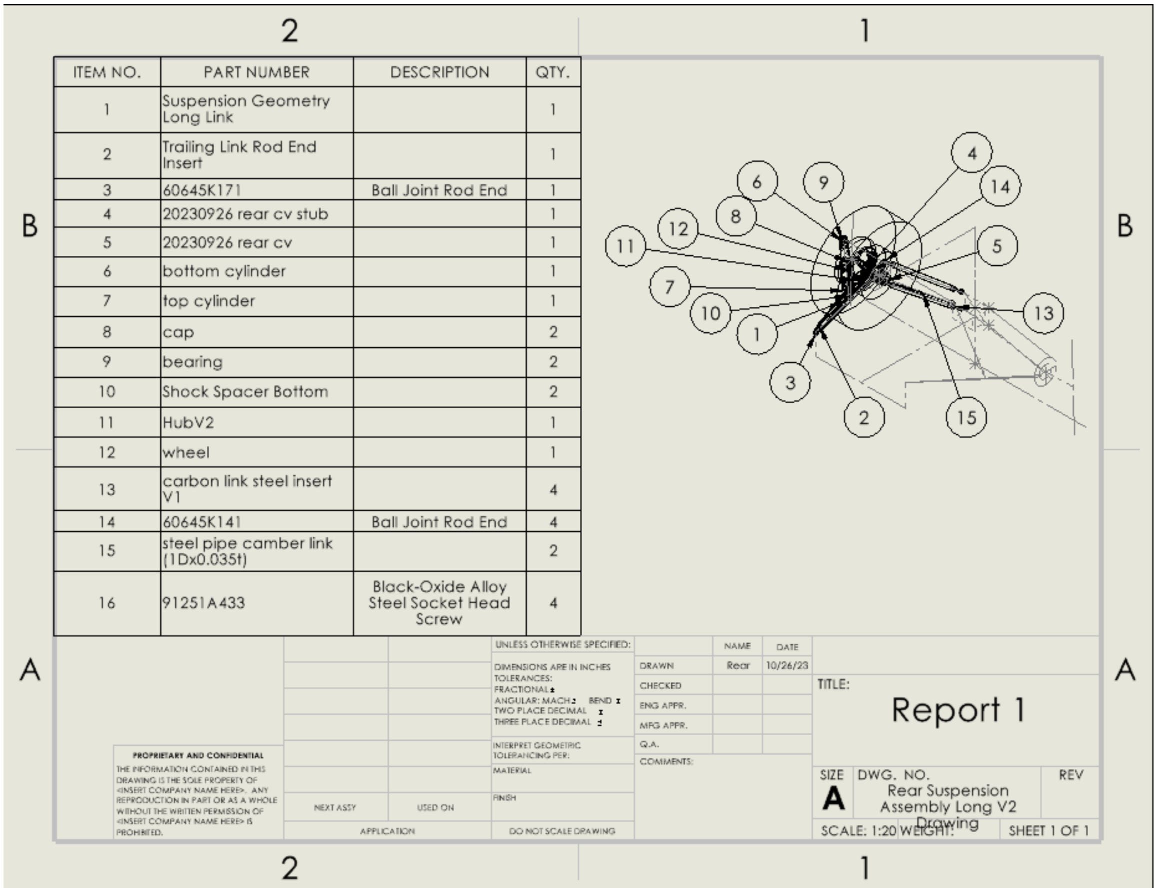


Figure 91: Rear Suspension Assembly

The current state of the rear suspension system in CAD is represented in Figure 91 and Figure 92. The assembly currently contains all necessary components to function, but it still in the refinement stage. The goal is to make all the components fit together better while still optimizing strength and weight of each component. The CAD will continue to be updated as the design process continues.



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	Suspension Geometry Long Link		1
2	Trailing Link Rod End Insert		1
3	60645K171	Ball Joint Rod End	1
4	20230926 rear cv stub		1
5	20230926 rear cv		1
6	bottom cylinder		1
7	top cylinder		1
8	cap		2
9	bearing		2
10	Shock Spacer Bottom		2
11	HubV2		1
12	wheel		1
13	carbon link steel insert V1		4
14	60645K141	Ball Joint Rod End	4
15	steel pipe camber link (1Dx0.035t)		2
16	91251A433	Black-Oxide Alloy Steel Socket Head Screw	4

Figure 92: Rear Suspension Assembly Drawing

4.4.3 Drivetrain

Based on the engineering calculations from Section 4.3.3, the team was able to finalize each concept variant for each subsystem.

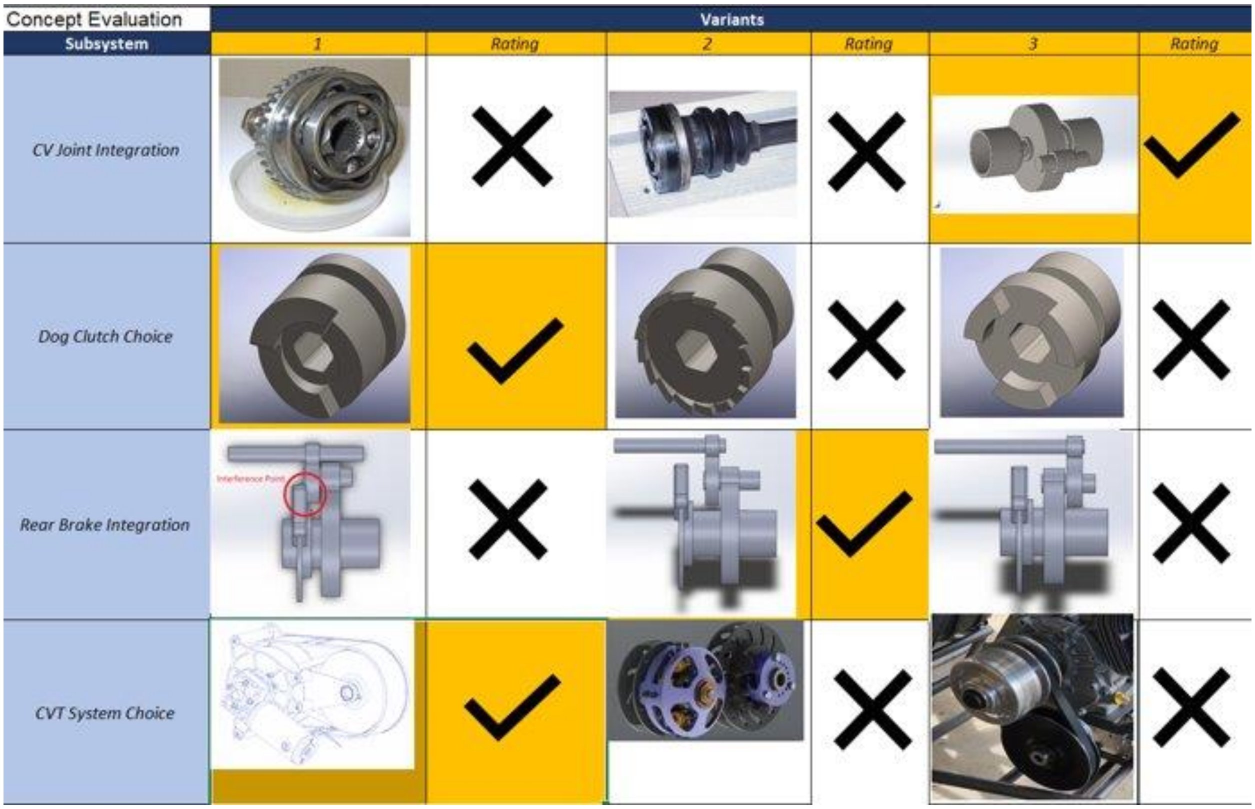
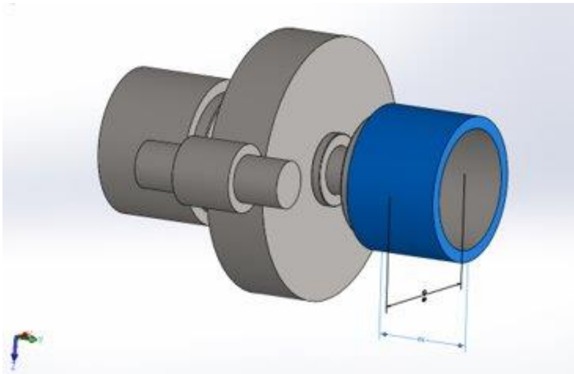


Figure 93: Drivetrain Concept Evaluation

The current design for each subsystem is presented in the figures below. The figures show a rough design of the CV joint integration on the output shaft, the dog clutch teeth, rear brake integration, and outline of the CVT system. These designs are not final, and they are subject to change due to several factors of integration, weight, and cost.



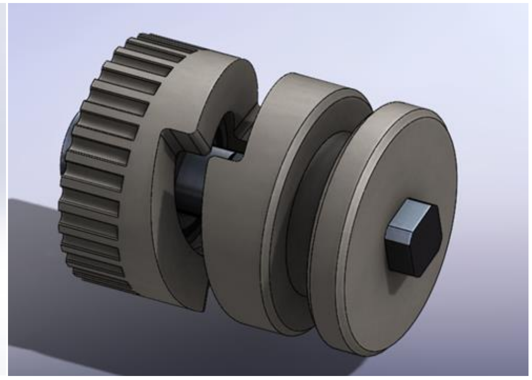
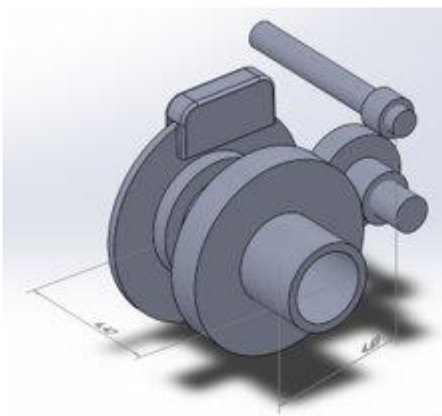


Figure 94: Front Gearbox CAD

Figure 95: Dog Clutch with Pulley



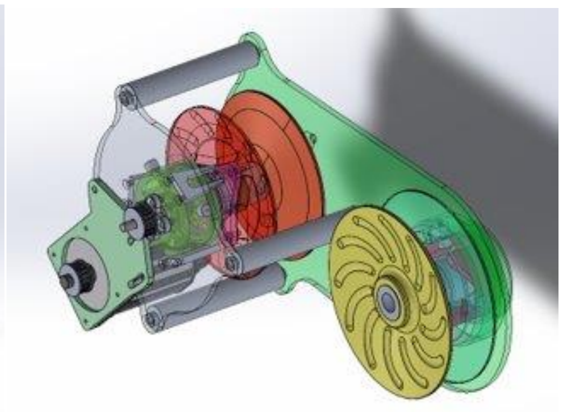


Figure 96: Rear Gearbox CAD

Figure 97: ECVT CAD

4.4.4 Frame

Through calculations made in section 4.3 Selection Criteria, the frame team was able to finalize all major design criteria and select the most appropriate options. The selected designs are shown below in Figure 98.

	Variants			
Subsystem	*1*	*Rating*	*2*	*Rating*
Frame Type	Front Brace	X	Rear Brace	✓
Fuel Tank Mount	Square Brackets	X	C-Brackets	✓
Seat Design	Slots	✓	Tabs	X
Side Impact Members	Straight	X	Flared	✓
Tube Material	AISI 4130	✓	AISI 1018	X

Figure 98: Frame Final Concept Selection

The current state of the frame in CAD is shown below in Figure 99. The design has been finalized for production, with integration of front and rear suspension teams being successful. After production of the frame, drivetrain components will be integrated, with the major integration components being completed already.

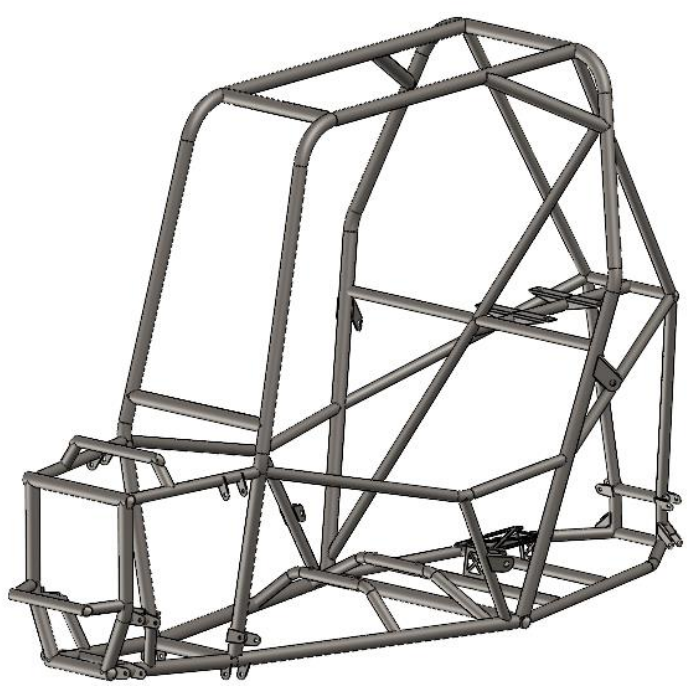


Figure 99: Frame Model

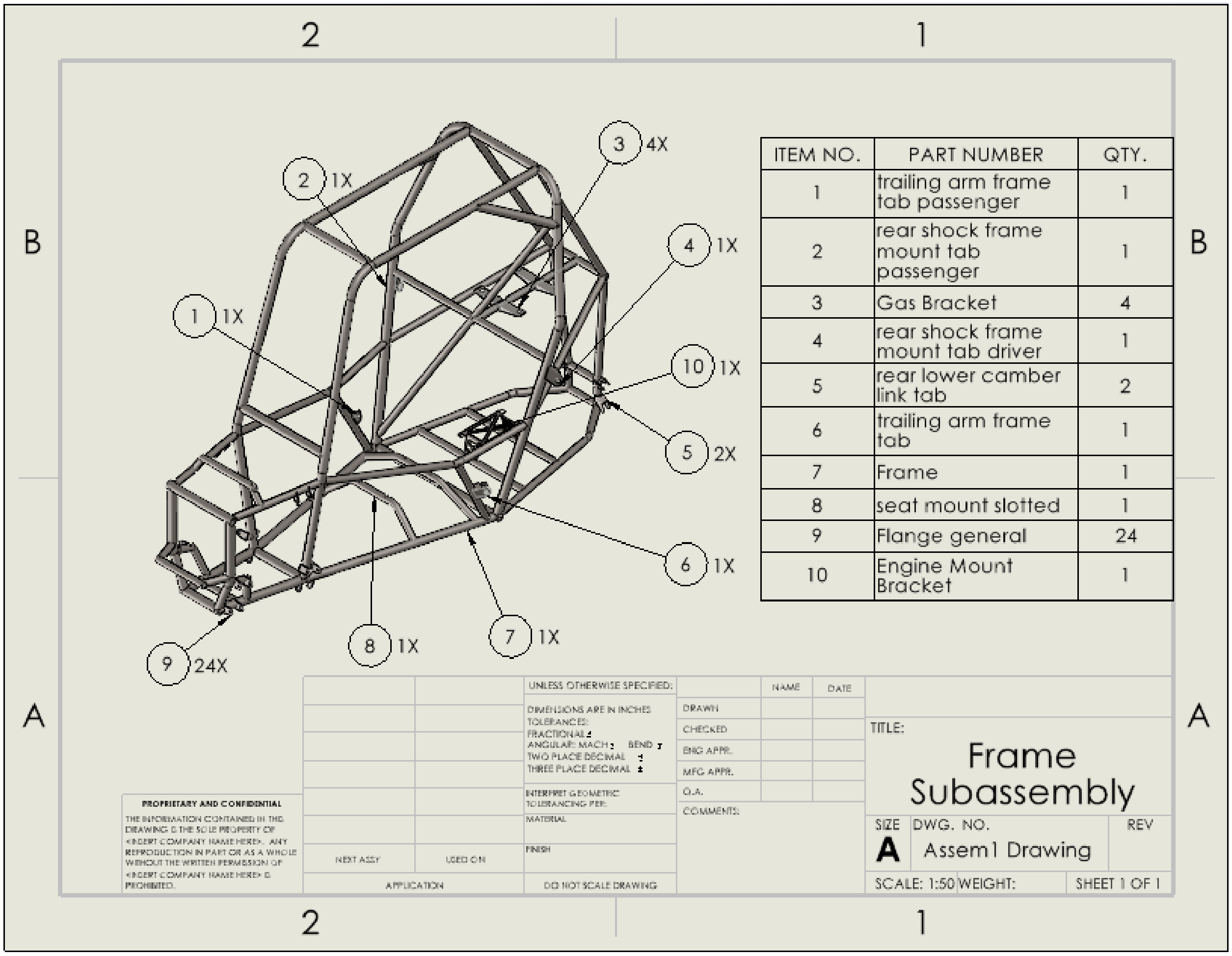


Figure 100: Frame Assembly Drawing

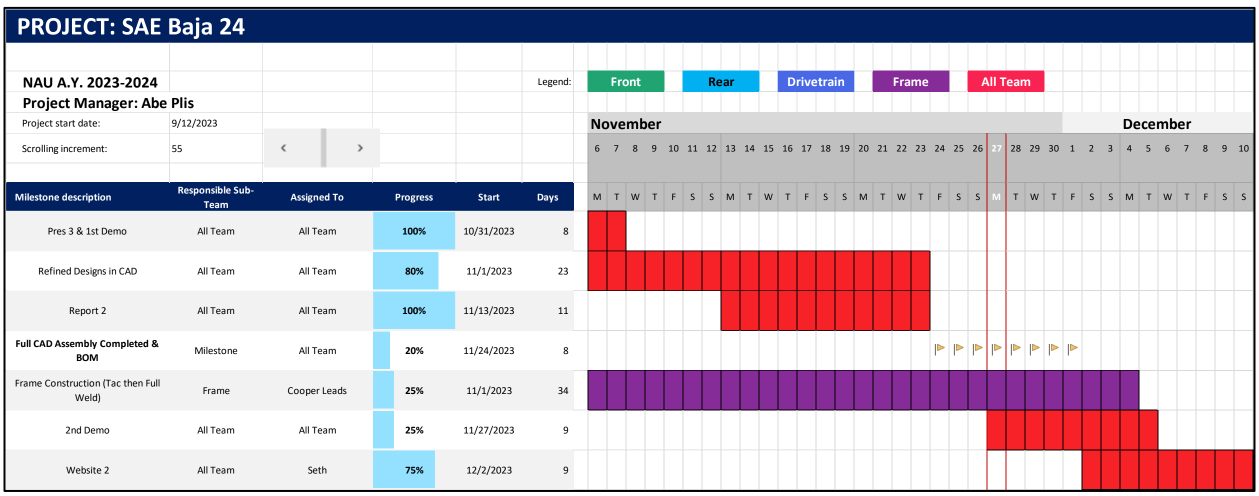
5 Schedule & Budget

5.1 Schedule

At the time of this report being written, the team has 3 weeks left in the semester and only a handful of outstanding tasks on the ME476C Gantt chart (Table 16). Presentation 3 and Prototype 1 have been completed, allowing the team to shift their focus towards the *Final CAD Design and Bill of Materials* deliverable as well as Prototype 2. The frame team is aiming to have a primary member frame tacked up by the end of the calendar year to satisfy deadlines established in several meetings with Dr. Willy. The rest of the team will focus on ordering raw materials and necessary components to begin construction of their respective sub-assemblies as the spring semester opens.

