-Initial Design Report-

SAE Baja 2023-2024

NORTHERN ARIZONA UNIVERSITY

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Project Sponsors: W.L. Gore, Monster Energy, NovaKinetics Aerosystems, Industrial Metal Supply

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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 October 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. All sub-teams have 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. As of October 28th, the team has a rough CAD assembly of the whole vehicle completed and has begun work on the prototyping stage of the initial design to meet the November 6th 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 4 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

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, Nova Kinetics, 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.

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

#	Engineering Requirement	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	Customer Opinion Survey: 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 0

Table 6: Rear End QFD

System QFD

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

Legend	
A	CWRUM
B	RIOT Baja
C	TS BAJA

	Decrease weight	Increase strength	increase rearward axle path	Increase linkage radii	Increase ground clearance	Vehicle width	Decrease CV axle angle
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			

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
		Technical Requirements							Customer Opinion Survey				
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

	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
			Technical Requirements							Customer Opinion Survey				
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	Decrease width of body	Decrease Cost	increase aerodynamics
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

		Technical Requirements						Customer Opinion Survey				
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*
Rigid	3	1	6	3	3	1	9				ÀBC	
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).

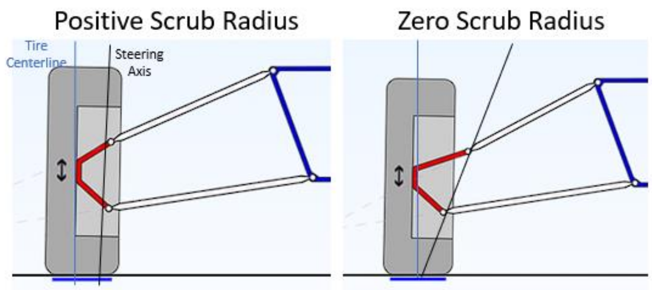


Figure 1: Scrub Radius Definition

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.

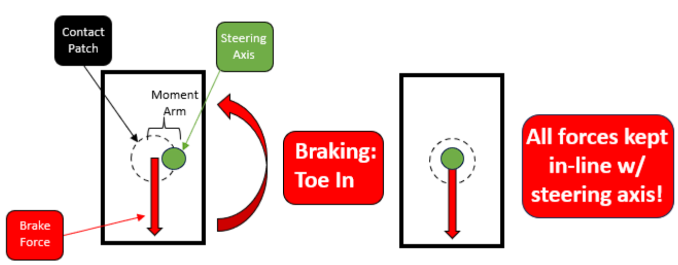


Figure 2: Scrub Radius Impact

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.

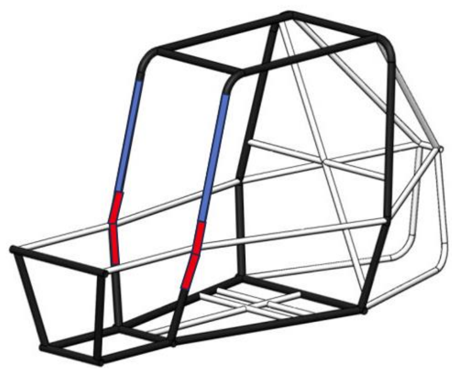


Figure 4: SAE Baja Chassis with Upper Front Brace in Blue and Lower Front Brace in Red

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.

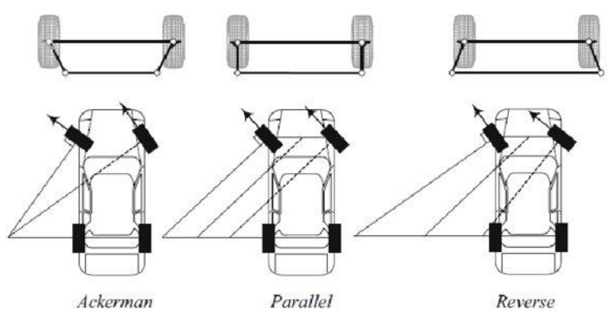


Figure 6 : Three Types of Steering Design

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 1st 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 3rd in suspension and 5th 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 2nd 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 3rd in maneuverability and 2nd overall. Cornell University utilized vertical frame mounts, as shown in the 3rd photo in Figure 9, which earned the team 2nd in maneuverability and 3rd 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.

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.

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.

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.

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

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.

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.

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.

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]

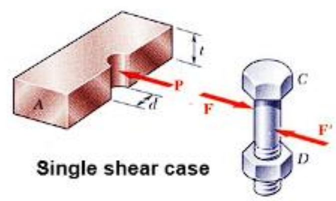


Figure 13: Hole Bearing Diagram

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].

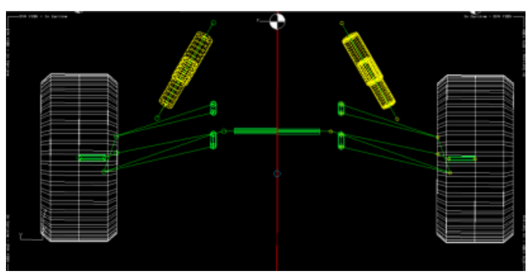


Figure 14: Lotus Shark Software Preview

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).

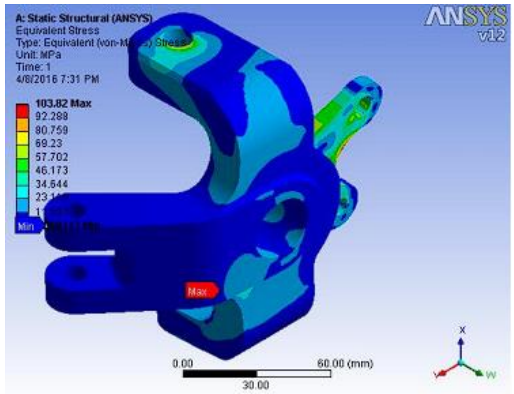


Figure 15: Steering Knuckle Example

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.

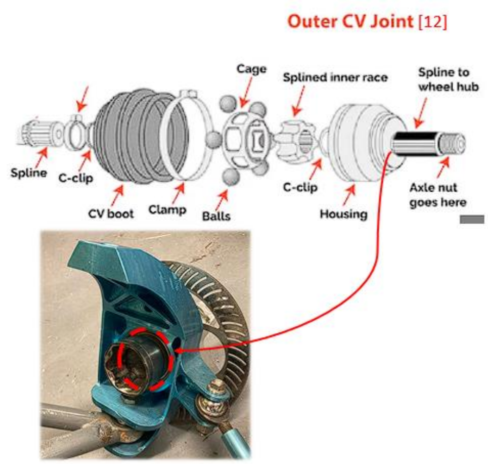


Figure 16: Steering Knuckle Modeling Diagram

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: Labeled Upper Front** A Control Armand will be the only areas with a force input.

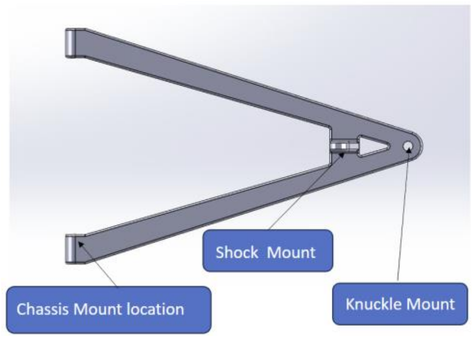


Figure 17: Labeled Upper Front A Control Arm

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.

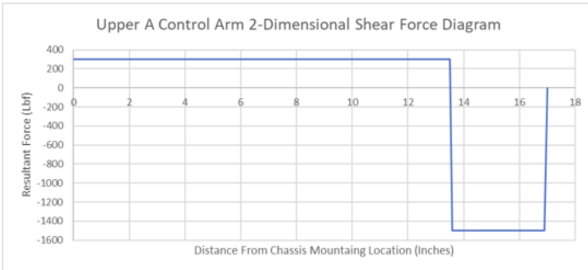
$$Shear\ Stress\ (Tao) = \sum \frac{Force}{Cross\ Sectional\ Area}$$

Equation 3: Instantaneous Shear Stress in a Member

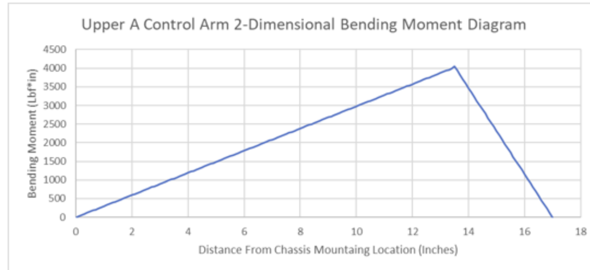
$$Bending\ Moment\ (M) = \int Shear\ Force\ Diagram\ (SFD)$$

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.



Upper A Control Arm 2-Dimensional Shear Force Diagram in the Vertical Direction



Upper A Control Arm 2-Dimensional Bending Moment Diagram in the Vertical Direction

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.

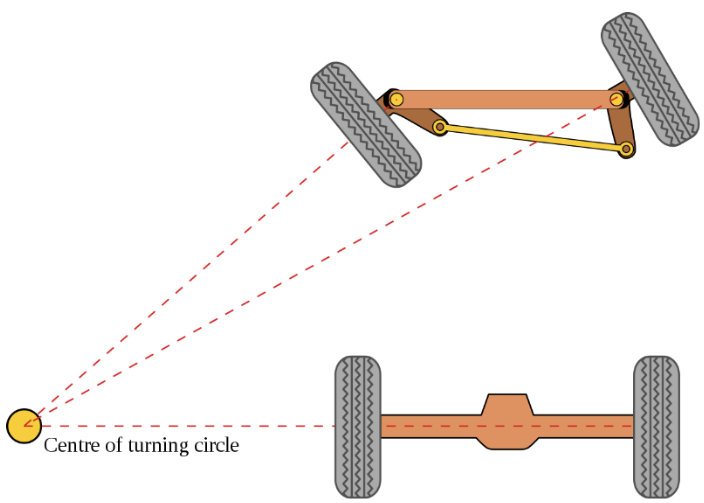


Figure 19 : Ackermann Steering

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

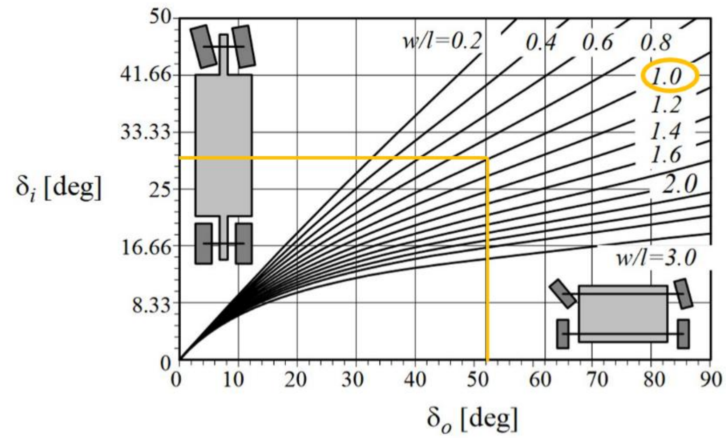


Figure 20 : Viable Steering Angles using Ackermann Steering

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

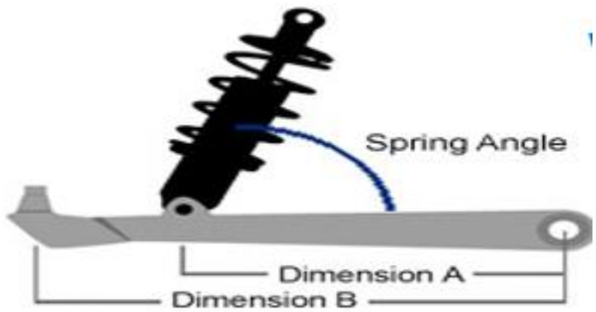
Understanding viable Ackermann angles as a function of slip angle and lateral force allows the team to construct a good steering car when developing steering geometry with the knuckle. In addition, insight to turning radius gives the team confidence that Ackermann steering is a good design and is viable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

