SAE NAU Mini Baja

Background Report

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DISCLAIMER

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1 BACKGROUND

1.1 Introduction

The Society of Automotive Engineers (SAE) is a U.S. based international organization that develops standards for various engineering practices. These engineering practices include manufacturing, design, testing, maintenance, etc. Included in SAE is a collegiate design series (CDS) that challenges students to engineer, fabricate, and compete their solution to a given engineering problem. For the 2016-17 school year, NAU SAE and the Mini Baja senior design team are working together to develop a vehicle to compete in SAE's Mini Baja competition in Gorman, CA on April 27th, 2017.

The senior design team will engineer, fabricate, and compete a competitive entry for the competition that satisfies the rules and regulations set by SAE International for the design series. This entry will be a brand new design, with no original system. The ultimate goal of this team is to compete in and finish all events in the Gorman, CA competition without failure. The importance of completing this goal is to reestablish NAU as a competitive entry in the SAE Mini Baja competition. In the last 3 years, NAU has failed to compete and finish all of the competition events. The 2016-17 NAU SAE Mini Baja team will be representing NAU's mechanical engineering department as well as NAU's SAE chapter.

This report will outline the project description and details as well as benchmarking and existing design concepts that have competed in previous Mini Baja competitions.

1.2 Project Description

The project description is outlined in the SAE 2017 Mini Baja Rules document as:

> *"Baja SAE® is an intercollegiate engineering design competition for undergraduate and graduate engineering students. The object of the competition is to simulate real-world engineering design projects and their related challenges. Each team is competing to have its design accepted for manufacture by a fictitious firm. The students must function as a team to design,* ***engineer****, build, test, promote, and compete with a vehicle within the limits of the rules. They must also generate financial support for their project and manage their educational priorities [1]."*

The rules document also provides all limitations and regulations that each design must abide by in order to compete in the competition. For details on these rules, please reference section 2 of this document.

2 REQUIREMENTS

This section will cover all of the parameters that the design will have to satisfy. The parameters are determined by Customer Requirements, Engineering Requirements, and their design links.

2.1 Customer Requirements (CRs)

Table 1: Customer Requirements

Customer Requirement	Description	Weighting	Relative Weight
Safety	The number one aspect of the competition is to have a vehicle that will keep the driver, and spectators safe	10	14%
Quick Acceleration	The vehicle must be able to reach higher speeds as soon as possible from a dead stop in order to make ground in the endurance competition	6	8%
Climbing Capabilities	The competition includes a hill climb event that vehicles must overcome	6	8%
Towing Capabilities	Each vehicle is required to have a tow point in the front and rear. In addition to this, there will be a tractor pull event that will utilize towing capabilities	6	8%
Durability	The vehicle will be cycled through harsh conditions throughout the competition and must be able to stay durable through each event	10	14%
Easy Maintenance	During competition the vehicle will need to be maintained and inspected	5	7%
Speed	In order to gain ground during the endurance race, the vehicle must overcome the speed of it's competitor's	4	6%
Reverse Capabilities	The winning-most vehicles throughout competitions have always had reverse capabilities	2	3%
Economical	The vehicle is being produced as a prototype for 4000 units per year	7	10%
Manufacturability	Part of the event must include ease of manufacturing for 4000 units to be mass produced per year	6	8%
Multi-Terrain Capable	The competition is on all terrains	10	14%

2.2 *Engineering Requirements (ERs)*

2.2.1 Drivetrain ERs

The list of ERs and their descriptions can be seen in Table 2. The justifications and target values for each ER is based on given parameters from the SAE Rules and Regulations, as well as weight and power requirements that stemmed from benchmarking the project.

Table 2: Drivetrain ERs

Engineering Requirement	Description	Target Value	Tolerance	Target Value Justification
Weight (Lb.)	Measuring the mass of the system	50	+/- 5	Must not exceed 10% of vehicles gross weight
Power Transmitted (Hp)	Measuring the horsepower outputed by the system	20	+/- 5	Must double engine Hp to produce required speed
Torque Transmitted (Lb-in.)	Measuring the torque outputed by the system	150	+/- 25	Must double engine torque to climb required terrian
Velocity (MPH)	Measuring the maximum velocity of the system	35	+/- 5	Must be competitive with previous winning teams
Engaed/ Disengage Time (s)	Measuring the time required for the system to start or stop	3	+/- 1	Must respond to users input in a timely manner
Material Strength (GSI)	Measuring the strength of the material of the system	30	+/- 3	Must withstand stresses of competition environment
Assembly Temp (F)	Measuring the heat increased in the system	180	+/- 20	Must not damage itself or subsystems around it
Cost (U.S. Dollars)	Measuring the cost of the parts within the system	800	+/- 50	Must not cost more than 13% of team's budget
Volume (Ft.^3)	Measuring the volume in the system	2	+/- 25	Must fit within frame
Manufacturing Time (hr)	Measuring the time required to manufacture the system	24	+/- 5	Must be able to be built within one semester's time
Gear Change Force (Lb.)	Measuring the force to change gears within the system	5	+/- 2	Must not exceed maximum instantaneous stress of gear
Lifetime Expectancy (Cycles)	Measuring the excepted cycles of the system before failure	1 Million	+/- 0.25%	Must meet AGMA standards for gearing life

2.2.2 Frame ERs

The ERs for the Frame must envelop all of the hoop and geometrical requirements determined by the SAE Rules and Regulations. Most of the frame's target values are already set by the rules and regulations, and as a result cannot be altered. Some of the ERs were unable to have a determined target value due to there being such a wide variety in that design parameter; therefore, a target value cannot be set until benchmarking for optimal parameters has been done. The Frame ERs can be seen in Table 3.

Table 3: Frame ERs

Engineering Requirement	Description	Target Value	Tolerance	Target Value Justification
Driver to Frame Clearance (inches)	Distance from driver to nearest cage member	6		Baja SAE Rules
Member Material (lb*in & lb*in²)	Bending Strength and Stiffness	3500 & 1000000		Baja SAE Rules
Straight Member Length (inches)	Longest length of any member	<40		Baja SAE Rules
Bent Members (degrees)	largest bend of member not at node	30		Baja SAE Rules
Primary Members (inches)	Total length of members			Strength/Weight optimization
Secondary Members (inches)	Total length of members			Strength/Weight optimization
RRH Minimum Lateral (inches)	Rear Roll Hoop minimum width	28		Baja SAE Rules
RHO/RRH Gusseting (inches)	Gusset regulations	5		Baja SAE Rules
$C_{R/L} \geq 12''$ from Seat Midpoint	Distance from Seat midpoint to point C			Baja SAE Rules
$C_{R/L}$ 41" above seat	Distance from Seat bottom to point C			Baja SAE Rules
SIM Distance from Seat Bottom (inches)	Distance between SIM and LFS	11	+/- 3	Baja SAE Rules
Welding (type)	Mig or TIG	Mig		Ease and time required
FDM & Vertical ≤ 45°	Max angle between FDM and Vertical			Baja SAE Rules
weight (lbs.)	Final weight of frame	70	+/- 5	speed/acceleration optimization
FAB ≤ 5" of B & ≤ 2" of A and S	FAB distance from points on RRH			Baja SAE Rules
Tube Dimensions (inches)	Diameter and thickness of tube			Strength/Weight optimization
Cost (Dollars)	Total cost of roll cage	150	+/- 50	Limited budget

2.2.3 Suspension ERs

Suspension has a wide range of ERs. One reason behind this is that there are specific ERs for both front and rear suspension alone, let alone ERs for both of the systems. The Justification for the target values are based on general engineering practices for certain types of suspension, as well as benchmarking based on what teams have done successfully in the past. The full list of Suspension ERs can be seen in Table 4.