For added reference, please see the comprehensive ME476C Gantt chart in *Appendix A: Project Management*.

Table 16: ME476C - Remaining Tasks on Gantt Chart



In addition to the live ME476C Gantt chart, the team has also generated a tentative Gantt chart for ME486C in the spring semester. All dates are subject to change, but the general outline will help the team narrow down manufacturing sequences and purchasing deadlines during the construction of the vehicle. Due to the size of this Gantt chart, please see *Appendix A: Project Management* for the entire copy. The generic structure will follow the tentative schedule, Table 17, provided by Dr. Willy in which a series of hardware checks are spaced out through the semester with a completed (and tested) car being the goal by the semester's end. For the competition deliverables, the WBS outlined in *Table 3: SAE Baja Competition Deliverables* will be strictly adhered to, ensuring the team's eligibility for competition in Gorman, CA between April 25th-28th. For a detailed discussion of ME486C's WBS with regards to purchasing deadlines and manufacturing responsibilities, please see Section 5.3

Table 17: Tentative ME486C Schedule

Week	Week Starts	In-Class Agenda	UGRADS (whole team)	Individual Assignments	Team Assignments
1	16-Jan	486C Kickoff meetings (team/staff)*		Self-Learning or Individual Analysis	Project Management
2	23-Jan	Team/Staff Meetings			Engineering Model Summary
3	30-Jan	Team/Staff Meetings			
4	6-Feb	Team/Staff Meetings			
5	13-Feb	Hardware Status Update - 33+% build		Peer Eval 1 due	
6	20-Feb	Team/Staff Meetings			
7	27-Feb	Team/Staff Meetings			Website Check #1
8	6-Mar	Hardware Status Update - 67+% build	UGRADS Registration	Peer Eval 2 due	*Start writing your testing plan!*
	13-Mar	***Spring Break!!!***			
9	20-Mar	Team/Staff Meetings	Draft of Poster		*Continue writing your testing plan! (order testing equipment)*
10	27-Mar	Team/Staff Meetings			Finalized Testing Plan (order testing equipment)
11	3-Apr	Hardware Status Update - 100% build	Final Poster & PPT	Peer Eval 3 due	Final CAD Packet
12	10-Apr	Initial Testing Results			
13	17-Apr	Product Demo & Final Testing Results			Final Report & Final Website Check
14	24-Apr	Practice presentations in class, Symposium on Friday, time TBD			
15	1-May			Peer Eval 4 due	Client Handoff - Spec Sheet & Operation/Assembly Manual
Finals	8-May				

5.2 Budget

A project of this magnitude requires a great amount of financial and logistical support to be successful. A concise budget will be provided for each sub-team to give the reader an increased understanding of the scope of this project's required resources.

5.2.1 Front

The front end team is responsible for the suspension system in the front of the car, the steering assembly, the braking system, as well as the pedals that allow for driver input during operation. This large list of tasks requires a significant number of resources that can add up quickly in cost. The team is predicting a base construction expense of $2649 with a spare parts budget of $500 to ensure successful performance at competition. Logistical expenses for the competition including registration and travel are predicted to total up to around $1,125. With a 5% contingency applied to account for any unpredicted expenses, the front end team's budget comes in at $4,674. This amount is larger than ideal and will be adjusted to a more manageable level with the assistance of sponsors presented in Section 1.1 Please see Table 18 below for an organized breakdown of this financial information.

Table 18: Front End Budget

	Category	Relevant Items	Approximated Cost
1	**Vehicle Expenses**	Brake System Control Arm Materials Rod-ends/Ball Joints Shock Rebuild Knuckle Material/Manufacturing **Estimated Total**	$1,000 $120 $50 $126 $1600 **$2649**
2	**Spare Parts**	Rod-ends, Bushings, Welding supplies, Hardware	$500
3	**Competition Expenses** Front Sub-team	Registration, travel (hotel rooms, vehicle rentals, gas, etc.)	$1,125
4	**Contingency (5%)**	Unpredicted Expenses	$400
		Total	**$4,674**

5.2.2 Rear

The rear end team is responsible for the rear suspension system and output drive to the rear wheels. This is relatively simple and requires the smallest budget out of all the sub teams. The rear suspension system is utilizing a few parts that are already in the workshop and belong to the Baja team to cut down on cost. This leaves the main cost for the suspension system being raw materials like steel that will need to be cut down and welded together. The drive system is a major cost for the rear end sub team because it consists of CV axles, bearings, and a CNC hub that will be produced in house. Including spare parts, travel expenses, and contingency the rear end budget comes out to $2,893.00 with the goal of leaving more money for the other sub teams.

Table 19: Rear End Budget

	Category	Relevant Items	Approximated Cost
1	**Vehicle Expenses**	Suspension System Drive System Prototyping **Estimated Total**	$410 $850 $50 **$1310**
2	**Spare Parts**	Camber links, rode ends, cv axles, hubs	$320
3	**Competition Expenses** Front Sub-team	Registration, travel (hotel rooms, vehicle rentals, gas, etc.)	$1,125
4	**Contingency (5%)**	Unpredicted Expenses	$138
		Total	**$2893**

5.2.3 Drive

The drivetrain team is responsible for transmitting the power from the engine to the wheels. There are four subsystems within the drivetrain team, consisting of the ECVT, front gearbox, rear gearbox, and 4-wheel drive system. The combined estimated vehicle expenses for all subsystems came out to be $6,359.12, $500 for spare parts, $1,125 for competition expenses, and contingency expense of $400. Adding up all the expenses, the drivetrain budget came out to be at $8,284.12. This is the highest budget within the other sub teams due to the required subsystems needed to operate the vehicle. To achieve a lower budget, the team has plans on contacting local companies for financial support.

Table 20: Drivetrain Budget

	Category	Relevant Items	Approximated Cost
1	**Vehicle Expenses**	Motor Front Gearbox Rear Gearbox ECVT 4WD **Estimated Total**	\$900 \$794 \$1,018.55 \$2,310 \$1,336.57 **\$6,359.12**
2	**Spare Parts**	Gears, CV Axles, Hardware	\$500
3	**Competition Expenses** Drivetrain Sub-team	Registration, travel (hotel rooms, vehicle rentals, gas, etc.)	\$1,125
4	**Contingency (5%)**	Unpredicted Expenses	\$300
		Total	**\$8,284.12**

5.2.4 Frame

The frame team is responsible for the roll cage, all parts required to integrate with other subteams, paneling, and driver safety. The scope of the team is very wide and includes many small features that can be difficult to account for; these small expenses can add up quickly. Fortunately, many of our sponsors alleviate many of the material costs for frame specific components, which helps lower the frame team budget. Currently, the team estimates \$496 for material expenses, \$200 for spare parts, \$1125 for competition and travel expenses, and \$100 as a contingency fund. These expenses bring our total expected cost of operation to \$1921. This number is much lower than the other teams due to our material sponsors, which allows other teams to use funds on improving their designs.

Table 21: Frame Budget

	Category	Relevant Items	Approximated Cost
1	**Vehicle Expenses**	Frame Material Paneling and Carbon Layup Safety Equipment Hardware **Estimated Total**	$400 $0 $46 $50 **$496**
2	**Spare Parts**	Welding supplies, Hardware, Tab Materials, Tubing	$200
3	**Competition Expenses** Frame Sub-team	Registration, travel (hotel rooms, vehicle rentals, gas, etc.)	$1,125
4	**Contingency (5%)**	Unpredicted Expenses	$100
		Total	**$1921**

5.3 Bill of Materials (BOM)

5.3.1 Front

The bill of materials for the front end tabulates the required materials, material cost, purchasing/manufacturing identification, vendor sourcing/part number, and manufacturer specific detail. A majority of the parts in the front end will be sourced from hardware suppliers, brake vendors (i.e. Wilwood), or existing shop stock. The remaining larger components will be manufactured in house via 4130 1" tubing or 6061 Aluminum billet to save the team on manufacturing costs. The entirety of the bill of materials for the front end can be seen in the table below. The total build cost is within accordance with the predicted budget for the front end specified in Section 5.2.1.

Table 22: Bill of Materials for SAE Baja Front

Part Name	Quantity	Cost	Total Cost	Material	Purchase/Made	Vender	Vender Part Number
Upper Control Arm	3	8.8	105.6	4130, 4'	Made	IMS	N/A
Lower Control Arm	3	8.8	105.6	4130, 4'	Made	IMS	N/A
Knuckle	2	450	900	6061-T6	Made	IMS	N/A
CV Axle/Nut	2	108	216	Steel	Purchased	Amazon	N/A
CV Spline	2	40	80	Steel	Purchased	Amazon	N/A
Hub	2	190	380	6061-T6	Made	IMA	N/A
CV Bearing	4	72.92	291.68	Steel	Purchased	SKF	6006-2RS1
Upper Swivel Joint Spacer	4	4	16	6061-T6	Made	IMS	N/A
Lower Swivel Joint Spacer	4	4	16	6061-T6	Made	IMS	N/A
Lower Knuckle Spacer	2	4	8	6061-T6	Made	IMS	N/A
Upper Control Arm Knuckle Bolt	2	14.76	14.76	Alloy Steel	Purchased	MMC	90044A124
Lower Control Arm Knuckle Bolt	2	14.89	14.89	Alloy Steel	Purchased	MMC	90044A125
Tie Rod Knuckle Bolt	2	11.99	11.99	Alloy Steel	Purchased	MMC	91251A542
Tie Rod Knuckle Spacer	4	4	16	6061-T6	Made	IMS	N/A
Delrin Bushing	12	1.5	18	Delrin	Made	MMC	7521T16
Control Arm Frame Bolt	4	18.22	18.22	Alloy Steel	Purchased	MMC	N/A
Rim	2	N/A	N/A	Steel	Owned	N/A	N/A
Tire	2	N/A	N/A	Rubber	Owned	N/A	N/A
Brake Rotor	4	N/A	N/A	Steel	Donated	IMS	N/A
Brake Caliper	2	Owned	Owned	N/A	Owned	HBS	N/A
Master Cylinder	2	220	440	N/A	Purchased	Tilton	78-625
Brake Hose	2	18.4	36.8	Steel	Purchased	Amazon	N/A
Brake Pedal	1	95	95	6061-T6	Made	IMS	N/A
Pedal Sensor	1	5	5	N/A	Purchased	Amazon	N/A
Throttle Pedal	1	30	30	Carbon	Made	N/A	N/A
Throttle Pedal Sensor	1	5	5	N/A	Purchased	Amazon	N/A
Steering Rack	1	20	20	Steel	Made	N/A	N/A
Rack End	2	10	20	Aluminum	Purchased	Joes Racing Products	N/A
Rack End Screws	2	5	10	Aluminum	Purchased	Joes Racing Products	N/A
Pinion Housing	1	100	100	Aluminum	Purchased	Joes Racing Products	N/A
Pinion Gear	1	100	100	Steel	Purchased	Joes Racing Products	N/A
Pinion Bearing	2	10	20	N/A	Purchased	Joes Racing Products	N/A
3/8 Rod End for Tie Rod	4	10	40	Steel	Purchased	McMaster Carr	60645K14
Tie Rod Threaded Insert	4	5	20	Aluminum	Made	N/A	N/A
Tie Rod	2	15	30	Carbon	Purchased	Amazon	N/A
Steering Column	3	15	45	Carbon	Purchased	Amazon	N/A
Aluminum Column Insert	6	2	12	Aluminum	Made	N/A	N/A
U Joint	2	40	80	Steel	Purchased	McMaster Carr	6443K106
Pinion Face Plate	1	5	5	Plastic	Purchased	Joes Racing Products	N/A
1/4"-20 Bolts for Steering Column	3	11.58	34.74	Titanium	Purchased	McMaster Carr	94081A112
1/4"-20 Nut for Steering Column	3	10.8	32.4	Titanium	Purchased	McMaster Carr	94528A117
1/2"-13 Bolt for Rack to Tie Rod	2	20.83	41.66	Titanium	Purcahsed	McMaster Carr	94081A749
1/2"-13 Nut for Rack to Tie Rod	2	10.32	20.64	Titanium	Purchased	McMaster Carr	90545A034

5.3.2 Rear

The bill of materials below shows the updated material and part list for the rear suspension setup. This considers every part/material that will be manufactured in house and just straight bought. The rear system has been split into four different groupings, CV Parts and Hardware, Camber Link Parts and Hardware, Shock Parts and Hardware, and Trailing arm / Knuckle Parts and Hardware.

Table 23: BOM for Rear End

Part No.	Part des	Qty.	Cost.	Total	Comments
	CV Parts and Hardware (Buying CV and Hardware and manufacturing the rest)				
1	Aluminum Blocks for hubs	2	$ -	$ -	using old baja stock
2	wheel	2	$ -	$ -	using old baja wheels
3	tires	2	$ -	$ -	using old baja tires
4	Yamaha Grizzly 350 CVs	2	$ 114.99	$ 229.98	
5	SKF Bearing 6006	4	$ 43.56	$ 174.24	
6	Spacer Steel	2	$ -	$ -	using old baja stock
7	Yamaha Grizzly 350 hubs for spline	2	$ 50.00	$ 100.00	from ebay
8	Medium-Strength Steel Hex Nut, Grade 5, Zinc-Plated, 3/4"-10 Thread Size	2	$ 0.79	$ 1.58	
	Camber Link Parts and Hardware (Mainly Bought)				
10	16 mm x 12 mm x 500 mm Carbon Fiber 1 in diameter	4	$ 15.50	$ 62.00	
11	Steel tubing 1 in x .93 x 17	4	$ -	$ -	
12	SS rod ends 696OT231	4	$ 13.75	$ 55.00	
13	ROD ENDS 60645k141	4	$ 6.60	$ 26.40	
14	Hex Screws 9157A657	8	$ 0.71	$ 5.69	
15	Locknuts 97135A419	8	$ 0.35	$ 2.77	
16	Aluminum inserts .62 x 1.4 in	4	$ -	$ -	using old baja stock
17	Steel inserts .75 x 1 in stock steel	4	$ -	$ -	using old baja stock
	Shock Parts and Hardware (Manufacturing Spacers only)				
18	Shocks	2	$ -	$ -	using Fox Evol
19	shock spacers	8	$ -	$ -	using baja stock
20	91271A646 Alloy-Steel 12-pnt Screw	2	$ 12.41	$ 24.82	
21	92018A111 high strength steel nylon insert flange	2	$ 2.27	$ 4.54	
22	5/16-18x1-1/4 Alloy 12-Point Flange Screws Black Oxide	2	$ 13.81	$ 27.62	
	Trailing arm / Knuckle Parts and Hardware (Manufacturing Trailing Arm and Knuckle)				
23	4130 1" x 0.095" Chromoly Round Tubing - 60"	1	$ 31.08	$ 31.08	
24	A36 1/4" Steel Plate - 30" x 8"	1	$ -	$ -	sponsored
25	Steel Round Bar - 3.25" x 2.6"	1	$ -	$ -	using old baja stock
26	Trailing Link Rod End Insert	2	$ -	$ -	using old baja stock
27	4237N107 Rod End with Nut - 5/8"-18	2	$ 14.78	$ 29.56	
28	91271A802 Alloy-Steel 12-Point Screw	1	$ 8.54	$ 8.54	
			Grand Total:	**$ 783.82**	

5.3.3 Drive

The bill of materials presented below show the required parts and quantities needed for both front and rear gearboxes. The price for each part is not listed due to an increase of necessary parts needed for the final design, however the budget for the entire drivetrain is listed in Section 5.2.3. The bill of materials shows the parts needed to be machined in house and the parts needed to be outsourced. The material that will be used will vary from 4130 steel and 4340 steel for the gears and shafts and purchasing the required amount will be done accordingly.