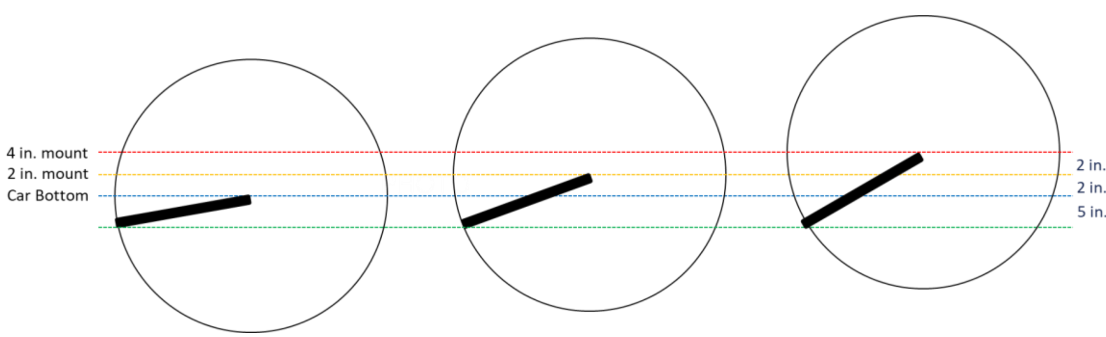


Figure 23: Trailing Link Configurations

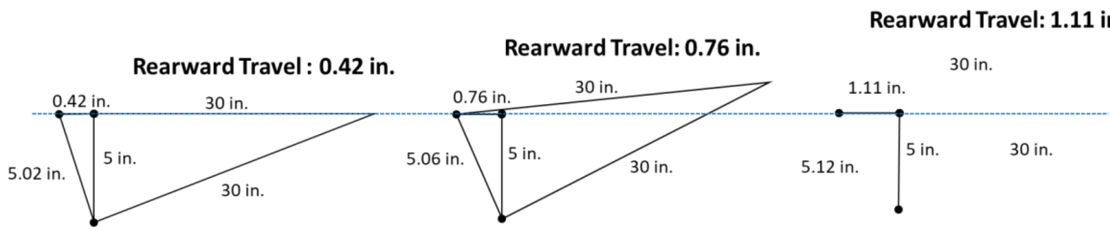


Figure 24: Rearward Axle Path Measurements

3.3.2.3 Joey Barta – Camber Gain

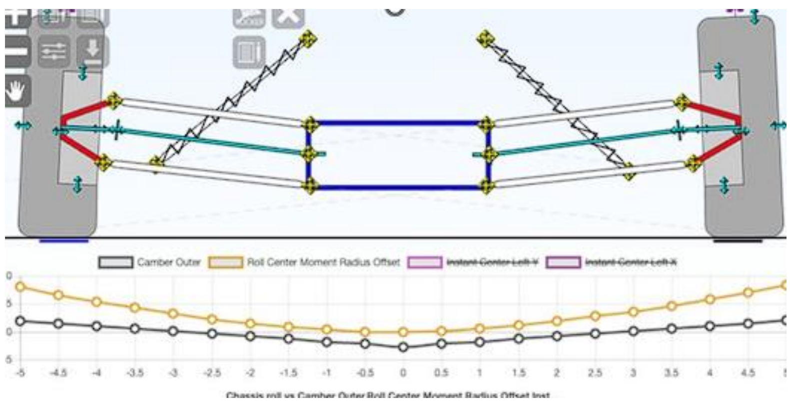


Figure 27: Camber vs. Roll Center

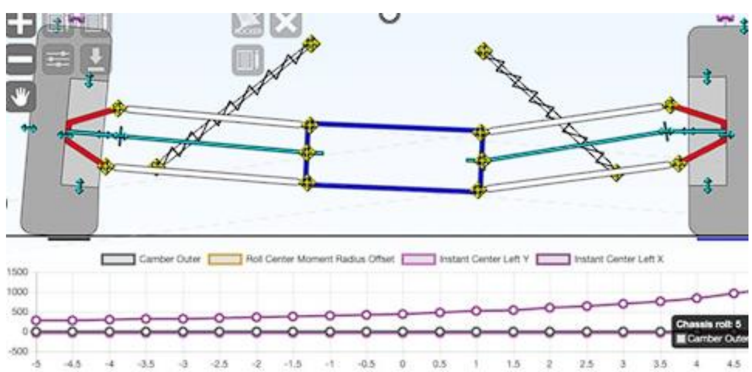


Figure 28: Camber vs. IC

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.

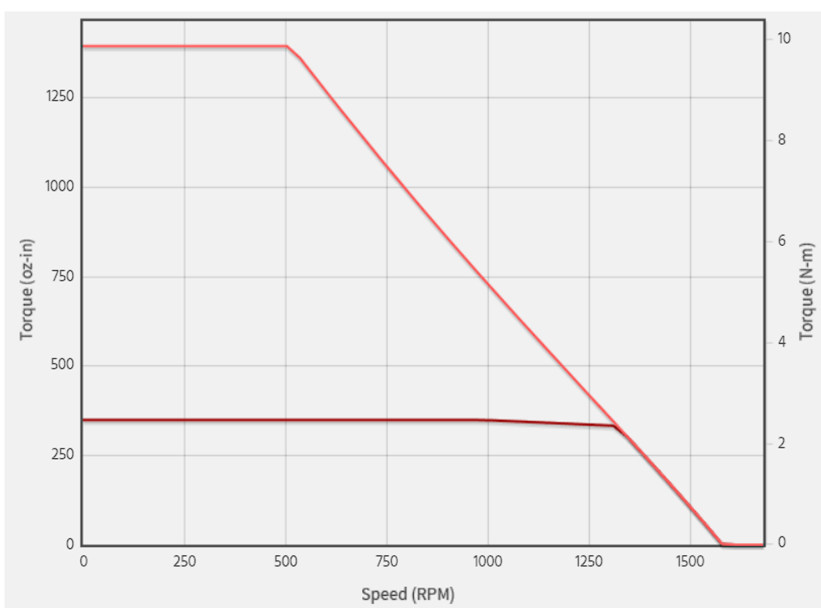


Figure 30: M-3432F-LS-08D Torque Curve

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.

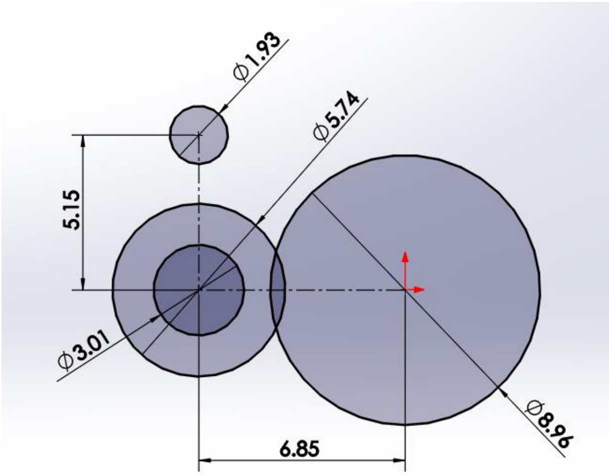


Figure 33: Chain Drive Geometry

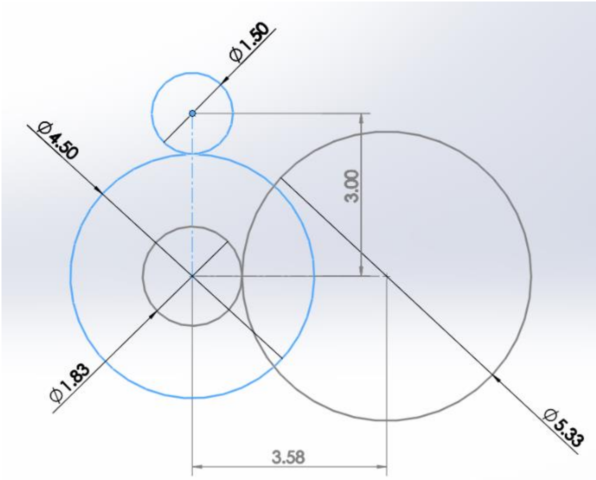


Figure 34: Gearbox Geometry

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

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

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.

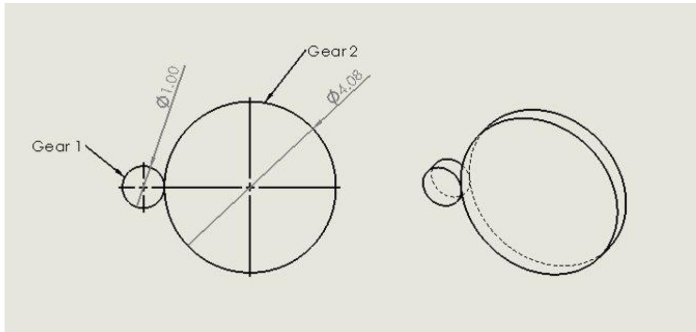


Figure 37: Preliminary Gear Design

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:

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

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

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

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

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.

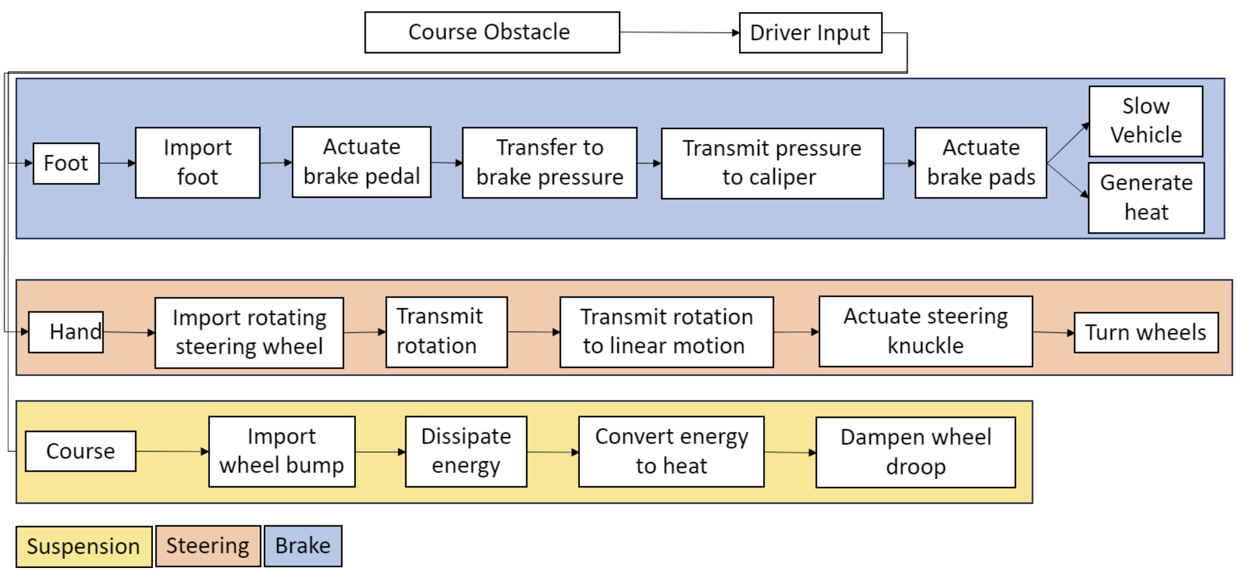


Figure 39: Front End Functional Model

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.

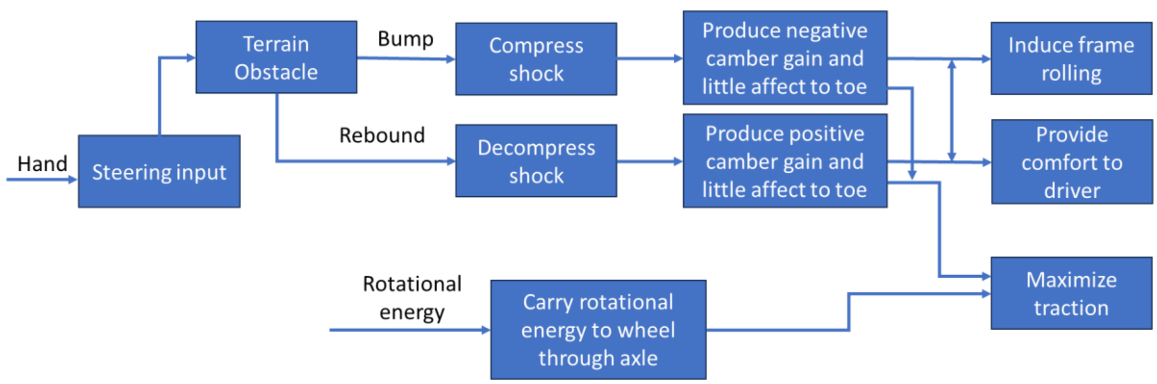


Figure 41: Rear Functional Model

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

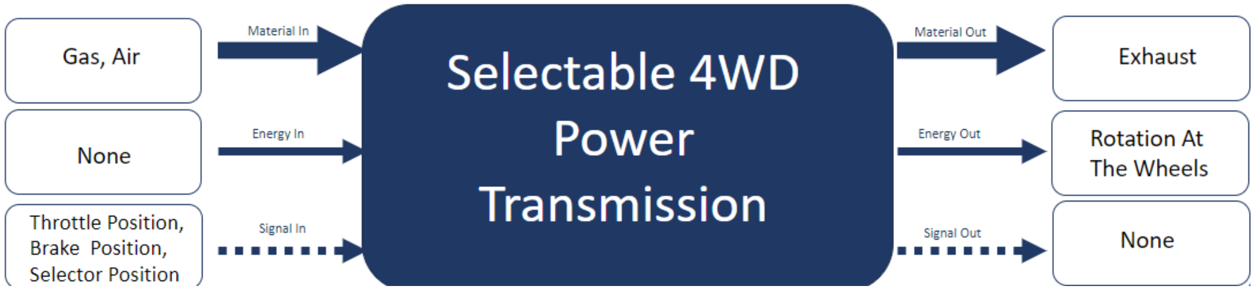


Figure 42: Drivetrain Black Box Model

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.

