Department of Mechanical Engineering
SAE Baja 24
ME486C - 001

To: David Willy

From: SAE Baja 24

Date: 01/26/2024

Re: Engineering Calculation Summary

1.0 Top Level Design Summary

This capstone team is operating under the engineering rules and regulations established by the SAE Baja organization with the goal of designing and manufacturing a single-seat, all-terrain Baja vehicle that will be used to participate in the April 2024 SAE Baja competition. The engineering problem established by this competition is to facilitate the construction of a maneuverable, reliable, and lightweight vehicle that satisfies all rules set forth in the 2024 SAE Baja competition rulebook. The 2023-2024 NAU SAE Baja capstone team is proud to present their up-to-date, but not complete, solution to this engineering problem, shown below in Figure 1. The design analysis will be performed by the four sub-teams that were responsible for the engineering of this vehicle in the subsequent memo.

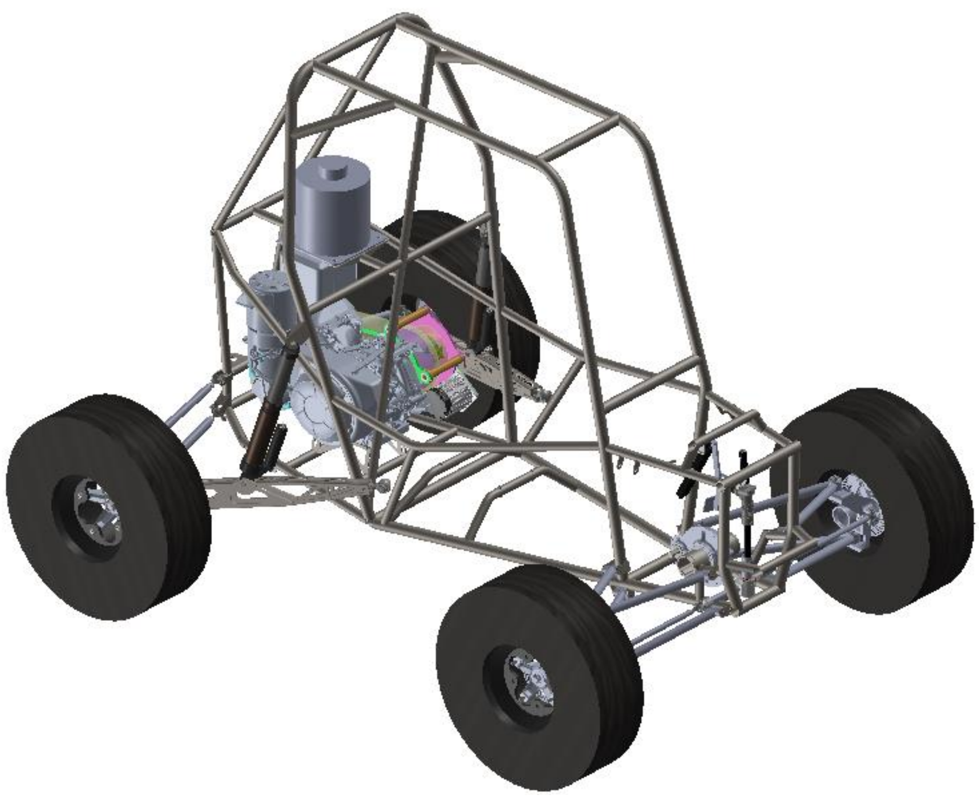

Figure 1: Full CAD Assembly

1.1 Front End

The front-end sub-team is responsible for designing a sub-system that allows the vehicle to steer, navigate rough terrain, and control driver inputs (braking, throttle, etc.) whilst adhering to the geometric constraints of the frame sub-team and the SAE Baja rulebook. As of the beginning of the spring semester, the front end solution has been designed and validated based on applicable engineering principles. The CAD assembly can be seen below in Figure 2.

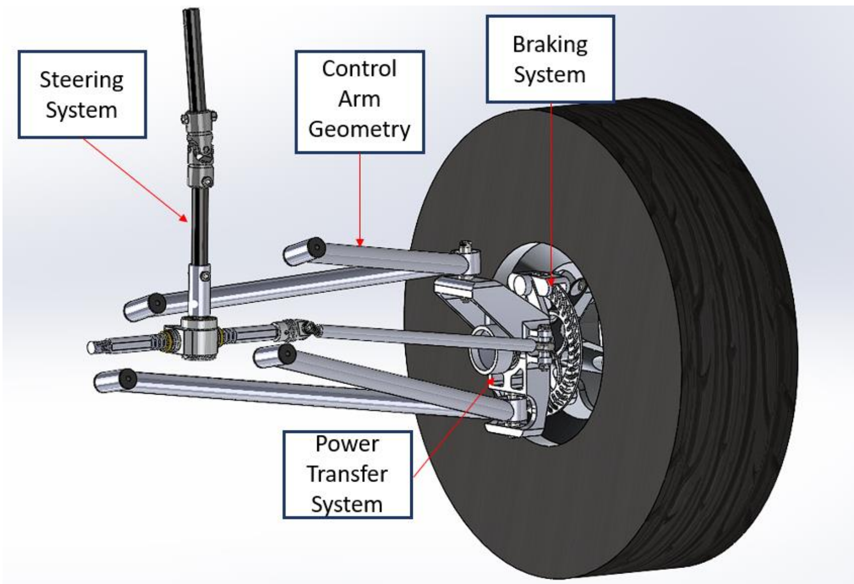

Figure 2: Front End Final CAD Design

This design in made up of several smaller systems that integrate to form the overall front end sub-system. First, the steering system will be discussed; the driver applies an input to the steering column via the steering wheel, which rotates a gear that is contained within the steering rack housing. This causes the steering rack to slide towards the chosen direction, which pushes on the tie rod that is connected to the steering rack and the steering knuckle. The knuckle is pushed, rotating along the steering axis created by the control arm ball joints, which causes the wheel to turn (facilitating the steering of the car during operation). Next, is the control arm geometry; the control arms are the triangular members created from 4130 steel tubing that connect to the frame (2 pivoting attachment points) and the knuckle (1 ball joint attachment point). These control arms guide the wheel through its travel as it encounters bumps and other debris on the track. The upper control arm also serves as an attachment point for a shock absorber that helps cushion impact perceived by the driver and maintains the vehicle's stability. After that, there is the power transfer system that is made up of the cv shaft, knuckle, hub, and wheel. The power is delivered from the front gearbox that routes down via a cv axle to the cv shaft. The cv shaft feeds through the knuckle and is connected to the hub via a splined shaft. The hub is attached to the wheel/tire assembly such that, as the cv axle receives power from the front gear box, the wheels of the vehicle rotate and transfer power to the ground. To control the speed of the car during operation, a braking system has been incorporated into the sub-system that functions in the following manner: the driver presses on the brake pedal (not pictured) to apply pressure throughout the brake lines; this pressure travels to the caliper and pushes the brake pads inwards, which applies clamping force to the rotor; the rotor is attached to the hub, so a clamping force on itself slows down the hub's rotation and, thus, the wheels of the vehicle. The sub-system, in its entirety, satisfies the rules established by the SAE regarding trackwidth (maximum 64”) and driver safety (all four wheels must lock up under braking pressure).

Aside from requirements established by the SAE, the team generated some general customer/engineering requirements (hereby referred to as “CRs” and “ERs”) that governed the design of the front end sub-system. Since the front end of the vehicle only has a single strict requirement established in the rulebook, many of these CRs are inferred based on desirable vehicle attributes and from extensive benchmarking research.

- 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.)

For each CR, a corresponding ER was generated with a quantifiable metric.

- 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 allowed 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/regulations.

The CRs and ERs that the front end worked with throughout the design cycle had a variety of interaction effects and, as such, were analyzed relative to each other as well as the design success of the car overall. The front end QFD (see Table 1) 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).

Table 1: Front end QFD

System QFD

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

#	Engineering Requirement	1	2	3	4	5
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 were 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

1.2 Rear End

The rear end sub-team deals with transferring rotational power to the rear wheels and ensuring the system maintains traction over rough terrain. The constraints for the rear end suspension system are defined by the frame dimension requirements, rear drivetrain location, and the SAE Baja rulebook. The rear end team has successfully completed the design phase of the solution to this problem and have begun working on the production and fitment of all the individual components. The finished assembly can be seen below in Figure 3.

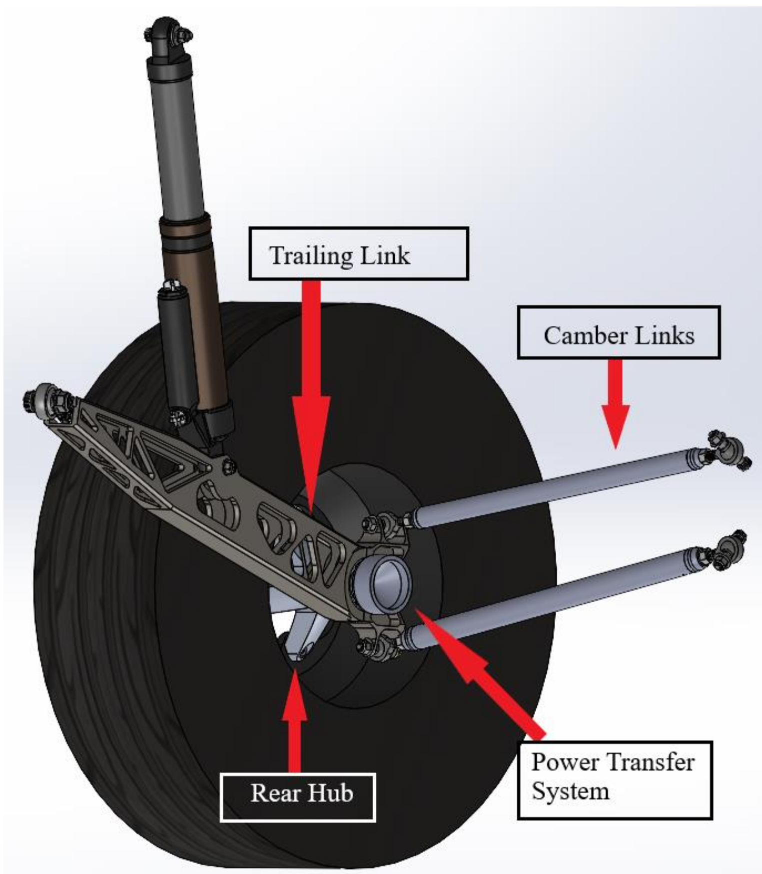

Figure 3: Rear End Final CAD Assembly

The rear-end subassembly connects the frame to the ground via six points of contact: two camber links, the CV axle, the trailing link, the shock, and the tire. The system originates from driver input via the throttle pedal. From this input, power surges at the engine, causing the cv to spin, transferring power from the gearbox to the ground. The cv cup is cradled by the knuckle which houses press-fitted ball bearings to ensure smooth operation and tight fitment under constant load. The cv spline is keyed to the hub, which is fastened by a nut on the cv threads. A spacer outside of the knuckle separates the CV bearing from the hub, which bolts to the wheel via four bolts. The wheel is encased by a tire which transfers power directly to the ground. Moving to the suspension, there are four pinned points of contact connecting the suspension to the frame. The trailing link serves as an arm that the tire pivots around. The trailing link design sees much more leverage than previous years trailing link suspensions, making for a high bending moment at the knuckle and middle of the link. Because of this, the link had to be rebuilt to withstand these loads. The knuckle is welded directly to the trailing link, maximizing strength and simplicity as its pivots at the camber link rod ends. Camber links connect the knuckle to the frame, allowing tunability of camber angle and cornering characteristics.

The trailing arm will be cut from 4130 steel and welded into a box with a milled steel insert at the frame side for the rod end to connect to. The plating is stripped of unnecessary weight validated by SolidWorks FEA resulting in a combined weight of 11.56 lbs including the knuckle. Cut at a radius, the knuckle, milled from the same strength steel, is welded directly to the trailing link at an angle allowing the camber links to be mounted vertical to each other, satisfying geometry proven to be successful in Lotus Shark. The camber links mount at the knuckle by 3/8" rod ends secured by 1.5" shoulder bolts, washers, and lock nuts. The mounting points allow for movement up to 3/16" forward or backward and the links themselves allow for total movement of 1.2", further promoting adjustability.