Table 24: Drivetrain BOM

Rear Gearbox		Rear Gearbox		Front Gearbox	
Part	Quantity	Part	Quantity	Part	Quantity
3 Tooth Spiral Jaw	1	Rear pasenger gearbox case	1	6656K151_Ultra-Thin Ball Bearing	2
1460T11_Threaded Black-Oxide Steel Track Ro	2	Dog clutch cap	1	60355K178_Ball Bearing	2
6391K33_Oil-Embedded Bronze Sleeve Bearin	1	Gearbox pully sheild	1	Front gearbox driver side	1
6656K236_Ultra-Thin Ball Bearing	4	Shift fork	1	Front gearbox passenger side	1
60355K151_Ball Bearing	6	Shift fork mount guard	1	Rear gear 4	1
60715K111_One-Piece Steel Thrust Ball Bearin	1	CVT gaurd cover	1	Rear gear 5 version 2	1
ECVT fixed primary sheave	1	Gearbox input shaft support	1	Rear gearbox intermediate shaft	1
ECVT mobile primary sheave	1	Limit switch activated	1	Front gear 5 output shaft	1
Primary shaft	1	Main CVT guard	1	SKF_65X80X8 CRW1 R	2
Secondary fixed sheave	1	Engagement cable mount	1	SKF_7438	2
Secondary moving sheave	1	Secondary guard	1	CV Cup	2
Gaged style secondary cam	1	Secondary gaurd 2	1	Sprag, GMN FK6205-2RS	2
Secondary cam nut	1	Secondary gaurd 3	1		
Secondary shaft	1	Disengagement cable mount	1		
Primary sliding shaft	1	M-343x	1		
Primary square bushing	1	Official Pulley Design	1		
Lead screw	1	PA-D22-077_reduced	1		
Lead screw nut	1	SKF_65X80X8 CRW1 R	2		
Lead screw nut flange	3	SKF_80X100X10 CRW1 R	2		
Primary moving sheave bushing	1	SKF_7438	2		
CVT backplate	1	Engine	1		
Exhaust clearance	1				
Rear CV	1				
Rear CV stub	1				
Control motor mount plate	1				
Control motor mount standoffs	3				
Lead screw bearing mount	1				
ECVT control pulley	2				
Motor mount plate	1				
Nut flange forks	1				
Rear driver gearbox case	1				
Rear gear 2	1				
Rear gear 3	1				
Rear gear 4	1				
Rear gear 5	1				
Rear gearbox intermediate shaft	1				
Rear input shaft	1				

5.3.4 Frame

Due to the nature of the frame team schedule, much of the material that the team is responsible for will not be decided on until later in the design process. The urgency of designing and manufacturing the frame itself means many of the smaller parts such as fasteners and housing materials are not yet decided on. These will be added after construction of the roll cage is completed. Other components such as various driver safety equipment may be able to be used from previous vehicles but must be inspected first. Many of the remaining components such as paneling, numbers, and aesthetics, are categorized as about the final 10% of the project. These costs will also be adjusted in the coming weeks.

Table 25: Frame BOM

Part	Description	Manufacturer	Qty	Units	Unit Cost	Cost	Obtained
Primary Members	4130 1.25x.065" tubing	IMS	45	ft	0.00	0.00	Y
Secondary Members	4130 1x.035"tubing	BMS	45	ft	8.00	360.00	N
Tabs	4130 0.1"sheet	VROOM Engineering	100	in^2	0.28	27.78	N
Side Panneling	Carbon Weave	Novakinetics	20	ft^2	0.00	0.00	N
Seat	Carbon Weave	Novakinetics	4	ft^2	0.00	0.00	N
Epoxy	Carbon resin epoxy	Novakinetics	1	gallon	0.00	0.00	N
Harness	Standardized	RaceQuip	1	unit	0.00	0.00	Y
Submarine Straps	Standardized	TBD	2	unit	0.00	0.00	y
Fire Extinguisher	5BC Standard	TBD	1	unit	0.00	0.00	Y
Extinguisher Mount Bracket	Drake FIREX-MNT-DAG	Drake	1	unit	0.00	0.00	Y
Fuel Tank Mounting Washers	McMaster Carr 94733A723	McMaster	8	50	0.00	0.00	Y
Mounting Hardware	Misc. nuts, bolts, washers needed	TMS Titanium	1	unit	0.00	0.00	N
Skid Plate	.06"HDPE	TBD	6	ft^2	5.00	30.00	N
Firewall	.02" sheet metal	TBD	9.5	ft^2	7.75	73.63	N
					Total Cost	491.40	

6 Design Validation and Initial Prototyping

6.1 Failure Modes and Effects Analysis (FMEA)

The first step in the validation process is for each sub-team to analyze their designs for critical potential failures and how these failures are being mitigated via clever engineering design.

6.1.1 Front

6.1.1.1 Upper Control Arm

The front end of the vehicle will feature a double A-arm style suspension system that consists of both an upper and lower control arm. This section will present the FMEA for the upper control arm; please see Section 6.1.1.2 for the FMEA of the lower control arm. The upper control arm features several potential failure points including the shoulder bolts that mount the arm to the chassis tabs, any welded areas (i.e. ball joint cup and any joined members), and any long lengths of tubing. The upper control arm could experience failure in a few modes: impact fracture, impact deformation, and impact fatigue. These failure modes will be mitigated in the design by running a larger diameter shoulder bolt (3/8”, see Section 6.3.1.1

Table 26: Upper Control Arm FMEA

Part # and Functions	Potential Failure Mode	Potential Effect(s) of Failure	Potential Causes and Mechanisms of Failure	RPN	Recommended Action
UCA Shoulder Bolt	Impact Fatigue	Erratic Operation, Poor Appearance	Overstressing	30	Use 3/8" Shoulder Bolts
UCA Shoulder Bolt	Impact Fracture	Erratic Operation, Poor Appearance	Impact Loading	30	Use 3/8" Shoulder Bolts
UCA Pivot Tubing	Impact Fatigue	No Longer Operational, Poor Appearance	Overstressing	9	Limit Length and Check Welds
UCA Pivot Tubing	Impact Fracture	No Longer Operational, Poor Appearance	Impact Loading	9	Limit Length and Check Welds
UCA Long Member(s)	Impact Fatigue	No Longer Operational, Poor Appearance	Overstressing	18	Limit Torsion on UCA and Check Welds
UCA Long Member(s)	Impact Deformation	No Longer Operational, Poor Appearance	Impact Loading	18	Limit Torsion on UCA and Check Welds
UCA Shock Mount	Impact Fatigue	No Longer Operational, Poor Appearance	Overstressing	30	Strengthen mount with addition weld/plates
UCA Shock Mount	Impact Fracture	No Longer Operational, Poor Appearance	Impact Loading	30	Strengthen mount with addition weld/plates
UCA Ball Joint Cup	Impact Fatigue	No Longer Operational, Poor Appearance	Overstressing	30	Orient properly relative to knuckle motion
UCA Ball Joint Cup	Impact Fracture	No Longer Operational, Poor Appearance	Impact Loading	30	Orient properly relative to knuckle motion

6.1.1.2 Lower Control Arm

The lower control arm will behave in a similar manner to the upper control arm in the scope of FMEA. The main failure locations, modes, and mitigation strategies are shared between the two parts which makes the discussion of the lower control arm’s FMEA redundant. The only difference seen is due to the fact that the lower control arm is much lower to the ground, thus increasing the likelihood of an impact from track obstacles or other debris. As such, a higher RPN appears in the rows mentioning the longer lateral members of the lower control arm. The mitigation strategy set in place to address this concern is to brace the long tubing perpendicular to its length as well as raising the ride height of the car. The increase in ride height will not only ensure the lower control arms are less likely to impact debris but will also work towards the vehicle’s main engineering requirement of increasing the ride height as much as possible (Section 2.1.1

Table 27: Lower Control Arm FMEA

Part # and Functions	Potential Failure Mode	Potential Effect(s) of Failure	Potential Causes and Mechanisms of Failure	RPN	Recommended Action
LCA Shoulder Bolt	Impact Fatigue	Erratic Operation, Poor Appearance	Overstressing	30	Use 3/8" Shoulder Bolts
LCA Sholder Bolt	Impact Fracture	Erratic Operation, Poor Appearance	Impact Loading	30	Use 3/8" Shoulder Bolts
LCA Pivot Tubing	Impact Fatigue	No Longer Operational, Poor Appearance	Overstressing	8	Limit Length and Check Welds
LCA Pivot Tubing	Impact Fracture	No Longer Operational, Poor Appearance	Impact Loading	8	Limit Length and Check Welds
LCA Long Member(s)	Impact Fatigue	No Longer Operational, Poor Appearance	Overstressing	64	Raise ride height and check welds
LCA Long Member(s)	Impact Deformation	No Longer Operational, Poor Appearance	Impact Loading	64	Raise ride height and check welds
LCA Ball Joint Cup	Impact Fatigue	No Longer Operational, Poor Appearance	Overstressing	30	Orient properly relative to knuckle motion
LCA Ball Joint Cup	Impact Fracture	No Longer Operational, Poor Appearance	Impact Loading	30	Orient properly relative to knuckle motion
LCA Bracing	Impact Fatigue	Flying Debris, Poor Appearance	Overstressing	15	Limit length and check welds
LCA Bracing	Impact Fracture	Flying Debris, Poor Appearance	Impact Loading	15	Limit length and check welds

6.1.1.3 Knuckle

The front steering knuckle is critical to maintaining vehicle steering and suspension performance. A failure of the upper control arm mounting interface, lower control arm mounting interface, or tie rod mounting interface would result in a complete failure of the vehicle. As a result, the knuckles critical mounting interfaces will see a very high-risk factor and will need to be inspected regularly to ensure proper and safe vehicle operation.

Table 28: FMEA for Front Steering Knuckle

Part # and Functions	Potential Failure Mode	Potential Effect(s) of Failure	Potential Causes and Mechanisms of Failure	RPN	Recommended Action
UCA Knuckle Shoulder Bolt	Impact Fatigue	Knuckle detatch from Control Arm	Overstressing	30	Increase Bolt Diameter to 3/8"
UCA Knuckle Shoulder Bolt	Impact Fracture	Knuckle detatch from Control Arm	Impact Loading	40	Increase Bolt Diameter to 3/8"
UCA Alignment Spacer	Impact Fatigue	Inconsistant Operation	Overstressing	8	Change material to steel from aluminum
UCA Alignment Spacer	Impact Fracture	Inconsistant Operation	Impact Loading	8	Change material to steel from aluminum
LCA Knuckle Shoulder Bolt	Impact Fatigue	Knuckle detatch from Control Arm	Overstressing	30	Increase Bolt Diameter to 3/8"
LCA Knuckle Shoulder Bolt	Impact Fracture	Knuckle detatch from Control Arm	Impact Loading	40	Increase Bolt Diameter to 3/8"
LCA Alignment Spacer	Impact Fatigue	Lower Suspension Effectivenenss	Overstressing	8	Change material to steel from aluminum
LCA Alignment Spacer	Impact Fracture	Lower Suspension Effectivenenss	Impact Loading	8	Change material to steel from aluminum
Tie Rod Shoulder Bolt	impact fracture	Knuckle detatch from Tie Rod	Overstressing	40	Increase Bolt Diameter to 3/8"
Tie Rod Shoulder Bolt	impact fatigue	Knuckle detatch from Tie Rod	Impact Loading	30	Increase Bolt Diameter to 3/8"
Tie Rod Bolt Spacer	impact fatigue	Lower Steering Effectiveness	Overstressing	12	Change material to steel from aluminum
Knuckle LCA Lower Mount	Impact Deformation	Knuckle detatch from Control Arm	Impact Loading	40	Increase material thickness to .5" from .3"
Knuckle LCA Lower Mount	impact fatigue	Knuckle detatch from Control Arm	Overstressing	40	Increase material thickness to .5" from .3"
Knuckle LCA Bolt Thread	impact fatigue	LCA Shoulder Bolt detatch from knuckle	Overstressing	54	Increase bolt thread to 3/8"x20
Knuckle LCA Bolt Thread	impact fracture	LCA Shoulder Bolt detatch from knuckle	Impact Loading	54	Increase bolt thread to 3/8"x20
Knuckle Tie Rod Mount Tab	impact deformation	Tie Rod detatch from knuckle	Impact Loading	40	Increase tab thickness to .3" from .2"
Knuckle Tie Rod Mount Tab	Impact Fatigue	Tie Rod detatch from knuckle	Overstressing	40	Increase tab thickness to .3" from .2"
Knuckle UCA Extension	Impact Deformation	UCA Detach from knuckle	Impact Loading	50	Increase CX area to decrease bending moment
Knuckle Bearing Bore	Impact Deformation	Lower drive effectiveness	Impact Loading	3	Harden inner surface of bearing bore
Knuckle Bearing seperation	Impact deformation	lower drive/steering effectiveness	Impact Loading	4	Increase thickness of bearing separation

6.1.1.4 Steering Assembly

The steering assembly is vital for the performance of the car and if damaged may hinder the vehicle unusable. To ensure that the system works, all parts must be validated including their construction. This ensures that each subsystem will stand up to the demands of the competition. The highest area for failure includes the rack and pinion. This is not a likely area of failure however if it were to break it would be an expensive part to replace. In addition to this the rack and pinion if it were to fail, would cause a failure of the entire driving system and would make the vehicle uncompetitive.

Table 29 - Steering Design Testing

Part # and Functions	Potential Failure Mode	Potential Effect(s) of Failure	Potential Causes and Mechanisms of Failure	RPN	Recommended Action
Knuckle Connection	Impact Fatigue	Broken Knuckle connection to steering arm	Overstressing	20	Conduct FEA on Knuckle tab
Knuckle Connection	Impact Fracture	Broken Knuckle connection to steering arm	Impact Loading	40	Conduct FEA on Knuckle tab
Steering Arm Carbon Tubing	Impact Fatigue	Broken Steering arm hindering steering capabilities	Overstressing	16	Limit Length
Steering Arm Carbon Tubing	Impact Fracture	Broken Steering arm hindering steering capabilities	Impact Loading	16	Limit Length and check Clearence and skid protection
Steering Arm Tubing Insert	Sheer Strength	Threaded insert pulls out of carbon steering arm	Overstressing through Tension	16	Use of Epoxy to increase tensile strength
Steering Arm Tubing Insert	Impact Deformati on	Threaded insert damaged from impact	Impact Loading	16	Check clearance and impact protection
Steering Column	Torsion	Breaking carbon steering column	Overstressing through Torsion	16	Recommend using a 16mm OD x 14mm ID tube
Steering Column	Sheer of Bolt	Bad steering performance and broken column	Overstressing of bolt	16	Epoxy spline insert to tube in addition to using bolt
Rack and Pinion	Contact Wear	Poor Steering Performance	Gradual wear of the rack and pinion assembly	60	Use of brass bushings with lubrication

6.1.2 Rear

6.1.2.1 Trailing Link

The trailing link construction was an area of interest that the rear end team wanted to examine with the design validation process. The potential failures in this system were identified as the rod ends, hardware, and weld points. The rod ends had the second highest RPN value of 42 and was addressed by using 5/8" rode ends in the final design. The hardware had the third highest RPN value of 24 and was resolved by incorporating high grade 5/8' hardware to attach the rod end to the baja frame. The highest RPN value belonged to the steel side plates of the trailing link that will be welded to the steel tube. The mitigation to this problem was to add more cross members and increase the welding surface area of the steel plate.

Table 30: Trailing Link FMEA

Part # and Functions	Potential Failure Mode	Potential Effect(s) of Failure	Potential Causes and Mechanisms of Failure	RPN	Recommended Action
Rod End	**Impact Fatigue**	Improper Geometry, Suspension Binding	Impact Loading	42	Use 5/8" Rod End
Rod End	**Abrasive Wear**	Improper Geometry, Suspension Binding	Poor Maintenance	42	Use 5/8" Rod End
Rod End Hardware	**Impact Fatigue**	No Longer Operational, Poor Appearance	Overstressing	24	Use 5/8" Hardware
Rod End Hardware	**Impact Fracture**	No Longer Operational, Flying Debris	Impact Loading	24	Use 5/8" Hardware
Steel Tubing	**Impact Fatigue**	Erratic Operation, Poor Appearance	Overstressing	8	Reinforce tubing with steel plate
Steel Tubing	**Impact Deformation**	Improper Geometry, Suspension Binding	Impact Loading	8	Reinforce tubing with steel plate
Side Support Steel Plate	**Impact Fatigue**	No Longer Operational, Erratic Operation	Assembly Errors	96	Maximize welding surface and use cross members
Side Support Steel Plate	**Impact Fracture**	No Longer Operational, Flying Debris	Impact Loading	96	Maximize welding surface and use cross members
Shock Hardware	**Impact Fatigue**	No Longer Operational, Poor Appearance	Overstressing	12	Choose higher grade hardware
Shock Hardware	**Impact Fracture**	No Longer Operational, Flying Debris	Impact Loading	12	Choose higher grade hardware

6.1.2.2 Hub/CV/Spacer/Wheel Mounts

The hub, CV, spacer, and wheel mounts were all validated as one system since they work off of each other. The hub spline had the third highest RPN value of 15 and was addressed by calculating the correct tolerance fit to ensure there is no slipping while the car is driving. The CV spacer had the second highest RPN value of 24 and was resolved by using steel over aluminum to ensure there is no room for failure. The highest RPN value for this system of 30 belonged to the steel hex nut at the end of the CV splines because it is responsible for keeping the bearings and knuckle in place. This issue was resolved by ensuring the right spacing of the hub on the CV spindle end.