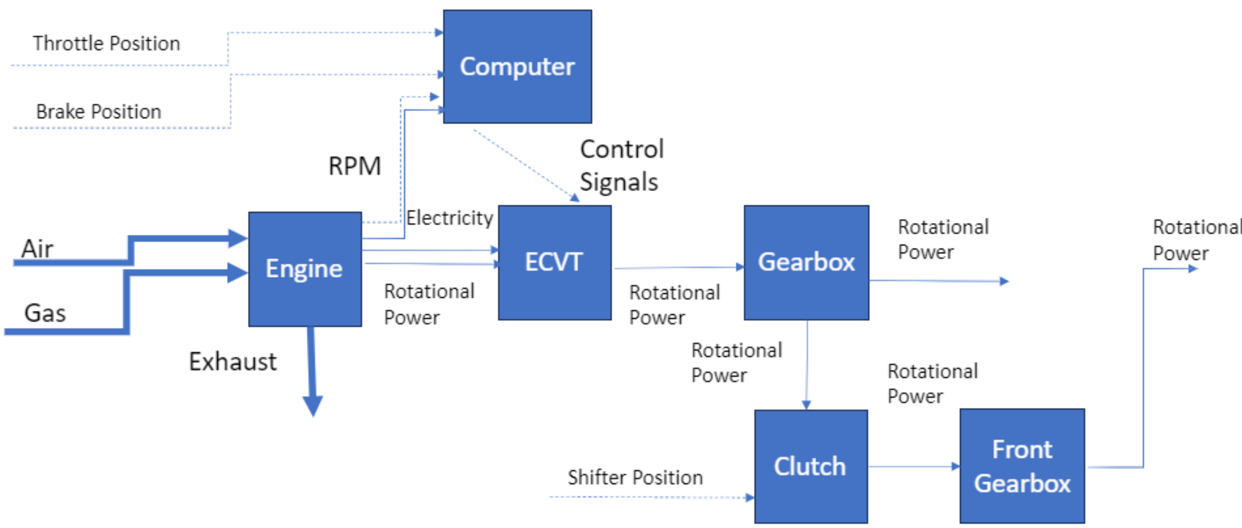


Figure 43: Drivetrain Functional Model

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.

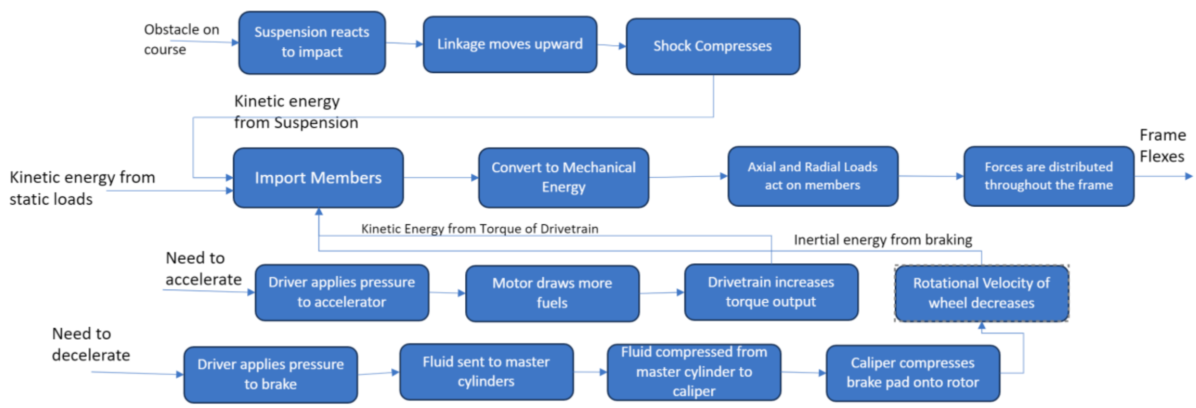


Figure 45: Frame Functional Model

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

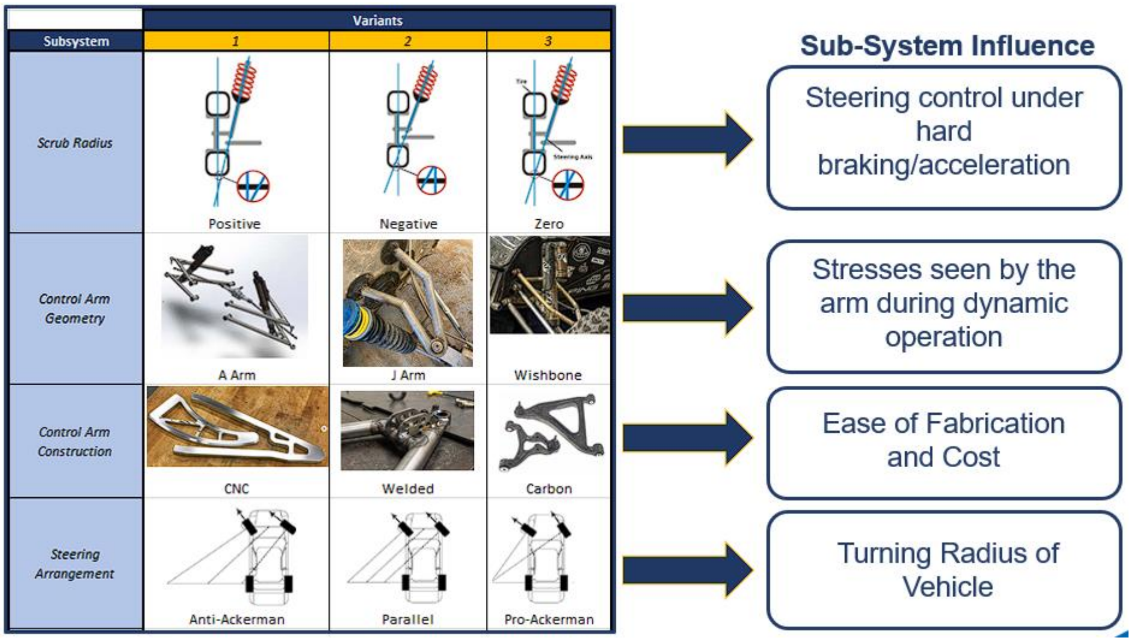


Figure 46: Front End Concept Generation 1

- *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.

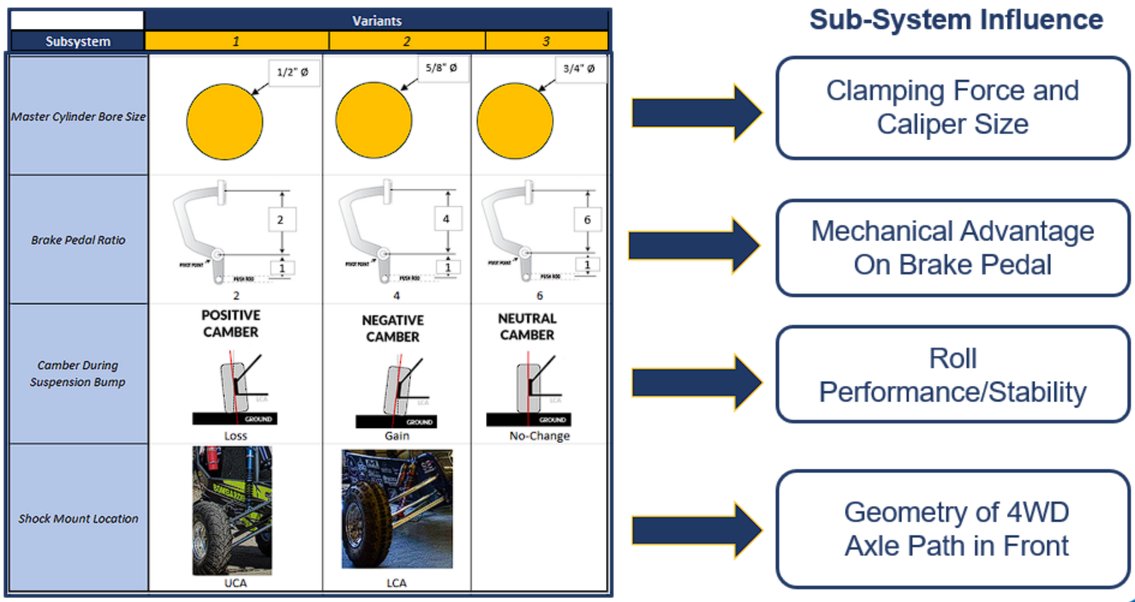


Figure 47: Front End Concept Generation 2

- *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

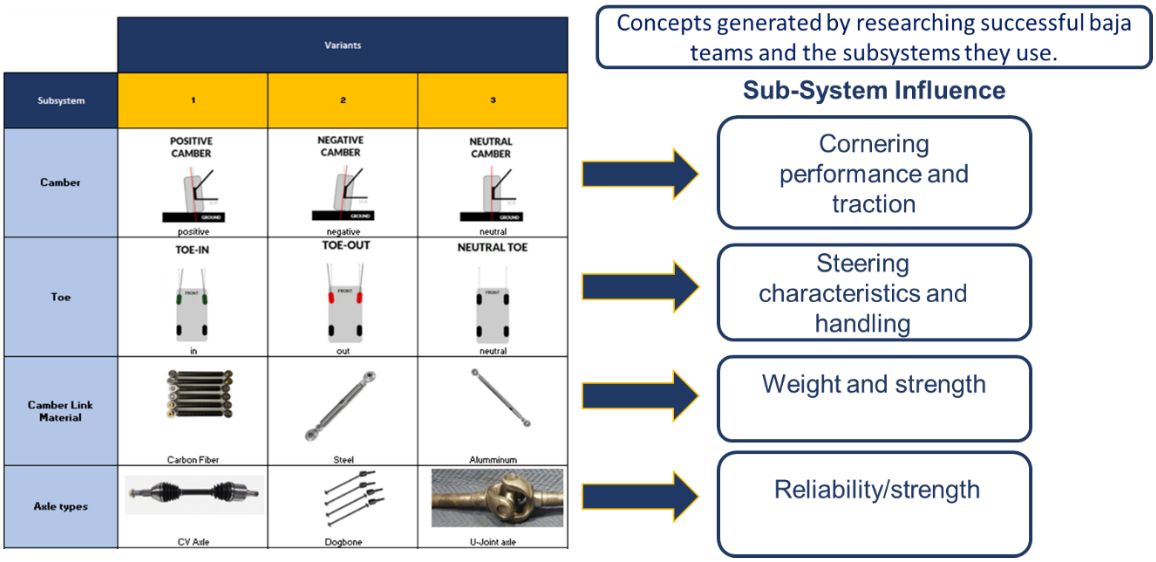


Figure 48: Rear Concept Generation Part 1

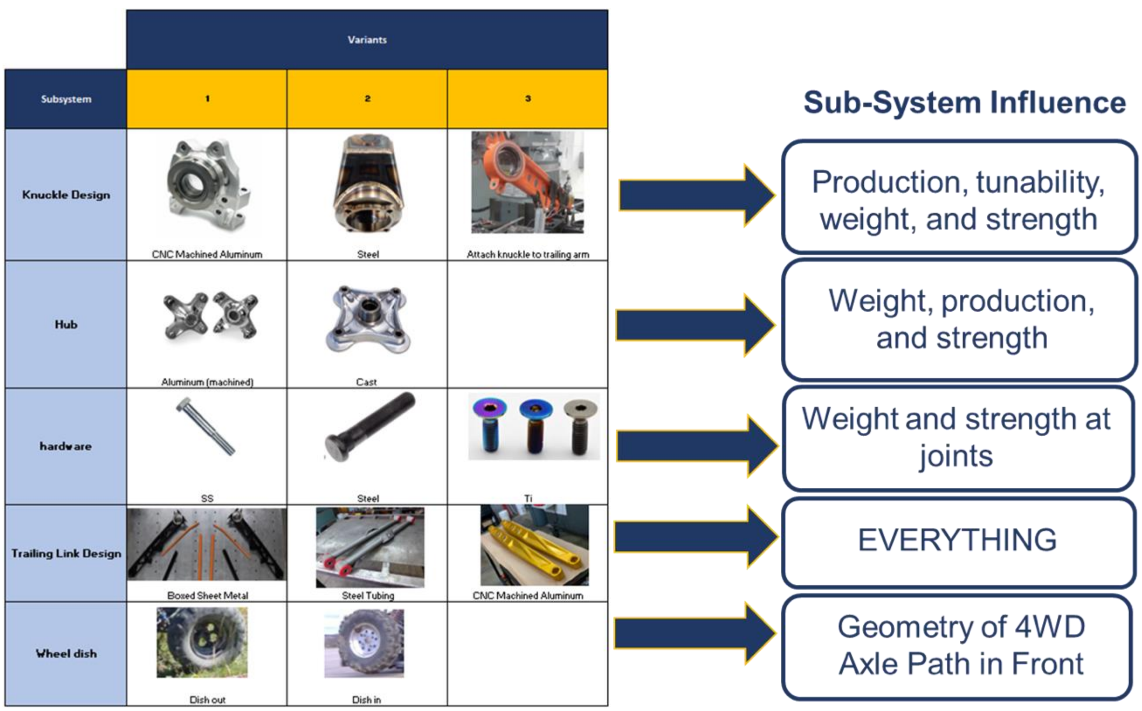


Figure 49: Rear Concept Generation Part 2

The key takeaways from the concept generation figures above are that strength, lightness, reliability, and performance are 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