Table 2: Rear End System QFD

System QFD

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

#	Technical requirement correlations	Decrease weight	Increase strength	Increase rearward axle path	Increase linkage radii	Increase ground clearance	Vehicle width	Decrease CV axle angle
1	Decrease weight							
2	Increase strength	-3						
3	Increase rearward axle path							
	Increase linkage radii	-1						
4	Increase ground clearance	-6	3	1				
5	Vehicle width	1	3		6	2		
6	Decrease CV axle angle		3	-2	1			

Legend	
A	CWRUM
B	RIOT Baja
C	ETS BAJA

#	Customer Requirements	Customer Weights	Decrease weight	Increase strength	increaseward rear 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				
1	Tunability	2	2	3	8	7	3	2	7	C	A			B
2	Servicability	2	2	6						A		BC		
3	Reliability	5	3	9	3	2			7			A	B	C
4	Ease of manufacturing	3	6	7	1	1		1	1		C	A		B
5	Low cost	5	9	9	3	3		1	3			A	C	B
6	Maximum Traction	2	7		8	8		3	1				B	AC
7	Maneuverability	4	5	1	8	6	5	7	1				AB	C
8	**Technical Requirement Units**		lb	Psi	in	in	in	in	degrees					
	Target Requirements		<50	NA	<1	20	>8	<64	180					
10	**Absolute Technical Importance**		120	133	97	82	26	46	73					
11	**Relative Technical Importance**		2	1	3	4	7	6	5					

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.

1.3 Drivetrain

The drivetrain sub-team is responsible for transmitting power from the engine to wheel, allowing the vehicle to move forward. A series of subsystems are required to transfer the power generated from the engine, which consists of the ECVT, front and rear gearboxes, and 4-Wheel Drive System. The drivetrain sub-team has successfully completed the design phase through extraneous calculations and validation testing. The team is now starting the production phase for all subsystems for the drivetrain. The finished CAD assembly for each drivetrain subsystem can be seen below in Figure 4.

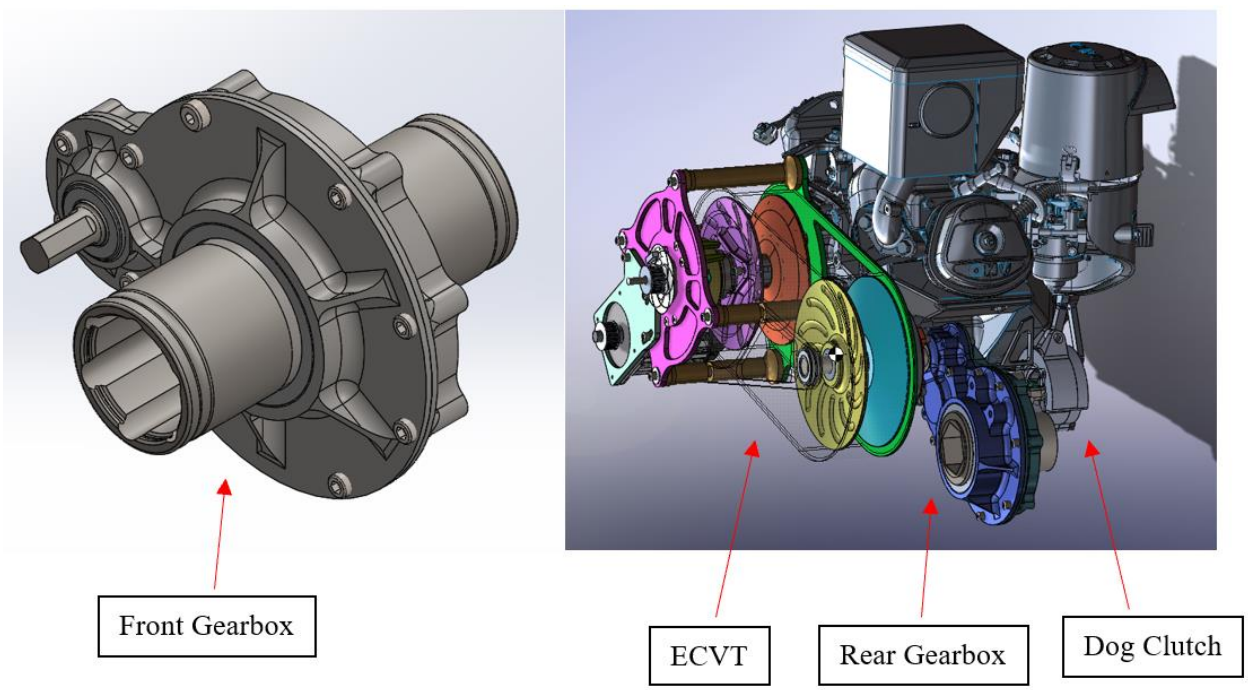

Figure 4: Drivetrain Final CAD Assembly

Table 3: Drivetrain QFD

	Technical Requirement	1	2	3	4	5	6	Legend	
1	top speed								
2	drivetrain efficiency								
3	torque to the wheels	6						Legend	
4	service life							A	Cornell 2023
5	total system weight (w/out engine)		3					B	NAU 2021 #21
6	total transmission range		6					C	NAU 2023 #74
7	Meets HROE Guard specifications					-9			

	Customer Needs	Customer Weights	top speed	drivetrain efficiency	torque to the wheels	service life	total system weight (w/out engine)	total transmission range	Meets HROE Guard specifications	1 Poor	2	3 Acceptable	4	5 Good
1	fast	5	9	6	6	3	6	6	1	C			B	A
2	High effeciency	3	9	9	6	6	3	6	3	C		B		A
3	fast acceleration	5	3	6	9	3	9	9	1	C		B		A
4	durable	1	1	6	1	9	1	4	9			AC		B
5	can crawl and go fast	4	9	6	9	3	3	9	1	C			B	A
6	NEEDS TO BE SAFE	5	1	1	1	6	1	1	9			C	B	A
7	Aesthetically Pleasing	3	9	1	9	3	9	9	3		C		A	B
8														
9														
10														
11														

	top speed	drivetrain efficiency	torque to the wheels	service life	total system weight (w/out engine)	total transmission range	Meets HROE Guard specifications
Technical Requirement Units	MPH	Unitless	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 drivetrain QFD shown in Table 3, there are certain characteristics that the drivetrain sub-team must follow to achieve an efficient and effective drivetrain. The customer requirements and engineering requirements were evaluated based on the importance of each criteria. The final rankings 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 produce an optimal design for all the subsystems for the drivetrain.

2.0 Summary of Standards, Codes, and Regulations

The engineering practices that were used to design this vehicle rely on a variety of standards, codes, and regulations from national and international professional engineering organizations. These standards apply to every part that is going to be machined and/or purchased for use on the vehicle. This section presents a compilation of these standards for each sub-team.

2.1 Front End

The industry standards, codes, and regulations that apply to the design of the front-end sub-system are as follows:

- Control Arm Swivel Joint
  - ANSI/ABMA 22.2
- Upper Control Arm Knuckle Bolt
  - ASTM A574
  - REACH (EC 1907/2006) (06/14/2023, 235 SVHC) Compliant
  - RoHS 3 (2015/863/EU) Compliant
  - Specialty Metals Compliant (252.225-7009)
- Lower Control Arm Knuckle Bolt
  - ASTM A574
  - REACH (EC 1907/2006) (06/14/2023, 235 SVHC) Compliant
  - RoHS 3 (2015/863/EU) Compliant
  - Specialty Metals Compliant (252.225-7009)
- Tie Rod Knuckle Bolt
  - ASTM A574
  - REACH (EC 1907/2006) (06/14/2023, 235 SVHC) Compliant
  - RoHS 3 (2015/863/EU) Compliant
  - Specialty Metals Compliant (252.225-7009)
- 4130 Steel Tubing for Control Arms
  - ASTM A1031 Grade 4130
- CV Bearing for Use in Knuckle
  - P6 class tolerances for radial bearings
- Delrin Rods for Bushing Manufacturing
  - This material is not hazardous under criteria of the Federal OSHA Hazard Communication Standard (29CFR 1910.1200)
  - RoHS 3 (2015/863/EU) Compliant
  - Specialty Metals COTS-Exempt
- 3/8” Rod End for Tie Rod
  - RoHS 3 (2015/863/EU) Compliant
  - REACH (EC 1907/2006) (01/17/2023, 233 SVHC) Compliant
  - Specialty Metals COTS-Exempt
- U-Joint for Steering Column
  - RoHS 3 (2015/863/EU) Compliant
  - REACH (EC 1907/2006) (06/14/2023, 235 SVHC) Compliant
  - Specialty Metals COTS-Exempt
- 1/4"-20 Bolts for Steering Column
  - RoHS 3 (2015/863/EU) Compliant

    - REACH (EC 1907/2006) (06/14/2023, 235 SVHC) Compliant
    - Specialty Metals Compliant (252.225-7009)
- 1/4"-20 Nut for Steering Column
    - RoHS 3 (2015/863/EU) Compliant
    - REACH (EC 1907/2006) (06/14/2023, 235 SVHC) Compliant
    - Not Specialty Metals Compliant
- 1/2"-13 Bolt for Rack to Tie Rod
    - RoHS 3 (2015/863/EU) Compliant
    - REACH (EC 1907/2006) (06/14/2023, 235 SVHC) Compliant
    - Specialty Metals Compliant (252.225-7009)
- 1/2"-13 Nut for Rack to Tie Rod
    - RoHS 3 (2015/863/EU) Compliant
    - REACH (EC 1907/2006) (06/14/2023, 235 SVHC) Compliant
    - Not Specialty Metals Compliant
- Knuckle Manufacturing/Tolerancing
    - AISI/ABMA 7-1995
    - ISO 68-1
- Hub Manufacturing/Tolerancing
    - AISI/ABMA 7-1995
    - ISO 68-1
- Throttle Sensor Limit Switch
    - UL (UL1054)/CSA (CSA C22.2 No.55)
    - VDE (EN61058-1)
        - Testing conditions: 5E4 (50,000 operations), T85 (0°C to 85°C)
- Brake Rotor, Chassis Mounting Tabs
    - ASTM A36/A36M Steel

The front-end design must also adhere to the rules and regulations established by the SAE Baja organization, which are outlined below:

- All-Terrain Capability
    - SAE Baja B.1.4.1 – Terrain Type
    - SAE Baja B.1.2.2 – Clearance and Traction
- Width Limitation
    - SAE Baja B.1.6 – Limitations
- Vehicle Controls Safety Compliance
    - SAE Baja B.7.1 – Brake System
    - SAE Baja B.7.1.4 – Brake Lines
    - SAE Baja B.7.2 – Throttle System
    - SAE Baja B.10.4.1 – Brake Light
- Universal Joint Shielding
    - SAE Baja B.8.7.2 – Universal Joints
    -

Adhering to the industry standards and the competition rules/regulations presented above will ensure that the design of the front end is safe for the driver and poses no threat to the safety of the engineers or other onlookers during manufacturing and operation.