Table 4: Suspension ERs

Engineering Requirement	Description	Target Value	Tolerance	Target Value Justification
Vertical Wheel Travel, Front (in)	Measured bump to full droop	10 inches	no less than 8"	Must have sufficient travel for rough terrain
Vertical Wheel Travel, Rear (in)	Measured bump to full droop	10 inches	no less than 8"	Must have sufficient travel for rough terrain
Travel Ratio, Front (up/total) (%)	Measured at ride height	60%	+/- 5%	To keep front end neutral in rough sections
Travel Ratio, Rear (up/total) (%)	Measured at ride height	75%	+/- 5%	For ground clearance and large impacts
Track Width (in)	Limited to 64" by rules	58 inches	no more than 64"	Narrow as suspension travel allows.
Wheelbase (in)	Ratio of 1.2 to 1.4 for WB:TW	75 inches	5 inches	Keep car short for quick turning
Caster Change	Target is desired value at ride height	7 deg	+/- 7*	Not extremely important, but change minimized
Camber Change	Target is desired value at ride height	+/- 2 deg	+/- 15*	Negative camber in bump desirable (body roll)
Toe (in)	Measured at ride height	0	+/- 1"	Can improve handling, adjustability desired
Bump Steer (deg)	Over full wheel travel	0	minimum	Design goal - 0 bump steer through travel
Steering Ratio (deg/deg)	Steering wheel input : wheel turn	6:01	4:1 to 10:1	Arm restraints require quick ratios
Scrub Radius (in)	Fixed by spindle choice, wheel choice	min		Effects steering effort/feedback, use offset to change
Height, Roll Center, Rear (in)	Only static roll center considered	higher than front		To create oversteer effect
Height, Roll Center, Front (in)	Only static roll center considered	min		For handling/dynamic considerations
Height, COG (in)	Estimate using CAD software	min		Design by mounting components lower
Roll Stiffness (lb-ft/rad)	Requires COG to analyze	unknown		Can be modified after completion if an issue
Drive Axle Plunge (in)	axial deflection of drive axle in travel	0	0"	Design goal - zero plunge through travel
Drive Axle Max Angle (deg)	Max angle of u-joint / CV joint	unknown	0*	Max operating angle, requires component selection
Total Unsprung Weight (lb)	Weight of wheel end and half of links	min		Keep to a minimum for weight and strength reasons

2.3 Testing Procedures (TPs)

2.3.1 Drivetrain TPs

1. **Weighing** – The overall Drivetrain package will be weighed, in order to determine the system's impact on the overall weight of the vehicle. The weight also helps estimate the cost, manufacturability, and peak velocity of the system.
2. **Dynamometer Testing** – The dynamometer will allow the team to measure the exact outputs to the wheels from the engine. This gives the team horsepower and torque outputs. In addition to this, the team can also measure the system's peak velocity. By utilizing a dynamometer, the team can tune the drivetrain package to the most efficient setup possible.
3. **Force Tests** – Force testing will involve measuring forces put onto the mounts until failure for the drivetrain packaging as well as the shifting forces required to change gears via a strain gauge. This will allow the team to optimize the shifter arm to minimize the required force to change gears, as well as set a goal for design on the drivetrain mounting points.

2.3.2 Frame TPs

1. **Finite Element Analysis (FEA)** – Due to the limitation of only having one frame to test, the team cannot perform physical impact or rollover tests on an actual built frame. To test against these situations, the team will perform iterations of a FEA on the frame to conservatively test the frame's strength.
2. **Exit Time** – The team will use a stopwatch to calculate exit times for the selected drivers. The test

will determine whether or not the driver can exit the cockpit within a window of 5 seconds as designated by the competition.

3. **Welding Penetration Depth** – Quality of the welds will be checked by SAE officials for frame construction to be certified. Welding penetration depth will be checked by cutting the member in half and inspecting that the weld fully penetrated through the entirety of the tube.
4. **Failure Testing and Analysis** – In order to prepare for the SAE Mini Baja competition's technical inspection, the team will replicate the welding coupon tests. A matching coupon will be made for each of the two tests and the team will subject the first to a tensile force enough to break it and then observe whether the break occurred in the weld or the tube. The second coupon will be sectioned along the length of the tube to test whether or not there was adequate welding penetration.

2.3.3 Suspension TPs

1. **Cycling Suspension** – This process consists of removing the shocks from the car, releasing all the air from the shocks, and reattaching them to the vehicle. In turn, this will allow the suspension to move easily with little force applied to the system. While cycling, measurements will be taken using a tape measure from the machine shop to calculate the vertical wheel travel, travel ratio, caster change, camber change, bump steer, drive axle plunge, and drive axle maximum angle.
2. **Weighing** – To determine the height of roll center, height of center of gravity, unsprung weight, and roll stiffness, it will require four scales supplied by the NAU SAE club. Each scale will be placed under a tire, resulting in all wheels being off the ground equally in height. The weight will be measured in each area and then the process will be repeated on a slight measurable incline.
3. **Static Measurements** – The remaining engineering requirements: track width, wheel base, toe, steering ratio, and scrub radius; can all be tested by taking simple measurements with a tape measure while the vehicle is fully prepared and resting normally on the ground.

2.4 Design Links (DLs)

2.4.1 Drivetrain DLs

Weight – The total mass of the engine mount, transaxle, CVT, and mounting frame is estimated as 46.8 pounds, which is around the weight goal for the system.

Power Transmitted – From the individual analysis, the power output is constant through the system. Therefore, the power to the wheels is still 10hp. Although this does not meet the ER target value, the system chosen is unable to increase the hp output to the wheels, which is okay for the system operation.

Torque Transmitted – The projected torque output to the wheels using the Comet 780 Series CVT coupled with a 10.15:1 Dana H-12 FNR Transaxle is 602.68 lb-ft. This exceeds the minimum requirement for the system.

Velocity – The estimated velocity from the individual analyses with the system mentioned above is 35.01MPH which exceeds the target value.

Engage/Disengage Time – The spec'd engage and disengage time from Schafer Industries is rated as 2.5s, which satisfies the target value.

Material Strength – The material to be used for the shifter (6061 Al) exceeds the amount of force required by Schafer Industries to change the gears of the transaxle.

Assembly Temperature – The provided lubricant for the transaxle, coupled with air cooled flanges keeps the temperature of the system within range of the Trumpler Requirements of less than 250 degrees.

Cost – The quote from Schafer Industries for a transaxle is $2400, which sets the system exponentially higher than the cost goal; however, Quality Drive Systems (QDS) sponsors SAE Mini Baja competitions, and gives a discount for this transaxle, which brings the cost down to \$1295 for the transaxle. The CVT, framing components, and mounting components are projected to be a total of around \$500. Therefore, the cost for the drivetrain sub-assembly is roughly estimated as \$1800. This is okay, due to an underestimate of the initial budget available from the NAU SAE Chapter.

Volume – By utilizing the engine mount, CVT, and transaxle onto a packaged frame, the volume is approximately equal to 2 cubic feet, which meets the ER target value.

Manufacturing Time – Because the only in-house fabricated items are the mounting plates and frame, the manufacturing time is minimized to below the target value of 24hrs.

Gear Change Force – The force to change gears as spec'd by Schafer Industries is 3 pounds, which satisfies the projected target value for the system.

Lifetime Expectancy – All of the components integrated into the assembly have a minimum catalog life of 10^9 cycles, which exceeds the target values.

2.4.2 Frame DLs

Driver to Frame Clearance – Research on the sitting height of a 95th percentile male yielded an average height of 38.2-inches. The current frame is designed to be 46-inches tall giving a buffer of roughly 8-inches from the top of the driver's head. The average shoulder width was found to be 20-inches and the current frame is designed to be 28-inches wide at shoulder height offering another 8-inch buffer. Each buffer fulfills the requirement to be greater than or equal to a 6-inch buffer.

Member Material – All primary members must be at least 1-inch diameter and 0.062-inches thick, while meeting a bending stiffness and strength of 1-inch 0.120-inch wall AISI 1018 steel. The current primary member choice of 1.375-inch diameter 0.065-inch wall DOM tube exceeds those regulations. The secondary members must be at least 1-inch diameter and 0.035-inch wall tube. The team will be using these secondary member dimensions in DOM tube.