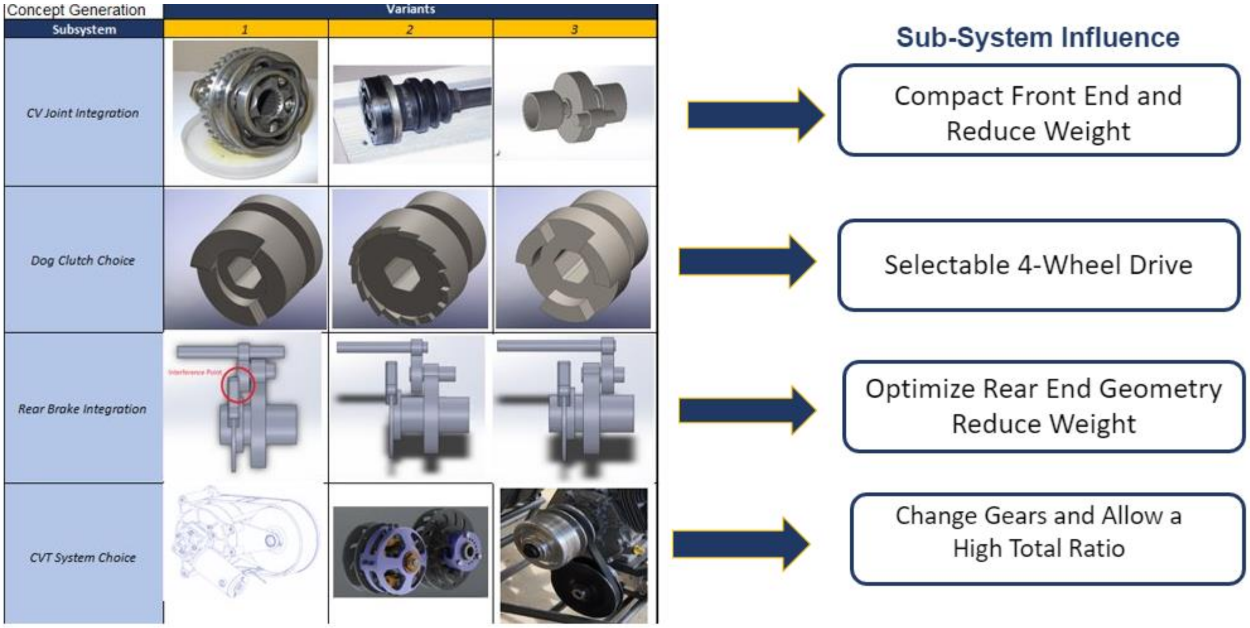


Figure 50: Drivetrain Concept Gen

Figure 51: Drivetrain Concept Generation

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

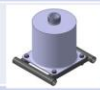
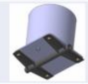
Subsystem	Variant 1	Variant 2	
Frame Type			Frame Integrity and Sub-System Integration
Fuel Tank Mounting			Time, money, effort in manufacturing and integrating
Seat Design			Seat mounting, driver safety
Side Impact Member			Overall frame length and structural integrity
Tube Material	AISI 4130	AISI 1018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

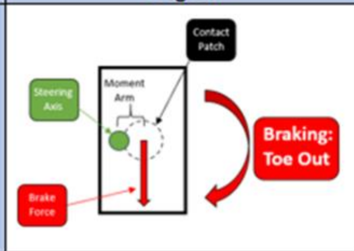
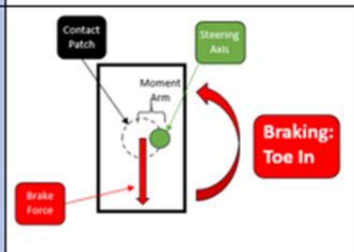
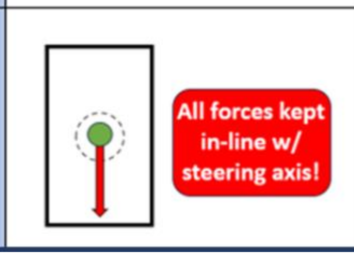
Scrub Radius - Wheel Torque Under Braking		
Variant	**Diagram**	**Result**
1 - Positive		64 lbf*ft
2 - Negative		64 lbf*ft
3 - Zero		0 lbf*ft

Figure 53: Scrub Radius Justification

The

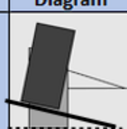
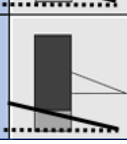
Camber During Bump		
Variant	**Diagram**	**Result**
1 - Gain		Better Roll Performance
2 - Loss		Worse Roll Performance
3 - No Change		Subpar Roll Performance

Figure 54: Camber Performance During Bump

The

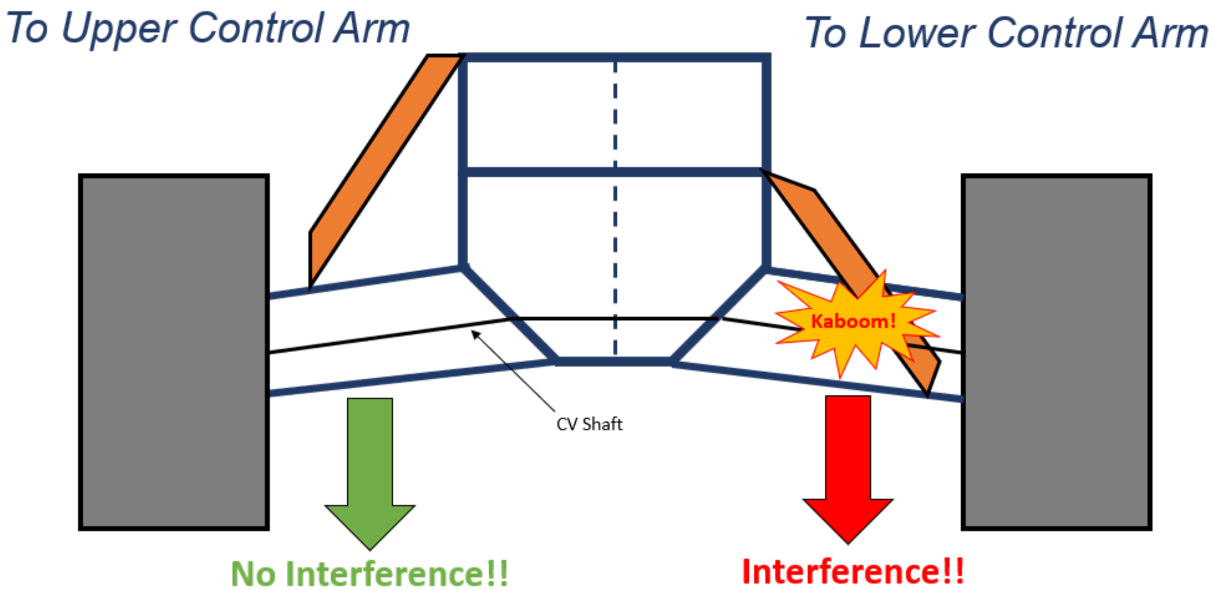
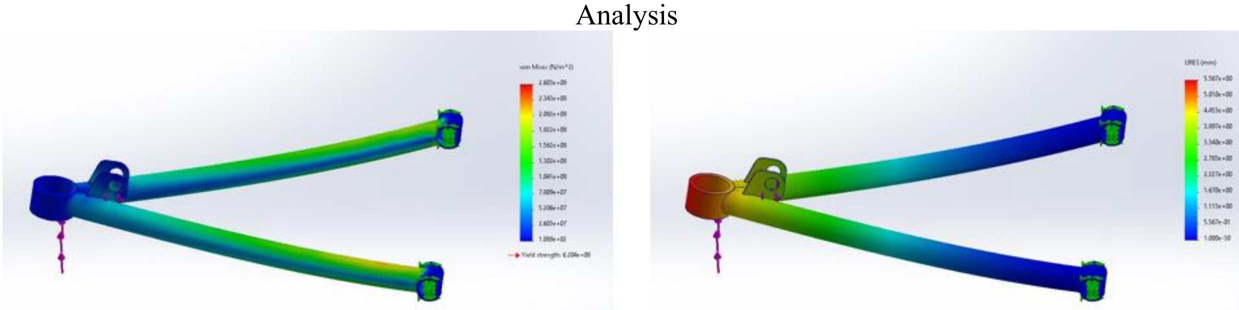


Figure 55: Shock Mount Location

As

4.3.1.2 Bryce Fennell

Analysis



Developed stress on front upper control arm

Displacement on front upper control arm

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

Sections

The

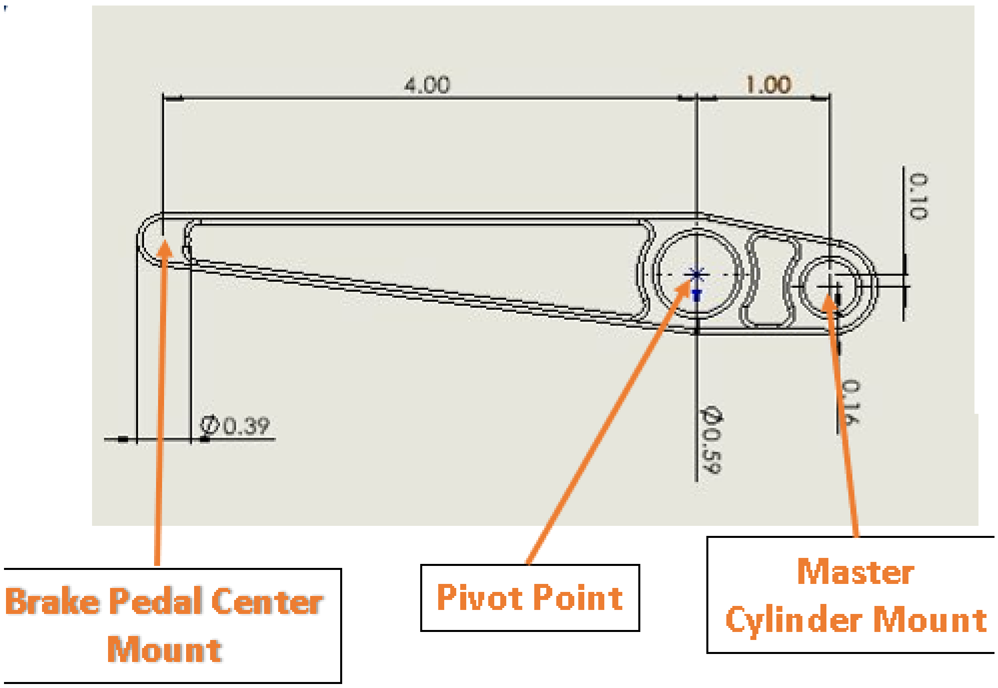


Figure 57: SolidWorks Drawing of Anticipated Pedal Dimensions

With

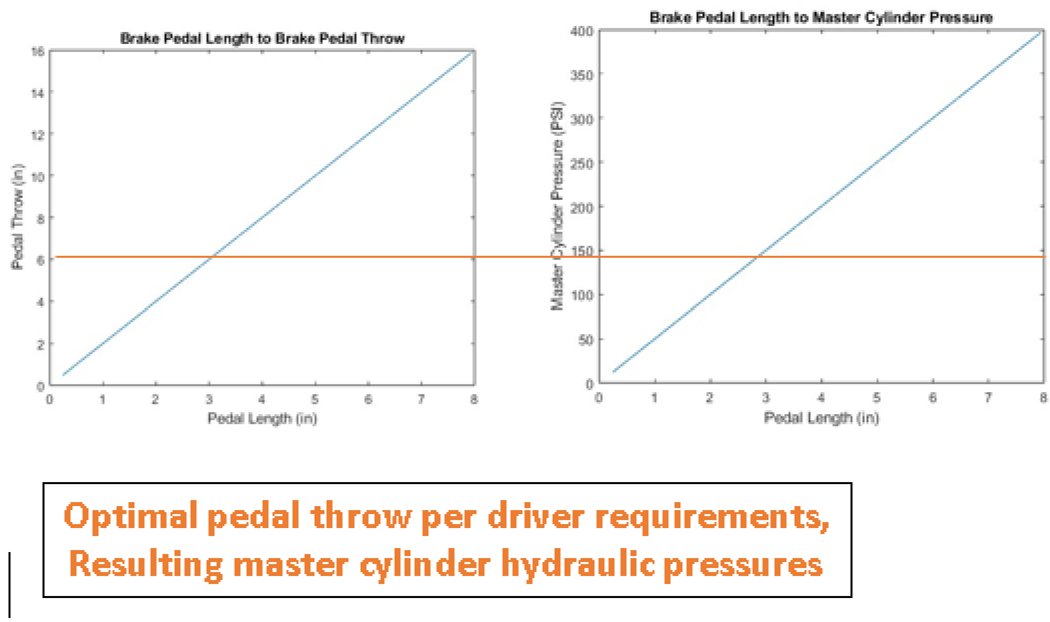


Figure 58: Left: Brake Pedal Length vs Brake Pedal Throw. Right: Brake Pedal Length vs System Pressure