2.2 Rear End

The industry standards, codes, and regulations that apply to the design of the rear end sub-system are as follows:

- 5/8” Rod End for Trailing Links
  - RoHS 3 (2015/863/EU) Compliant
  - REACH (EC 1907/2006) (01/17/2023, 233 SVHC) Compliant
  - Specialty Metals COTS-Exempt
- 5/8” Shoulder Screw for Trailing Link Pivots
  - ASME B18.3, ASTM A574
  - RoHS 3 (2015/863/EU) Compliant
  - REACH (EC 1907/2006) (01/17/2023, 233 SVHC) Compliant
  - Specialty Metals Compliant (252.225-7009)
- ½”-13 Flange Locknut for Trailing Link Pivots
  - RoHS 3 (2015/863/EU) Compliant
  - REACH (EC 1907/2006) (01/17/2023, 233 SVHC) Compliant
  - Not Specialty Metals Compliant
- 3/8” Shoulder Screw for Shock Mounts and Camber Link Pivots
  - ASME B18.3, ASTM A574
  - RoHS 3 (2015/863/EU) Compliant
  - REACH (EC 1907/2006) (01/17/2023, 233 SVHC) Compliant
  - Specialty Metals Compliant (252.225-7009)
- 5/16”-18 Flange Locknut for Shock Mounts and Camber Link Pivots
  - RoHS 3 (2015/863/EU) Compliant
  - REACH (EC 1907/2006) (01/17/2023, 233 SVHC) Compliant
  - Not Specialty Metals Compliant
- 3/8” Rod End for Camber Links
  - RoHS 3 (2015/863/EU) Compliant
  - REACH (EC 1907/2006) (01/17/2023, 233 SVHC) Compliant
  - Specialty Metals COTS-Exempt
- CV Bearing for Use in Knuckle
  - P6 class tolerances for radial bearings
- Knuckle Manufacturing/Tolerancing
  - AISI/ABMA 7-1995
  - ISO 68-1
- Hub Manufacturing/Tolerancing
  - AISI/ABMA 7-1995
  - ISO 68-1

The rear end design must also adhere to the rules and regulations established by the SAE Baja organization, which are outlined below:

- All-Terrain Capability
  - SAE Baja B.1.4.1 – Terrain Type
  - SAE Baja B.1.2.2 – Clearance and Traction
- Width Limitation

    - SAE Baja B.1.6 – Limitations

2.3 Drivetrain

The industry standards, codes, and regulations that apply to the design of the drivetrain sub-system are as follows:

Rear Gearbox

- Bearings for gearbox
    - Bearing Trade Number R12
- Gear and Housing Manufacturing/Tolerancing
    - AISI/ABMA 7-1995
    - ISO 68-1

Front Gearbox

- Bearings for gearbox
    - Bearing Trade Number R10
- Gear and Housing Manufacturing/Tolerancing
    - AISI/ABMA 7-1995
    - ISO 68-1

4 Wheel Drive System

- Pulley Manufacturing/Tolerancing
    - AISI/ABMA 7-1995
    - ISO 68-1
- Dog Clutch Manufacturing/Tolerancing
    - AISI/ABMA 7-1995
    - ISO 68-1
- Shifter and Housing Manufacturing/Tolerancing
    - AISI/ABMA 7-1995
    - ISO 68-1

SAE Baja rules and regulations for drivetrain are listed below:

- Powertrain Guards
    - SAE Baja B.9.3.2.1 - Belt, Gear, and Chain Drives
    - SAE Baja B.9.4 - Drivetrain Breather / Vent System

3.0 Summary of Equations and Solutions

During the design of the vehicle, each sub-team identified critical load cases that were relevant to the operation of their sub-system. This section outlines the selection of these load cases, the calculations performed during these load case simulations, resulting design Factor of Safety (FoS) tables, and a summary of notable design revisions.

3.1 Front End

For the front end of the vehicle, the main operational focus is on the forces generated during a sudden, hard impact to the vehicle's suspension components. 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 hub 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 was used:

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

Equation 1: Front End Suspension Impact Force

This calculation resulted in a maximum impact force of 2160 lbf being seen by all suspension components during a worst-case scenario.

Aside from a drop-type impact, the largest opportunity for impact on the control arms will occur during a high-speed impact with a rigid object. A scenario involving a direct, sudden impact on the tire from the front will drive force through the ball joint mount on the control arms which will induce the largest bending moment on the long lengths of tubing.

The following equation for calculating impact force was used to iterate different impact speeds in the FEA simulation covered in the latter portion of this section.

$$F = \frac{mv^2}{2d} = \frac{vm}{t}$$

Equation 2: Control Arm Impact Force

where,

F = collision impact force
m = mass of object (550 lbs. with driver)
v = velocity of object (5, 10, and 20 mph)
d = distance of impact (N/A)
t = time of impact (0.25s for collision, start to finish)

This simple equation will be iterated in Excel to calculate impact forces for collisions at 5, 10, and 20 mph. These impact forces will be fed into an FEA program that will place a bearing force through the ball joint cup and yield a factor of safety for the lower control arm with a variety of bracing orientations.

Now, a variety of calculations derived from the worst-case load conditions will be presented below by all engineers on the front end team:

Abraham Plis

Utilizing Equation 1, a bearing stress of 1080 psi can be calculated to be seen by the knuckle across its 2-inch thickness during the drop impact 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.

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

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

Equation 3: Force Resistance of Bolt

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

Equation 4: Area of Bolt

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

Equation 5: Factor of Safety

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

Following that calculation, an analysis concerning the impact resistance of the control arms was performed for a variety of impact speeds, as described in the beginning of this section. The main intention of this calculation was to determine the efficacy of installing short bracing members parallel to the control arm pivot axis to increase the bending resistance of the control arms under impact (example provided in Figure 5).

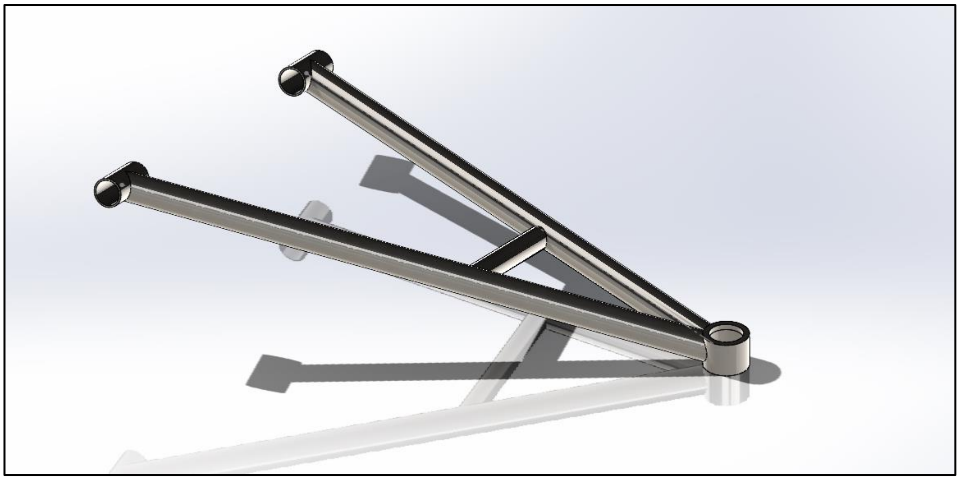

Figure 5: Front End Control Arm Bracing Example

To initialize the FEA testing, the lower control arm model with the 8" bracing was added into a new simulation. The pivots were fixed in space to provide the worst-case scenario and all force was directed through the lower control arm at once. The impact force would realistically be absorbed by the tie rod, upper control arm, knuckle, hub, and shock which would reduce the impact force significantly. The FEA was first run with the full impact force (Equation 2) to determine which bracing orientation yielded the highest FoS with the expected FoS to be well below 1.0. Then, the impact force was scaled down to 25% of its original value to account for the realistic force dissipation that would take place in the front suspension assembly during an impact to get a realistic gauge of the lower control arm's FoS that would, hopefully, be above 1.0 up to 10 mph. As a sanity check, the lower control arm was also tested for normal operating functionality during a single wheel landing which was determined to impart an impact force of 500 lbf (25% of the impact seen by the upper control arm due to shock placement). The same process was repeated for the upper control arm, though 75% of the impact force was seen in this case due to the shock placement routing much of the suspension related force to the upper control arm. The results from this worst-case scenario calculation for both the upper and lower control arm, along with the optimized bracing locations, can be seen below in Table 4 and Table 5.

Table 4: LCA Bracing FEA Results

Lower Control Arm w/ 6" Brace					
Impact Scenario	**Force Derivation**	**Applied Force**	**Orientation of Force**	**Max Displacement**	**Min FoS**
Worst Case Vertical Load for LCA	EQ1	500 lbf (quarter of force will be seen by LCA)	Upwards through ball joint cup	0.0045 in	1.43
Realistic Case Impact Load at 5mph w/ 0.25 s impact duration	EQ2	550 lbf	Backwards through ball joint cup	0.011 in	2.3
Realistic Case Impact Load at 10 mph w/ 0.25 s impact duration	EQ2	1100 lbf	Backwards through ball joint cup	0.023 in	1.1
Realistic Case Impact Load at 20 mph w/ 0.25 s impact duration	EQ2	2200 lbf	Backwards through ball joint cup	0.046 in	0.57

Table 5: UCA Bracing FEA Results

Upper Control Arm w/ 4" Brace					
Impact Scenario	**Force Derivation**	**Applied Force**	**Orientation of Force**	**Max Displacement**	**Min FoS**
Worst Case Vertical Load	EQ1	1620 lbf (three quarters of force will be seen by UCA)	Upwards through ball joint cup	0.002 in	2.2
Realistic Case Impact Load at 5mph w/ 0.25 s impact duration	EQ2	1650 lbf	Backwards through ball joint cup	0.0012 in	4.3
Realistic Case Impact Load at 10 mph w/ 0.25 s impact duration	EQ3	3300 lbf	Backwards through ball joint cup	0.0025	2.2
Realistic Case Impact Load at 20 mph w/ 0.25 s impact duration	EQ4	6600 lbf	Backwards through ball joint cup	0.005 in	1.1

From this process, it was determined that the upper and lower control arms would be structurally sound up in collisions nearing 10 mph, with collisions exceeding that speed being a concern. The implications of this result will be discussed later in this section.

<u>Bryce Fennell</u>

Using impact force estimates from equation 1 an impact force of 2200lbf transmitted through the knuckle is calculated to be used for the following engineering analysis of the front knuckle, hub, and mounting hardware.

Calculations to determine the minimum shoulder bolt size for the upper control arm mounting interface with the knuckle were performed under the assumption of a 2200lbf impact to the knuckle with all vertical impact force directed through the upper control arm/knuckle mounting interface. Calculations performed utilizing a grade 8 shoulder bolt of either .25-inch or .375-inch shoulder bolt diameter with a tensile strength of 150,000 psi. Using equations 3-5 listed above, a minimum factor of safety for a .25in shoulder bolt came to 2.4, while a minimum factor of safety for a .375-inch shoulder bolt came to 5.42. Due to the marginally increased weight for an improved factor of safety, a .375-inch shoulder bolt will be used when mounting the upper control arm to the knuckle.

Determining the lower control arm mounting bolt followed a similar process to the calculation’s foe the upper control arm with the distinction of replacing equation 3 with equation 6 as detailed below.

$$F_{shear,bolt} = \frac{S_{tensile} * (0.6) * (0.9) * A_{bolt}}{2}$$

Equation 6: Shear of Shoulder Bolt in Double-Shear Mode

The main difference between the upper and lower control arm impact forces is caused by all vertical forces being directed through the upper control arm as this is where the shock is mounted. The only forces the lower control arm faces are in the horizontal axis as the control arm keeps the wheel from bending outwards or inwards from the vehicle. Finally, because of the knuckle geometry, the lower control arm/knuckle mounting bolt will act in double shear as there are 2 supporting locations of the bolt as opposed to the single supportive location for the upper control arm as shown below.

Calculating using equations 4-6, a factor of safety for the lower control arm bolt using a .25-inch shoulder bolt came to 4.8. Given the higher factor of safety it was determined to not increase bolt size to .375-inches.

The disk brake will be mounted to the front hubs using 4 .25-inch grade 8 bolts. The maximum brake torque outputted from the braking system was found to be 2764 inch-pound force acting at a radius of 1.5 inches from the hub center resulting in a shearing force per bolt of 1036 pound-force. Factor of safety calculations were performed using equations 3-5 as listed above resulting in a factor of safety of 5.1 for all 4 bolts. Due to the high factor of safety, the decision to not increase bolt size was agreed upon.

The results from the above calculations indicate the knuckle and all mounting hardware will be capable of sustaining a bottom out impact of 2200 pound-force directed through the knuckle with a minimum factor of safety of 4.8. Improvements to the minimum factor of safety through increasing bolt size or grade were not needed and the design will move forward to production.

Evan Kamp

To meet specifications outlined by SAE Baja, the brake pedal must be able to withstand a pushing force of 450 lbs at the pedal. Because this is the minimum braking force that must be exerted on the pedal, the pedal could be optimized within its design to meet this requirement while also remaining as light as possible. Using 6061 T-6 aluminum, the brake pedal was designed hanging from the top of a top brace member. This orientation is shown in the figure of the brake subassembly below.