Straight Member – Currently, all of the frame's straight members fall within tolerance of the 40-inch limit with the exception of the roll cage overhead members which are 43 in long. Therefore, the team will add a gusset member or change the angle of the FAB front members.

Bent Members – All of the bends occur at frame nodes with an exception of the bend in the SIM members which is 30°, thus meeting the design requirements.

Primary Members – Based on the template given by the SAE Baja Rules, the team has all the required nodes and primary members for the frame (Color coded red in Figure RC1 of the SAE Rules and Regulations).

Secondary Members – Based on the template given by the SAE Baja Rules, the team has all the required nodes and secondary members for the frame (Color coded green, yellow, blue, and orange in Figure RC1 of the Rules and Regulations).

RRH Minimum Lateral – The minimum lateral measurement of the current frame is 29.7-inches, which fulfills the requirement of at least 28-inches.

RHO/RRH Gusseting – Currently, the frame has no incorporated gussets, therefore this requirement is not a factor for the design.

C_R/L ≥ 12" from the Seat Midpoint – With the seat midpoint estimated at 10-inches from the bottom of the RRH, the frame currently has a 16-inch buffer to points on the C_R/L, which satisfies the 12-inch requirement.

C_R/L ≥ 41" from the Seat Bottom – The seat bottom is estimated to be between 1 to 2-inches above the bottom of the frame, the points for C_R/L currently sit 44-inches from the seat bottom of the frame, which meets the 41-inch requirement.

SIM Distance from Seat Bottom – The SIM members on the frame are located 10-inches above the seat bottom, thus meeting the requirement of being at a distance of 8 to 14-inches.

Welding – Due to the material selection and manufacturability requirements, the team will be using a MIG welding technique to assemble the frame.

FDM & Vertical ≤ 45° – The FDM member of the frame makes a 35° angle with the vertical member, which satisfies the 45° requirement.

Weight – Analysis of the CAD model created in SolidWorks estimates the frame to weigh 76.1 pounds. This estimate does not consider the added weight from welding; however, the target goal was conservative based on the top performing teams from previous competitions, so while the team hopes to stay as close to the target goal as possible, having a little extra weight is not of high concern.

FAB ≤ 5" of B & ≤ 2" of A/S – The frame has been built to have the FAB members in both the front and rear of the car mesh into the RRH precisely on top of points B, A, and S so the range requirements are satisfied for the proposed design.

Tube Dimensions – The selected primary members for the design have an outer diameter of 1.375-inches and a thickness of .065-inches. While there is no direct limit on the diameter of the tube, the thickness fits the requirements of being greater than .062-inches.

Cost – The team's running estimate from Alro Metals to buy 72 feet of the primary and secondary members would cost $206 and $114, respectively, with $100 in shipping; bringing the total frame cost to an estimated $420. While this estimate is almost twice the target value, the team is still waiting for quotes from local warehouses with possible sponsorships that might bring the total cost closer towards the target value.

2.4.3 Suspension DLs

Wheel Travel – The proposed front suspension design can achieve over 12 inches of suspension travel while meeting alignment engineering requirements. Cycling of the system using the final prototype and final system cycling during fabrication phase will identify clearance issues and set the limited travel to around 10 inches of wheel travel, by means of limiting straps and bump stops. Preliminary calculations of the rear suspension design using conservative link lengths found over 11 inches of travel within a standard CV joint operating angle. Similarly, cycling will limit travel to prevent driveshaft damage, however 11 inches is a reasonable estimate. Both systems exceed the target value of 10 inches, front and rear.

Travel Ratios & Ride Height – Both front and rear suspension were designed to ride at a 14" frame height, with an available bump travel ratio of at least 75%. Clearance is to be built into the front I-beam members in order to achieve a minimum 60% travel ratio in limited form. The rear suspension will satisfy this requirement as droop travel will be limited by CV operating angle, not bump travel. Setting of ride heights after fabrication will require only changing air pressures due to the selection of FOX FLOAT 3 shock absorbers which use infinitely adjustable air springs, offering an easy preload adjustment.

Track Width – The front suspension was designed for a 60-inch track width using the specified tire size and wheel offset. This meets the target value established for the vehicle. The rear suspension allows for narrower track widths; however, it will also measure 60 inches for maximum wheel travel with smaller CV operating angles.

Wheel Base – The wheelbase of the vehicle is dependent on the integration of frame and suspension systems, in order to locate the transmission, cockpit, and suspension pivot points correctly. The final wheelbase selected was 78 inches. This puts the vehicle at a 1.3 wheelbase to track width ratio allowing for a better turn radius, as well as ease of initiating the turn.

Alignments – Camber and caster change are described in detail in section 5.2.3. The front suspension system's camber and caster change over suspension travel are kept within the tolerances of 30 and 14 degrees, respectively. Toe is to be set using adjustable tie rods. The twin camber arm design of the rear suspension virtually eliminates camber change over suspension travel, and allows for setting of both camber and toe by adjusting camber arm lengths.

Bump Steer – Bump steer is successfully eliminated in the front suspension system by locating the tie rod pivots directly in front of the I-beam pivots through the use of a swing set steering system. This places the instant center of the wheel and the point of rotation of the tie rod on the same axis, creating equal dynamic radii and preventing bump steer in any direction due to incorrect arcs.

Steering Ratio – The steering system proposed provides a 6:1 steering ratio, based on the components selected in the steering analysis. This system was chosen by minimizing steering ratio as directed by the engineering requirement, while observing other dynamic relationships, for different component selections. This 6:1 ratio is right on the target value.

Scrub Radius – The scrub radius was minimized to the extent possible by the selection of wheel offsets, which place the wheel center closer to the frame.

Roll Center – By observing the instant centers of the I-beam in the front and the camber arms in the rear, it was noted that shorter links with steeper angles would raise the roll center. The camber arms are around half the length of the I-beam, so the roll center of the rear is higher than that of the front, allowing the roll axis of the vehicle to slope downwards towards the front of the vehicle, resulting in oversteer tendencies.

COG and Roll Stiffness – The height of the center of gravity of the vehicle may only be found through iterative computer modeling of every system, or through weighing the vehicle in race ready condition as noted in testing procedures. At this stage, the height of the center of gravity was minimized for performance by a conscious effort to locate weight lower in the vehicle. Roll Stiffness requires the COG height to develop target values, but was built into the vehicle by placing the shocks as outboard as possible. The rear suspension design allows for easy modification with an anti-roll bar if roll stiffness becomes an issue in testing.

Drive Axle Plunge & Operating Angles – The rear suspension allows for a near zero driveshaft plunge over the full range of travel due to the use of the twin camber arm system. This removes the need for plunging axles or CV joints. Maximum CV joint operating angle was estimated as 45 degrees during rear suspension design, and still achieves the target wheel travel values.

Total Unspring Weight – Due to the advantages of the simple analysis and structure of the TIB front suspension, lightweight members may be used, as maximum loads occur in tension and compression. Elimination of rod ends or Heim joints saves weight in heavy steel, welds in tube nuts, and joint weight. The rear suspension uses two arms transferring the spring force which must be made amply stiff, but does not require an upright. The tension and compression camber arm members may be made lighter than similar triangulated members.

2.5 House of Quality (HoQ)

For the HoQ, the team decided the most efficient construct of the table is to have each sub-system team create their own engineering requirements based on the needs of their system. This resulted in 3 individual HoQ's. In order to maintain a relationship between all of the sub-systems, the same customer requirements are used for each individual HoQ. The detailed HoQ results can be seen in Tables 5, 6, and 7 for Drivetrain, Frame, and Suspension, respectively.