The

The

To

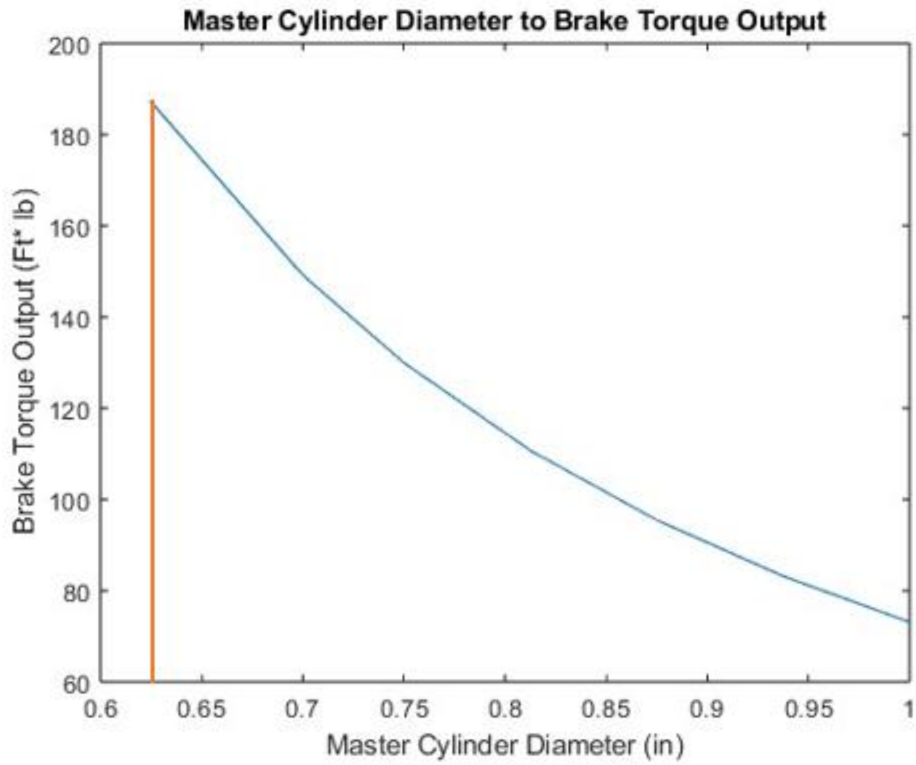


Figure 59: Master Cylinder Bore Size Versus Brake Torque Output at The Wheel

Analyzing

4.3.1.3 Evan Kamp

With

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

Due

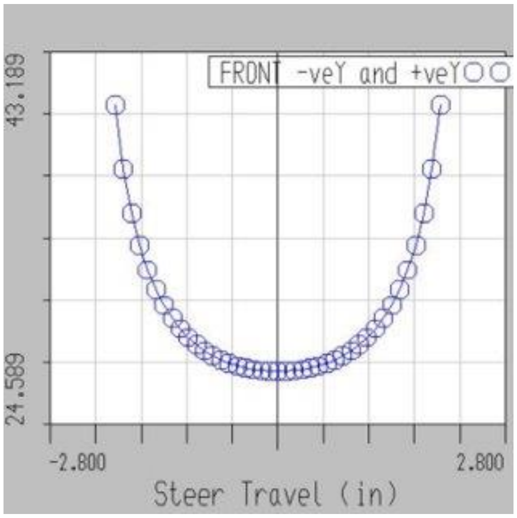


Figure 60: Percent Ackerman

Using

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

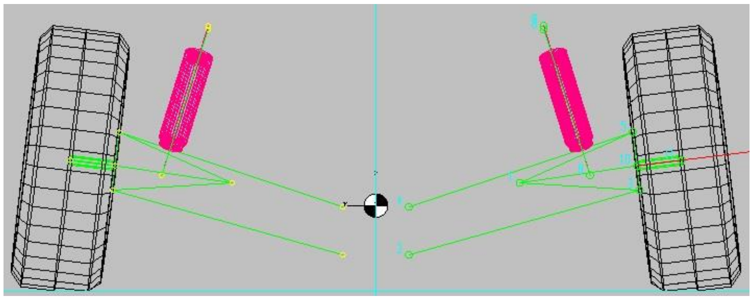


Figure 61: This is the suspension system under full compression.

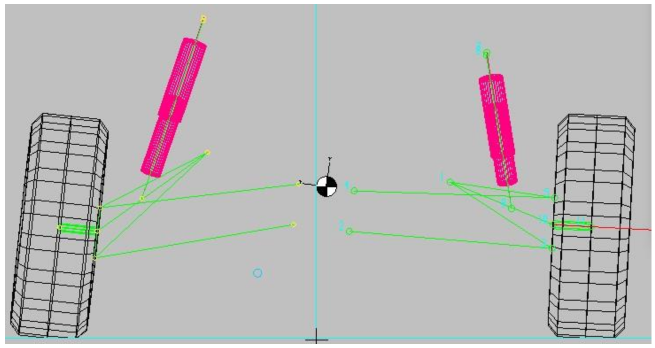


Figure 62: This is the system under full roll or cornering.

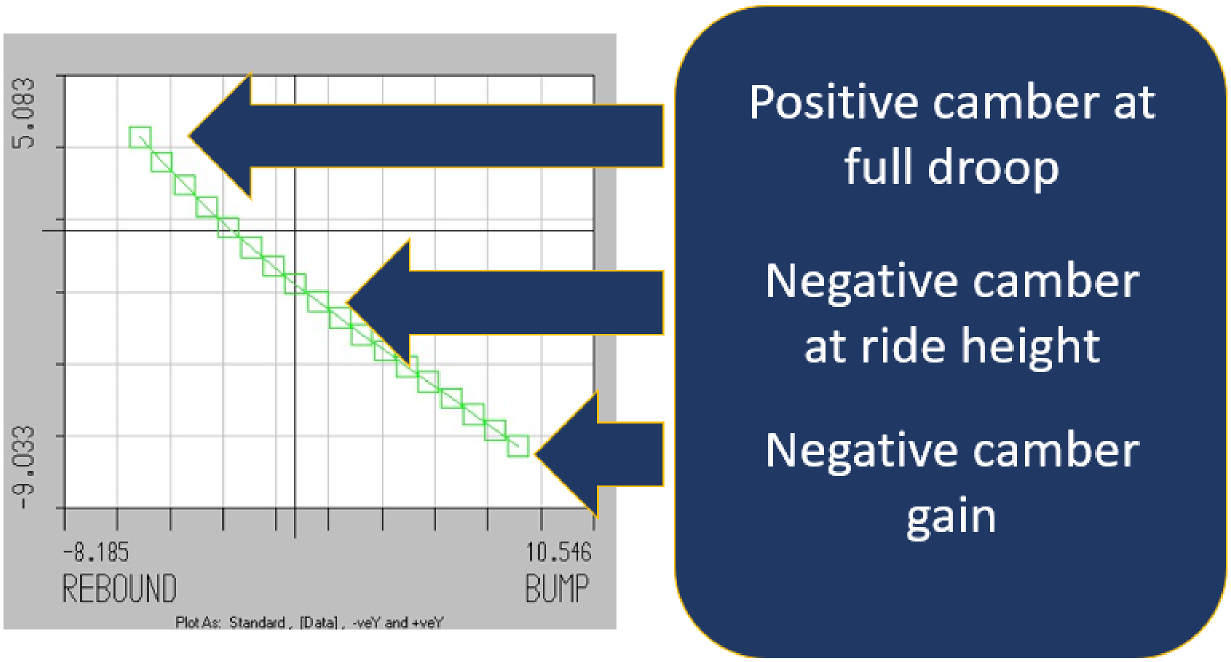


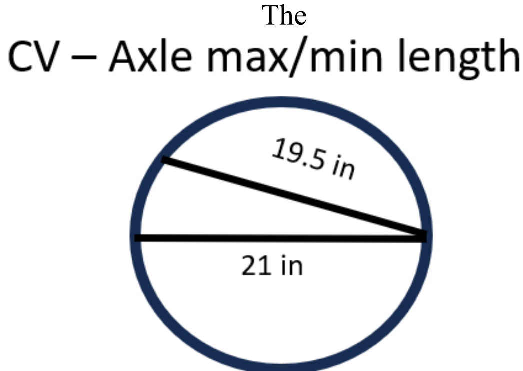
Figure 63: Graph of the camber angle at different points of travel. Listed are main takeaways from the graph.

This
The

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



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

A

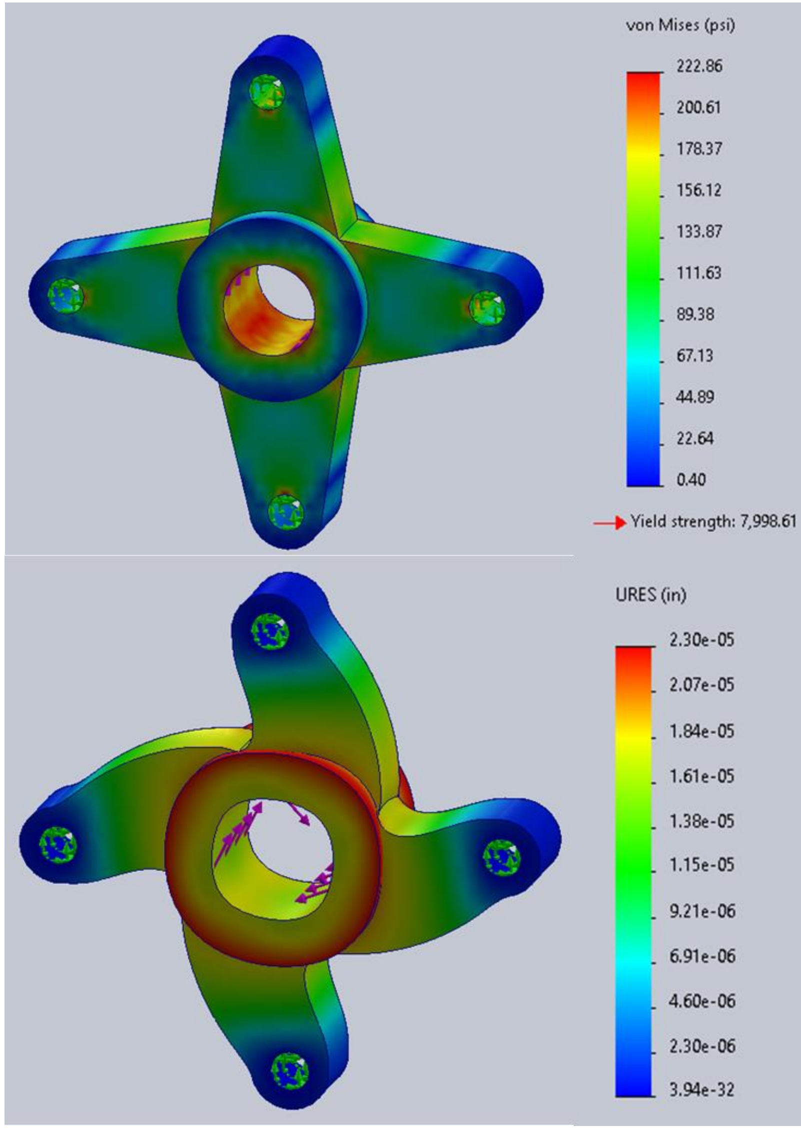


Figure
This

4.3.2.2 Lars Jensen

After

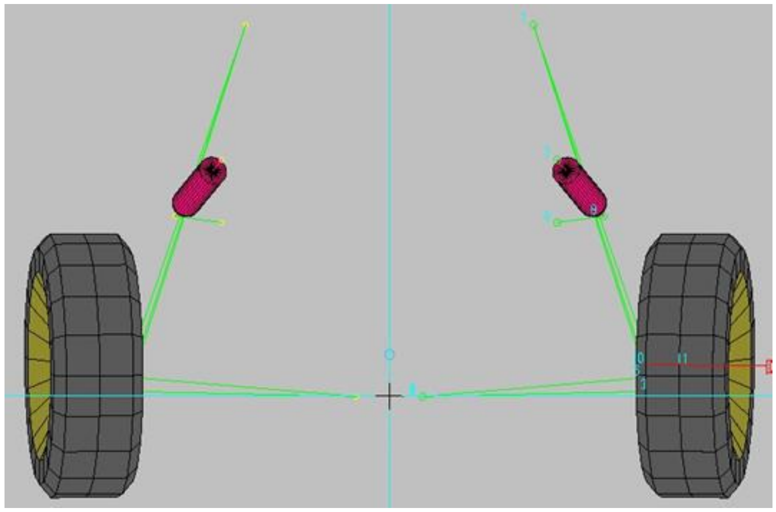


Figure 67: SHARK Top View

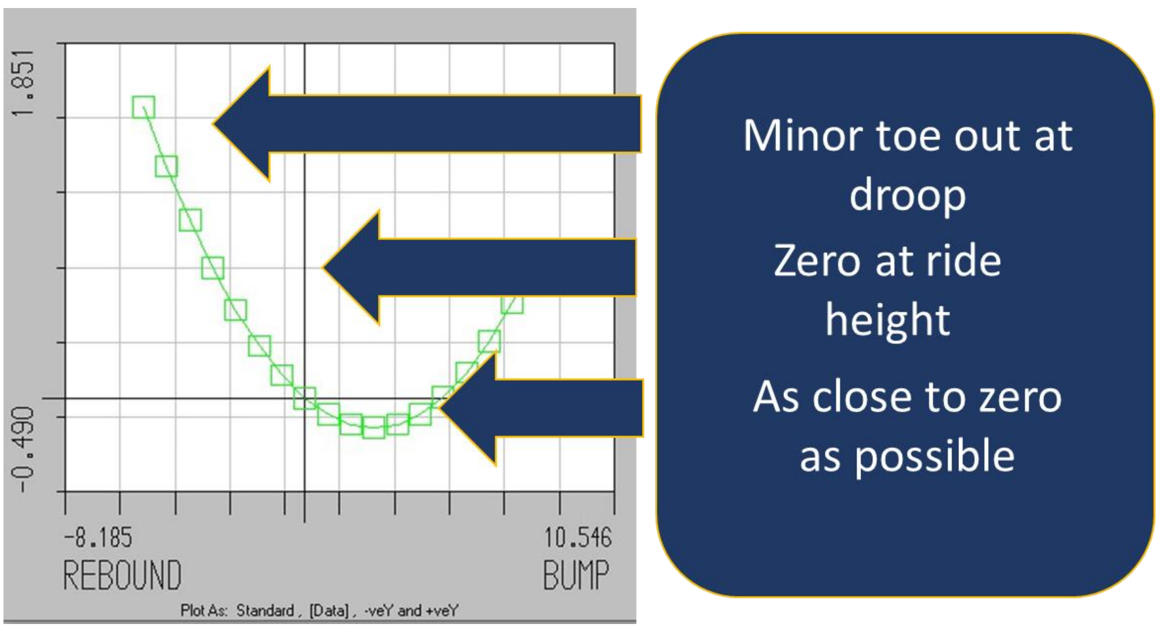


Figure 68: SHARK Toe Analysis

The

Equation 21 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.