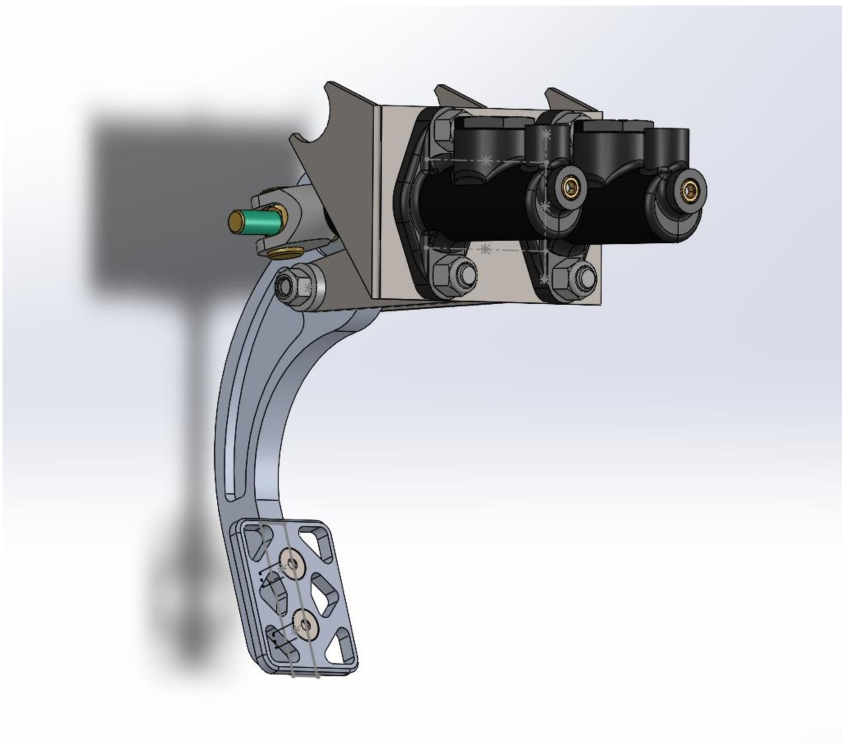

Figure 6 - Brake Assembly

This configuration of the brake system allows for less clutter on the bottom of the car which already has the gearbox and rack and pinion system by placing the master cylinders above the drivers feet. This is done by using a welded tab with the master cylinders mounted horizontally facing back towards the driver. Because of this configuration, the brake pedal must be curved to accommodate the drivers foot position. These design requirements played major factors in the overall design of the pedal.

To calculate the length of the pedal, a MATLAB script was generated to calculate braking force. Concluding that the desirable length from the pivot point was 6 inches, the pedal was designed to accommodate these dimensions. The pedal was at this time also chosen to be made from 6061 T6

Aluminum due to its high strength, low weight, and relatively low cost and ease of manufacturing. With the machine shop on campus, the pedal could be easily produced with either the CNC Haas or Tormach readily available for manufacturing.

Taking the design and placing it within SolidWorks FEA simulations, the brake pedal was tested using a hinge fixture at the pivot point and a fixed point at where the pedal interfaces with the brake balance bar. The pedal was tested at this configuration yielding a minimum factor of safety of 1.921. With the force tested being a required SAE requirement, it is vital that the pedal meets this test as to ensure that the component passes inspection at competition. With an almost double factor of safety, however, the pedal can be optimized further to remove material and ensure that it is as light as possible. The following figure shows the stress diagram produced by the FEA simulation.

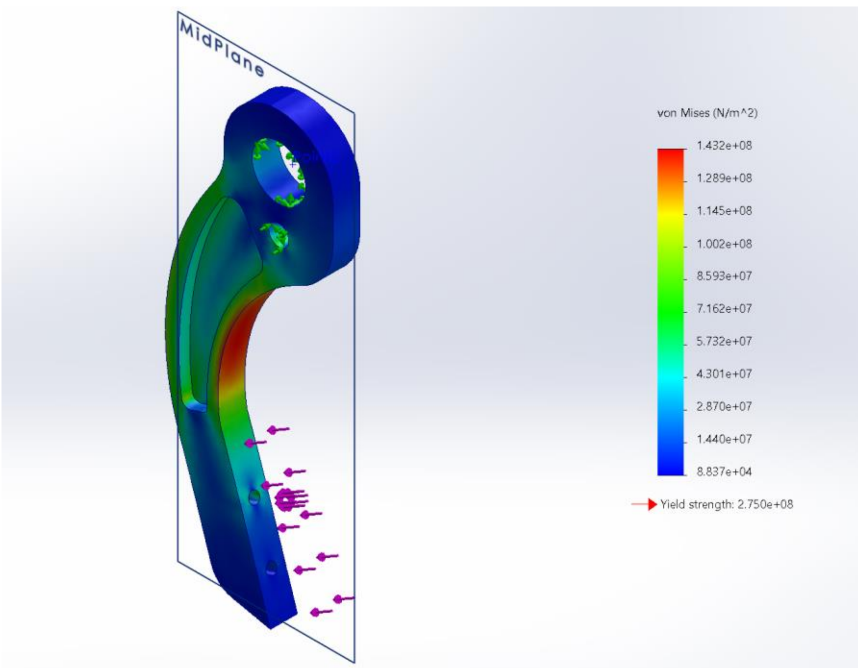

Figure 7 - FEA Stress Analysis of Brake Pedal

Looking at the results produced by the simulation, the brake pedal meets all requirements outlined by the SAE Baja competition. The simulation will be used as a tool during further optimization to see where the areas of greatest stress occur. This helps when finding which areas of the pedal can lose material without sacrificing strength. The factor of safety of the brake pedal was also produced and is visualized in the figure below.

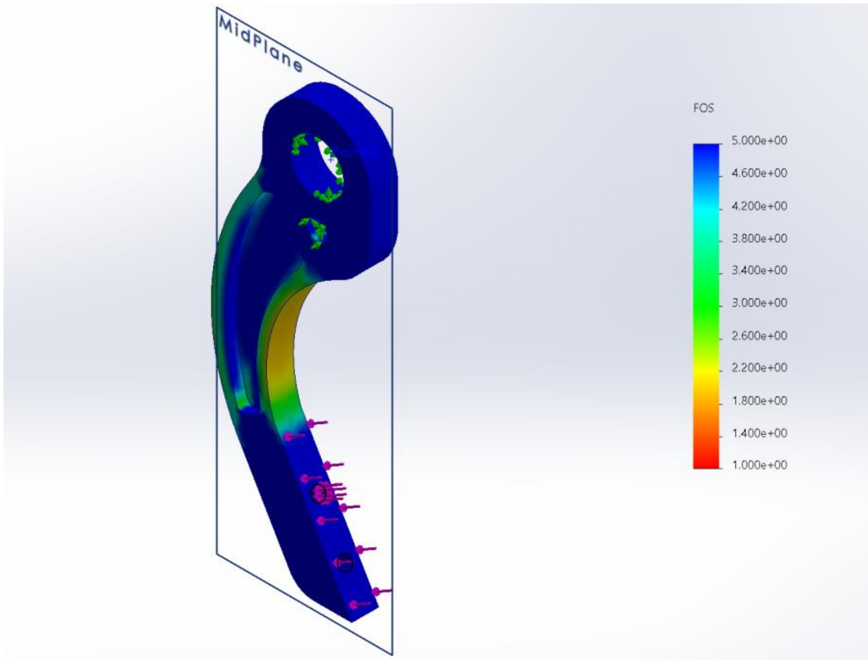

Figure 8 - Factor of Safety for Brake Pedal

The brake pedal will be revised to be lighter through loss of material. This will make sure that the pedal still meets SAE standards while also being as light as possible keeping weight down within the front end subsystem.

With all essential engineering calculations being performed above, it is appropriate to present a factor of safety table that summarizes all the major designed components and their associated safety ratings.

Table 6 - Factor of Safety for Front End Sub-System

Component	Component Description	Min FoS
Control Arm Shoulder Bolt (3/8")	Secures control arms to frame	8.28
Lower Control Arm	Articulates motion of knuckle/hub assembly	1.1 at 10 mph, 0.57 at 20 mph
Upper Control Arm	Articulates motion of knuckle/hub assembly	2.2 at 10 mph, 1.1 at 20 mph
Brake Rotor	Controls rotation of hub/wheel	1.48
Brake Pedal	Controls the Braking of the Car	1.921 with 450lbs of Pressure
Lower Control Arm Bolt	Secures control arm to the knuckle	4.8
Disk brake bolts	Secures the brake rotor to the hub	5.1
Upper Control Arm Bolt	Secures the control arm to the knuckle	2.4

This FoS table points out the relative safety that the front end has been designed with, as evidenced by most major structural components having a factor of safety comfortably over 1. There is one component, the lower control arm, that does display a factor of safety below 1 during a 20-mph collision. Though this number may be overtly concerning, it is important to contextually realize the implications of a 20-mph “dead stop” collision. In this scenario, injury will occur to the driver, tie rods, wheel, frame, etc.; some damage is expected to occur to the lower control arm in this case, especially due to its length. There are no further design revisions that can be made at this time, other than bracing, that can allow this control arm to withstand a 20-mph collision. All other parts have been revised according to insight gained during engineering calculations (increased bolt sizing, general brake pedal design, etc.) with the final goal of increasing factors of safety and decreasing manufacturing expenses. With a single part having a factor of safety below 1 because of a highly traumatic collision scenario, the design of the front end has been deemed safe and justifiably operable. There is no further need for design revisions at this point.

3.2 Rear End

Joey Barta

The strength of the upper carbon fiber camber links was analyzed during a simulated roll over where all of the weight of the vehicle is concentrated on the lower outer wall of one rear tire. In this scenario, the upper link is in tension and the lower link is in compression of a 2333lb bearing load applied from the knuckle tabs, as shown in Figure 9. The chosen super swivel rod ends allow for 55° of swivel, meaning that the knuckle can flex 25° before the rod end maxes out its travel. This analysis assumes that the rod end has maxed out and the torsional force is extended through to the insert.

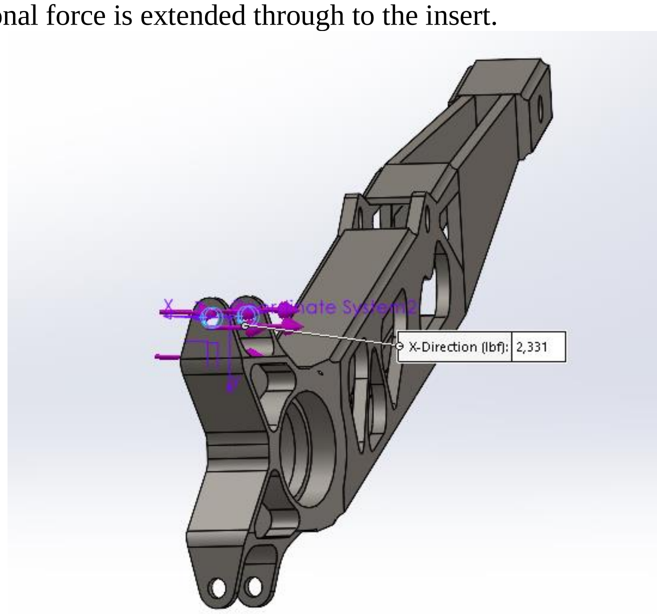

Figure 9: Applied Bearing Load

Because this is a bearing load, the upper link will experience the greatest stress in torsional shear. The rod end joins the link via a T6 aluminum insert. This insert will be knurled and glued with high strength composite epoxy to ensure it will endure the torsional forces during a crash. Under this load, the connection between the aluminum insert and the epoxy's shear strength of 7000 psi is the limiting factor of the design. With a 0.5" diameter aluminum insert, that extends 1in into the link, there is $1.5in^2$ of shear area. With this, the part has a critical force in torsion of 10995 lbs which yields a factor of safety of 1.57. Given that this calculation does not account for the friction factor of the knurling, this is acceptable. In order to increase this value, the surface area of the insert can easily be increased to 1.5".