2.5.1 Drivetrain HoQ

Table 5: Drivetrain HoQ

Customer Requirement	Weight / Engineering Requirement	Weight (lb.)	Power Transmitted (hp)	Torque Transmitted (lb-in.)	Velocity (MPH)	Engage/Disengage Time (s)	Material Strength (GSI)	Assembly Temp. (F)	Cost (U.S. Dollars)	Volume (ft.^3)	Manufacturing Time (hr)	Gear Change Force (lb.)	Lifetime Expectancy (Pinion Cycles)
Safety	10	3	1	1	3	9	9	9	9	9		9	9
Quick Acceleration	6	9	9	9	9	3		1	9	3		3	
Climbing Capabilities	6	9	3	9	1	3	3	1	9	3		1	
Towing Capabilities	6	9	9	9	1	1		1	9	1		1	
Durability	10	3					9	9	9		9	9	9
Easy Maintenance	5	3					9	9	9	9	9	9	9
Speed	4	9	9	3	9	9		1	9	3		3	
Innovation	2	9	9	9	9	3	3	3	9	9	9	3	9
Economical	7	9				9	9	9	9	9	9	9	9
Manufactureability	7	9					9	9	9	9	9	9	9
Multi-Terrain Capable	10	3	9		3		3	1	9	3			
Absolute Technical Importance (ATI)		447	280	202	180	237	405	389	657	363	279	399	369
Relative Technical Importance (RTI)		2	8	11	12	10	3	5	1	7	9	4	6
Targets with Tolerance [Value,+- Tolerance]		50,5	20,5	150,25	35,5	3,1	30,3	180,20	800,50	2,.25	24,5	5,2	1M,0

The top three RTI for Drivetrain's HoQ is cost, weight, and material Strength. These results make sense as the top three RTI due to drivetrain having a high impact on the overall cost and weight of the vehicle, as well as failure modes for the vehicle. The last NAU Mini Baja team experienced catastrophic failure in their drivetrain mounts; therefore, material strength will be key during the Drivetrain design process.

2.5.2 Frame HoQ

Table 6: Frame HoQ

Customer Requirement	Weight / Engineering Requirement	Driver to Frame Clearance (in.)	Member Material (Mpa)	Straight Member Length (in.)	Bent Members (Degrees)	Primary Members (#)	Secondary Members (#)	RRH Minimum Lateral (in.)	RHO/RRH Gusseting (#)	CR/L ≥ 12" from Seat Midpoint	CR/L 41" above seat	SIM Distance from Seat Bottom (in.)	Welding (Mpa)	FDM & Vertical ≤ 45°	Weight (lb.)	FAB ≤ 5" of B & ≤ 2" of A and S	Tube Dimensions (in. - Dia.)
Safety	10	9	3	1	1	3	1	1	1	1	1	1	9	1		1	3
Quick Acceleration	6		9										1		9		3
Climbing Capabilities	6		3										3		9		1
Towing Capabilities	6		9										9		3		1
Durability	10		9	1		1			1				9		1		3
Easy Maintenance	5		1										1		1		1
Speed	4		9										1		9		1
Innovation	2		1		3	3	3		1				1		9		1
Economical	7		9		1	1	1						1		9		1
Manufactureability	7		3	3	3	3	3		1				9		1		1
Multi-Terrain Capable	10		3										1		3		
Absolute Technical Importance (ATI)		90	403	41	44	74	44	10	29	10	10	10	349	10	295	10	115
Relative Technical Importance (RTI)		5	1	9	7	6	7	11	10	11	11	11	2	11	3	11	4
Targets with Tolerance [Value,+- Tolerance]		>6	>365	<40	<30	>9	>5	>29	min	min	min	11,3	>370	min	<70,5	min	>1

Frame's top three RTI include material strength, welding strength, and weight. The frame will set the base for how much the overall vehicle will weigh. Typically, vehicles weighing over 600 pounds are not successful during the endurance race for competition. In addition to this, material and weld strengths cannot fail for the Frame at any time during assembly and competition. This is why they are ranked so high on the RTI scale for the Frame HoQ.

2.5.3 Suspension HoQ

Table 7: Suspension HoQ

Customer Requirement	Weight / Engineering Requirement	Vertical Wheel Travel, Front (in)	Vertical Wheel Travel, Rear (in)	Travel Ratio, Front (up/total) (%)	Travel Ratio, Rear (up/total) (%)	Track Width (in)	Wheelbase (in)	Caster Change	Camber Change	Toe (in)	Bump Steer (deg)	Steering Ratio (deg/in)	Scrub Radius (in)	Height, Roll Center, Rear (in)	Height, Roll Center, Front (in)	Height, COG (in)	Roll Stiffness (lb-ft/rad)	Drive Axle Plunge (in)	Drive Axle Max Angle (deg)	Total Unsprung Weight (lb)
Safety	10	1	1	3	3	3		3	3	3	9	1	3	9	9	9	3	9	9	3
Quick Acceleration	6															3				3
Climbing Capabilities	6	3	9				3			1	3	3		1	1	3				3
Towing Capabilities	6		3		3		3	3												
Durability	10	9	9	9	9		1	1	9	1	9		1			1	3	9	9	9
Easy Maintenance	5		1											1	1		1	9	9	1
Speed	4	9	9	1	1	3	3	3	9	3	9	3	9	9	9	9	9			3
Innovation	2	9	9			1	1	1	3	3	9	3	3	9	9	9	3	9	9	3
Economical	7	3	3								1	9	3	1	1	9	3	3	3	9
Manufactureability	7	3	3								3	9	3	3	3	1	3	3	3	9
Multi-Terrain Capable	10	9	9	9	9	3	9	1	3	1	3	3	3	3	3	3	3	1	1	1
Absolute Technical Importance (ATI)		304	363	214	232	74	150	82	192	74	310	202	154	213	213	290	179	295	295	315
Relative Technical Importance (RTI)		4	1	9	8	18	16	17	13	18	3	12	15	10	10	7	14	5	5	2
Targets with Tolerance [Value,+- Tolerance]		10,2	10,2	60,5	75,5	58,6	75,5	7,5	0,10	0	0	20,5	0,2	min	min	min		0	20,5	min

The top three RTI for the Suspension team are rear wheel vertical travel, total weight, and amount bump steer. Weight is another major factor for suspension, because weight sets the standard for what type of requirements will be needed from the suspension to allow the car to operate as desired. The amount of bump steer experienced by suspension as well as the vertical wheel travel will allow the vehicle to operate as efficiently and comfortably as possible.

3 EXISTING DESIGNS

3.1 Design Research

In order to design an efficient, fully functional vehicle, the team must perform extensive research and benchmarking for the project. The team plans on continuously benchmarking previous Mini Baja competitions and looking at what worked or did not and why. In addition to this, research will be performed on different types of subsystem level assemblies for vehicles that can apply to the Mini Baja design.

One major benchmark for the team is NAU's 2015 Mini Baja vehicle, which was not competed due to the previous team's failure to register. Particularly, the team looked at the frame and suspension of the vehicle.

Regarding suspension, the 2015 NAU Baja vehicle is equipped with a double wishbone (A-Arm) front suspension, and a single trailing arm rear suspension. During analysis, the front end was found to have too stiff of a spring with geometry that did not mesh with other components. Toe and Camber exceeded tolerances through suspension travel, and mounting points were not designed for maximum strength. Benchmarking of the front suspension suggests more of a focus on dynamics and ride quality during design is needed, as well as efficient manufacturing techniques. According to previous reports, when the drivetrain is assembled, there are clearance issues with the drive axle. The orientation of the trailing arm indicates large amounts of axle plunge when full travel is achieved. Benchmarking of the rear suspension indicates the importance of packaging and integration with other vehicle systems.

Regarding the frame of the vehicle, it is light weight, structurally sound and great for competition. A big part of this is due to the TIG welding of the frame, which increases strength and reduces weight when compared to a MIG weld. This frame also utilizes additional secondary members that increase the strength and integrity of the vehicle. Points of weakness in the frame design include the pedal assembly and seat. The pedal assembly of the existing car is not adjustable for different size drivers. This is a design the 2017 Mini Baja team would like to implement so that the car can be driven by multiple people and score higher in the sales and design evaluation at competition. The seat used in the previous years incorporated excess material and weight compared to others available and used by top teams at competition.

3.2 System Level

Most system level components, as well as designs, come from the SAE rules and regulations of the Mini Baja CDS. Because the rules are so limiting, there is little to no room for the addition or removal of existing system components. As a result, most competition teams contain a similar function decomposition for their designs. For the results of the decomposition, please reference section 3.3 of this document.