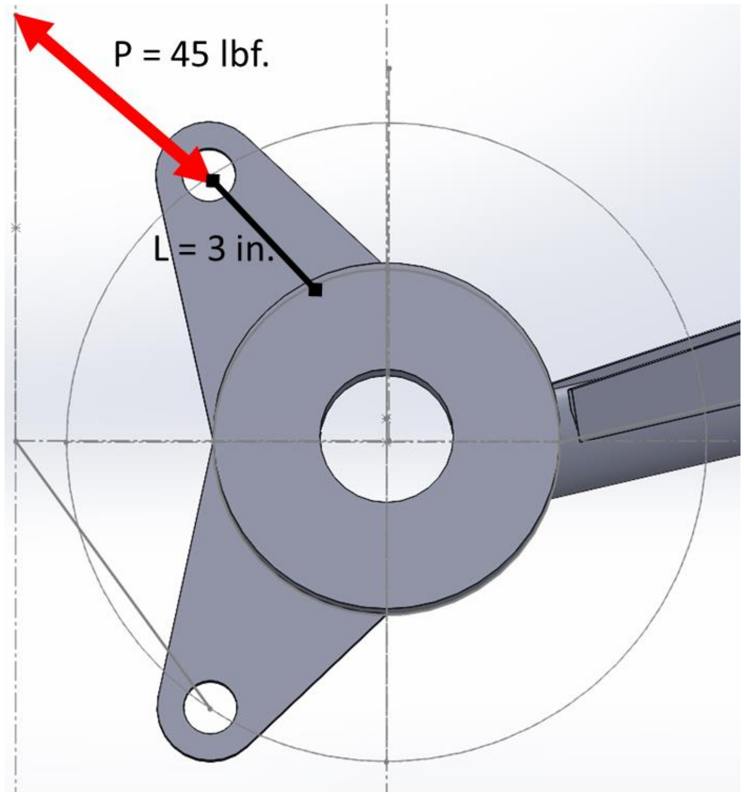


Figure 69: Knuckle Design

- 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.

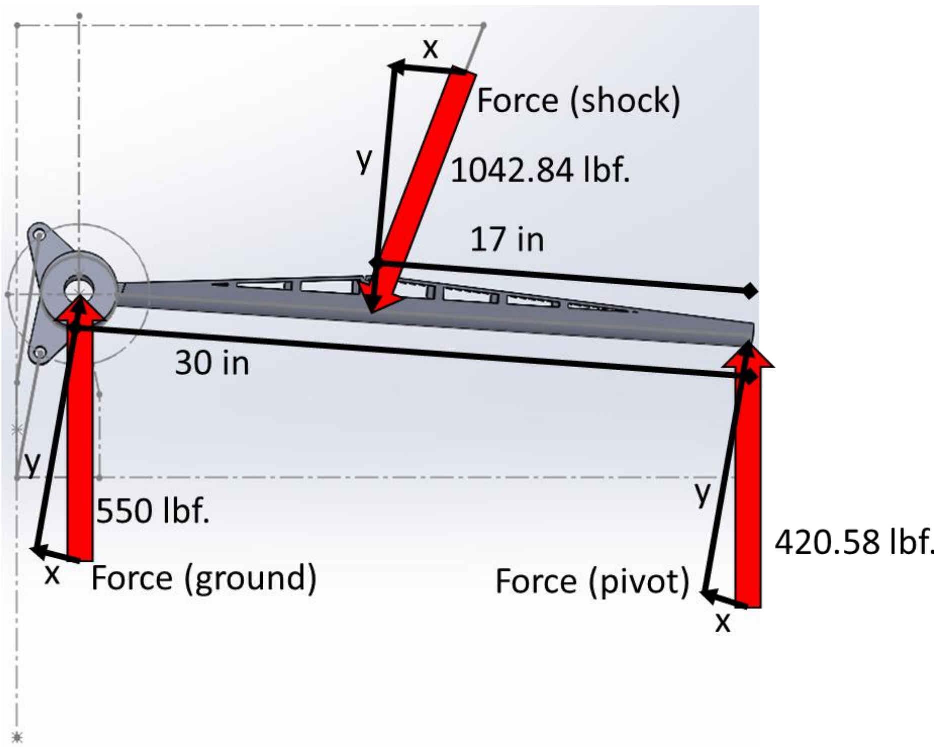


Figure 70: Trailing Link Design

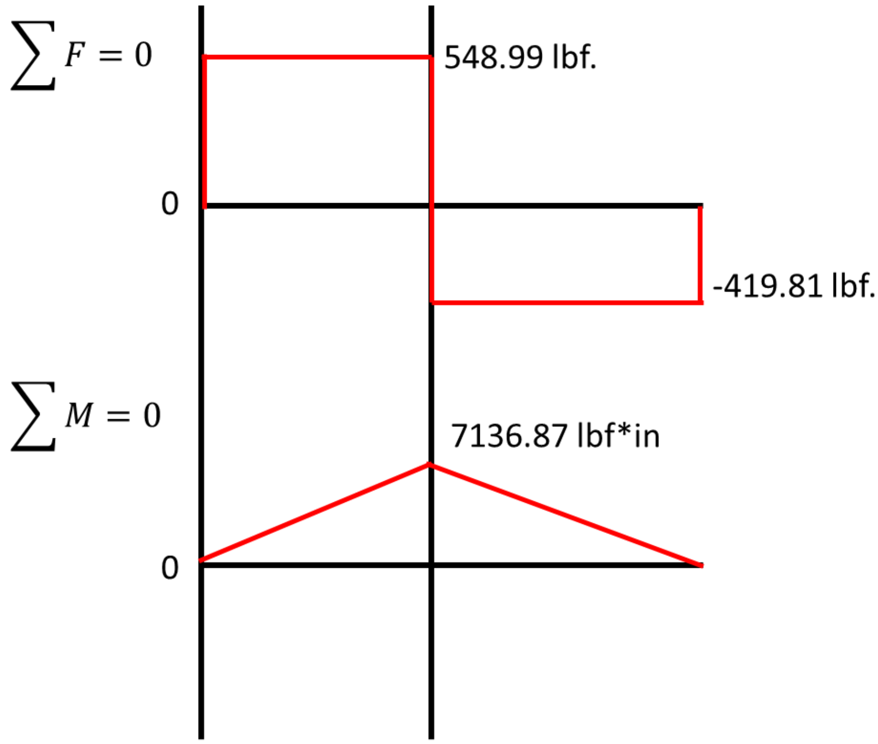
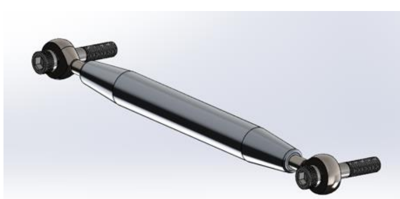
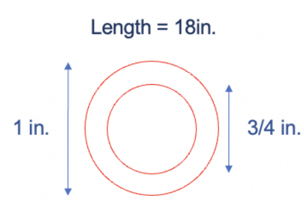


Figure 71: Shear and Moment Diagram

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:

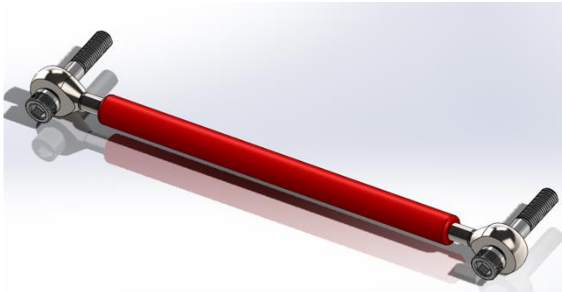


Figure 72: Carbon-Fiber Camber Link

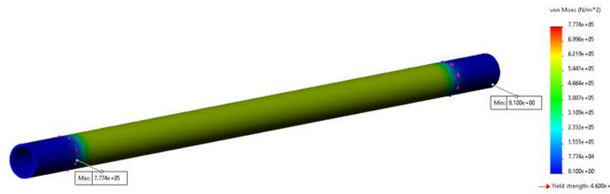


Figure 73: Steel Tube FEA

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

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 78, 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 78, 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 78. The equations used in the calculations are pictured below in Figure 79. 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 80. 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.

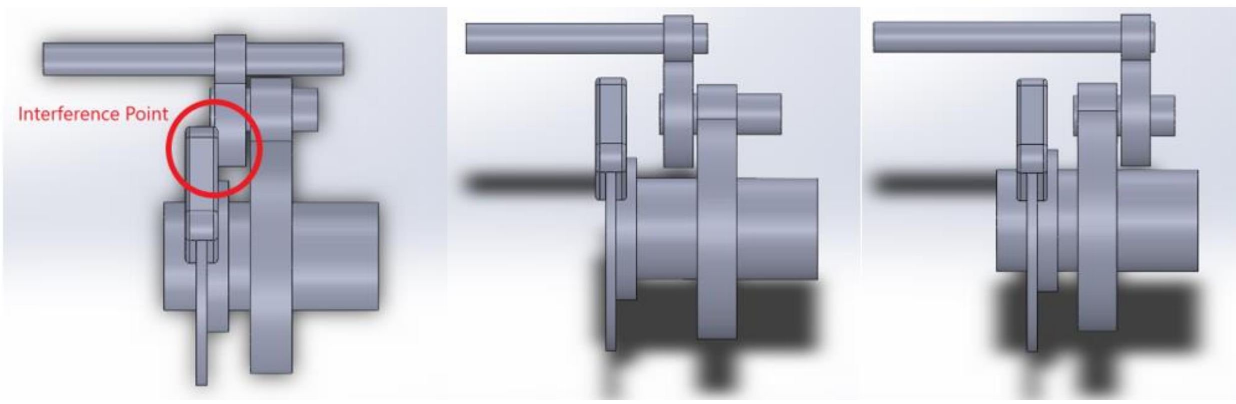


Figure 78: Gearbox Design Alternatives

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 79: 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 80: 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.

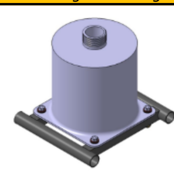
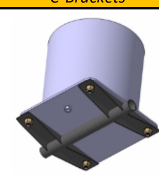
Figure 81: Potential CVT System Choices

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.

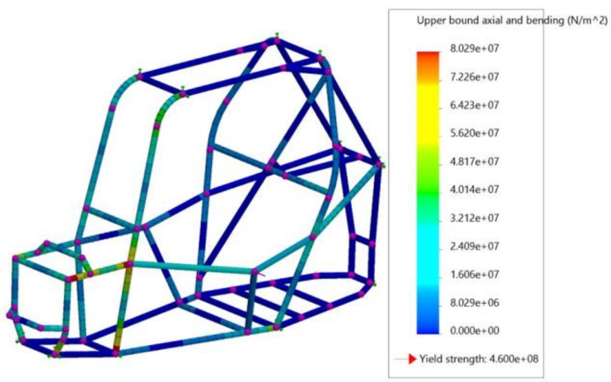


Figure 82: FEA of Flared SIM

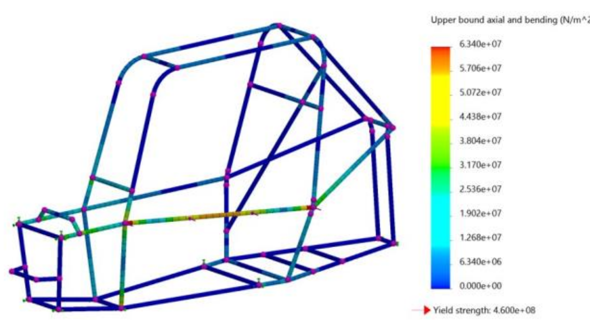


Figure 83: FEA of Straight SIM

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

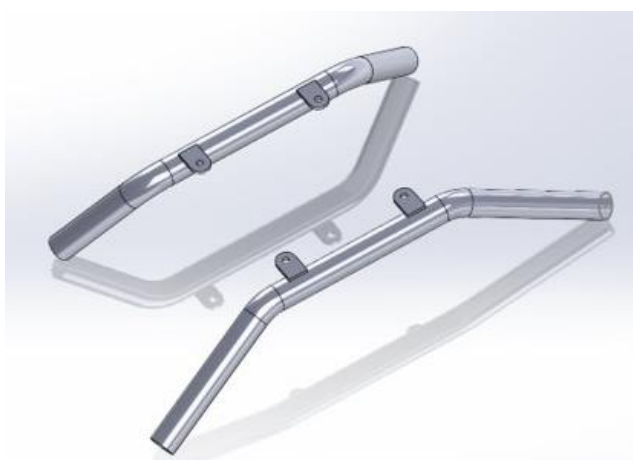


Figure 86: Tab Mounts

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 84: 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.