Table 31: Hub/CV/Spacer/Wheel Mounts FMEA

Part # and Functions	Potential Failure Mode	Potential Effect(s) of Failure	Potential Causes and Mechanisms of Failure	RPN	Recommended Action
CV Axle End	**Impact Fatigue**	Erratic operation, Poor performance	Overstressing	0	Will not happen.
CV spline	**Slipping**	Loss of power transferred to the wheel	Too high of tolerance	15	Ensure tight fit for hub and spline
Bolts from hub - wheel	**Shearing**	Flying Debris/ No longer operational	Material selection	14	Choose high quality hardware
Nuts	**Stripping**	Flying debris/ No longer operational	Assembly error	10	Choose high quality hardware
Arms to wheel	**Impact Deformation**	Flying debris/ No longer operational	Impact Loading/ Overstressing	30	Choose high quality hardware
Steel Hex Nut	**Stripping**	Flying debris/ No longer operational	Overstressing/ Assembly Error	30	Choose high quality hardware
CV Spacer	**Impact fatigue**	Moving parts on CV/ Poor performance	Overstressing	24	Use steel instead of aluminum

6.1.2.3 Camber Links

The camber links are a major component of the rear suspension system and required a thorough validation process to eliminate potential failures. The RPN value of 45 belonged to the carbon fiber tubing that will be used in the upper camber links. The carbon fiber tubes will be tested in tension and compression to ensure that they will survive the baja competition. Titanium bolts had the same RPN value of 45 and the potential failures will be mitigated by ensuring the adequate diameter bolt is used in each necessary location. The rod ends had the highest RPN value of 54 mainly due to the fact that they are a connection point between the solid camber link tube and the rod end. It is essential that the correct epoxy or welding procedure is used on the rod ends to prevent significant failure.

Table 32: Camber Link FMEA

Part # and Functions	Potential Failure Mode	Potential Effect(s) of Failure	Potential Causes and Mechanisms of Failure	RPN	Recommended Action
Carbon Fiber tube	**Impact Fracture**	Detrimental Failure of Rear Suspension	Impact Loading	45	Extensive Testing Under Different Loads
Aluminum Composite threaded Insert	**Surface Fracture**	Detrimental Failure of Rear Suspension	Overstressing	12	Extensive Testing Under Different Loads
Steel Tubing	**Impact Fatigue**	Difficult and Unpredictive Performance	Impact Loading	36	Strong Welds
Rod End	**Impact Wear**	Difficult and Unpredictive Performance	Impact Loading/Improper Maintenence	54	Proper Lubrication
Black-Oxide Screws	**Impact Fatigue**	Difficult and Unpredictive Performance	Overstressing	30	High Diameter, Small Pitch Screws
Titanium Screws	**Impact Fatigue**	Difficult and Unpredictive Performance	Overstressing	45	High Diameter, Small Pitch Screws
High Strength Glue	**Surface Fracture**	Detrimental Failure of Rear Suspension	Impact Loading	36	Extensive Testing Under Different Loads

6.1.2.4 Knuckle

The knuckle design is crucial for the rear suspension system to function properly when fully assembled meaning it was necessary to validate this part. The third highest RPN value of 18 was assigned to welds between the knuckle and the trailing link. To mitigate this failure, point more reinforcement was added to this location with more surface area to weld with. The second highest RPN value of 24 belongs to the CV bearings which will experience a high amount of radial loading during operation. Maintaining the bearings with proper maintenance and ensuring the proper tolerance fit is machined into the knuckle will ensure that the bearings last the life of the car. The steel round bar makes up the main structure of the knuckle and received a RPN value of 42. If the steel round bar fails than the car become inoperable meaning that enough material needs to be maintained in the knuckle to prevent catastrophic failure.

Table 33: Knuckle FMEA

Part # and Functions	Potential Failure Mode	Potential Effect(s) of Failure	Potential Causes and Mechanisms of Failure	RPN	Recommended Action
Steel Round Bar	**Impact Fatigue**	Erratic Operation, Poor Appearance	Overstressing	42	Maintain adequate amount of support material
Steel Round Bar	**Impact Fracture**	No Longer Operational, Flying Debris	Impact Loading	42	Maintain adequate amount of support material
Camber Link Mounts	**High-Cycle Fatigue**	Erratic Operation, Poor Appearance	Overstressing	8	Maximize welding surface
Camber Link Mounts	**Buckling**	No Longer Operational, Flying Debris	Overstressing	8	Maximize welding surface
Camber Link Hardware	**Impact Fatigue**	No Longer Operational, Poor Appearance	Overstressing	6	Choose high quality hardware
Camber Link Hardware	**Corrosion Fatigue**	Erratic Operation, Poor Appearance	Poor Maintenance	6	Choose high quality hardware
CV Bearings	**High-Cycle Fatigue**	No Longer Operational, Flying Debris	Impact Loading	24	Use oversized single roller bearings
CV Bearings	**Abrasive Wear**	Erratic Operation, Poor Appearance	Poor Maintenance	24	Use oversized single roller bearings
Trailing Link Weld	**Impact Fatigue**	Erratic Operation, Poor Appearance	Overstressing	18	Reinforce contact area with additional steel plate
Trailing Link Weld	**Impact Fracture**	Flying Debris, No Longer Operational	Impact Loading	18	Reinforce contact area with additional steel plate

6.1.3 Drive

6.1.3.1 ECVT

Table 34: ECVT FMEA

Part # and Functions	Potential Failure Mode	Potential Effect(s) of Failure	Potential Causes and Mechanisms of Failure	RPN	Recommended Action
Electronics Computation Module	Physical damage to computers	Incorrect Control Signals, damage to CVT primary sheaves due to overtravel	Debris ingress, mount failure, vibration fatigue of connections	84	ensure computation module housing is sealed to debris, mount housing with vibration dampening mounts
Electronics Motion Module	Encoder loss of position	Erratic CVT movement, damage to CVT primary sheaves due to overtravel	Extreme vibration, high electrical interference, lose cable connections	112	Ensure tight cable connections, route high amperage wires away from signal wires
Primary Sheave Assembly	Main Shaft fatigue failure	Loss of power transmission, damage to control motor, damage to belt	Lack of maintenance and inspection, higher than anticipated loads, wear from debris	30	Change main shaft to steel, properly inspect components for wear and replace within service life
Secondary Sheave Assembly	Fatigue failure of cam followers	Reduced max torque transfer, increased belt temperature leading to belt failure	Poor structural design, repeated high rpm CVT engagements from stopped	30	Inspect secondary moving sheave before use, replace within service life
Support Structure	Impact failure	catastrophic system failure, damage to all components	Massive crash, fatigue stress build over time leading to weakened structure	20	Don't crash

6.1.3.2 Rear Gearbox

For the rear gearbox, an FMEA analysis was done on all components to identify all potential failure modes, their effects, the severity of that failure mode, and the best action that could be taken to mitigate these. In the rear gearbox, the component with the highest RPN's were the shaft bearings, the gearbox seals, and the intermediate shaft. The shaft bearings have the highest value of RPN because their failure was based on a total failure involving a full lockup of the bearing. In this case there would be complete drivetrain failure with no power being transferred to the wheels, and the issue would be nearly impossible to diagnose without disassembling the rear gearbox. The solution to this failure is simply using ball bearings that are overrated for their life and load requirements. The second highest RPN value was the gearbox seals because a failure in this component would result in oil leaking form the gearbox, which is a black flaggable offense according to the Baja competition rules. The solution to this is also to use seals with a high factor of safety for their purpose to ensure that this failure does not occur. The third highest value of RPN was the intermediate shaft of the gearbox and this was due to the complexity of the shaft as well as the fact that its critical location is inside of the gearbox which would make diagnosis difficult. The remedy to this failure mode is to use a high strength steel (heat treated 4140 or 4340) and significant amounts of math to ensure that the shaft meets design requirements.

Table 35: Rear Gearbox FMEA

Part # and Functions	Potential Failure Mode	Potential Effect(s) of Failure	Potential Causes and Mechanisms of Failure	RPN	Recommended Action
Brake Components	**Fatigue Failure, Component Destruction**	Brake Failure	Brake line severed by terrain, excessive use	40	Protect brake components
Gearbox Seals	**Fatigue**	Transmission Fluid Leak	Overheating the gearbox, bearing failure	96	Use seals with a high FOS to mitigate failure risk
Shaft Bearings	**High-Cycle Failure**	Bearing Lock, Increased Friction Resistance	Cycle exceed design life	168	Use oversized ball bearings
Gears	**Contact Stresses, Fatigue Failure**	Teeth Wear/Striping, Higher Inefficiency, Failure to Transmit Torque	Material failure due to overuse, overheating, or unforeseen stress	40	Use heat treated 4140 steel for optimal strength
Input Shaft	**Fatigue Failure**	Material Yielding	Unforeseen stresses causing material failure	20	Use heat treated 4140 steel for optimal strength
Intermediate Shaft	**Fatigue Failure**	Material Yielding	Unforeseen stresses causing material failure	60	Use heat treated 4140 steel for optimal strength
Output Gear with Integrated CV Cups	**Fatigue Failure**	Material Yielding	CV cup failure or gear teeth failure due to overuse or unforeseen stresses	20	Use heat treated 4140 steel for optimal strength and employ a high FOS

6.1.3.3 Front Gearbox

An FMEA analysis was preformed to identify the potential modes of failure for the components of the front gearbox. The analysis, shown in Table 21, shows that the CV cup will fail due to overstressing and both input and output shafts have higher chances of fatigue due to overuse. To mitigate these failures, it is necessary to choose strong materials that can be heat treated so that it can improve wear resistance. Additionally, it is recommended to use oversized bearings for both input and output shafts so that the load is evenly distributed.

Table 36: Front Gearbox FMEA

Part # and Functions	Potential Failure Mode	Potential Effect(s) of Failure	Potential Causes and Mechanisms of Failure	RPN	Recommended Action
CV Cup	**Abrasive Wear**	Erratic Operation, Poor Appearance	Impact Loading, Overstressing	75	Increase material wall thickness
Sprag	**High-Cycle Fatigue**	No Longer Operational, Flying Debris	Impact Loading, Overstressing	30	Use oversized sprag clutch
Gear Bearings	**High-Cycle Fatigue**	No Longer Operational, Flying Debris	Impact Loading, Overstressing	45	Use oversized ball bearings
Output Gear	**Contact Fatigue**	Erratic Operation, Flying Debris	Overstressing	60	Increase hardness by heat treatment
Input Gear	**Contact Fatigue**	Erratic Operation, Flying Debris	Overstressing	60	Increase hardness by heat treatment
Output Shaft	**High-Cycle Fatigue**	No Longer Operational, Flying Debris	Assembly Errors, Overstressing	30	Increase hardness by heat treatment
Input Shaft	**High-Cycle Fatigue**	No Longer Operational, Flying Debris	Assembly Errors, Overstressing	30	Increase hardness by heat treatment

6.1.3.4 4 Wheel Drive

Each part involved in four-wheel drive integration will be made of either AISI 4130 or 4340 steel. The calculations that it took to design each part, check stress, forces, and factor of safety, and interactions with other parts were made with material in mind and the results is an assortments of parts designed to run together with no failure. However, we do not live in a perfect world. The FMEA table below has each item rated and what their largest potential modes of failure for those parts are. Shaft bearings noticeably have the higher RPN number. This is because it is one of the parts interacting with 3 different parts all at once, however it is the only parts that has forces related in just as many directions. Hence, why a potential cause could be an exceeded cycle life. In addition, no other part has a dangerous RPN rating and is within spec to withstand the potential cause of failure. The recommended action for the system as a whole is surprisingly simple, and that would be just to make sure that each part is exactly in the place it should be, secured in those locations and to use the system during the moments where the least amount of overall strain will be.

Table 37: 4 Wheel Drive FMEA

Part # and Functions	Potential Failure Mode	Potential Effect(s) of Failure	Potential Causes and Mechanisms of Failure	RPN	Recommended Action
Driving Clutch	**Material Failure**	Non-Operational	Tooth Shearing	40	Engage while off the throttle
Driven Clutch	**Material Failure**	Non-Operational	Tooth Shearing	10	Engage while off the throttle
Rear End Pulley	**High Force Failure**	Non-use of 4 Wheel Drive	Driving side clutch warps material	30	Engage while off the throttle
Front End Pulley	**Load Failure**	Non-use of 4 Wheel Drive	Belt load moves Circlip	120	Use heavy duty clip
Timing Belt	**High-Cycle Failure**	Non-use of 4 Wheel Drive	Exceed Tension Rating	24	Tension to specifications
Rear End Shaft	**Fatigue Failure**	Clutch will not be able to engage	Assembly Error	7	Lock down all non-moving parts
Shaft Bearing	**High-Cycle Failure**	Bearing Lock, Increased Friction Resistance	Cycle exceed design life	168	Use oversized ball bearings

6.1.4 Frame

6.1.4.1 Frame Members

All frame members will be made of AISI 4130 steel in their respective primary and secondary dimensions.

Some of the members of the roll cage have a higher likelihood of impact and have a higher chance of failure. These members include the front bumper, roll hoop overhead members, and side impact members. Other portions of the frame, while unlikely to have impact, are in high wear areas or are vitally important to the control of the vehicle. These include the seat mount, steering column mount and seat itself. These six failure points of the vehicle were subjected to an FMEA analysis. From the FMEA, the team concluded that the most important areas to consider in design analysis was the seat. If the seat were to fail during operation, the driver would have a very difficult time staying in control of the vehicle. The results of the FMEA for frame members is shown below in Table 38.

Table 38: Frame Member FMEA

Part # and Functions	Potential Failure Mode	Potential Effect(s) of Failure	Potential Causes and Mechanisms of Failure	RPN	Recommended Action
Bumper	Impact Fracture, Impact Deformation, Impact Fatigue	No Longer Operational, Poor Appearance	Assembly Errors, Impact Loading, Manufacturing Defect	18	Ensure Proper Assembly
Roll Hoop Overhead Members	Impact Fracture, Impact Deformation, Impact Fatigue	No Longer Operational, Poor Appearance	Assembly Errors, Impact Loading, Manufacturing Defect	10	Ensure Proper Assembly
Side Impact Members	Impact Fracture, Impact Deformation, Impact Fatigue	No Longer Operational, Poor Appearance	Assembly Errors, Impact Loading, Manufacturing Defect	10	Ensure Proper Assembly, Optimize Supportive Geometry
Seat Mount	Impact Fracture, Impact Deformation, Impact Fatigue	No Longer Operational, Poor Appearance	Assembly Errors, Impact Loading, Manufacturing Defect	16	Ensure Proper Assembly, Limit Impact Opportunities
Steering Column Mount	Impact Deformation, Impact Fatigue	Erratic Operation	Assembly Errors, Impact Loading, Manufacturing Defect	12	Ensure Proper Assembly, Optimize Geometry
Seat	Impact Fracture, Impact Wear	Safety Hazard, No Longer Operational, Uncomfortable	Manufacturing Defect, Impact Loading	24	Thicken material, Avoid High Stress Geometry

6.1.4.2 *Front Shock Mounts*

The main failure points for integration points with other subsystems in the vehicle are the mounting locations for the front and rear suspension platforms. Through the analysis in Table 39, the shock mounting tab was shown to be the highest risk failure due to the loss of control of the vehicle that would happen if the tab failed. Prevention for failure in this case would involve certifying all welders and double checking all welds afterwards to ensure proper installation.

Table 39: Front Suspension Mounts FMEA

Part # and Functions	Potential Failure Mode	Potential Effect(s) of Failure	Potential Causes and Mechanisms of Failure	RPN	Recommended Action
Front Shock Tab	Impact Fatigue, Loop around bolt Failing, Deformation	Non-functional Vehicle, Incorrect front geometry Poor handling	Poor welds, Incorrect placement	45	Verify Welds, Ensure proper placement before final welds occur
UCA Tabs	Impact Fatigue, Loop around bolt Failing, Deformation	Non-functional Vehicle, Incorrect front geometry Poor handling	Poor welds, Incorrect placement	30	Verify Welds, Ensure proper placement before final welds occur, In line with other UCA tab
LCA Tabs	Impact Fatigue, Loop around bolt Failing, Deformation	Non-functional Vehicle, Incorrect front geometry Poor handling	Poor welds, Incorrect placement	30	Verify Welds, Ensure proper placement before final welds occur, In line with other UCA tab

6.1.4.3 *Rear Shock Mounts*

Similar to the front mount points, the rear suspension FMEA analysis in Table 40 compare all mounting points for the rear suspension assembly. The rear shock tab was found to be highest risk, followed closely by the tab for the trailing link. These two points are the main two mounting locations for the assembly and would lead to catastrophic failure of the subsystem if they were to fail. As with the front suspension points, mitigations involved ensuring all welds are properly done and installation of the tab is up to standard before driving the car.

Table 40: Rear Suspension Mounts FMEA

Part # and Functions	Potential Failure Mode	Potential Effect(s) of Failure	Potential Causes and Mechanisms of Failure	RPN	Recommended Action
Rear Shock Tab	Impact Fatigue, Impact Wear, Surface Fatigue Wear	Poor Handling, No Longer Operational, Poor Appearance	Incorrect Assembly, Overstressing	64	Ensure Proper Assembly, Use Thicker Material
Trailing Link Tab	Impact Fatigue, Impact Wear, Surface Fatigue Wear	Poor Handling, No Longer Operational, Poor Appearance	Incorrect Assembly, Overstressing	56	Ensure Proper Assembly, Use Thicker Material
Upper Camber Link Tab	Impact Fatigue, Impact Wear, Surface Fatigue Wear	Poor Handling, No Longer Operational, Poor Appearance	Incorrect Assembly, Overstressing	32	Ensure Proper Assembly, Use Thicker Material
Lowe Camber Link Tab	Impact Fatigue, Impact Wear, Surface Fatigue Wear	Poor Handling, No Longer Operational, Poor Appearance	Incorrect Assembly, Overstressing, Impact Loading	32	Ensure Proper Assembly, Use Thicker Material

6.2 *Initial Prototyping*

As the next step in engineering design validation, each team constructed a series of physical and virtual prototypes to demonstrate elements of their sub-system with the goal of resolving outstanding design questions.

6.2.1 Front

The first design question that the front end had was centered around the behavior of the suspension geometry that was established in the team's suspension software, Lotus *Shark*. This geometry was generated over dozens of iterations until it was fully optimized and reacted in a manner approved by all sub-team members. These geometric hard points were converted from *Shark* into SolidWorks and realistically modeled with control arms, a steering rack, a knuckle/hub combo, and a functional shock (Figure 101).