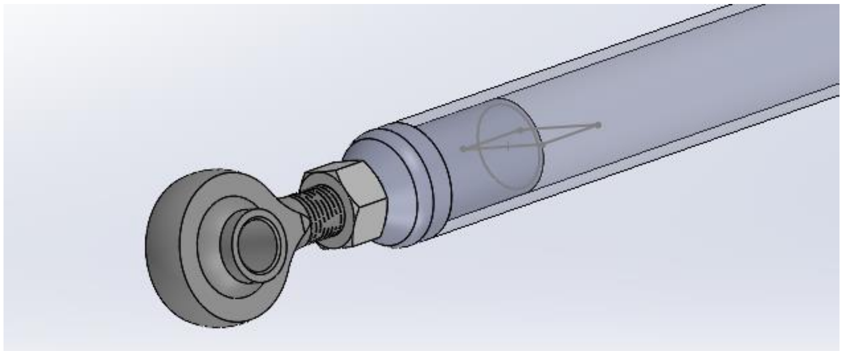

Figure 10: Carbon Fiber Camber Link

<u>*Seth DeLuca*</u>

The team decided on the worst-case scenario to be 3g force landing on one of the rear suspensions. In addition to this force the hub may be likely to experience torque from braking. Drivetrain estimates there to be a maximum of 250 lb/in of torque applied to the hub. This is the maximum load that the hub is likely to see, this scenario simulates if the vehicle were to jump of a good-sized jump landing on one side of the rear and apply the brakes before the vehicle hit the ground.

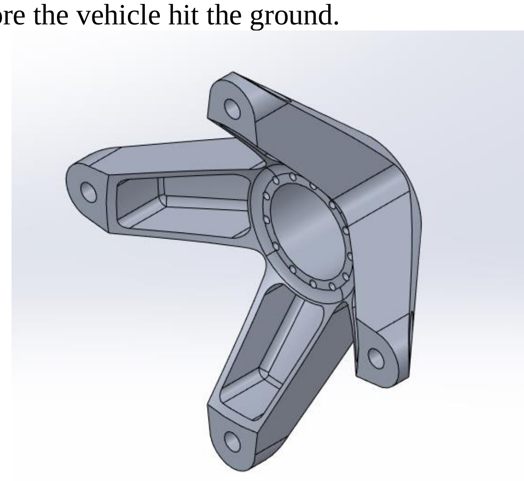

Figure 11: Hub for Rear Suspension

These forces will be applied as shown in the figure below, with the torque being applied to the center of the hub as well as a bearing load applied to the center of the spline. In addition to these a fixture was added to the four bolt holes.

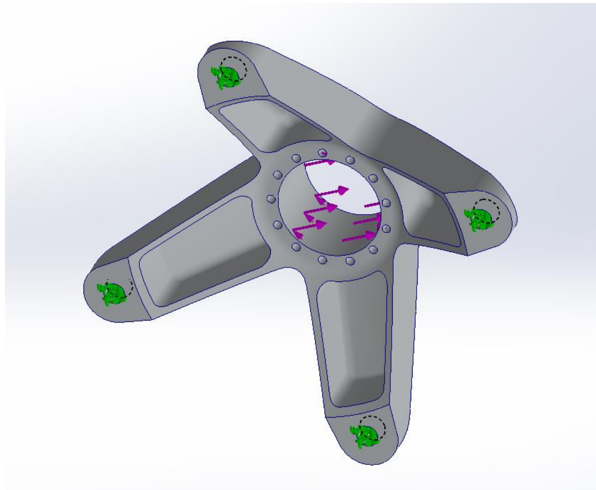

Figure 12: Forces acting on the hub under maximum conditions.

This study was conducted and reconducted until the product was at the point of being finalized. These iterations were critical to the team as the hub changed frequently to ensure enough tire spacing was available. When design this part for the vehicle, the factor of safety (FoS) goal was around two since this was a rather critical piece of the vehicles purpose. After the iterations of the hub were conducted the results for the final study are shown below.

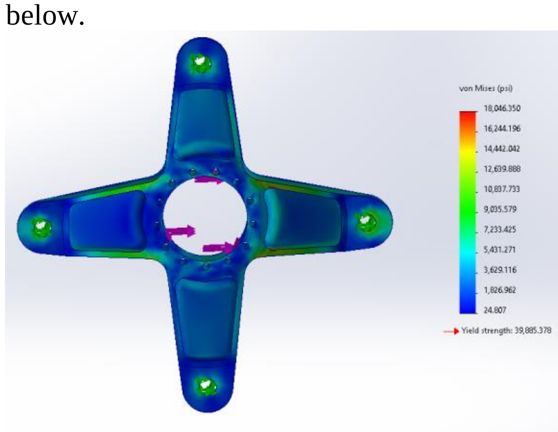

Figure 13: Results from study for the final iteration of the hub.

This image shows the critical points, in red, that will experience the most load in the hub. In this case red does not mean bad, it is great because the FoS at the red points is 2.21, this is super close to the desired FoS. This part was analyzed utilizing T6 6061 Aluminum, which is a common aluminum therefore is accessible and low cost.

<u>*Lars Jensen*</u>

The rear suspension of the baja car will be experiencing some extreme loads during the competition which needed to be considered during the design process. The entire team agreed on the assumption that the final baja car will weigh around 500 pounds giving the suspension teams an idea of loads that can be applied to the suspension system. The rear end team decided to design for a 3g force being applied through one rear wheel during complete bottom out. This would be the maximum expected loads that the baja car would experience during the competition if it landed unbalanced off a jump or large obstacle. This assumption was confirmed by a student who has been to the competitions in the past and has three years of experience with this project increasing the overall confidence of the design.

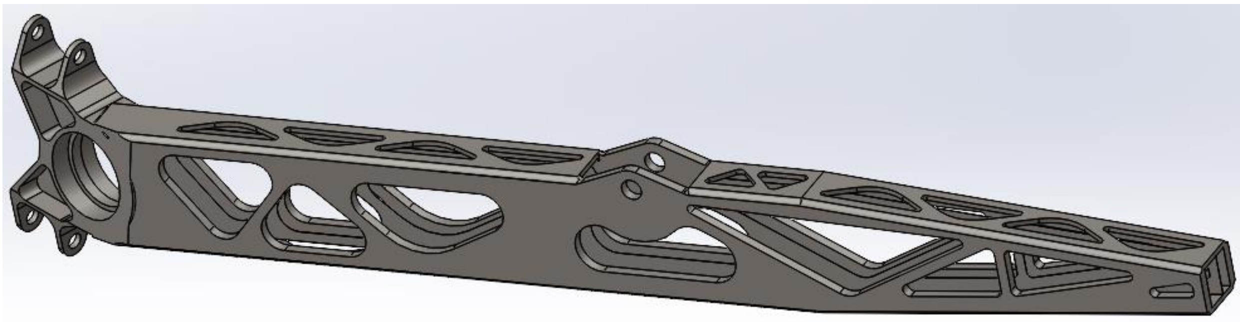

Figure 14: Trailing Link CAD Design

The 3g force is applied through the bearing carriers of the knuckle from the contact point between the bottom of the tire and the ground. This was achieved by using the remote load feature in SolidWorks and a defined coordinate system origin. With the 3g force the load applied at this point is calculated to be 1,500 lbf and is evenly distributed between the two bearing carriers. This external load is the best representation of a full bottom out scenario where all the force is from resistance with the ground. The FEA model also accommodated for the forces being applied through the camber links in the opposite direction of the load through the wheel. By analysis it was determined that the camber link pivot points on the knuckle will be experiencing 2,331 lbf loads against the force from the wheel. This was represented in the FEA model using two bearing loads through the center points of the camber link tabs on the knuckle in the opposite direction away from the frame. The shock mount point and front pivot are treated as fixed hinges to complete the FEA model and get the correct movements from the trailing link.

The minimum Factor of Safety for the trailing link shown above in Figure 14 was calculated to be 0.33 using the FEA software in SolidWorks. This model was run using ASTM A36 steel and the location of the minimum Factor of Safety was internally located in the knuckle. The main steel plate structure of the trailing link had minimum Factor of Safety values closer to one using a mixture of 1/8” and ¼” thickness plates. The knuckle is going to be CNC machined out of 4130 steel stock that the team already has at the machine shop meaning the strength will be much higher than in this model. Running the same FEA model again with an AISI 4130 Steel applied as the material produced an overall minimum Factor of Safety value of 0.62 in the same location as the first model. Changing the material did increase the strength of the knuckle and resolved this issue of failure during this load case. The locations of the minimum Factor of Safety were the same for both models and appear to be an issue with how the CAD model was created meaning it will be fixed during the production process.

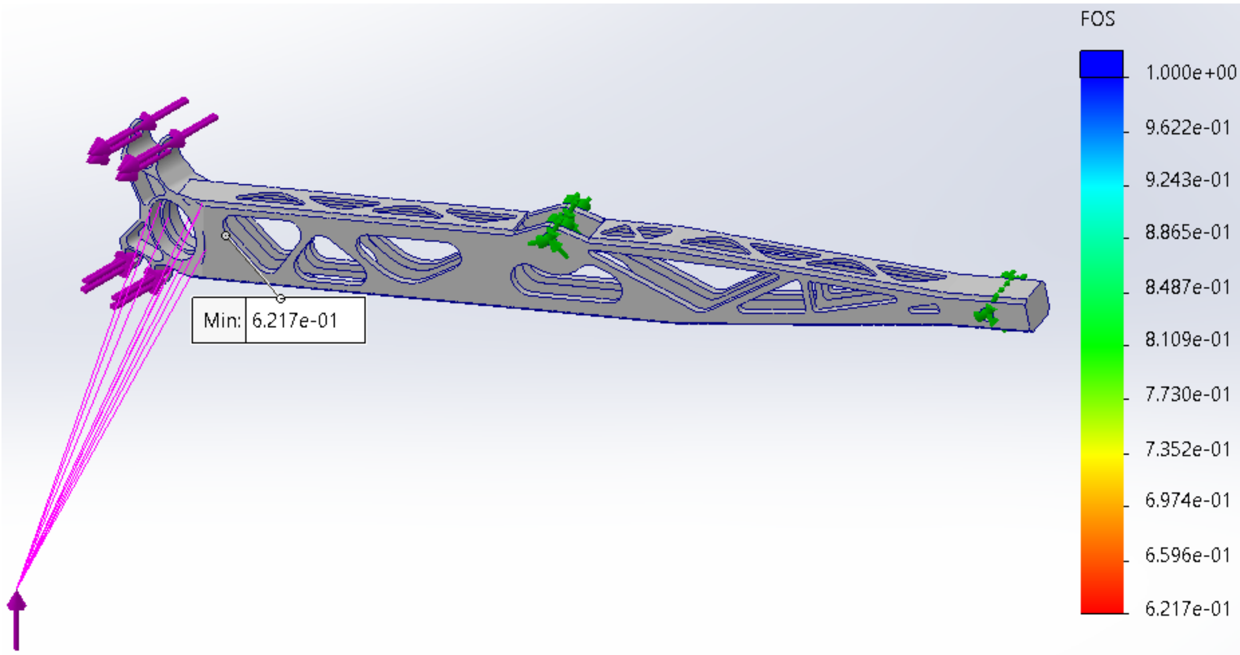

Figure 15: AISI 4130 Steel FEA Model

Applying FEA models to the design process of the rear trailing link aided in selecting materials and finding locations where the weight could be reduced. The FEA models showed that it is necessary for the knuckle to be produced from AISI 4140 steel to prevent failure during the most extreme load case that the suspension system may experience during the competition. The steel plate can be A36 and the two main structural supports that hold the shock need to be ¼" thick while the boxed in sections can be 1/8" thick. The FEA models also allowed for different variations of cutouts to be tested to find the lightest and strongest version of the trailing link for the final design.

3.3 Drivetrain

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, 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 17 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 18 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.