3.2.1 Existing Design #1: Drivetrain

The drivetrain is responsible for how the vehicle will be put into motion. This is achieved by translating the engine's rotational energy into linear motion. Drivetrains for the Mini Baja competition typically involve a continuously variable transmission (CVT) linked to a type of axle. These systems provide an efficient, easy, and lightweight solution for providing drive from the engine to the wheels. For these reasons, the team plans on utilizing a CVT to act as a clutch for a gearbox that will increase torque ratios for the team. The type of axle will be determined when more details are received regarding the 2017 CA competition events.

3.2.2 Existing Design #2: Frame

The frame of the vehicle is comprised of primary and secondary members which all other system assemblies are mounted onto. The frame also provides the housing for the driver. The driver is harnessed into a seat decided upon by the frame team to minimize weight and space. Nearby to the driver a fire extinguisher, kill switch, and lap transponder are mounted to the frame. A heat shield is mounted behind the driver that protects the driver from drive train components. The heat shield is designed to be light weight while having low emissivity and high reflectivity. Lastly, a pedal assembly will be adjustable to accommodate different height drivers while maintaining throttle and braking performance.

3.2.3 Existing Design #3: Suspension

There are four types of suspension that were researched and considered for the design. In previous competitions, the suspension most commonly used by previous teams is the “Double wishbone” or “Double A-arm” suspension. This is because it can be modified in multiple ways to save weight and improve performance. This suspension is comprised of an upper and lower control arm that attaches the frame to the hub with a strut aligned between them to control vertical movement. Another type of suspension similar to that is a Macpherson Strut; this eliminates the upper control arm helping reduce weight, but cannot withstand as much force in the direction the car is moving. The multi-link system is made up of multiple pieces attaching the frame to the hub. It can be designed to a specific class of driving and allows in depth tuning for racing or other designated environments. Multi-link is too advanced for this project and will not be needed. A Twin I Beam (TIB) suspension is used in the front suspension of some Ford models. It uses two beams connected to the frame on opposite sides of each other, each member crosses the center axis to attach to the hub on the other side, creating a “Scissor-jack” like effect. TIB is strong and reliable, but uses a lot of material causing it to be expensive and increases the overall weight of the vehicle. The best options to achieve the team goals of reliability and weight reduction are the Double – A – Arm or TIB designs. These designs will be further analyzed and compared using calculations to determine the best option.

3.3 Subsystem Level

In order to decompose the very complex design process for a vehicle, the team utilized the categories of the SAE Mini Baja rules and regulations [1]. Per these rules and regulations, main components of the vehicle’s design include the drivetrain, frame, and suspension. Because of this, the team decided to make these the three main subsystems of the overall vehicle’s design. From there, each subsystem has its own decomposition of components that make up the actual assembly of the system. The decomposition is summarized in Figure 1.

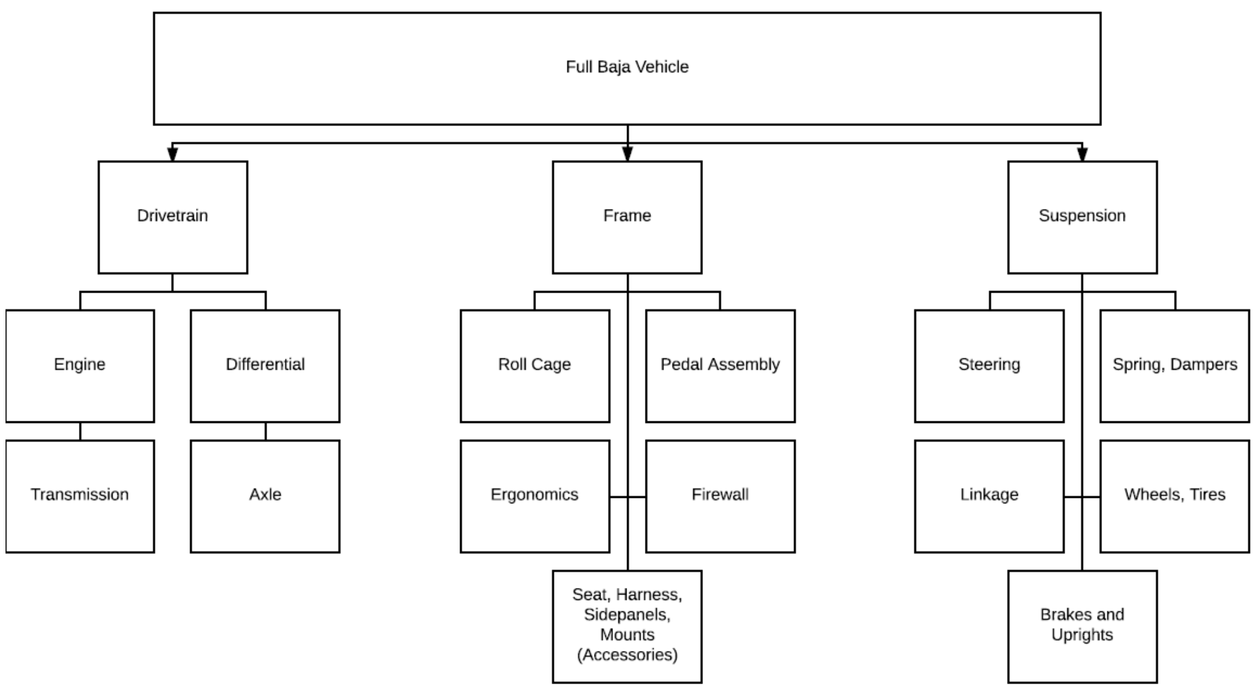

Figure 1: Project Decomposition

3.3.1 Subsystem #1: Drivetrain

The drivetrain is the system that transmits power from the engine to the wheels. The engine converts chemical energy into thermal and mechanical energy. The mechanical energy can then be measured in the form of power and torque. The power and torque is transferred through an output shaft to a CVT. The CVT can increase power or torque, while decreasing the other. The rate at which the CVT achieves this is dependent upon the gear ratios being utilized. At higher RPM's there will be a higher power output, with a low torque output, and vice versa for low RPM's. The energy is then sent through the output shaft of the CVT into an axle. The axle delivers energy into the wheels that convert to the linear motion of the vehicle.

3.3.1.1 Existing Design #1: Constant Velocity (CV) Axle

A CV-axle allows the wheels connected to the output to move independent of each other. The axle is not encased in a shaft like a live axle. By allowing the axle to spin freely without a cover, the weight of the output shafts decrease which makes the vehicle lighter. More weight loss can be seen when utilizing a CV-axle due to the design no longer needing an axle housing. The CV-axle can be designed to withstand any torque required to compete. A weakness of the CV-axle is the lack of torque transfer over obstacles encountered by individual wheels. In addition to this, open components make the CV-axle vulnerable to impact forces that could cause failure.

3.3.1.2 Existing Design #2: Continuously Variable Transmission (CVT)

A CVT is a system that can be adjusted for an infinite amount of gear ratios. A CVT operates from a belt that is spread across two pulleys. The pulleys work in unison to change power and torque ratios relative to each other. The two pulleys are designed to increase or decrease their radius to change the ratio of pulley revolutions. When engine rpms increase, the driven pulley starts to compress, increasing its radius; while the driving pulley starts to expand to alternatively decrease the radius. At lower engine rpms, the CVT can create a high amount of torque, but as the engine rpms reach a maximum, the CVT will output a higher amount of horsepower. The weakness of a CVT is they require a minimum rpm before it engages the driveshaft. This can cause lag in response from the driver engaging the gas pedal.

In this design application, the CVT will be linked to a jackshaft, that allows the CVT to transmit power and torque to the axles. A jackshaft is an intermittent shaft that couples to different rotational systems.