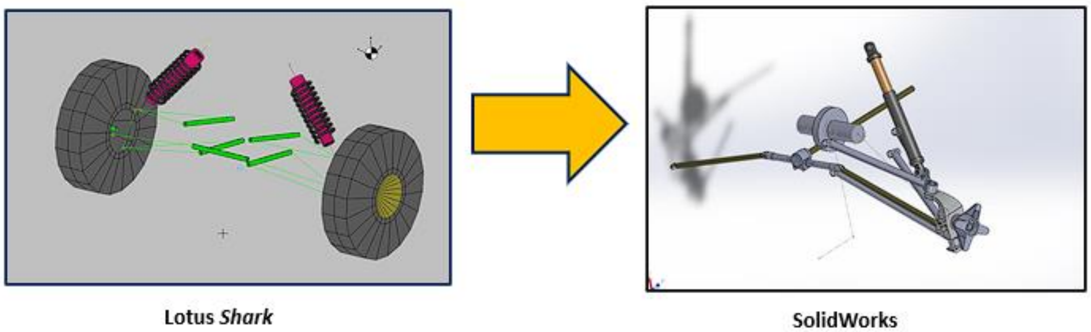


Figure 101: Front End Suspension Geometry

This geometry behaved well in theory but needed to be validated in the real-world. To do so, the team constructed a mounting board that simulated the driver's side of the chassis, constructed PVC control arms (see Section 6.3.1.1 and 3D printed mounting tabs, a knuckle, and misalignment spacers for the spherical joints on the top and bottom of the knuckle assembly. The design was fully assembled using realistic hardware and functioned in a realistic manner complete with a mock shock that slid within the design's expected range of motion. The prototype revealed that the ball joint cups on the upper and lower control arms needed to have their mounting angles altered to avoid interference with the knuckle mounting surfaces during full compression and full extension motions. Moving forward, the team will be revising these ball joint cups to account for full mobility of the system. See Figure *102* for an annotated graphic of this discussion as well as documentation of the physical prototype itself.

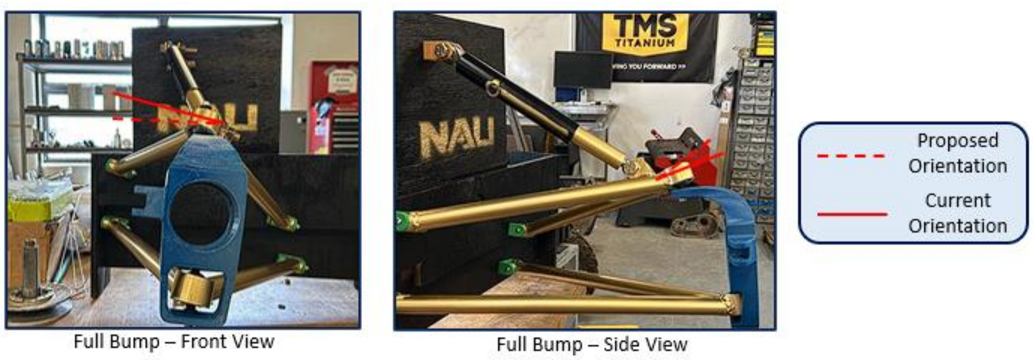


Figure 102: Front End Physical Prototype

The team updated its CAD to reflect the lessons learned from the physical prototype and performed a virtual motion study to prototype the assembly's full articulation. The design question this time is to see how all parts in the updated front end assembly mesh and move relative to each other during compression and extension of the shock with the intention of interference identification. A motion study was conducted in SolidWorks that verified the functionality of the updated assembly; though this animation could not be presented in this document, see Figure *103* below for an annotated static example.

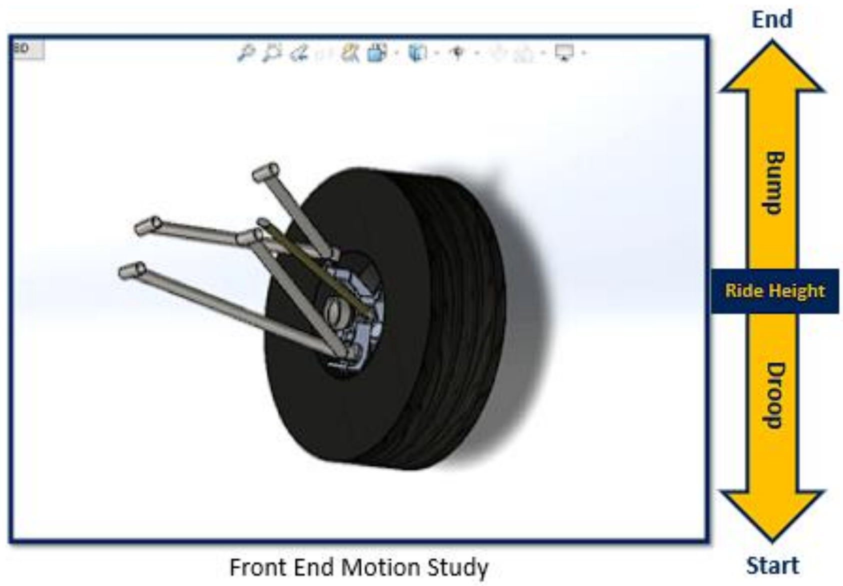


Figure 103: Front End Virtual Prototype

This virtual prototype informed the team that their design was operating successfully at this point in the semester, meaning all effort going towards geometric confirmation could be diverted towards completion of collective design optimization and the design remaining smaller components in the assembly (brake rotor, gas/brake pedal, etc.)

6.2.2 Rear

Similar to the front team, Rear also prototyped the whole suspension geometry as seen in CAD. This allowed the team to be able to visualize better and have a physical item to look at. This was utilized to reinforce the paths of the wheel under bump and droop conditions. The team hoped to confirm a lot of the characteristics or ideas the team has been thinking about all semester. As well as finding holes in some of these engineering designs and ideas.

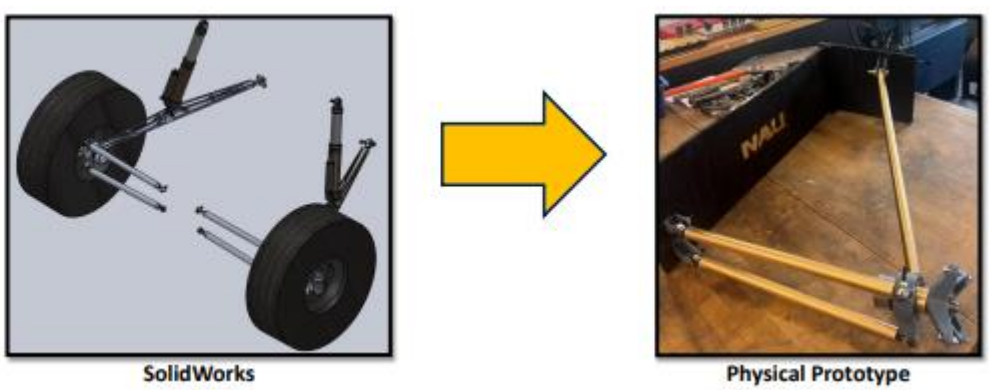


Figure 104: Rear end physical prototype

The team purchased PVC for this prototype. Once all of the parts were 3D-printed and assembled the team was able to see some minor issues with the system we have been working on. One of the main issues included the mounting bracket on the knuckle for one of the camber links. The bracket had a proposed changing the angle to be in line with the link. This will allow a smoother motion for the system throughout the travel.

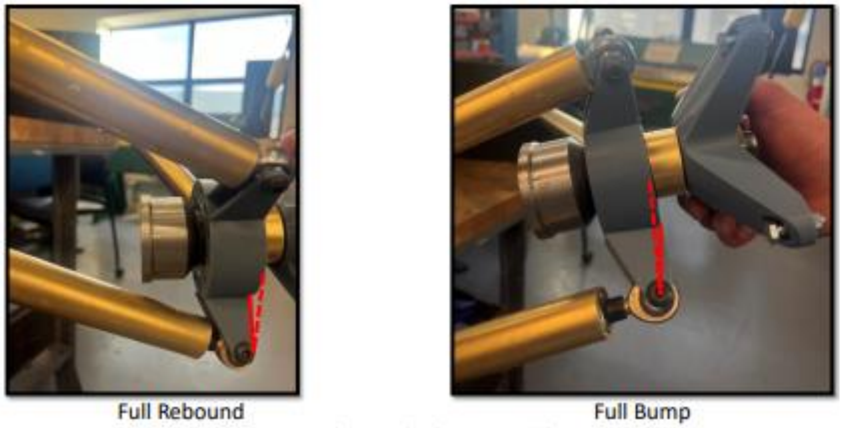


Figure 105: Proposed fix to the mounting tab.

The next minor issue uncovered was in the hub. This was seen during the assembling process. The wheel bolts had issues inserting into the wheel. Also, the holes were way small. The fixes for this were to measure the wheel more accurately and update the CAD and use 3/8 instead of 5/16 bolts.

Figure 106: Issue where the holes on hub do not line up to wheel holes. Also, the camber link's rod end is were too big for the knuckle.

The teams virtual prototype simulated the motion of the suspension member in Lotus SHARK. This allowed for the team to ensure that the design the team has been modifying in SolidWorks was feasible. The team refreshes the points in SHARK every now and then to double check the design is still operating in good fashion.

6.2.3 Drive

6.2.3.1 ECVT

6.2.3.2 Rear Gearbox

Below, in Figure *107*, and Figure *108* are the rear gearbox prototype profiles. These 3D prints were done based on the results of the MATLAB script described below. The goal of this model was to determine if the current design is adequate in terms of manufacturing concerns, spacing, and sizing. It also gave the team a better understanding of the overall size of the design as well as how well it will integrate into the other system in which it is connected. The gearbox housing required further optimizing at the time of the 3D print, so it was not included in the initial prototype as the team was more interested in the design of the internal components than the package in which they are going to be stored.

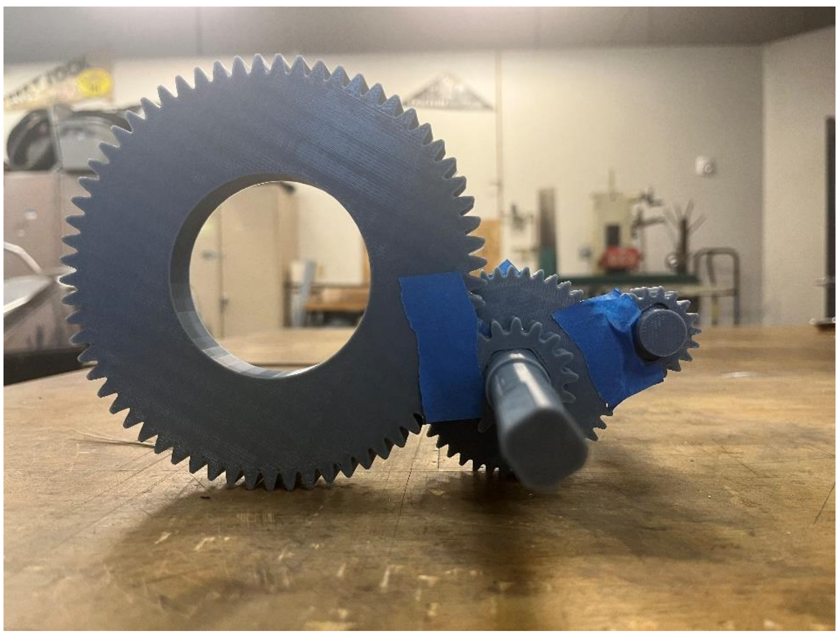


Figure 107 : Rear Gearbox Prototype Side View

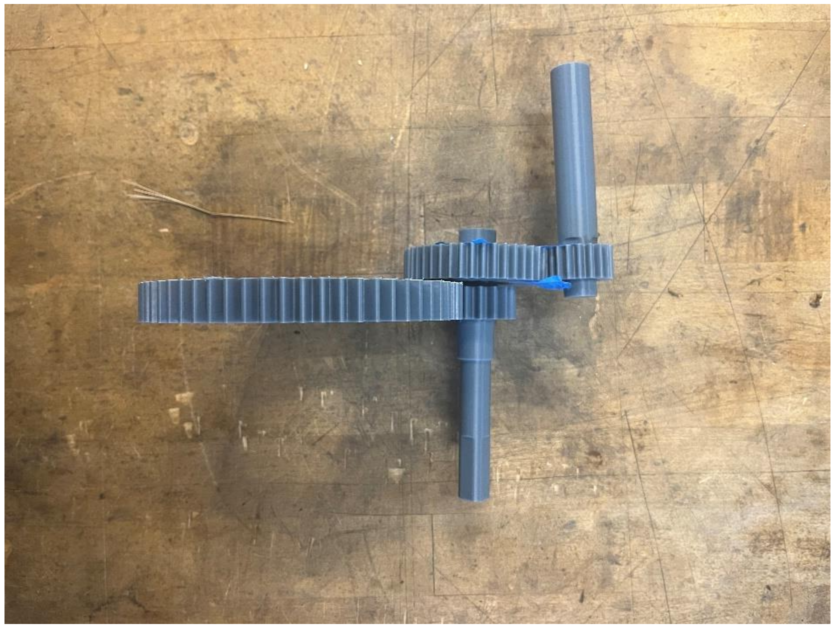


Figure 108 : Rear Gearbox Prototype Top View

6.2.3.3 Front Gearbox

One the necessary components that the team has designed is an integrated CV cup that would attach to the output shaft of the front gearbox. Eliminating the current CV cup and designing one around the front gearbox has influenced many challenges in achieving the correct plunge geometry. The CV cup was first modeled in SolidWorks by taking precise measurements of the current CV cup. Verifying the correct wall thickness, an FEA analysis was conducted on the stresses that would occur, which is detailed in Section 6.3.3.3. After extensive calculations and determining correct plunge geometry, the model (Figure 109) was then 3D printed and tested.

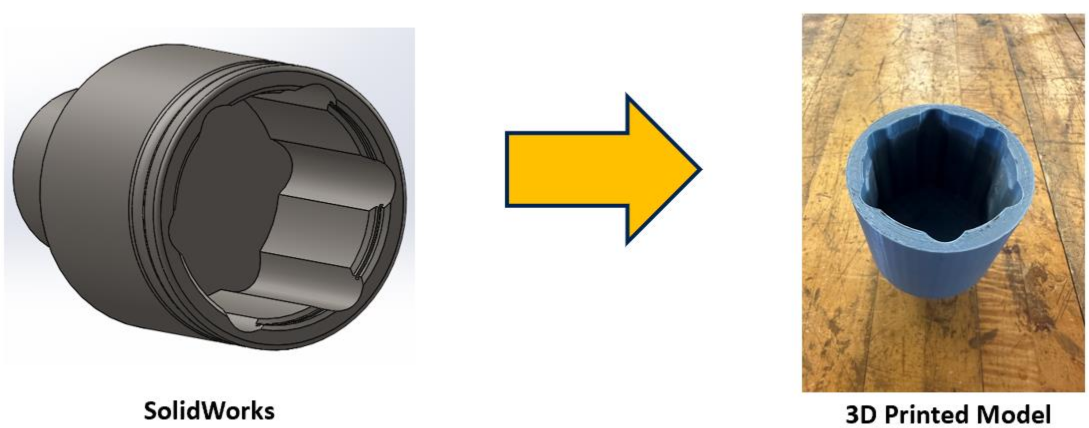
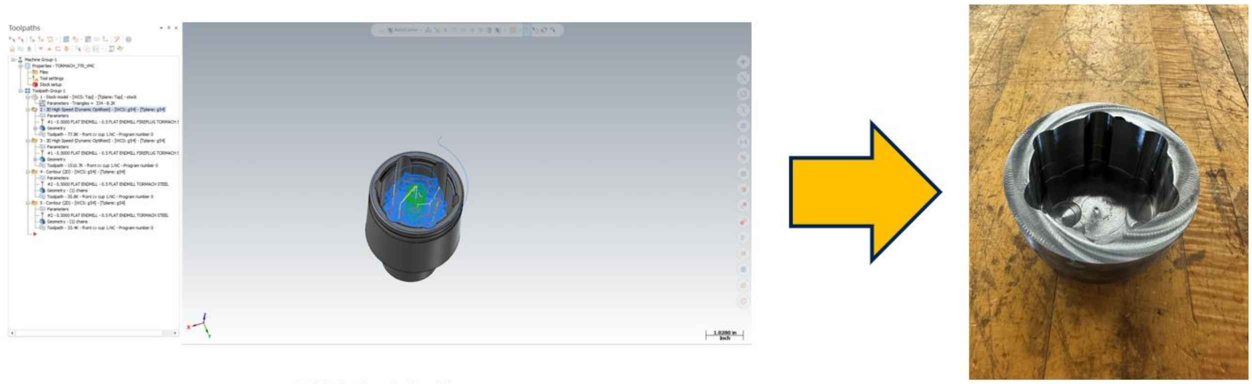


Figure 109: CV Cup Prototype

Based on the prototype, the inner race was fitted tightly into the CV cup and had achieved the team's goal of determining the correct plunge geometry. However, due to the nature of 3D printing and its reliability of being exact, the team has opted to increase the pitch diameter of the center circle so that there is less of a tight fit. To test the new geometry, the team decided to machine the part using CAM Tool Paths, shown in the figure below.