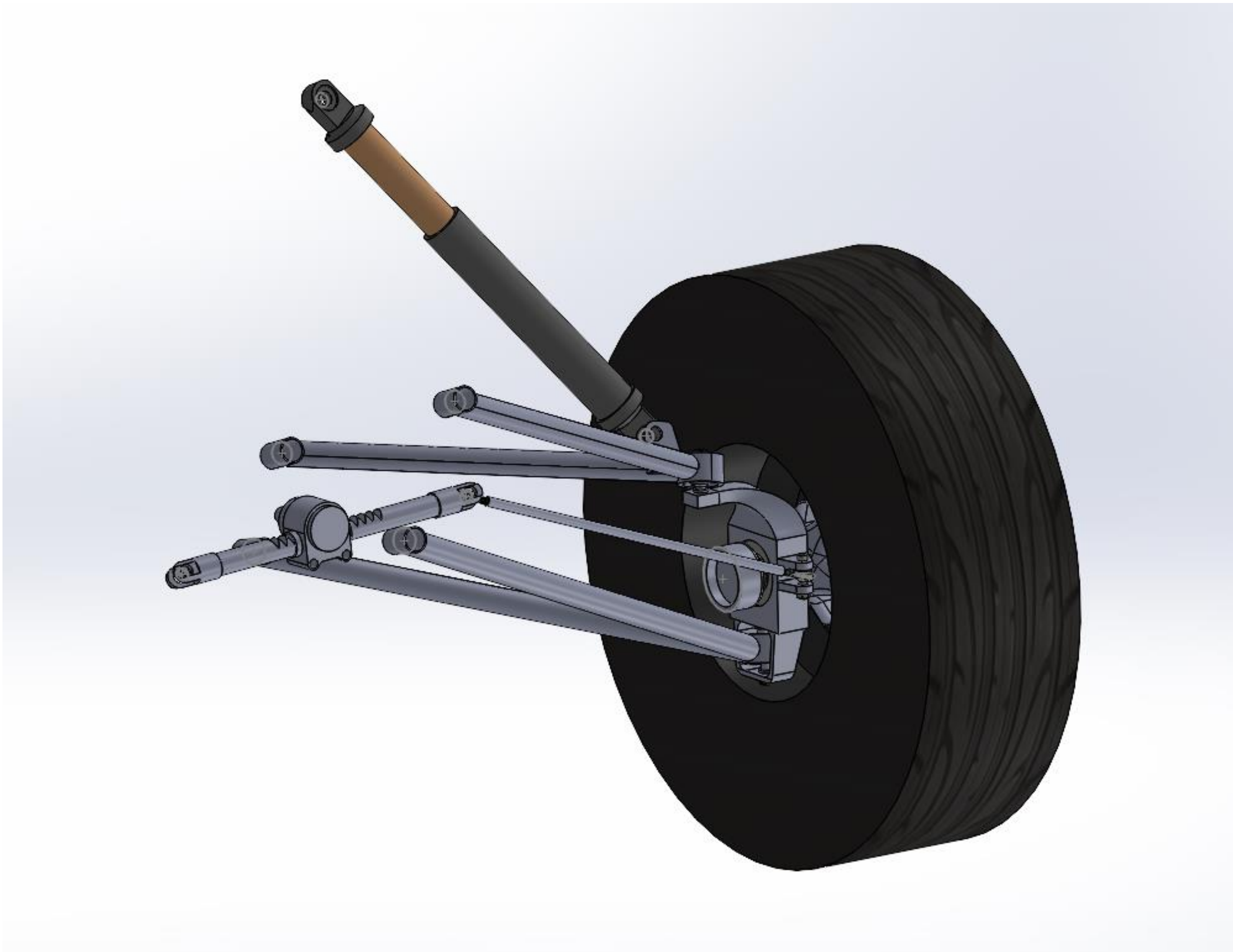
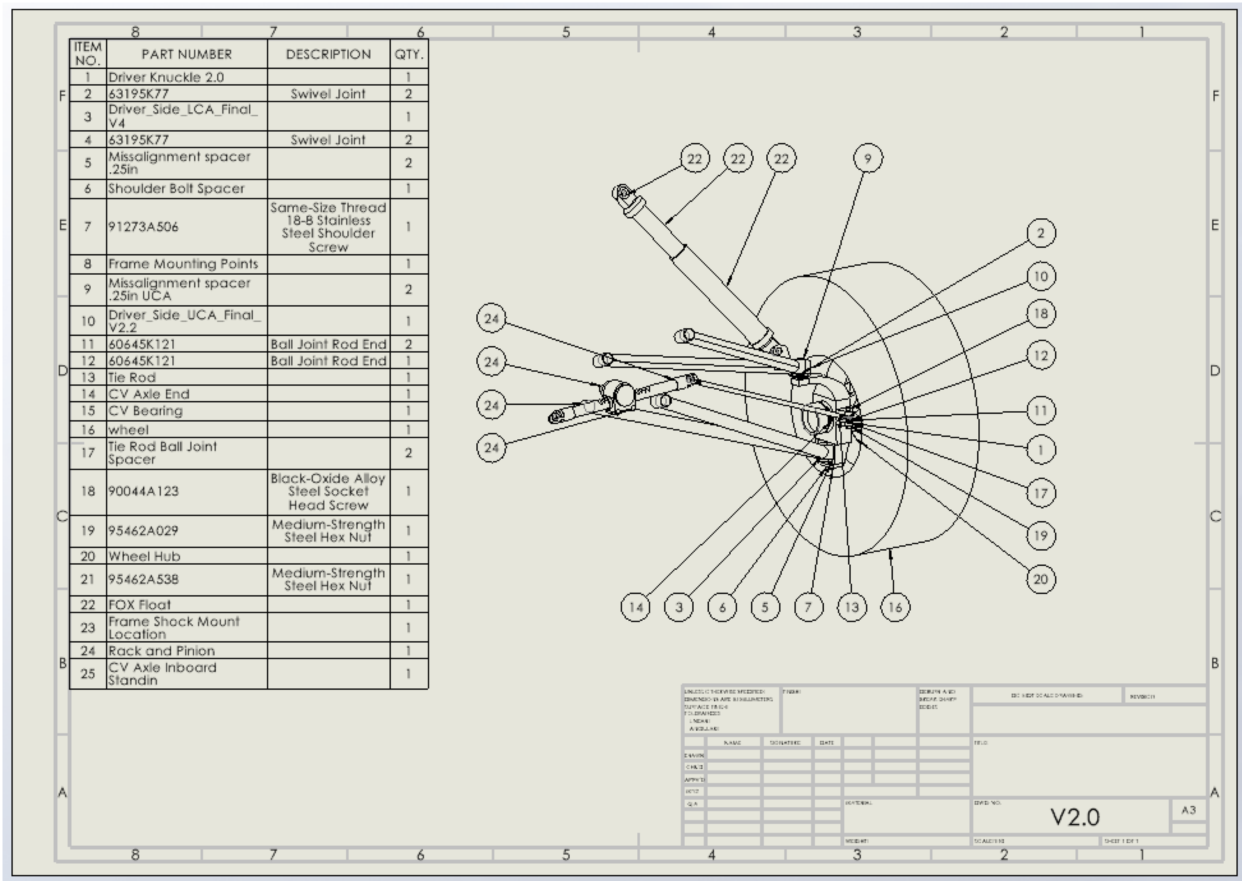


Figure 85: Current SolidWorks Model for Baja Vehicle Front Suspension and Steering



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 86: 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 87: Rear End Decision Matrix

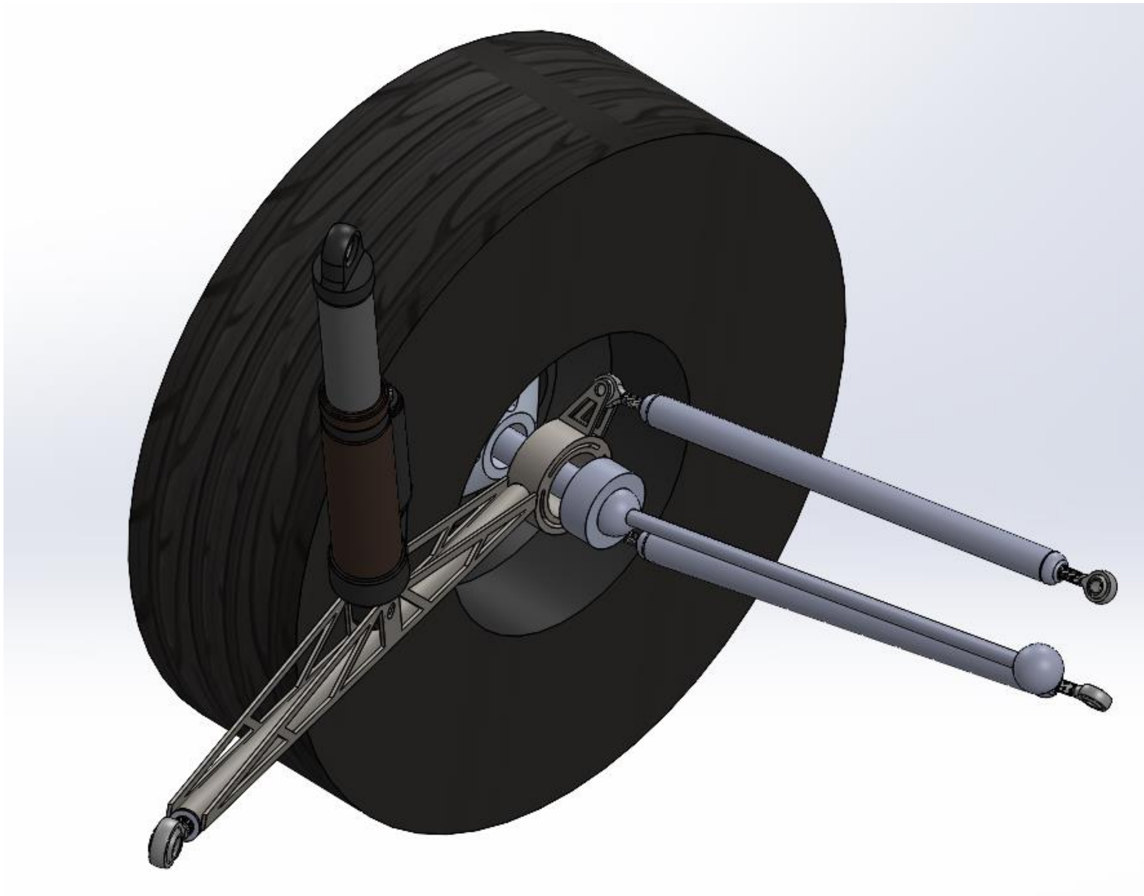
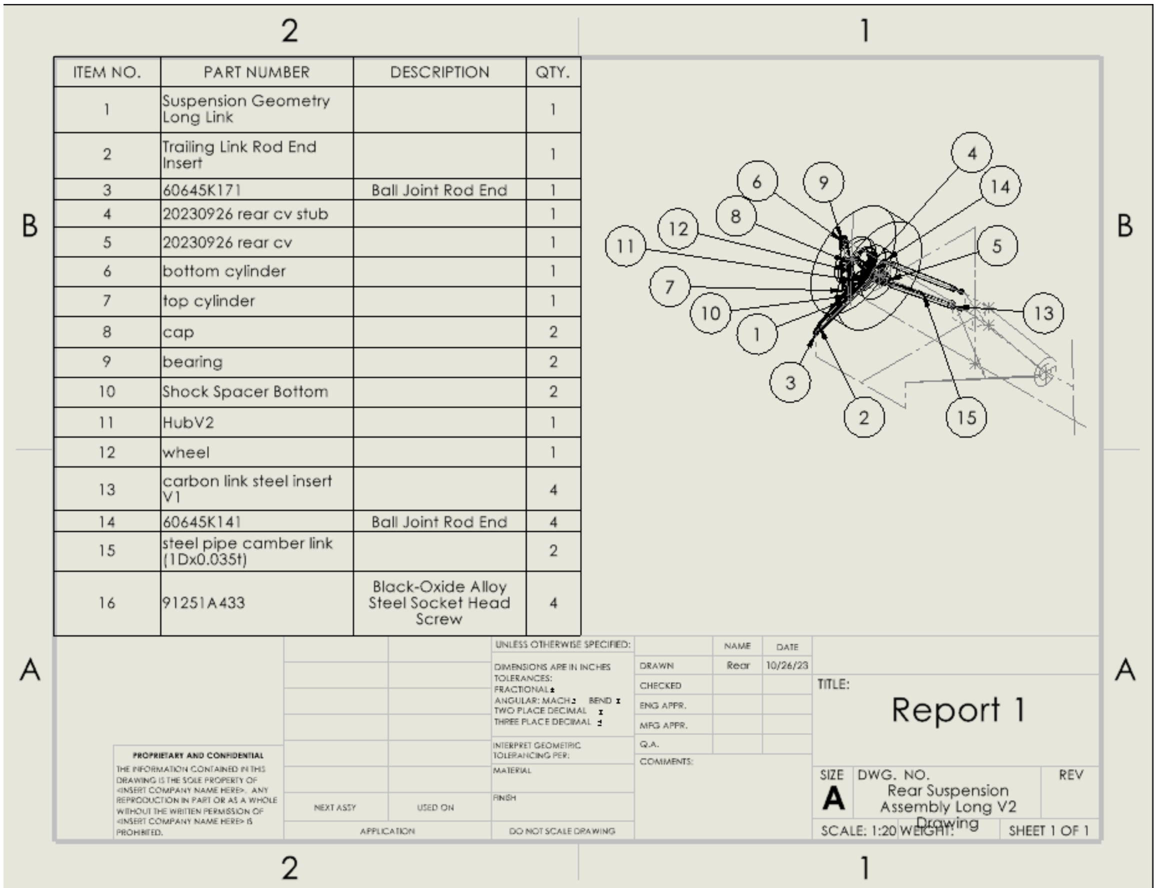


Figure 88: Rear Suspension Assembly

The current state of the rear suspension system in CAD is represented in Figure 88 and Figure 89. 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 89: 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.