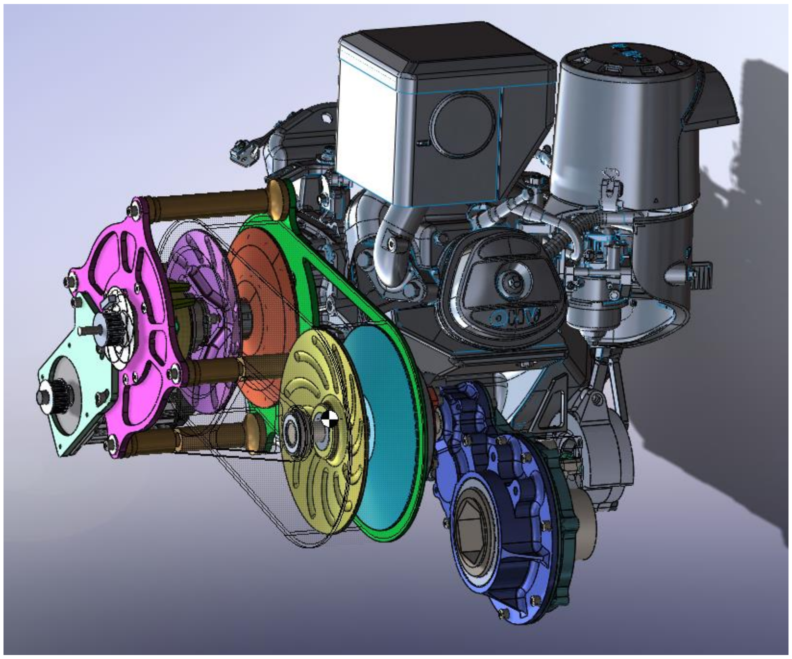

Figure 16: ECVT Motor Assembly

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 17: 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 18: Lead Screw Equations

Ryan Fitzpatrick

The rear gearbox was designed based on maximum values for torque and horsepower outputs from the engine and then through the eCVT. In order to ensure that the gearbox would be able to handle operation, all calculations and design decisions were based upon continuous gearbox operation at the maximum operational torque value. Because of this major assumption, there is a major design factor of safety already incorporated into the gearbox design, but it was still imperative to ensure that the gearbox would not fail even under these conditions. In order to be able to iterate the design continuously without having to recalculate dozens of design criteria, a MATLAB script was created that is programmed to calculate all necessary design values and output values that are useful in then calculating design FOS for each component. For the sake of brevity, the MATLAB script is not attached in the Appendix because it is over 300 lines long, but it has been submitted as a part of the last reports and presentations. The initial inputs into this MATLAB script are number of teeth in each gear, and their diametral pitch which were both previously determined to achieve the ideal reduction through the gearbox and to minimize the wear on each gear during operation. With some other inputs into the script (engine torque, engine horsepower, and eCVT ratio) the forces between the gears as well as shaft reaction forces were then determined. Using this information, and shaft layout geometry, the bearing reaction forces for each shaft were determined with sum of the forces and moments equations. With all pertinent information now calculated, the MATLAB script generated shear and bending moment diagrams for each of the shafts which were then used in shaft stress for fatigue calculations to determine minimum diameters and factors of safeties at each of the critical locations. The equations for bearing life and shaft analysis are given below along with discussion of their use. Figure 19 shows the Shigley's equation used to calculate the bearing life rating for design selection. This equation is used to calculate the catalog load rating for a bearing based on desired life and desired loading on that bearing. The bearing reaction forces calculated in the MATLAB script were input into this equation for a desired life of 40 hours of operation based upon team discussion of what the bearing needs to withstand. The resulting bearing selection for the design yielded bearings that had catalog load ratings much higher than the calculated values, which means that the selected bearings should outperform what the team needs them to do.

$$C_{10} = F_R = F_D\left(\frac{L_D}{L_R}\right)^{1/a} = F_D\left(\frac{\mathcal{L}_D n_D 60}{\mathcal{L}_R n_R 60}\right)^{1/a} \qquad (11\text{–}3)$$

Figure 19: Bearing Life Equation

Below are the equations used in the shaft stress design for infinite life. Figure 20 shows the Von Mises fluctuating stress equations used in the analysis of the gearbox. Figure 21 shows the factor of safety equation for a Modified Goodman analysis which is a conservative estimate of the factor of safety which predicts infinite life in the component. It is important here to note that for components larger than a 2" cross section, an ultimate tensile strength of 164 kpsi was used, and for components smaller than 2" cross section, 224 kpsi was used. These values come from prior research as well as conversations with the company that will be doing the team's heat treatment on the gearbox components. The important result of these equations is the FOS that this method predicts. Using this analysis, a FOS of 1.0 predicts that the component will have an infinite life. This means that the component is not predicted to fail due to fatigue in operation. Each of the gearbox components resulted in a FOS of greater than one in this analysis which means that infinite life is predicted for the operation of the gearbox. For the purpose of brevity, I will include only the main equations used from Shigley's textbook, but the complete analysis used can be found in Chapters 6, 7, and 18.

$$\sigma_a' = (\sigma_a^2 + 3\tau_a^2)^{1/2} = \left[\left(\frac{32K_f M_a}{\pi d^3}\right)^2 + 3\left(\frac{16K_{fs} T_a}{\pi d^3}\right)^2\right]^{1/2} \quad (7\text{–}5)$$

$$\sigma_m' = (\sigma_m^2 + 3\tau_m^2)^{1/2} = \left[\left(\frac{32K_f M_m}{\pi d^3}\right)^2 + 3\left(\frac{16K_{fs} T_m}{\pi d^3}\right)^2\right]^{1/2} \quad (7\text{–}6)$$

Figure 20: Fluctuating Von Mises Stress Equations

$$\frac{1}{n} = \frac{\sigma_a'}{S_e} + \frac{\sigma_m'}{S_{ut}}$$

Figure 21: Modified Goodman Factor of Safety

Jarett Berger

For the front gearbox, an FEA was conducted on the driver side gearbox casing seen in Figure 17. Due to the forces acting upon the input and output gear, there will stress acting on the gearbox casing. In order to ensure that the casing will be strong enough to withstand these forces, ribs were added on the outside of the casing where the bearings are placed to hold both input and output shafts. This will improve the stiffness of the gearbox and thus, achieving a higher factor of safety.

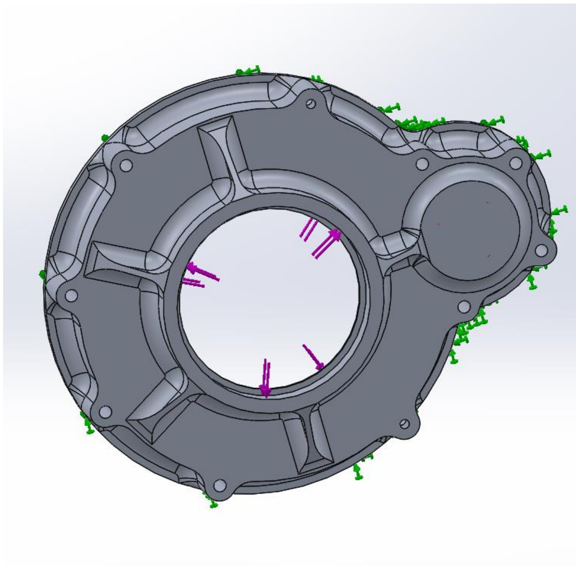

Figure 22: FEA Front Gearbox Driver Side Casing

The calculated FEA on the driver side casing was 2.08 using the material 6061-T6 aluminum. This factor of safety is adequate for this design since it is strong enough to encase both input and output gears while operating at a high torque. The benefits of using 6061-T6 aluminum will decrease the weight of the gearbox since both gears and shafts are made from 4340 annealed steel, which carries a significant amount of weight.

Donovan Parker

Most parts in the four-wheel drive subsystem take a shape relating to a cylinder and they are all rotating. Due to this, Force, Stress, and safety calculations were mostly the same and used to make sure that each part of the system interacting with each other were copacetic. Recently the factor of safety of some parts were calculated recently and can be found in the Factor of Safety tables for drivetrain. There were also precautions the drivetrain team must take regarding the Hazardous Release of Energy (HROE) specifications in the SAE Baja Rule Book.

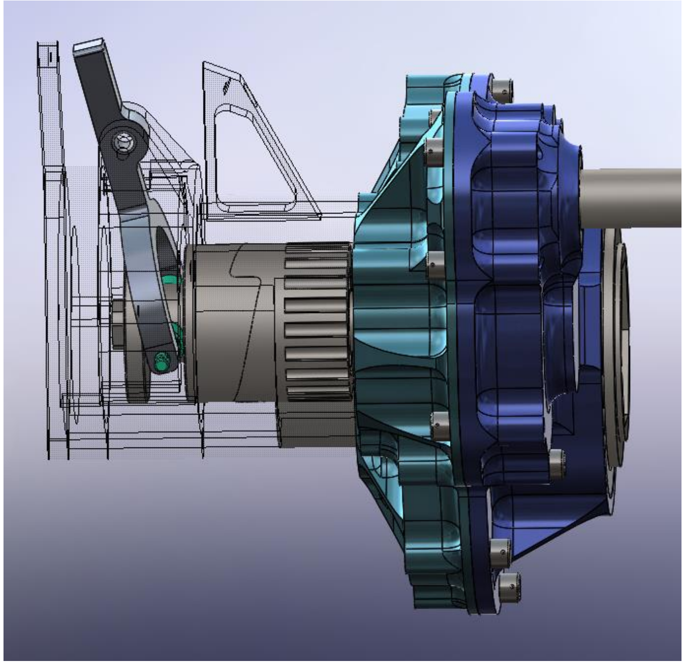

Figure 23: Rear Gearbox and Clutch Front

Below are the equations and figure 19 as an example of what each part of the four-wheel drive system went through. The equations were used in the initial calculation to determine the feasibility of the parts that were designed. For example, the parts for the clutch and the clutch teeth themselves had a factor of safety of 28.84 followed by an idling acceleration of 553.35 ft/s^2, which in turn would be the necessary acceleration for ideal engagement as well. Furthermore, the FEA ran in Figure 24 is the analysis of the front-end pulley, though it was first calculated and checked again in the simulation in SolidWorks. The factor of safety calculated for a pulley of that design made of 4140 annealed Steel was 1.2 and, in the simulation, ran in SolidWorks confirms such. The factor of safety measurement in Figure 24 is red<1<blue meaning a factor of safety of 1 and above Will show as blue and anything under 1 will show as red. The red regions do not denote failure of any kind but have a factor of safety of 0.98 or 0.99 depending on the location, however those locations will only serve as an issue if there is over-tensioning of the belt drive.

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

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

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

$$F = \frac{T}{r_i/12}$$

$$\sigma = \frac{F}{A}$$

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

$$a = \frac{F}{m}$$

Where:

F_1 = Tension side force (pounds)

F_2 = Slack side force (pounds)

T = Torque (foot · pounds)

F = Force (pounds)

σ = Normal Stress on Teeth (pounds per square inch)

A = Surface Area (inches^2)

a = Acceleration (feet per second^2)

r = Radius, d = Diameter (inches)

m = mass (pounds)

S = Material Strength (pounds per square inch)

n = Factor of Safety (unitless)