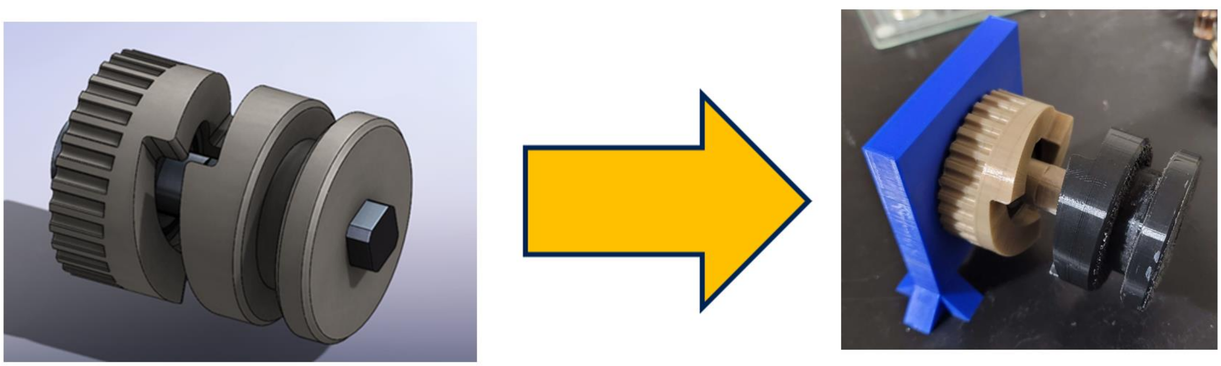


CAM Tool Paths **CNC Model**

Figure 110: CV Cup Machined Prototype

As a result, the team noticed that there was more side to side movement when the inner race was placed inside the CV cup. This is not optimal since it can cause premature wear of the inner grooves. Therefore, the team has decided to use the original geometry and when machining the final iteration, the team will carefully machine material off the side walls and test to see if the inner race has minimal movement.

6.2.3.4 Dog Clutch



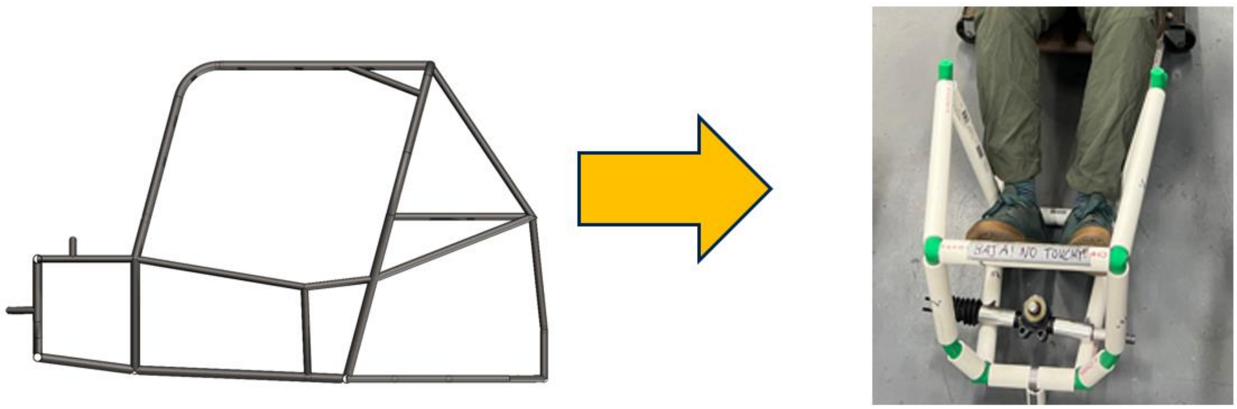
SolidWorks Physical Prototype

Figure 111: Clutch and Coupling Prototype

Based off of how the prototyped behave and what we could get it to do we ow know that the pitch on both of the pulleys are spot on and should mesh with the actual belt just fine. However, we notice that just 3D printed that with an ample amount of clearance the driven side of the shaft does not move as smoothly as we would like it too. To fix that we took a look at the material we are going to be using and know for a fact that if we leave the driven clutch side and the shaft alone that steel will not like nor let more steel rub up against it so we are looking into slightly changing the driven side clutch design to insert a brass fitting.

6.2.4 Frame

The primary job of the frame sub team is to combine all other teams in a manner that conforms to the strict rules set by the SAE governing body. One of these integrations involves a set of rules pertaining to the driver of the vehicle during race events. In order to ensure that all requirements for driver restrictions were met, the frame team built a 1:1 model of the roll cage to conduct accurate measurements. The main concern for the team was the width of the footbox and if it was too narrow for proper use.



SolidWorks **PVC Model**

Figure 112: Frame Footbox Prototype

Through the prototyping process, the team found that while the current design of the footbox provides adequate width for proper usage, some other dimensions in the seating area had changed and no longer met specification. Theses discoveries led the team to increase the height of the overhead roll hoop bars and increase the width of the roll hoop to accommodate specific rules set by SAE for driver clearance. After these changes, the team has finalized the design of the roll cage and is now in the process of production.

6.3 Other Engineering Calculations

In a similar manner the previously mentioned engineering calculations, further engineering calculations were performed to identify design issues and resolved them via analytical assessment.

6.3.1 Front

6.3.1.1 Abraham Plis

The first engineering calculation performed for front end was centered around the shoulder bolt sizing that will be used on the pivots of the control arms. Debates over bolt sizing came up in the prototyping stage with the two main options being a 1/4" diameter shoulder bolt or a 3/8" diameter shoulder bolt. Most reasonably priced shoulder bolts are constructed out of grade 8 steel, which has a tensile strength of 150,000 psi, a ratio of shear to tensile strength of 0.6, and a dynamic load factor of 0.9. The max impact load case that will be used is 2160 lbf for which the derivation can be seen in Section 3.3.1.1

$$F_{shear,bolt} = S_{tensile} * (0.6) * (0.9) * A_{bolt}$$

Equation 28: Force Resistance of Bolt

$$A_{bolt} = 2 * \frac{\pi}{4} * d_{bolt}^2$$

Equation 29: Area of Bolt

$$FoS = \frac{F_{shear,bolt}}{F_{shear,impact}}$$

Equation 30: Factor of Safety

Plugging in all numbers from the information presented above yields a factor of safety of 3.68 for the 1/4" shoulder bolt and 8.28 for the 3/8" shoulder bolt. These results illustrate that a 1/4" bolt could be utilized at the pivots but a 3/8" bolt will be much stronger and increase the team's confidence in their design in the long run.

Another engineering calculation is presented in the form of annotated engineering drawings of the upper and lower control arms (Figure *113*). These drawings were generated as part of the construction phase for prototyping and allowed the team to assemble control arms that were nearly identical to their counterparts in SolidWorks. Attributes such as tubing length, joint angle, and ball joint cup offset were called out that facilitated physical construction of the arms using PVC tubing.

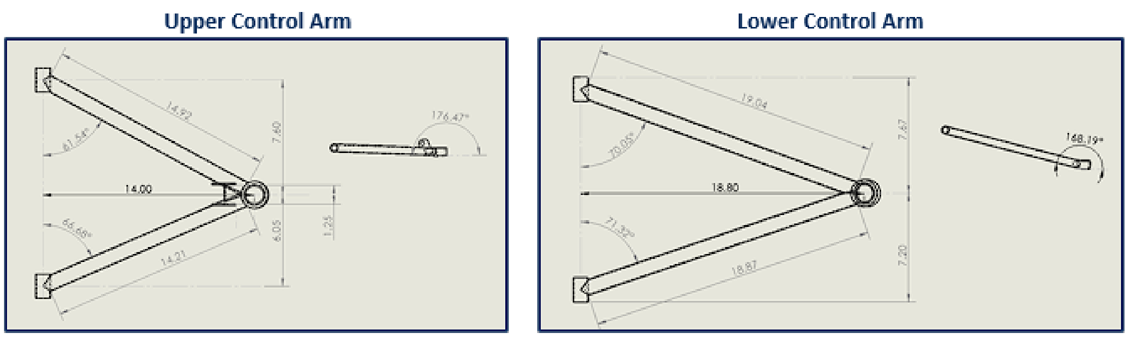


Figure 113: Annotated Engineering Drawings of UCA and LCA

6.3.1.2 Bryce Fennell

Engineering calculations were performed on the interface between the upper control arm and the knuckle. At this interface a ¼"-20tpi bolt will fix the upper control arm swivel joint spacer to the upper neck of the knuckle with a fixturing nut attached on the lower end. This design can be seen in the figure below displaying the interface.

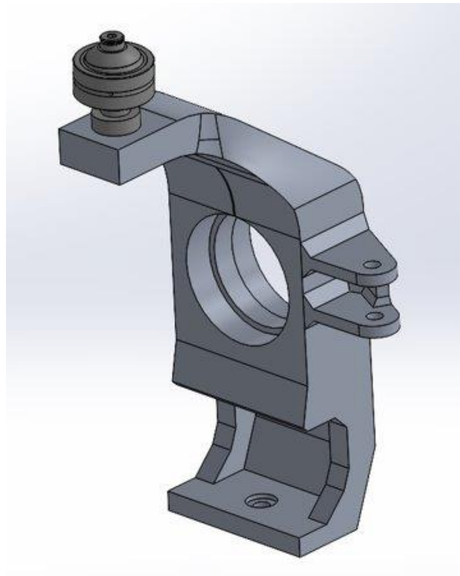


Figure 114: Front Driver Knuckle With Upper Control Arm Mounting Interface

Using 6061-T6 aluminum billet for the knuckle material with an impact force of 2200lbf through the wheel calculations were performed to determine the Factor of safety of the neck of the knuckle. Using the shear and cross-sectional area formulas seen below, an expected shear force is calculated.

Equation 31: Equations for Shear Stress and Cross Sectional Area

$$\boldsymbol{F_{shear,Aluminum}} = \mathrm{S_{shear}} * \mathrm{A_{Minimum\ Cross\ Section}}$$

$$\boldsymbol{A_{Minimum\ Cross\ Section}} = base * height \quad \boldsymbol{FoS} = \frac{F_{shear,bolt}}{F_{shear,impact}}$$

Using the shear strength of 6061-T6 aluminum at 3770 KPsi a factor of safety of 1.45 was determined. This factor of safety being above 1 indicates the knuckle will withstand the impact; however, this lower factor of safety will be improved upon to keep the vehicle safe.

6.3.1.3 Evan Kamp

To validate the steering design and in particular the carbon steering arm, the forces acting on the tie rod must be calculated. To do this the force acting on the rod is calculated assuming that the car is at rest and on asphalt.

Table 41 - Steering Force Calculation

Hypothetical Weight of Vehicle with Driver	450lbs / 205kg

½ (450lbs) = 225lbs on the front two tires
Under the Assumption that there is a perfectly centered center of gravity

Cornering Mass on One Front Wheel	112.5lbs / 51.25kg

Friction Force Calculation

$$f = \mu N$$

Friction Coefficient for Asphalt	.9
Normal Force (N)	3622.5 lb(ft/s^2)
Force of Friction on front two wheels	3260.25 lb(ft/s^2)

Contact Patch (Yellow)	7.5in / 0.625ft
Torque due to friction force on Wheel	2037.5 lb(ft/s^2)ft
Lateral Push Distance (Orange)	4 in / .333ft

Equation 32 - Torque due to Lateral Push

$$T_{lpush} = f_t * Distance\ from\ Tire\ Rod\ to\ Kingpin\ Axis$$

These calculations will be further applied to the calculations for shear force for an epoxy gnarled insert that will be used in the carbon construction of the tie rod.

6.3.2 Rear

6.3.2.1 Seth DeLuca

This calculation is the full FEA on the hub. Below is the final hub:

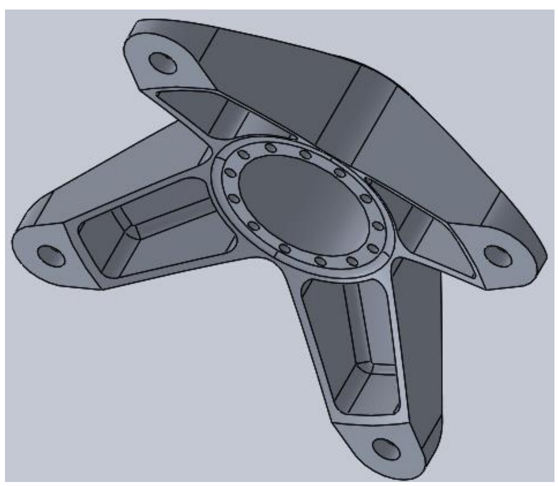


Figure 115: Final Hub Model

The first thing to consider when evaluating FEA is the forces acting on the hub for "the worst-case scenario", this is when all of the forces are going to be at their peak. This is assumed when the car is landing on one of the rear wheels and while braking. This force coming down was reasonably assumed to be 4Gs. This was applied to the center hole as a bearing force. This applies more of a parabolic force as opposed to a distributed force. A torque was then applied simulating the braking torque of 250 lbf ft. Finally, the mounting holes are all fixed in the simulation. The Factor of safety could theoretically be one since the likely hood of the "worst case scenario" is kind of exaggerated, however the team decided to shoot for a factor of safety of 2. The FEA study was ran giving a FoS of 8. This was a little high, so material was subtracted and manipulated until this was lowered. After this iterative process was finalized the FoS was found to be 2.27. This provides a little bit of extra strength to the hub while ensuring this will be a light part. The results from the final analysis are shown:

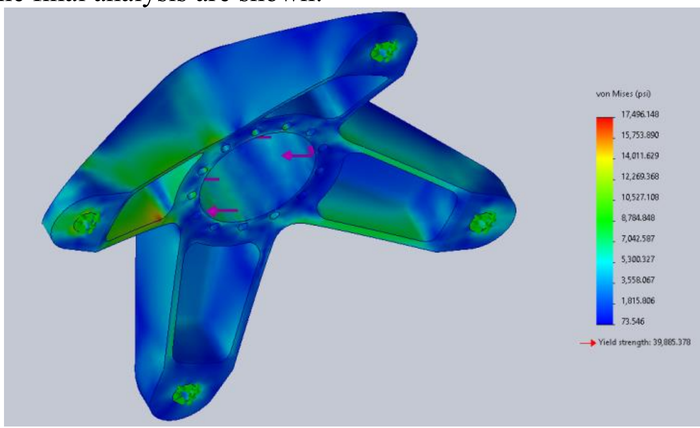


Figure 116: FEA analysis with final iteration of the hub.

This shows the highest stress points are around 18 KSI. This gives a factor of safety of 2.27 using 6061-T6 Aluminum. This FoS is what the team wanted now it was time to look at fatigue life. This is from showing conditions the hub will see repeatedly. For this the braking torque was the same however the bearing load was reduced to 500 lbs. This study was ran and shown below:

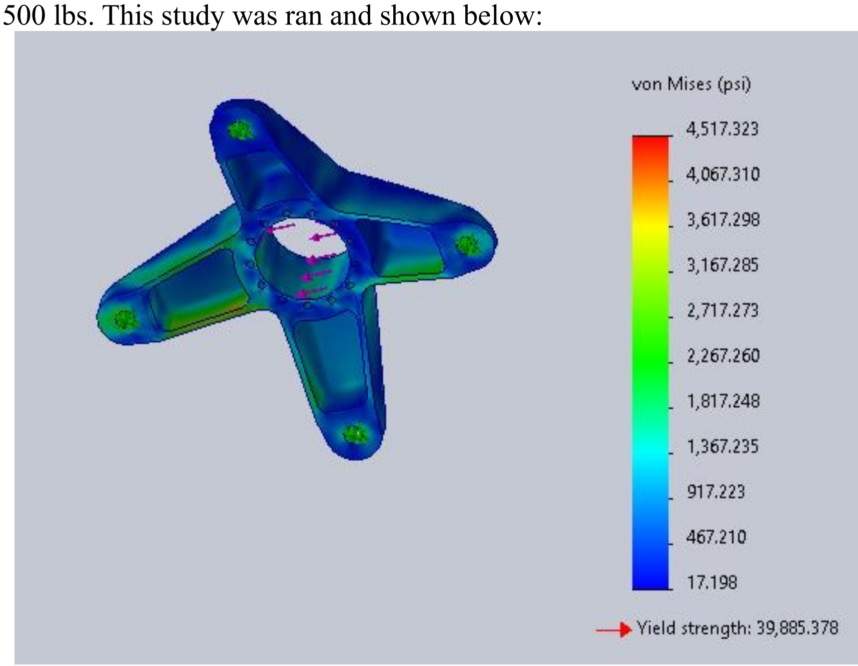


Figure 117: Fatigue FEA on the hub

For this analysis the team is less interested in the factor of safety and more interested in number of cycles the hub could see. The team wants the hub to have infinite life to evaluate this the team used a SN diagram, shown below.

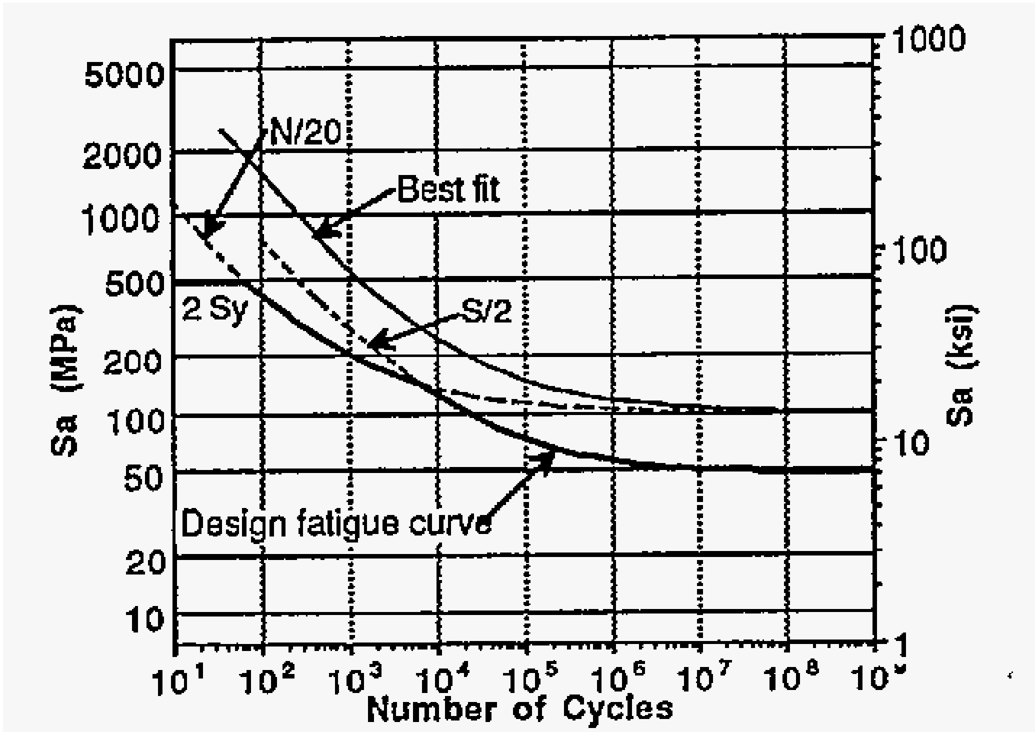


Figure 118: SN diagram of 6061-T6 Aluminum

This SN diagram shows infinites cycles need be less the 7 KSI. From Figure 117 the highest stress is 4.5 KSI. This shows we will have near infinite life on the hub.