3.3.1.3 Existing Design #3: Gearbox

A gearbox is a sub-system that utilizes gear ratios to broaden the vehicle's torque and power capabilities. The gearbox can change the amount of torque or horsepower to the wheels depending on which gear ratio is being engaged. The driver of the vehicle can switch the gearing ratio by moving a fork that can disengage or engage gears inside of the sub-system assembly. The gearbox also allows the team to design for a reverse gear. The gearbox allows the team to have a reverse gear, which makes it a highly desirable design. However, in order for the gearbox to change gears, it must be used in conjunction with a CVT in order to disengage the assembly from the engine.

3.3.2 Subsystem #2: Frame

The subsystems assigned to the frame assembly include the seating, pedal system, and the type of welding. These components all drive the design for the frame assembly, and directly affect the analysis of the frame.

3.3.2.1 Existing Design #1: Seating

Research of previous competition leaders yielded mixed results for seat design. The two overall leaders of the 2016 CA competition, University of Michigan and Cornell University, both selected suspension seats. The four runners-up for the same competition all used conventional style seats. The suspension style seat allows the user to adjust the height of the seat through a strap assembly which may be beneficial in creating a multiuser friendly car; whereas a conventional seat may be uncomfortable for different body types. Further research needs to be done on the possible weight impact of such selections.

3.3.2.2 Existing Design #2: Frame Geometry

Geometry of the frame is a very flexible aspect of the car and research of previous competitors yielded many different designs. A common trend among the competition leaders was the lack-of or minimal 'nose' section of frame. The "Nose" section being defined by points G and E from Figure RC5 of the Official Baja 2017 Rules. Leading cars also had a widening bend in their side impact members (SIM) to allow for ease of access into and out of the vehicle. The team will look at possibly implementing a similar design to meet the competition requirements of vehicle exit times.

3.3.2.3 Existing Design #3: Welding Method

The types of welding available to the team for this competition include MIG and TIG. Since the overall goal is to reduce frame weight while also creating a structurally sound frame, both type of welds have their pros and cons. MIG welding is the most common form due to its ease of application. This type of welding can be done in a short amount of time; however due to the auto-feeding of material by the machine, additional weight accumulates for each weld. When TIG welding, the wire is hand-fed by the user while the material is being heated, leading to less overall material, longer weld times, and deeper weld penetration.

3.3.3 Subsystem #3: Suspension

The suspension allows the wheels of the vehicle to travel vertically with respect to the frame in order to provide a stable and comfortable ride. Each wheel fits to a hub, which is housed in an upright member. The upright member must be restricted to only two degrees of freedom: vertical travel, and rotation about the vertical axis for steering. The suspension ties the frame to the support points of the upright, and acts as a mechanical linkage to create a path of wheel travel. The steering system controls the angular location of the wheels about the vertical axis, if necessary. Steering input is done through a shaft connected to a steering wheel, and into a mechanical device to convert it to linear motion. A spring and damper system is used to support the sprung mass of the vehicle and control oscillations in the vertical direction. Controlling the rebound of the spring prevents loss of control of the vehicle.

3.3.3.1 Existing Design #1: Spindles

Spindles contain the hub and wheel mounting assembly for independent suspension systems. This includes mounting points for linkage and steering systems, as well as portions of the braking and powertrain systems. Spindle design can vary greatly depending on the desired applications and manufacturing methods.

The front suspension spindles must have an integrated braking system and knuckles for the steering system input. This is a common design, typically found on a variety of sport ATVs. Categories driving the design of spindles are weight, mounting point dimensions, and brake size.

For the rear suspension, the spindle must house a driven hub, which then integrates with a drive axle system. This design is utilized by an upright spindle that has similar applications to that of the final drive system, such as the rear spindles on the 2008-2011 Polaris Outlaw 525IRS. These uprights also have optimal linkage mounting locations for their associated drive axles and CV joints.

3.3.3.2 Existing Design #2: Steering

Steering on off-road vehicles is typically achieved through a steering rack. A steering rack takes a steering wheel's input and converts it to translational motion. The length of the steering knuckle determines final steering ratio and steering effort. Proper U-joint design on the steering shaft input is required to avoid binding. A rack from an off-road class of vehicle would perform well in a Baja vehicle. Steering racks are available in a variety of lengths, ratios, and pinion (input) locations. In a Baja vehicle, a steering rack provides the best design in regards to weight when compared to other options.

3.3.3.3 Existing Design #3: Springs and Dampers

Springs and dampers usually packaged in the form of a coil-over shock or an air shock. FOX Shocks Power Sports division offers a FLOAT series air shock to SAE Mini Baja teams at discounted prices. These springs are progressive and adjusted through air pressure, while maintaining a weight of under 5 pounds. A more vertical shock position would be beneficial to minimize change in shaft velocity.

The three air shock options FOX offers to SAE Mini Baja competitors are: FLOAT 3, FLOAT 3 EVOL R, and FLOAT 3 EVOL RC2. The FLOAT 3 is a single chamber progressive air shock with no reservoir, and weighs 2.25 pounds. The EVOL R model adds a piggyback external rebound damping adjuster, at 3.5 pounds. The RC2 model adds: a reservoir with a secondary air chamber for bottoming, external rebound adjusters, and two external compression adjusters for high and low speed. The RC2 model weighs 4.75 pounds. This gives the team options for design in budget and tuning abilities.

4 DESIGNS CONSIDERED

4.1 Drivetrain Designs

Below are the Drivetrain designs considered. Each design has a advantages/disadvantages table that considers the given ERs, CRs, and rules of the competition.

4.1.1 CVT Linked w/ F/R Gearbox to Locking Differential

This system begins with the CVT receiving an input from the engine. When the engine is in idle, the CVT acts as a clutch that disengages the engine from the rest of the drivetrain system. During this phase, the driver can easily switch between the forward or reverse gears. Once a gear is selected, the driver can then raise the RPM's which the engine is running at, which in turn expands the CVT to engage into the gearbox. The gearbox then transfers that rotational energy into the differential. The differential will then transfer the energy to the CV axles, which allow the tires to convert the rotational energy into translational energy. This design can be seen in Figure 2.

Table 8: CVT - Gearbox - Locking Diff. Advantages/Disadvantages

Advantages	Disadvantages
Adjustable Gears	Weight
Reverse Capabilities	Difficult Maintenance
	Complex Linkages

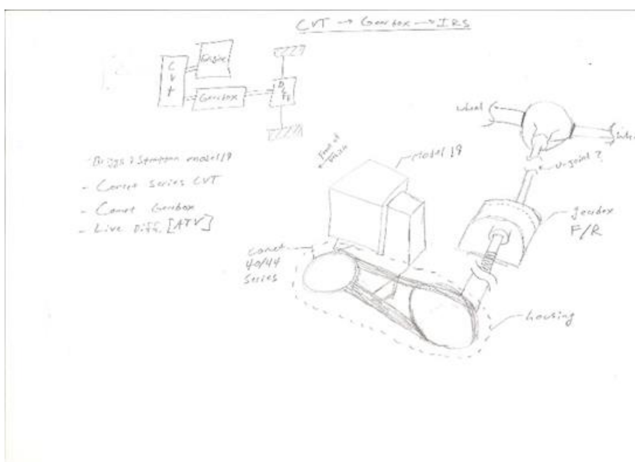

Figure 2: CVT w/ F/R Gearbox to Locking Diff

4.1.2 CVT Linked by Jackshaft to a Solid Axle

The CVT will receive it's input from the engine. The CVT will then continuously change RPM ratios delivered to a jackshaft based off of the engine RPM. The jackshaft will be chain driven with sprockets linked to the CVT and a solid Axle. As the jackshaft is engaged, the chain driven sprockets will rotate the axle, which allows the tires to convert the rotational energy into translational energy for the vehicle. This design can be seen in Figure 3.