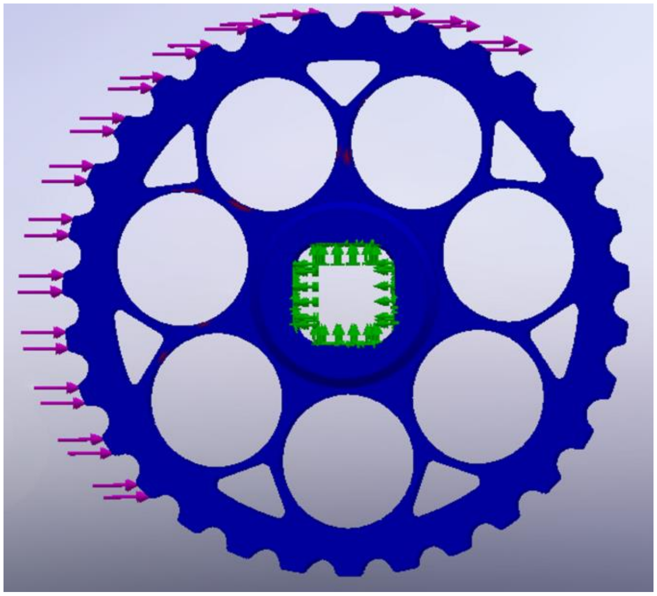

Figure 24: Front Gearbox Pulley FEA Analysis

FOS Table

Table 7: Drivetrain Factor of Safety Table

Component	Component Description	Min FoS
Font End Pulley	Engagaes Front Gear box for 4WD	0.98
Front and Rear Output Gear	Output Gear	3.07
CV Cup	Connects CV Axle from Gearbox	7.37
Driving Clutch	Connect or Disconnect 4WD Power	28.84
Driven Clutch/ Rear End Pulley	Driven Clutch Side connected to pulley	28.84
3280-8m-20 Tming Belt	Belt Transfers Power from Rear to Front Gear box	1.0+ (2000% Operational Life)
Clutch Bearing	Keeps the Driven side clutch from rotating with the saft	3.59
Front Gearbox Driver Side Case	Driver side housing for front gearbox	2.08
Front Gearbox Passenger Side Case	Passenger side housing for front gearbox	1.87
CF4/KR12 Cam Follower	Track Roller for tangental rotating objects	1.10
Primary mobile sheave	Primary sheave for ECVT	1.21 for 10^9 cycles
Primary fixed sheave	Fixed sheave for ECVT	1.21 for 10^9 cycles
Secondary mobile sheave	Secondary sheave for ECVT	.875 for 10^9 cycles
Secondary fixed sheave	Fixed sheave for ECVT	.875 for 10^9 cycles
CVT standoffs	Standoffs for CVT	1.67
CVT outside partial standoff	Outside standoff for CVT	2.53
CVT inside partial standoff	Inside standoff for CVT	1.67
Primary shaft	Shaft for ECVT	1.81
Secondary shaft	Shaft for ECVT	5<
Secondary cam	Secondary cam for CVT	3.55
Secondary cam mount plate	Secondary cam mount plate for CVT	3.39
Backplate	Backplate for CVT	5<
Frontplate	Frontplate for CVT	1.53
Motor mount plate	Plate for motor	1.21
Square bush	Bushing for CVT	5<
Gearbox shaft support	Gearbox shaft support for CVT	2.65
Anti-rotation forks	Anti-rotation forks for CVT	5<
Lead screw nut flange	Lead nut for CVT	4.18
Cast Iron Acme Round Nut	Nut for CVT	5<
316 Stainless Steel Acme Lead Screw	Screws for CVT	2.23
Rear Gearbox Input Shaft	Input Shaft into Rear Gearbox	1.18
Rear Gearbox Intermediate Shaft	Intermediate Shaft in Rear Gearbox	1.19
Rear Gearbox Output Shaft	Output Shaft in Rear Gearbox	1.5
Rear Gearbox Bearings	Bearings for Shaft Supports in Rear Gearbox	2.15
Front Gearbox Input Shaft	Input Shaft for Front Gearbox	1.35
Front Gearbox Input Gear	Input Gear for Front Gearbox	1.17
Front Gearbox Output Shaft	Output Shaft for Front Gearbox	3.21

4.0 Flowcharts and Other Diagrams

To facilitate proper sub-system design and integration, each sub-team developed flowcharts and other diagrams to illustrate the intended functionality of their sub-system. These flowcharts and diagrams for each sub-team are presented below.

4.1 Front End

Before the front end was able to be designed, the function of the car was deconstructed to allow for a deeper understanding of the subsystems that work together to ensure good function for the car. The team used this decomposition when generating design as well as geometry to ensure that the front end steering and suspension worked accordingly.

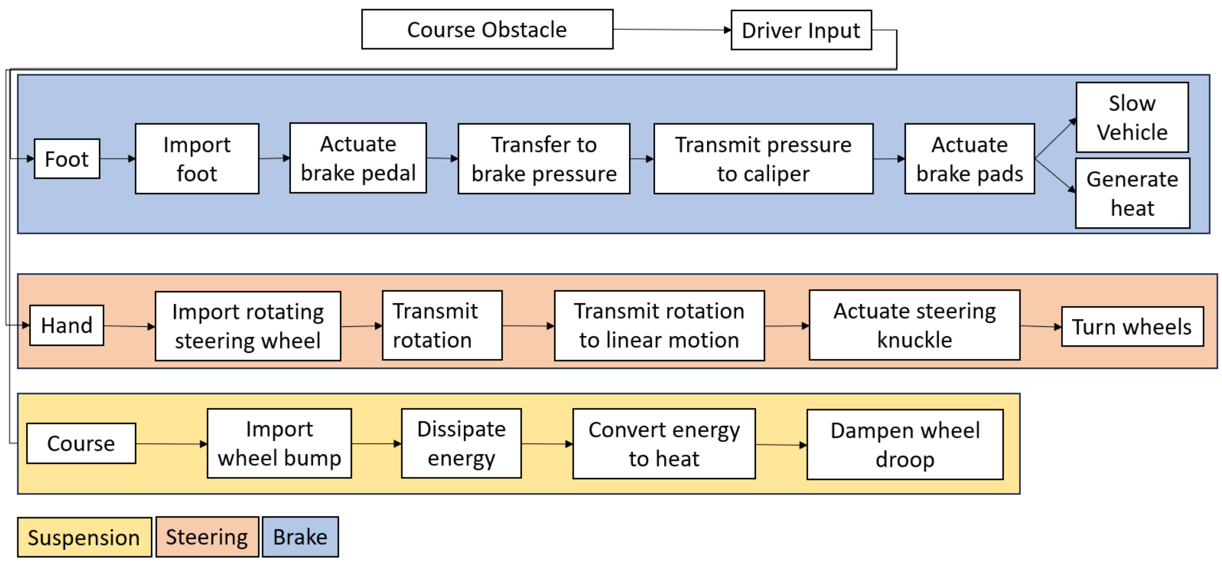

Figure 25 - Function Decomposition

Using Lotus SHARK software, the team designed a front-end suspension system that would meet the demands of the SAE Baja competition. Using the software, the team had countless trials dialing in geometry to improve steering and suspension performance. One of the important performance systems dependent on geometry is the steering. The following figure displays the percent Ackerman produced by the steering geometry. The percent Ackerman describes the cars steering behavior as the inside wheel turns at a factor more than the outside wheel. Because this is nonlinear the figure below displays the percent Ackerman of the vehicle.

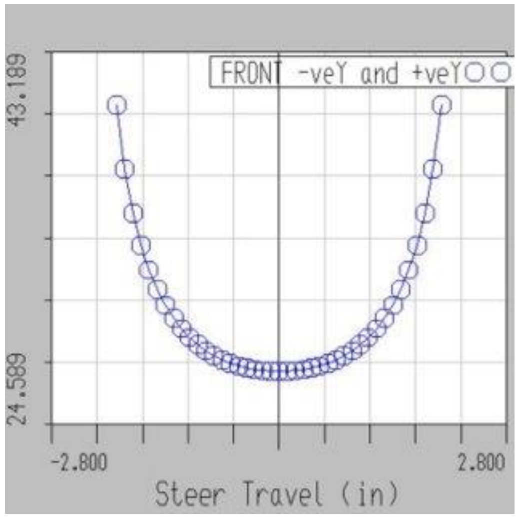

Figure 26 - Percent Ackerman

These geometric points were conveyed over to the frame sub team so that integration between the subsystems could be done during the initial design process. Initial modeling was conducted and tested using SolidWorks FEA software to serve as a proof of concept as well as to check tolerancing between parts.

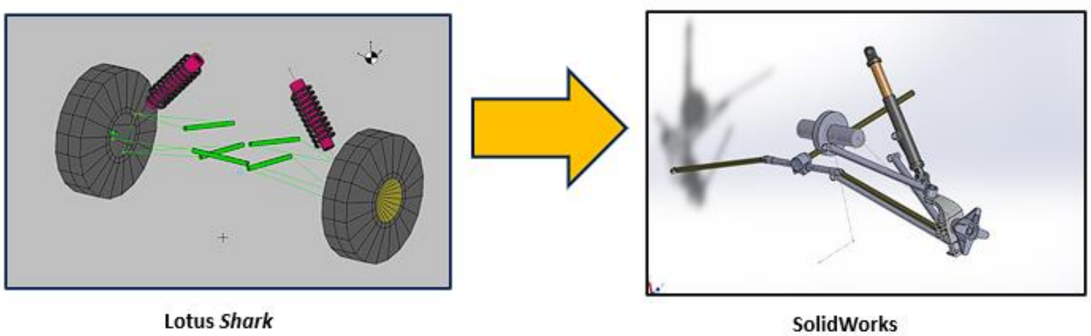

Figure 27 - Initial Modeling

Since this integration has been at the forefront of front-end design to ensure that the subsystem fits within what the frame team has designed. This was kept in mind when designing subsystems such as the brake system previously talked about and shown in Figure 6 as well as the throttle design shown in the figure below.

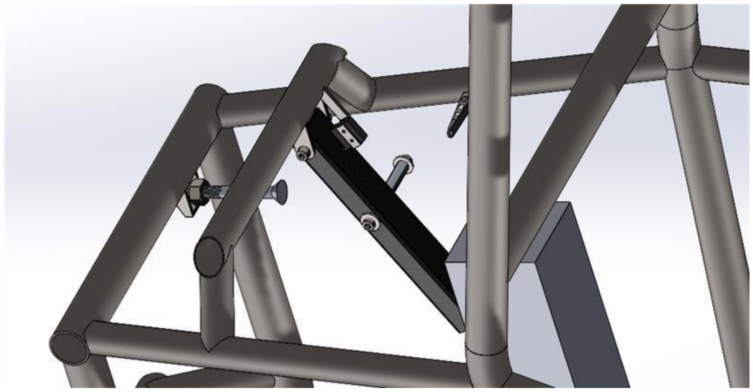

Figure 28 - Throttle Design

With the front end geometry generated in shark translated into SolidWorks, the frame, drivetrain, and front end was constructed in CAD. Using CAD allows for integration before parts are produced allowing for the saving of money and proper hardware and tabs can be chosen. The use of CAD software also checks tolerances ensuring that the front end gets full range of motion without interference between parts.

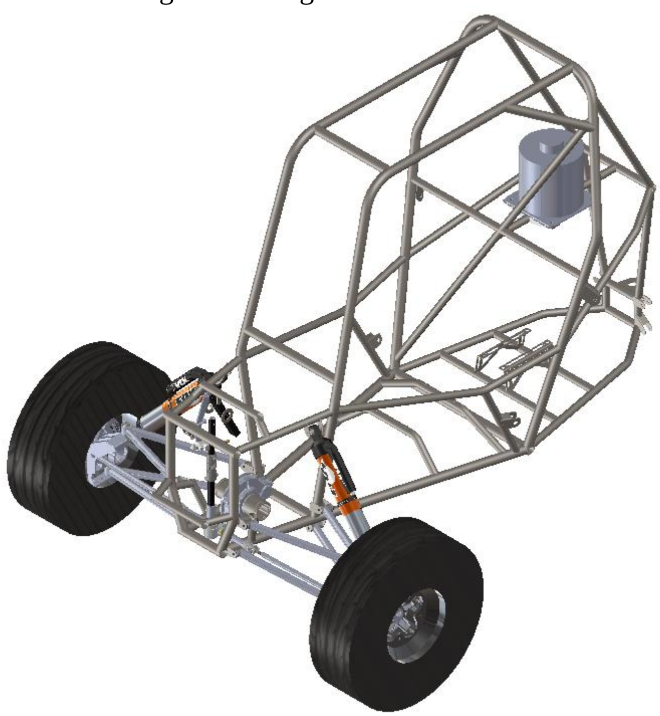

Figure 29 - Front End Assembly

4.2 Rear End

During the first semester the Rear Sub-team decided upon a geometry to base the design off moving forward. This was using the CAD software called Lotus SHARK, a suspension analysis software. The greatest part of the design process was when the team finally got the design to a point where the team can compare the design (Figure 30) to the SHARK geometry based off (Figure 31).