6.3.2.2 Joey Barta

Assumptions and B.C.

$$R_o = 0.31495in$$
$$R_i = 0.2362in$$
$$L = 16.13\ in$$
$$L_{eff} = L \times k = L$$
$$V = 2.1994\ in^3$$
$$A_{cs} = 0.136355\ in^2$$
$$I_z = \left(\frac{m}{2}\right) \times \left(R_o^2 + R_i^2\right) = 0.010191\ in^4$$
$$R = 0.27338$$

$$E = 50{,}763{,}199.98\ psi$$
$$\sigma_y = 1{,}547 - 467{,}000\ psi$$
$$\rho = 0.0614 \frac{lb}{in^3}$$
$$m = 0.131510\ lb$$
$$Pinned - Pinned \gg k = 1$$

$$If\ S < S_{crit}\ use\ Johnsons\ Formula$$

$$Johnsons\ Formula = \sigma_y \times A_{cs}\left[1 - \left(\frac{\sigma_y}{4\pi^2 E}\right)\left(\frac{LE}{R}\right)^2\right]$$
$$Eulers\ Formula = \frac{\pi^2 EI}{L_{eff}^2}$$

Equation 33: Johnson's and Euler's Formulas

Figure 119: CF Rods

Results

- $F_{crit} = 1{,}547\ lbf \gg 26{,}257\ lbf$
- With an assumed minimum σ_y of 11,600 psi, the minimum F_{crit} is 1,547 lbf. The significance of this force will be explained in the Design Validation Slide.

The engineering calculations leading max stress calculations were governed by both, the Johnson's and Euler's formulas. The dimensions of the rod shown above were paired with an estimated modulus of 50,763,199.98 psi and an estimated yield strength of 11,600-467,000 psi (large gap from comparing tensile to compressive strength). Assuming a pinned member, Euler's formula was used with the low yield strength and Johnson's was used with the max yield strength. These formulas outputted critical forces of 1,547 lbf – 26,257 lbf. Assuming a max load of 4g's, this gives a FoS of 1.52 which is ample room for error at this time.

6.3.2.3 Lars Jensen

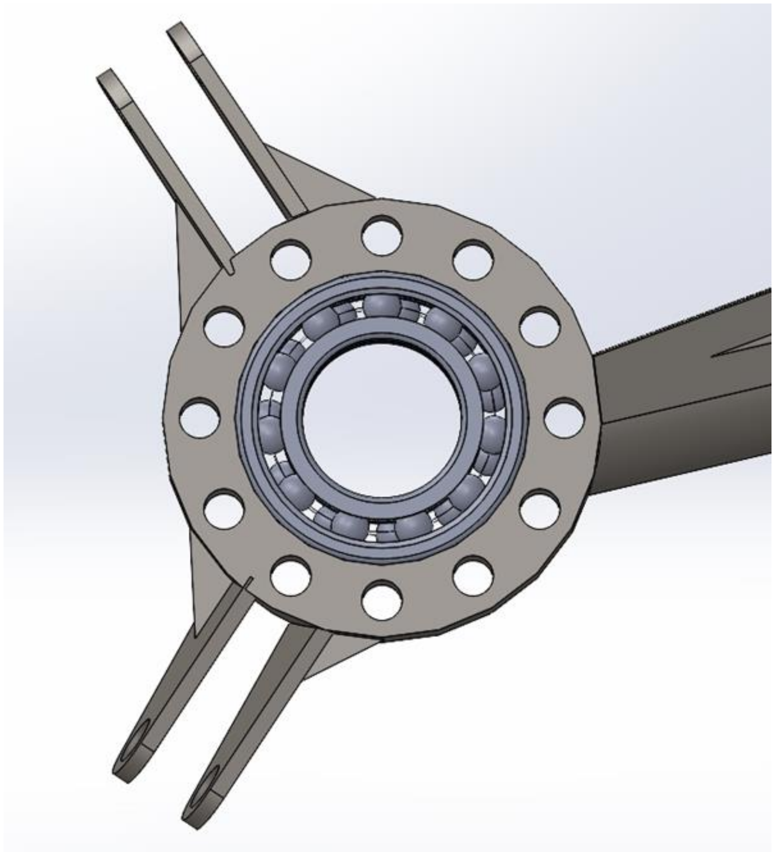


Figure 120: Rear Knuckle Bearings

Equation 34: Max Bearing Fit Diameter

$$D_{max} = D + \Delta D = 2.165\ in + 0.0012\ in = \mathbf{2.1662\ in}$$

Equation 35: Min Bearing Fit Diameter

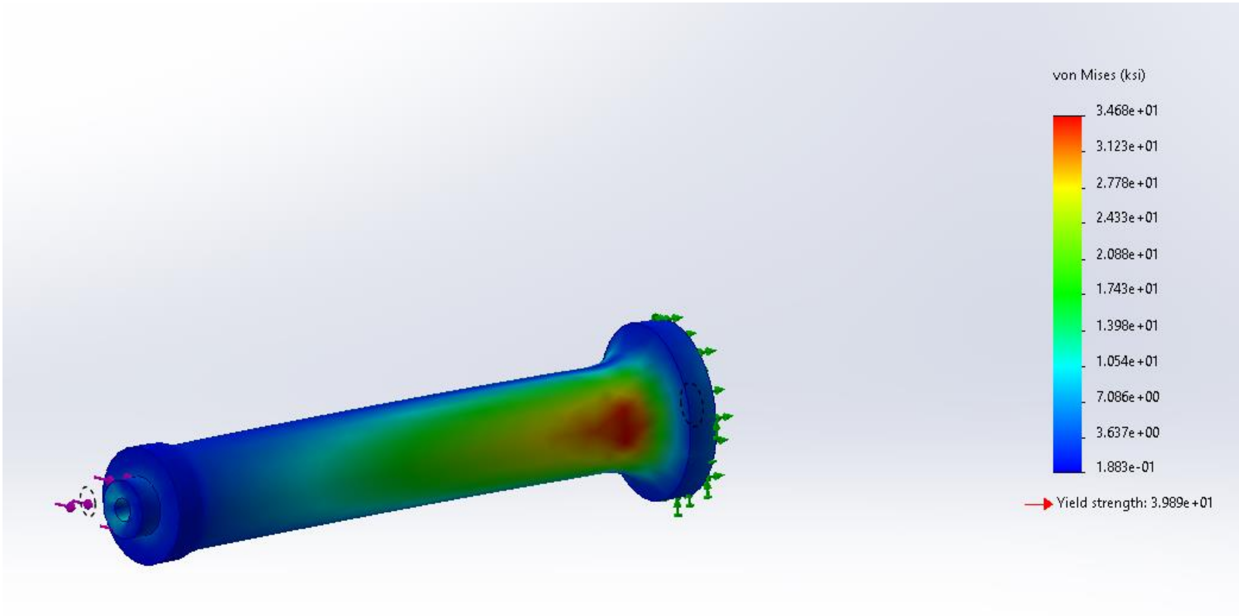
$$D_{min} = D = \mathbf{2.165\ in}$$

The engineering calculation shown above in Figure 120 explores the diameter that should be machined into the knuckle to ensure the correct fit for the two SKF 6006 deep groove ball bearings. The outside bearing diameter is 2.165 in, and the parameters are a force fit, H7/u6, and IT7. With the defined parameters, it was determined that the minimum diameter for the machined hole is 2.165 in and the maximum diameter for the hole is 2.1662 in. This information will be used during the machining process of the rear suspension knuckles.

6.3.3 Drive

6.3.3.1 Henry Van Zuyle

Calculations for the strength of the standoffs that support the motor mount plate were performed to ensure the motor will be stable. It also confirms that the deflection won't be greater than the clearance between the drive nut and primary main shaft. The force selected was equivalent to a 20g impact.



6.3.3.2 Ryan Fitzpatrick

For the initial prototyping, the calculation that went into making the design that was then 3D printed were conducted in MATLAB. The MATLAB script takes in the inputs of the geometric spacing of the shaft components for the input, intermediate, and output shafts, as well as the calculated values for reaction forces on that shaft and generates shear and bending moment diagrams for each of the shafts. The script also outputs the critical locations on the shafts and using these calculated the minimum diameters required at this critical point for the shaft to handle the bending moment and torque. Below in Figure 121, Figure 122, and Figure 123 are examples of the output of this MATLAB script. As can be seen in Figure 122, the factors of safety for the input and intermediate shafts, FOS A = 1.0921 and FOS B = 1.0429, respectively, are as close to one as possible. The reason for this design choice is because there are numerous other design factors of safeties built in already. The most significant factor of safety already built into the design is that all of the components are designed around the maximum values for torque, horsepower, and bending moment that the system can possibly experience based on engine and eCVT output. In its operation, the system is only going to experience these maximum values very briefly and rarely during moments of maximum acceleration and it is never going to experience max torque and HP at the same time because the two occur at different points in the engines powerband. Getting the FOS for each shaft critical location as close to 1 as possible minimizes material and decreases weight which improves the performance of the design and the car as a whole.

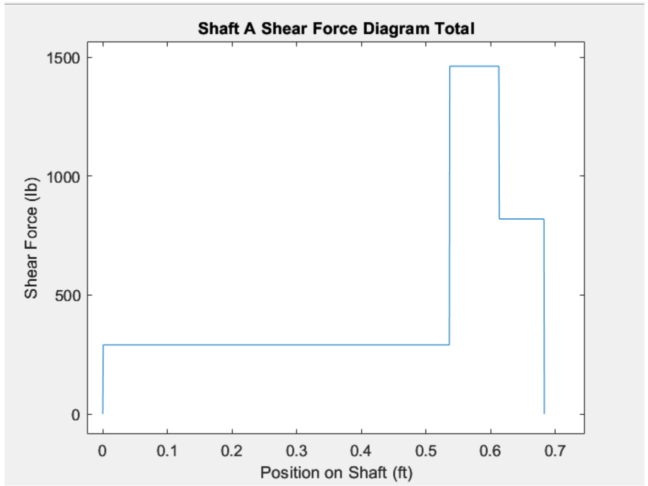


Figure 121 : Rear Gearbox MATLAB Shear Force Diagram

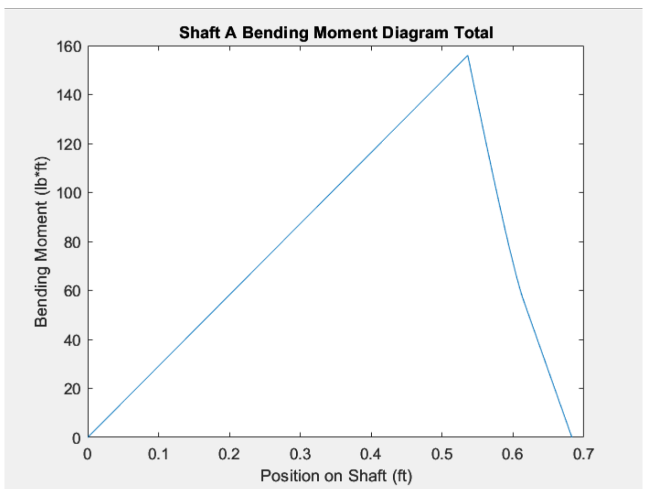
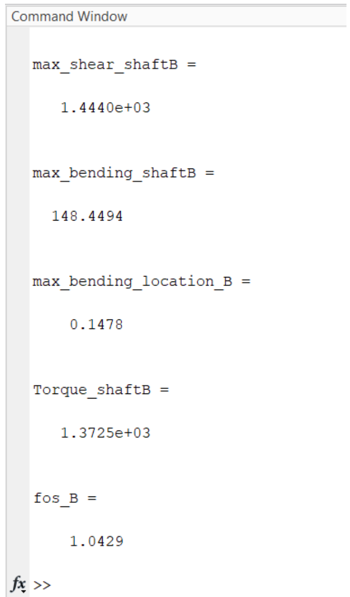


Figure 122 : Rear Gearbox MATLAB Bending Moment Diagram



```
Command Window

max_shear_shaftB =

   1.4440e+03


max_bending_shaftB =

  148.4494


max_bending_location_B =

    0.1478


Torque_shaftB =

   1.3725e+03


fos_B =

    1.0429

fx >>
```

Figure 123 : Rear Gearbox MATLAB Script Outputs

6.3.3.3 Jarett Berger

Conducting additional engineering calculations for the front gearbox, an FEA was performed on the CV cup (Figure 124) using the built-in feature in SolidWorks. As the CV Cup rotates, internal forces can be caused by the plunging effect, which can degrade the material inside the CV Cup. A force of 400lbs was applied to each inner groove where the ball bearings from the inner race would sit.

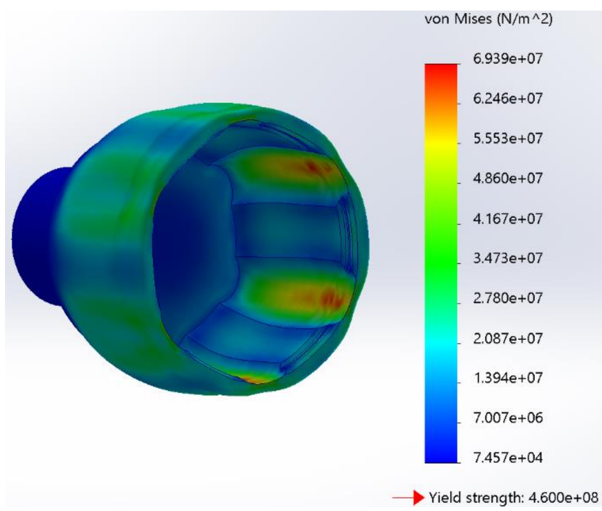


Figure 124: CV Cup FEA

Based on the FEA analysis, a factor of safety of 6 was achieved. This factor of safety is adequate for this design due to the balance between weight and strength optimization.

6.3.4 Frame

6.3.4.1 Gabriel Rabanal

Additional engineering calculations were performed on the tabs of mounting locations for suspension to the frame. The mounts for the shocks were found to be higher stress and were evaluated using FEA modeling to ensure adequate design. The Rear left shock tab was evaluated for displacement and stress analysis when a force of 550 lbs is applied directly through the hardware to the tab.

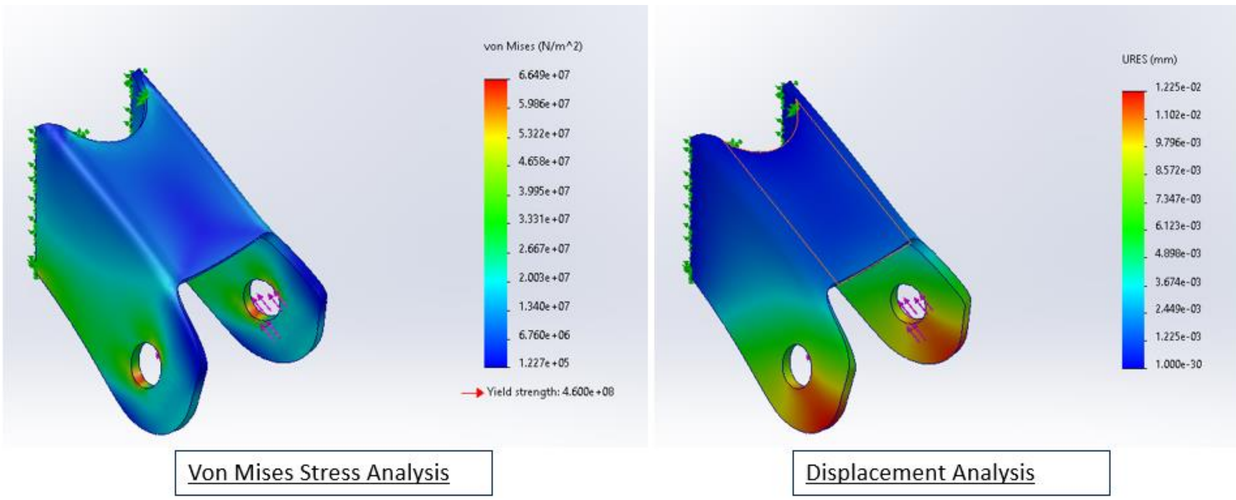


Figure 125: Shock Tab FEA

Shown in Figure 125, the stress and displacement analyses both proved successful, with a calculated factor of safety of roughly 18 for the material. Because this number is so high, any failure at this location would likely be a result of improper assembly and welding to the frame.