Table 9: CVT - Jackshaft - Locking Diff. Advantages/Disadvantages

Advantages	Disadvantages
Less Components	Weight
Simple Linkages	No Reverse Capabilities
Adjustable Sprockets	Weak Sprockets
Small Volume	Open Rotational Components
Easy Maintenance	

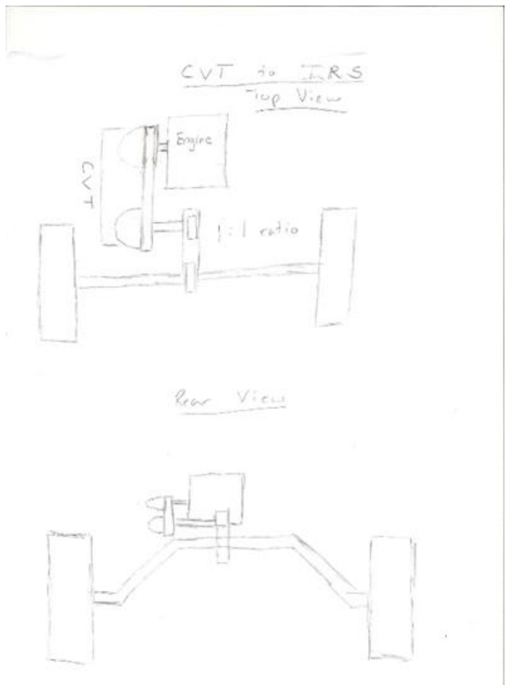

Figure 3: CVT - Jackshaft - Locking Diff.

4.1.3 CVT Linked to F/R Transaxle to CV Axles

The CVT will receive an input from the engine's output. The CVT will then act as a disengage between the engine and transaxle when the driver decides to switch between the forward and reverse gears. Once a gear is selected, the driver will increase the engine's output, which will engage the CVT and begin to transmit rotational energy from the engine to the transaxle at continuously varying ratios. The transaxle will then change the power and torque ratios depending on its gear set and send the rotational energy into the CV axles. The CV axles will then transmit the energy to the tires, which convert rotational energy into the translational motion of the vehicle. This design can be seen in Figure 4.

Table 10: CVT - Transaxle - CV Axles

Advantages	Disadvantages
Adjustable Gears	Complex Internal Components
Reverse Capabilities	Limited Max Torque
Lightweight	
Less Components	
Simple Linkages	
Easy Maintenance	
Small Volume	

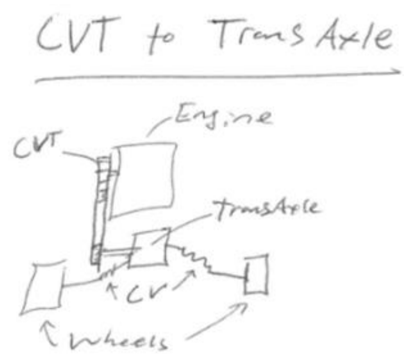

Figure 4: CVT - Transaxle - CV Axles

4.2 Frame Design

The design space for the frame is very limited within the list of rules given by SAE Baja. The team started by creating a parametric model of the cockpit based off of the model given by the competition rules. The basic cockpit model was comprised of the required primary members and some secondary members including the side impact members (SIM), fore and aft bracing members (FAB), and the under seat members (USM). Once a base model was created, the team began to research past competitions to find areas that teams had modified, but still fell within competition parameters. A common design that was noticed during benchmarking was a frame that lacked a "nose" section. The frame team proposed this design to the suspension team and found that the lack of a nose would not impact their designs so the frame team trimmed off the nose to save weight. The second area of modification was in the side impact members, which designate the top of the "Tub" that the driver sits in. The team chose to add a bend in these side members to make them curve out from the cockpit. This design was inspired by two things: (1) review of the previous competition winner, Michigan State University, who incorporated a similar design; and (2) hands on experience with NAU's previous Baja car which proved difficult to get out of with the straight side members. Overall the goal of modifying these side members is to increase the ease of entry and exit of the vehicle so the driver can operate within the 5 second exit window designated by the competition.

Rule specifications state that any materials used for primary members on the frame must meet the bending stiffness and bending strength of 1-inch diameter, .120 inch wall tubing made of 1018 steel. A range of tubing in multiple sizes and materials were compared to find options that would exceed the stiffness and strength requirements, while reducing the overall weight of the vehicle. Based upon these restraints, the material choices were narrowed to 4 types due to weight savings, manufacturability, bending strength, stiffness, availability from material providers, outer diameter to increase strength of attachment points, and time available to weld members. Table 6 (Section 5.1.2) displays not only the datum for bending strength and stiffness, but also shows two options in drawn over mandrel (DOM), but also 4130 Chromoly tubing. Industrial Metal Supply (IMS) has worked with NAU Baja teams in the past in providing metal for the project, the possible sizes carried by IMS were compared with what is available according to the company's website. No tubing has been selected thus far, as each tube option has several positive and negative aspects to account for based on the Drivetrain and Suspension team's design.

Figures 5, 6, 7, and 8 show the existing frame CAD designs for the team.

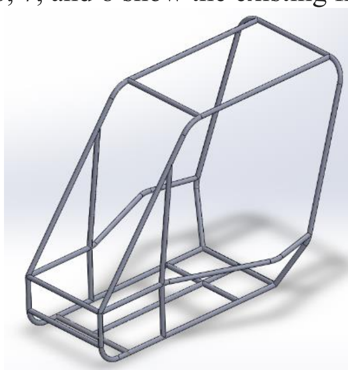

Figure 5: ISO View

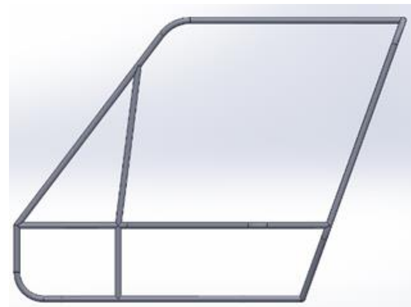

Figure 6: Right View

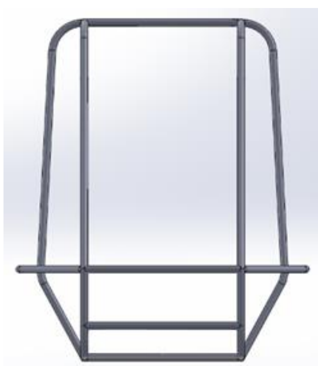

Figure 7: Front View

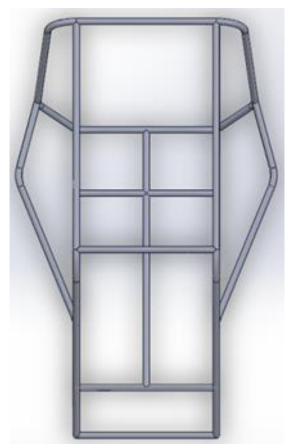

Figure 8:Top View

4.3 Suspension Designs

4.3.1 Front Suspension

The front suspension system is integrated with braking and steering systems. Benchmarking identified some designs in standard use for Baja SAE, as well as alternative designs. Each design has relative strengths and weaknesses and was compared using competition, customer, and engineering requirements in the selection process.

4.3.1.1 Control Arm Systems

Equal length and Short-Long Arm (SLA) control arm systems were considered. These are illustrated in Figure 9. Equal length, parallel arms offer the simplest geometry with no caster or camber change over wheel travel. SLA configurations offer some beneficial alignment curves to combat issues such as body roll and bump steer [9]. Both designs would have many degrees of freedom in geometry making for complex dynamic analysis. The number of articulating joints, usually rod ends, is a reliability issue. A rack and pinion, shown in Figure 10 offers a simple steering system design with little bump-steer.