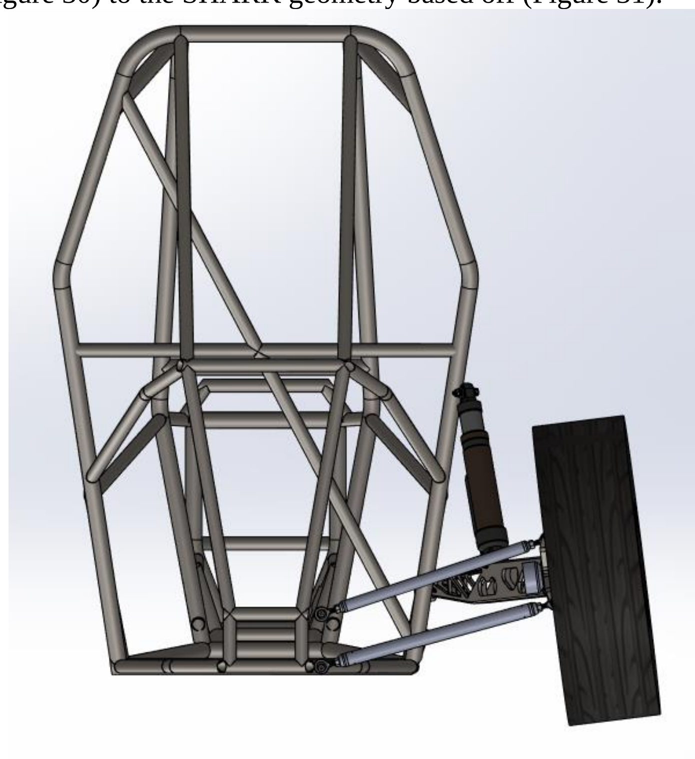

Figure 30: Rear suspension design at full compression

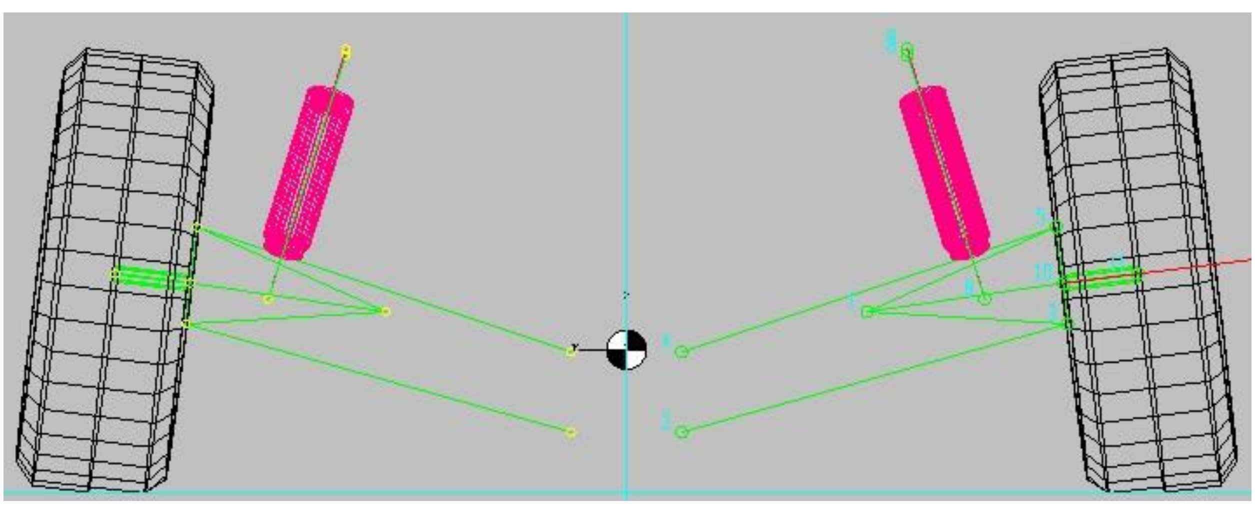

Figure 31: Rear suspension geometry at full compression from first semester

As one can see the camber angle at full compression in the design is very similar to the shark software. The toe angle may even be better at full compression.

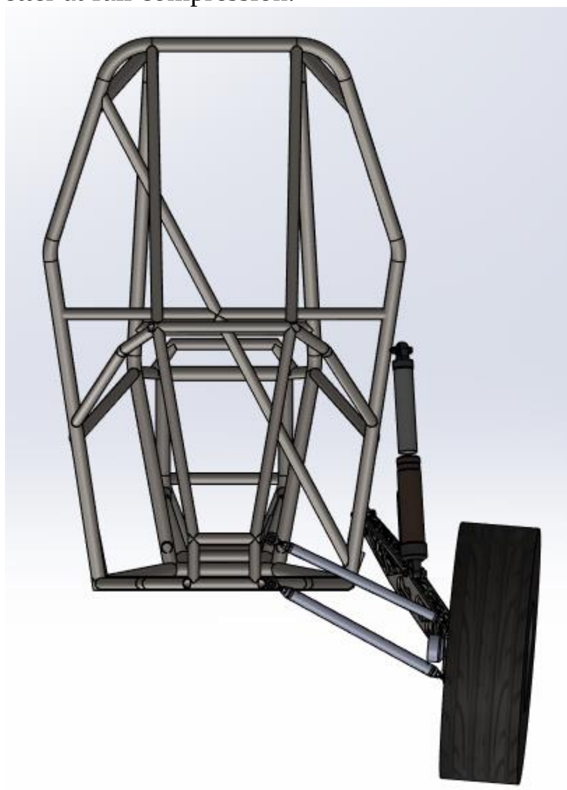

Figure 32: Rear suspension design at full droop.

The Shark software had a positive angle under full rebound which we have a very similar angle to the design. These suspension analyses are extremely helpful in the design process of the vehicle. It allows for the team to determine a bunch of design requirements and use those in parallel to the design of the vehicle. This software aids the sub-team in creating a proper design for the vehicles and team requirements.

4.3 Drivetrain

The CAD models of the drivetrain systems can be seen below in Figure 33, Figure 34, and Figure 35. In the system, power is transferred from the engine through the electronically controlled CVT and into the rear gearbox. The rear gearbox supplies power to the rear wheels and is attached to the four-wheel drive clutch and timing belt system which, when engaged by the operator, supplies power to the front gearbox, thus supplying power to all four of the wheels. This timing belt configuration can be seen below in Figure 34. Each subsystem had FEA individually ran on it to simulate a subsystems interaction with another subsystem. The overall analysis was deemed successful, and subsystem testing after manufacture is the next stage of the design.

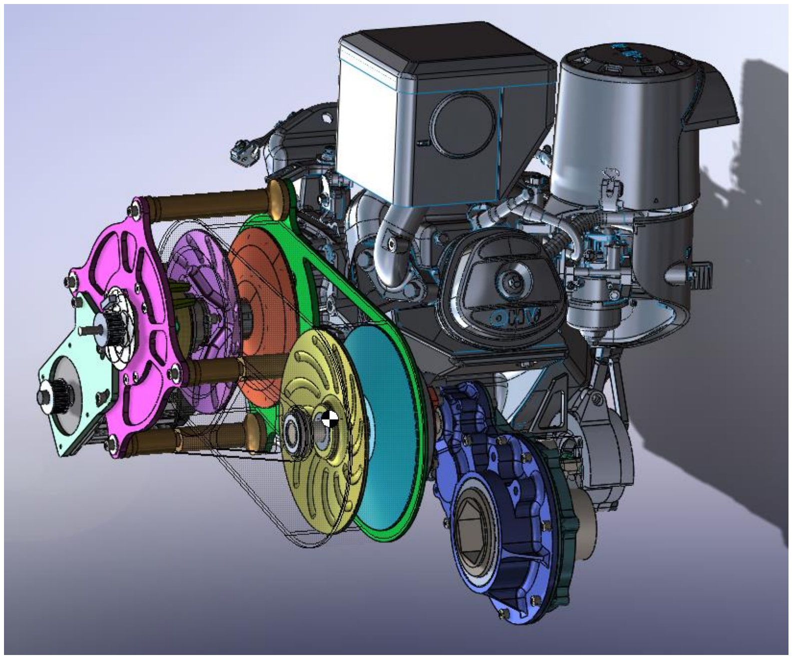

Figure 33: CVT, Motor, and Rear Gearbox Assembly

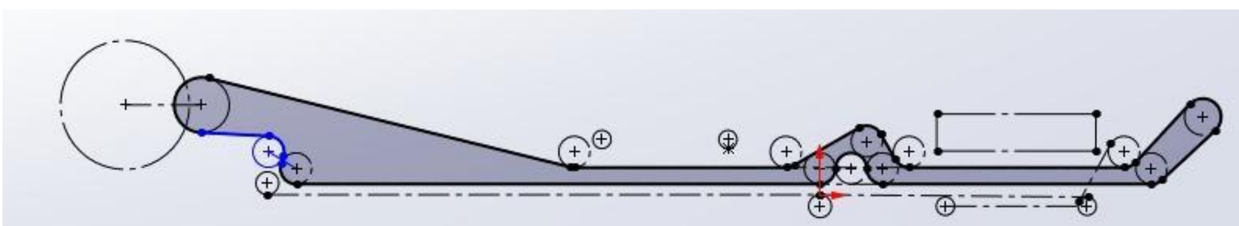

Figure 34: Belt Routing Under the Vehicle

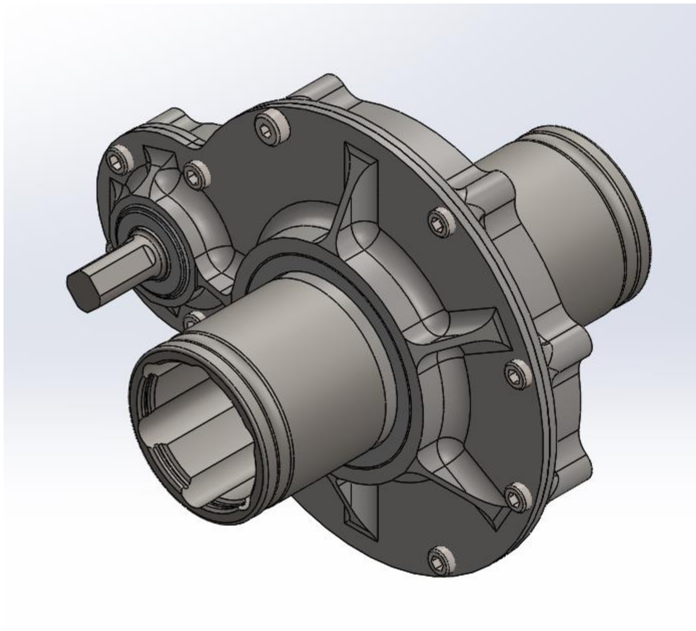

Figure 35: Front Gearbox Assembly

5.0 Moving Forward

As the semester progresses, the team will fully shift its attention towards manufacturing the vehicle's components; however, some small design/premanufacturing tasks will need to be completed before all parts are toleranced and ready to be machined/purchased. These design tasks and manufacturing preparation steps are outlined below for each sub-team.

5.1 Front End

Moving forward the front-end will need to complete the final calculations on the mounting tabs for the steering rack mechanism to ensure the tabs will not fail in the event of a severe impact. At this point all other relevant calculations to the front end have been completed and with calculated minimum factor of safety values above values set out in the initial design steps, all completed components have been cleared mechanically and were ready for design review.

A design review has been performed on the knuckle, hub, brake system, and control arms to ensure a sound design and ease of manufacturability. All listed components have undergone interference checks and have tolerances associated with all machined dimensions. With this review being completed the components above have been cleared to move into the production phase ahead of schedule.

A final design review needs to be completed on the steering assemblies to cover tolerancing of machined parts, checking for mechanical interferences with moving parts such as between the knuckle and tie rods at full bottom out and full rebound, and validating the maximum angle the CV axle can operate at. After this design review is completed and all models have been updated to the corrected standards, all front end design will be completed.

5.2 Rear End

The final design calculations have been completed for the rear end sub-team and the system is ready to be produced. The manufactured and purchased components have been checked against multiple load cases in FEA to ensure that they do not fail during the competition. The system has also been checked for clearances as the shock is cycled through its travel using the final CAD assembly in SolidWorks.

The rear end sub-team is on track to get the rear suspension system manufactured and tuned before the competition. The team has started producing components for the rear hub assembly and is about ready to join the aluminum hub with the steel cut out splines that attach to the CV axle. The production of steel camber links has also started with the rod ends fixtures being turned in the lathe at the machine shop. The knuckle has been finalized and sent to the machine shop for CAM development and production.

The next steps of the process will be sending the correct steel plate files to the laser cut sponsor who will get the multiple components cut out and ready for welding. With the plates back in the machine shop, the rear end team will be able to fit all the components in a jig and complete the welding of the trailing links. Another big step that needs to be completed soon is making an order for all the necessary hardware that will be used in the rear suspension system. The hardware will allow the team to fully assemble all of the individual components that we have manufactured into one smoothly operating system.

5.3 Drivetrain

The drivetrain sub-team has completed all necessary calculations for each drivetrain subsystem. The factor of safety has been calculated for each component and this will ensure that each subsystem component will not fail during operation. At this point, all components have been thoroughly analyzed and are ready for the next process of the design review.

For the design review, each machined component for both front and rear gearboxes and the pulleys, shifter, and shifter housing will need to have the correct tolerances. This will ensure proper fitment when assembled. As for the machined CV Cups, there will be more testing needed to ensure the correct fitment. Completing this design review, each component will be ready for the final design review.

The final design review will overlook any imperfections of the front and rear gearbox components and 4-Wheel Drive System components. Validating the correct fitment and tolerances will be necessary before it gets machined. After clearing this design review, all components for the drivetrain subsystem will be set for machining and no further design will be required.