6.3.4.2 Antonio Sagaral

An analysis was done on the member that the front shock attaches to. The analysis was done to determine if the member would deform under a serious impact aimed at a front right or left side wheel. The following equations are the governing equations for the analysis.

$$\delta_{max} = \frac{Fb(3L^2 - 4b^2)}{48EI}$$

Equation 36: Deflection Equation

$$I = \frac{\pi}{64}(D^4 - d^4)$$

Equation 37: Moment of Inertia

Where,

F = 525lb

a = 11 in

b = 3 in

E = 29,000 ksi

I = 0.0426

L = 13.6 in

The maximum deflection was found to be 0.0236 in. This is well within the acceptable range and a dramatic over exaggeration of the force that would be scene from such an impact. The shock would absorb much of the energy greatly reducing the force.

6.3.4.3 Cooper Williams

Expanding on the analysis of the Side Impact Member (SIM), additional deflection calculations were preformed to validify the structural integrity of each member of the SIM assembly, numbered 1 through 3. By treating each of the three members as a cantilever with intermediate loading, the deflection of the 4130 steel tubing can be calculated. Below is a list of governing equations, relevant values, and assumptions:

$$y_{AB} = \frac{Fbx}{6EIl}(x^2 + b^2 - l^2)$$

Equation 38: Cantilever Deflection with Intermediate Loading

$$I_y = I_x = \frac{\pi}{64}(D^4 - d^4)$$

Equation 39: Moment of Inertia of a Tube

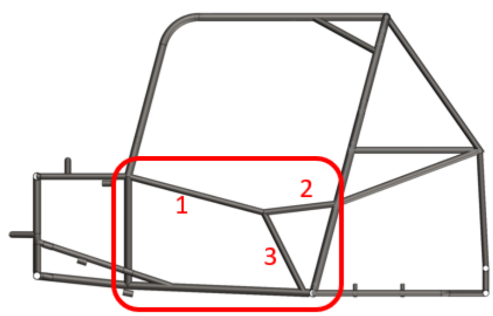


Figure 126: SIM Labeling

Assumptions:
Simply Supported Beam with an Intermediate Load
a = $\frac{1}{2}l$, b = $\frac{1}{2}l$
$x = 0.5l$
F = 300 lbf

Figure 127: Deflection Assumptions

Relevant Values:
D = 1.00 in
d = 0.93 in
L_1 = 22.977 in
L_2 = 12.049 in
L_3 = 14.501 in
E = 29000 kpsi

Figure 128: Relevant Deflection Values

Given these equations, assumptions, and found values, the deflection of each of the pieces of tubing can be calculated and compared.

Maximum Deflection per Member	
δ_1	0.0212 in
δ_2	0.0030 in
δ_3	0.0053 in

Figure 129: SIM Deflection Values

From these calculated values, found using Equation 38 and Equation 39, it is apparent that the shortest sections of tube deflect the least. This was expected and validates the calculations. Additionally, these values are a fraction of an inch and provide some mathematical evidence that the driver would be completely safe using this material at the given lengths for Side Impact Members.

6.4 Future Testing Potential

6.4.1 Front

Testing of each subsystem will be completed in a variety of ways. Due to the expensive nature of the knuckle and other components within the front-end assembly, the team will use both online FEA Modeling as well as physical testing. The team has broken each subsystem into three groups labeled control arm construction, steering system, and knuckle construction. Each sub team will have its own test that is specific to the construction and application of the system. The control arms have been tested thoroughly by Ansys FEA Software. With adjustments and further calculations, additional FEA testing will be completed. The construction of the control arms will also be tested. To pass tech inspection, welds must be certified thus proving the construction of the control arm. The steering system will be tested once the car is constructed with digital angle gauges and the turning radius of the vehicle will be tested. Design testing was already completed thoroughly in Lotus Shark Software. The tensile strength of the tire arms may be tested to ensure that the threaded insert mate adequately with the carbon tubing used. Due to the expensive nature of the knuckle, the majority of testing will be conducted virtually.

6.4.2 Rear

The three areas that the rear end team hope to test in the future include trailing links, CV end spacing, and camber links. The trailing links will need to be placed in a jig while a hydraulic press applies a vertical force on the knuckle. The welded and finished trailing link will be analyzed with a strain gauge to ensure there is an acceptable amount of deflection and zero failure before install on the vehicle. The CV spacing has been tested using the first prototype and will be double checked during the assembly of the final system to the car. The upper carbon fiber camber link will need to be placed in a jig and experience a baseline tension/compression force of 1,547 lbf. Additional crash testing with the camber links will be performed when the car is fully operational before the competition to have adequate time to make changes if needed.

6.4.3 Drive

Future testing for the drivetrain team will focus on evaluating the performance of materials needed for the drivetrain components. This will ensure that the components will last for the expected life. Selecting the appropriate material will be planned accordingly based on strength and impact resistance needed to enhance the overall vehicle performance; however, keeping in mind the cost to purchase the necessary materials. The team will also continue using FEA analysis on all drivetrain components to make sure that all parts are fully optimized. Additionally, drivetrain testing may extend beyond the mechanical components to include the human ergonomics of the dog clutch shifter. Future design and testing may be incorporated, ensuring that the interaction between the driver and the drivetrain is intuitive.

6.4.4 Frame

The frame team is on an accelerated schedule relative to some of the other sub teams and Capstone teams. Due to this, testing newer prototypes or providing proof of concept is not a realistic goal. If the frame sub team designs and manufactures the frame to meet SAE BAJA regulations, the frame will need no testing to meet structural requirements. Thus far, the team has tested the frame design against SAE BAJA regulations, and it has been adjusted appropriately, resulting in a final frame. However, after the frame and vehicle are manufactured, there will be impact testing to assess the strength of multiple different components, including the frame. This will test the integrity of the frame. Another test for the frame is the weld tests that Professor Willy will examine. The quality of these test welds will demonstrate the ability to assemble the frame in a competent and competitive manner.

7 CONCLUSIONS

The objective of this project is to develop and build a competitive and functional single-seat Baja vehicle. This capstone must fulfill the requirements of NAU's capstone course and meet all standards outlined by the Society of Automotive Engineers' Baja competition guidelines. To meet these requirements, the team of thirteen has been split into four sub teams to optimize each subcategory. Thought the requirements for frame are strict and limit variables such as tube-length and geometry, many sub teams had much more freedom within their design. This required the analysis of the competition itself to see what performance requirements would have to be met and what would have to be done within the design to meet these benchmarks. This opened communication with previous NAU Baja teams and comparison with past vehicles. In addition, the frame team met with its competition driver to ensure that all cockpit sizing was adequate and comfortable. While meeting the requirements of SAE Baja, the team will also strive to represent its sponsors during competition with the best vehicle possible. The team is excited to have Gore, Nova Kinetics, IMS, TMS Titanium, and Monster Energy onboard this year and will continue to fundraise to meet the twenty-thousand-dollar fundraising goal. The separation of sub-systems does generate its own difficulties as the integration of these individual systems is vital in ensuring that the entire car fits together. To ensure integration, the team has communicated and met extensively outside of regular capstone hours. With already over a thousand hours tracked since the start of the semester, the team is designing a SAE Baja competition car with the attributes shown in the figure below. Prototyping has allowed insight into the construction of the vehicle as well as integration between subsystems. Construction of the vehicle has begun and the necessary tools to do so such as the jig is being produced.

Figure 130: Baja Full Assembly

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9 APPENDICES

9.1 *Appendix A: Project Management*

Table 42: Appendix Gantt 1 – ME476C

PROJECT: SAE Baja 24

NAU A.Y. 2023-2024

Project Manager: Abe Plis

Project start date: 9/12/2023

Scrolling increment: 0

Legend: Front, Rear, Drivetrain, Frame, All Team

Milestone description	Responsible Sub-Team	Assigned To	Progress	Start	Days
Frame Completion in CAD	Milestone			10/31/2023	1
Presentation 1	All Team	N/A	100%	9/12/2023	7
Major Sub-System Decisions	All Team	Team Leads	100%	9/15/2023	7
Wire Frame	Frame	Cooper Lead	80%	9/12/2023	15
Define Front Suspension Points & Begin CAD	Front	Bryce Lead	100%	9/20/2023	7
Define Rear Suspension Points & Begin CAD	Rear	Seth	75%	9/20/2023	7
Define Drivetrain Points & Begin CAD	Drivetrain	Henry Lead	100%	9/20/2023	7
Measure Hailey & Design Rollcage	Frame	Cooper Lead	100%	9/18/2023	10
Concept Generation & Selection	All Team	Team Leads	100%	9/26/2023	11
Presentation 2	All Team	N/A	100%	10/3/2023	7

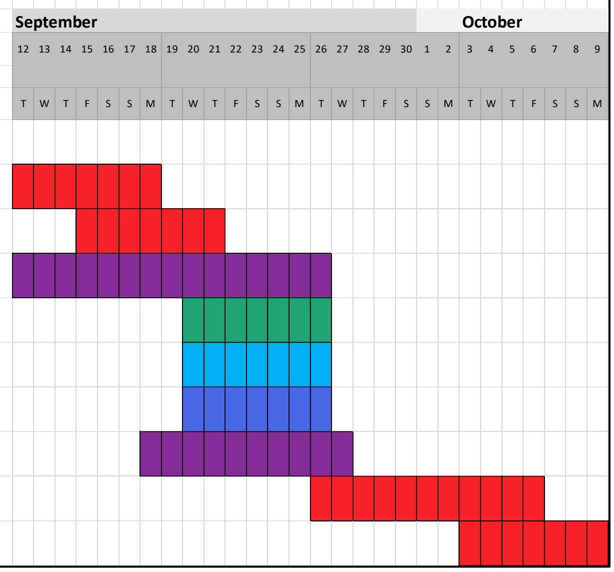

Table 43: Appendix Gantt 2 – ME476C

PROJECT: SAE Baja 24

NAU A.Y. 2023-2024

Project Manager: Abe Plis

Project start date: 9/12/2023

Scrolling increment: 28

Legend: Front | Rear | Drivetrain | Frame | All Team

Milestone description	Responsible Sub-Team	Assigned To	Progress	Start	Days
Frame Completion in CAD	Milestone			10/31/2023	1
Presentation 1	All Team	N/A	100%	9/12/2023	7
Major Sub-System Decisions	All Team	Team Leads	100%	9/15/2023	7
Wire Frame	Frame	Cooper Lead	80%	9/12/2023	15
Define Front Suspension Points & Begin CAD	Front	Bryce Lead	100%	9/20/2023	7
Define Rear Suspension Points & Begin CAD	Rear	Seth	75%	9/20/2023	7
Define Drivetrain Points & Begin CAD	Drivetrain	Henry Lead	100%	9/20/2023	7
Measure Hailey & Design Rollcage	Frame	Cooper Lead	100%	9/18/2023	10
Concept Generation & Selection	All Team	Team Leads	100%	9/26/2023	11
Presentation 2	All Team	N/A	100%	10/3/2023	7
Packaging Integration (Wheelbase, car length, etc.)	All Team	Cooper & Henry Lead	90%	10/3/2023	15
Report 1 & Webiste 1	All Team	Seth	30%	10/20/2023	8
Finalize Frame (footbox, lower rear triangle, rollcage)	Frame	Cooper Lead	60%	10/11/2023	22
Subsystem Designs	Milestone			11/23/2023	1
UCA, LCA, Knuckle, Rack, Brakes, Hubs	Front	Bryce Lead	50%	10/11/2023	15
Trailing Arm, Camber Links	Rear	Seth Lead	30%	10/11/2023	15
eCVT, Front/Rear Gear Box, Belt Power Transfer, Brakes	Drivetrain	Henry Lead	50%	10/11/2023	17
Rough Designs Fully in CAD	All Team	All Team	0%	10/11/2023	22
Analysis Memo	All Team	All Team	0%	10/28/2023	7

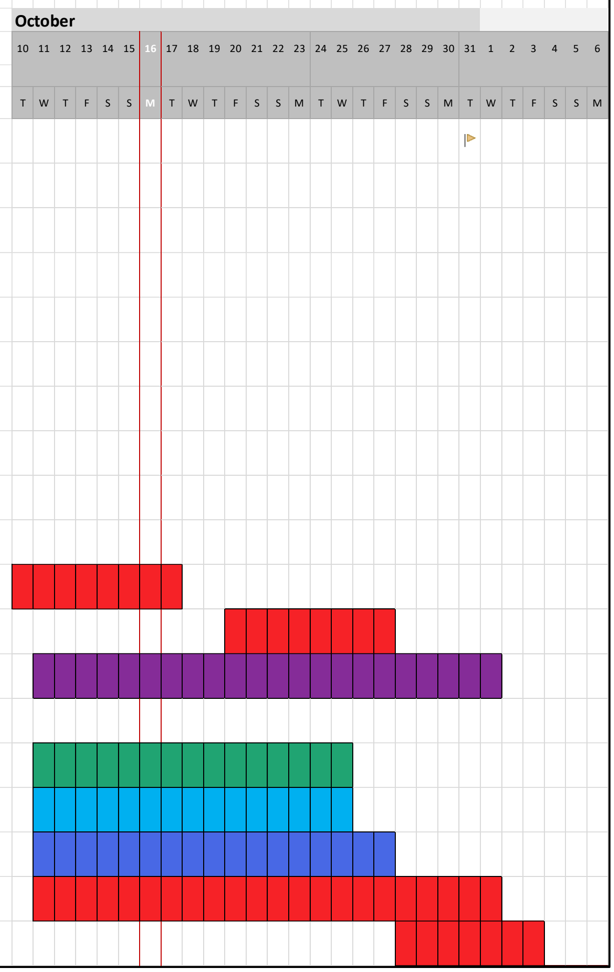

Table 44: Appendix Gantt 3 – ME476C

PROJECT: SAE Baja 24

NAU A.Y. 2023-2024
Project Manager: Abe Plis
Project start date: 9/12/2023
Scrolling increment: 55

Legend: Front, Rear, Drivetrain, Frame, All Team

November / December

Milestone description	Responsible Sub-Team	Assigned To	Progress	Start	Days
Pres 3 & 1st Demo	All Team	All Team	0%	10/31/2023	8
Refined Designs in CAD	All Team	All Team	0%	11/1/2023	23
Report 2	All Team	All Team	0%	11/13/2023	11
Full CAD Assembly Completed & BOM	Milestone	All Team	0%	11/24/2023	8
Frame Construction (Tac then Full Weld)	Frame	Cooper Leads	0%	11/1/2023	34
2nd Demo	All Team	All Team	0%	11/27/2023	9
Website 2	All Team	Seth	0%	12/2/2023	9

Table 45: Appendix Gantt 4 - ME486C

PROJECT: SAE Baja 24

NAU A.Y. 2023-2024
Project Manager: Abe Plis
Project start date: 9/12/2023
Scrolling increment: 126

Legend: Front, Rear, Drivetrain, Frame, All Team

January / February

Milestone description	Responsible Sub-Team	Assigned To	Progress	Start	Days
Hardware - 33%	Milestone	All Team	0%	2/13/2024	1
Frame Welded with Tabs	Frame	Cooper Lead	0%	1/16/2024	7
UCA,LCA, Steering & Begin Knuckle	Front	Bryce Lead	0%	1/16/2024	21
Start CVT Housing, Gears, and Pully	Drivetrain	Henry Lead	0%	1/16/2024	28
Camber Links & Hub	Rear	Seth Lead	0%	1/16/2024	21

Table 46: Appendix Gantt 5 - ME486C

PROJECT: SAE Baja 24

NAU A.Y. 2023-2024
Project Manager: Abe Plis
Project start date: 9/12/2023
Scrolling increment: 155

Legend: Front, Rear, Drivetrain, Frame, All Team

February / March

Milestone description	Responsible Sub-Team	Assigned To	Progress	Start	Days
Hardware - 67%	Milestone	All Team	0%	3/16/2024	1
Begin Ergo & Safety	Frame	Cooper Lead	0%	2/14/2024	28
Hub, Brake Rotor, Calipers, Plumbing, CV Axle, Steering	Front	Bryce Lead	0%	2/14/2024	31
Finish eCVT and Mount, Assemble Rear Gears and Dog Clutch/Front Gearbox Assembly	Drivetrain	Henry Lead	0%	2/14/2024	31
Brake Routing and Communicate with Drive	Rear	Seth Lead	0%	2/14/2024	21

Table 47: Appendix Gantt 6 - ME486C

PROJECT: SAE Baja 24

NAU A.Y. 2023-2024
Project Manager: Abe Plis

Project start date: 9/12/2023
Scrolling increment: 171

Legend: Front | Rear | Drivetrain | Frame | All Team

March

Milestone description	Responsible Sub-Team	Assigned To	Progress	Start	Days
Hardware - 100%	All Team	All Team	**0%**	4/3/2024	1
Drivable Car	All Team	All Team	**0%**	3/16/2024	7
Complete Ergo & Safety	Frame	Cooper Lead	**0%**	3/16/2024	14
Dial Suspension & Steering	Front	Bryce Lead	**0%**	3/16/2024	18
Finish and Verify Power Transfer, Arduino Code for eCVT	Drivetrain	Henry Lead	**0%**	3/16/2023	18
Dial Suspension	Rear	Seth lead	**0%**	3/16/2024	14
Poster Draft	All Team	All Team	**0%**	3/1/2024	19

Table 48: Appendix Gantt 7 - ME486C

PROJECT: SAE Baja 24

NAU A.Y. 2023-2024
Project Manager: Abe Plis

Project start date: 9/12/2023
Scrolling increment: 190

Legend: Front | Rear | Drivetrain | Frame | All Team

March | April

Milestone description	Responsible Sub-Team	Assigned To	Progress	Start	Days
Testing	All Team	All Team	**0%**	4/17/2023	3
Initial Testing	All Team	All Team	**0%**	4/3/2024	7
Final Testing	All Team	All Team	**0%**	4/10/2024	7
Final Poster	All Team	All Team	**0%**	3/20/2024	22
Competition & Symposium					
Gorman Comp	All Team	All Team	**0%**	4/25/2024	4
UGRADS	All Team	All Team	**0%**	4/26/2024	1