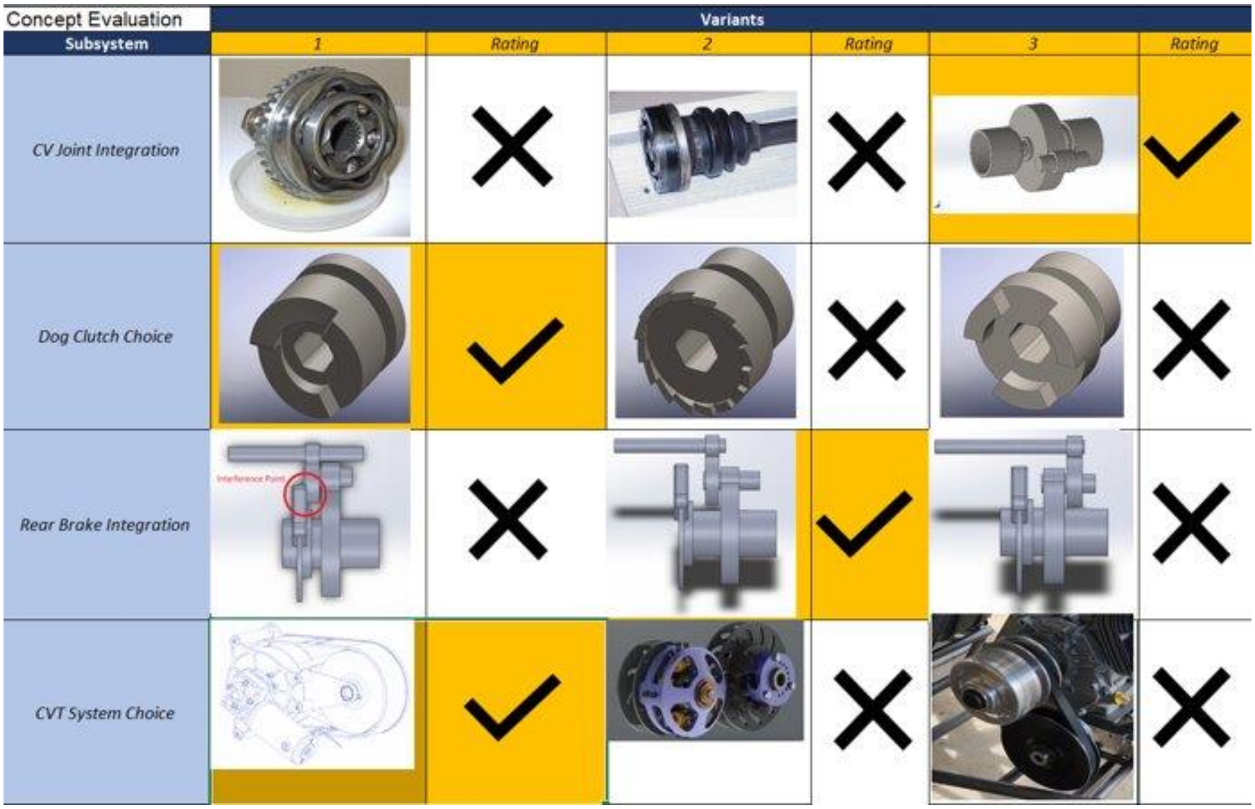


Figure 90: Drivetrain Concept Evaluation

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.

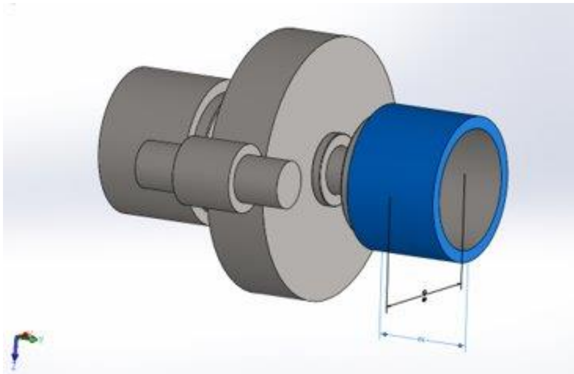


Figure 91: Front Gearbox CAD

Figure 92: Dog Clutch with Pulley

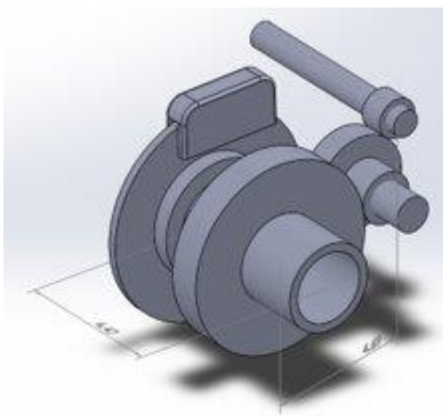


Figure 93: Rear Gearbox CAD

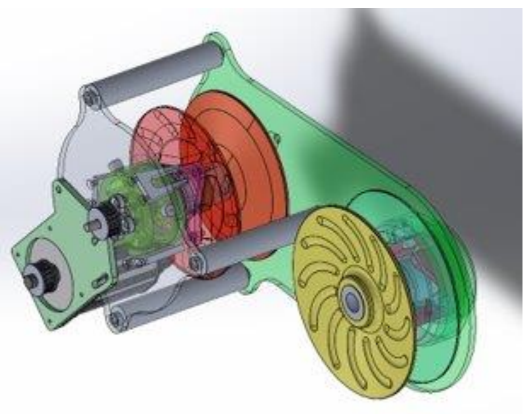


Figure 94: ECVT CAD

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 95.

	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 95: Frame Final Concept Selection

The current state of the frame in CAD is shown below in Figure 96. Currently, the team is working on integration with other sub teams, mainly front and rear suspension. Rear suspension tabs are next to be integrated, followed by body paneling and drivetrain integration.

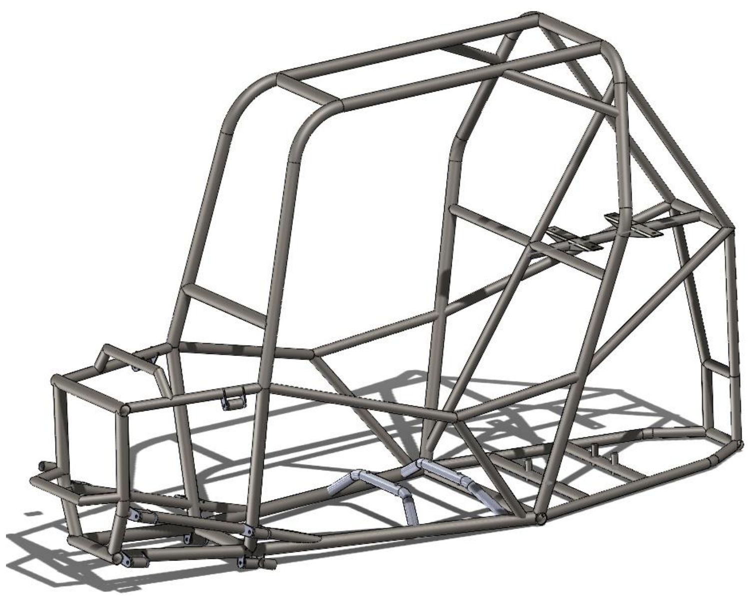


Figure 96: Frame Model

5 CONCLUSIONS

The objective of this project is to develop and build a competitive and functional single seat Baja vehicle. This capstone not only must meet the requirements of NAU's capstone course but also must 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, 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 Figure 97 below.

Figure 97: Baja Full Assembly

6 REFERENCES

[1] "Nau Sae Baja team on Instagram: 'so excited to see where our team members end up in the next few years...and that's a wrap!,'" Instagram, https://www.instagram.com/p/COd0nAALO0O/?utm_source=ig_web_copy_link&igshid=MzRlODBiNWFlZA (accessed Sep. 12, 2023).

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7 APPENDICES

7.1 *Appendix A: Project Management*

Table 16: Appendix Gantt 1

PROJECT: SAE Baja 24

NAU A.Y. 2023-2024

Project Manager: Abe Plis

Project start date: 9/12/2023

Scrolling increment: 0

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

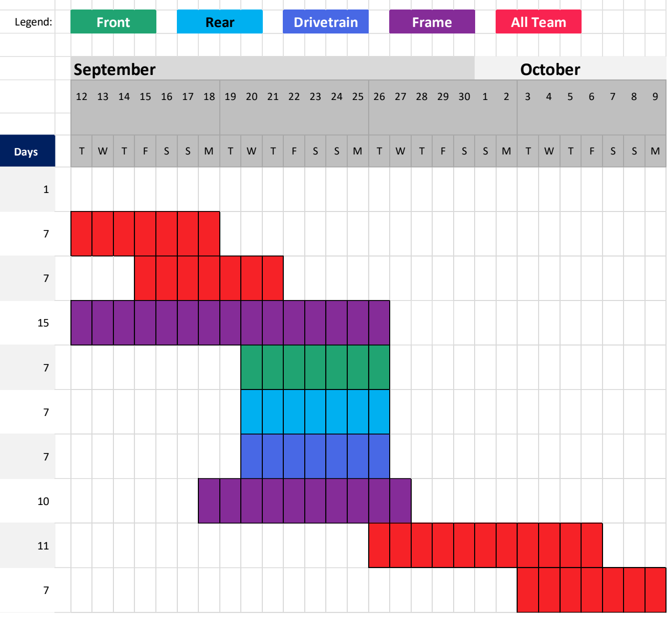

Table 17: Appendix Gantt 2

PROJECT: SAE Baja 24

NAU A.Y. 2023-2024

Project Manager: Abe Plis

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

Project start date: 9/12/2023

Scrolling increment: 28

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

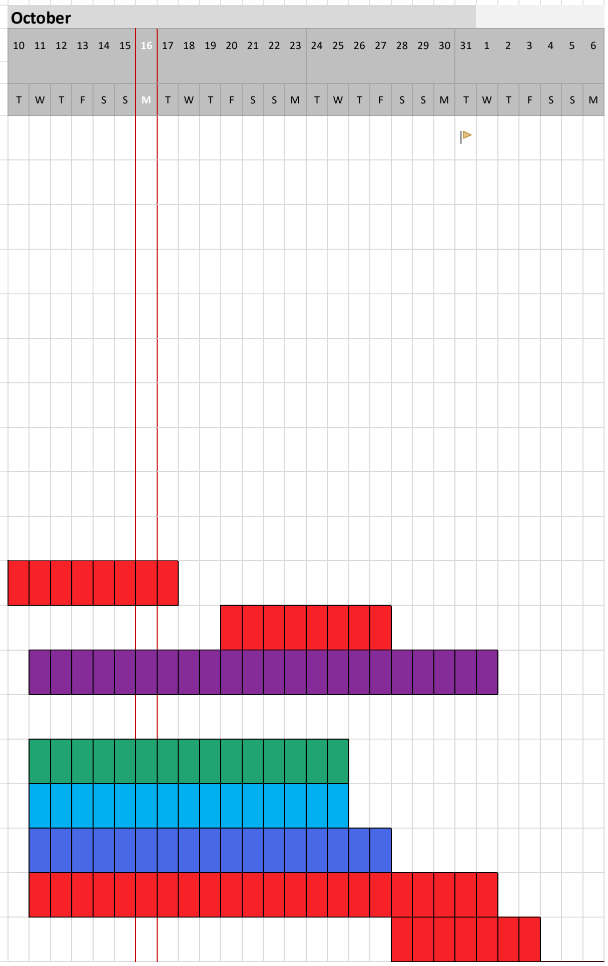

Table 18: Appendix Gantt 3

PROJECT: SAE Baja 24

NAU A.Y. 2023-2024

Project Manager: Abe Plis

Project start date: 9/12/2023

Scrolling increment: 55

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