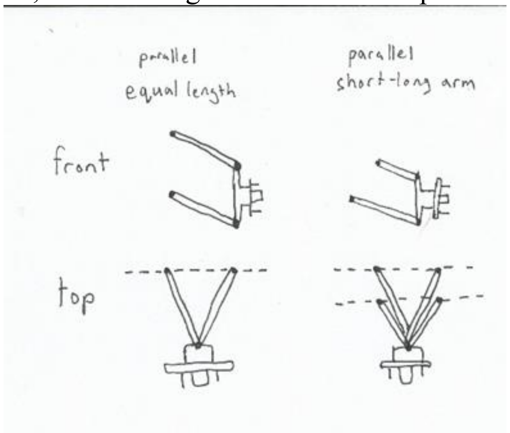

Figure 9: SLA Control Arms

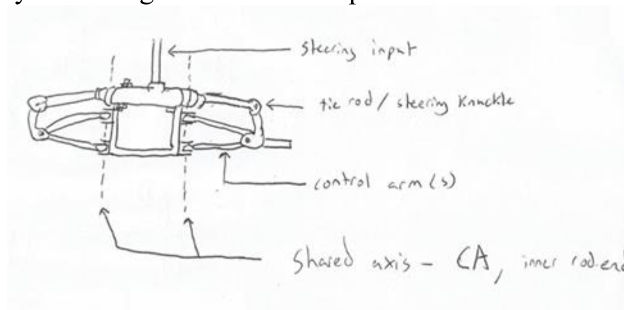

Figure 10: Rack and Pinion

4.3.1.2 TIB Design

A Twin-I Beam design, as in Figure 11, was considered as an alternative to traditional control arm designs [10]. The advantages of this sytem are negative camber gain in bump to combat rolling and minimize slip while turning, and simplicity. Two spherical bearings are used per wheel, and some spindles may offer the use of a kingpin design to replace ball joints. Force and dynamic analysis are simplified.

The drawbacks of this design are the possibility of extreme wheel alignment changes in travel and the need for a special steering system to combat bump-steer. Figure 12 shows a typical "swing set" steering system. This accomplishes the correct location of tie rod pivots, but is large, complex, and heavy, and requires the use of a steering box. Figure 13 is an alternative design that simplifies packaging by steering from the front of the spindle. A traditional rack and pinion is used and linkage translates rack input to tie rod output. This design also allows for easy tuning of steering ratios through "swinger" lengths and pivot locations.

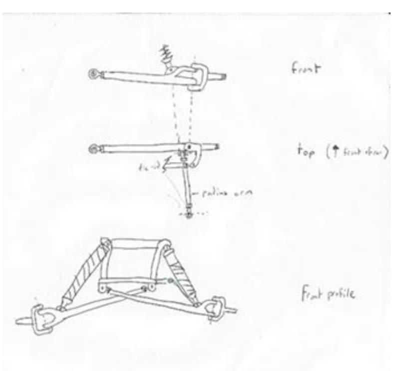

Figure 11: Twin-I Beam (TIB)

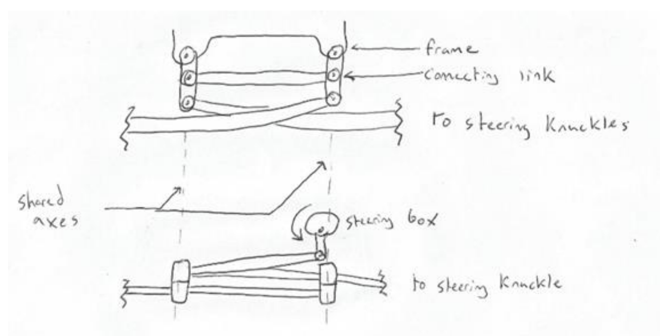

Figure 12: Swing Set Steering

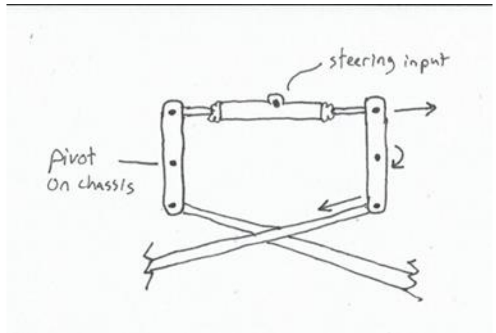

Figure 13: Steering Box

4.3.2 Rear Suspension

Rear suspension is integrated with drivetrain and braking systems. Primary design considerations will be the drive axle plunge and operating angles. Maximizing wheel travel within those constraints, and maintaining wheel alignment will be used to evaluate designs in the selection phase.

4.3.2.1 Triangulated Trailing Arm

Triangulated trailing arms or semi-trailing arms as pictured in Figure 14 a rear suspension specific design. This design is the strongest option, and likely the lightest. In order to package this system, excessive lengths must be avoided, which can result in limited suspension travel. Maintaining wheel alignment and designing for drive axle geometry would be difficult with triangulated trailing arms.

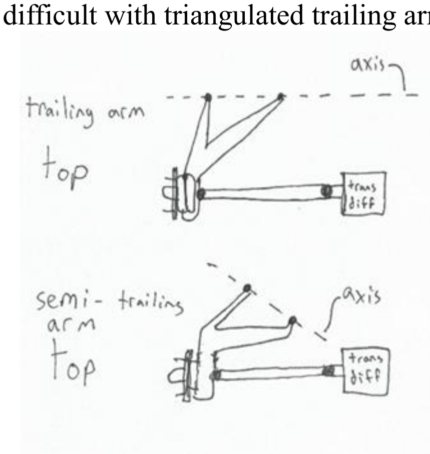

Figure 14: Triangulated Trailing Arms

4.3.2.2 Equal Length Parallel A-Arm

This design is similar to the A-Arm design for the front suspension; however, it utilizes a spindle within an integral drive hub. Equal length arms allow designs that minimize axle plunge and maximize travel within goal track widths. Wheel alignment can be kept consistent through travel, and toe adjustment is easily achieved. This option is very light, but strength is still an issue due to the number of pivots within the system. This can also be seen in Figure 9.

4.3.2.3 Trailing Arm with Camber Arms

This suspension design was based on benchmarked systems seen on high-end UTVs. As seen in Figure 15, a single trailing arm runs longitudinally and houses the hub assembly. Two camber arms, in the form of links with offset rod ends, constrain the wheel laterally. When the camber arm pivots are placed in line with drive axle joints, zero plunge can be achieved. Long trailing arms with spring mounts allow for maximum wheel travel, and can be mounted on the lower side chassis members. This option requires design of a strong, but light trailing arm.

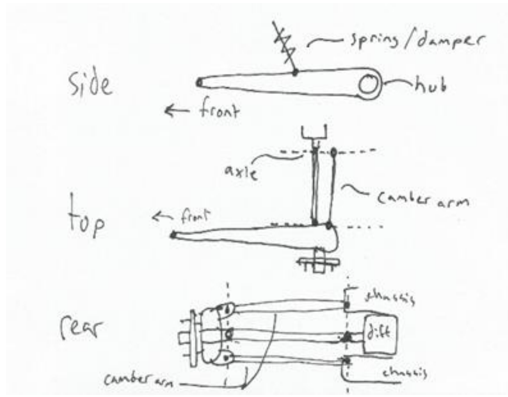

Figure 15: Trailing Arms w/ Camber Arms

5 DESIGN SELECTED

This section will review the analyses performed by each sub-assembly, and how the selection process for each sub-assembly was completed.

5.1 *Rationale for Design Selection*

All design selections were completed by analyzing each concept in a decision/selection matrix. The results of these matrices are discussed in this section.

5.1.1 Drivetrain Selection

When looking at the first two designs considered, the Drivetrain team realized both concepts had great ideas that would be idealized when combined. As a result, the third design considered combined the ability of a reverse gear from the gearbox design, as well as the simple linkage that came from the CVT jackshaft design. This resulted in the selected design being the CVT to Transaxle to CV axle system. This design gives the team weight reduction, simplicity, ease of maintenance, and allows for the Drivetrain to be packaged into a smaller assembly. The next step in the selection process for Drivetrain is to formulate a spec sheet to compare different parameters of components that meet the system's requirements. The decision matrix and results can be seen in Table 11.

Table 11: Drivetrain Decision Matrix

CRs	Weighting	CVT - Gear - Diff.	CVT - Jackshaft - Axle	CVT - Transaxle - CV's
Quick Acceleration	6	3	4	4
Climbing Capabilities	6	5	3	4
Towing Capabilities	6	5	2	4
Durability	10	4	3	4
Easy Maintenance	5	1	4	4
Speed	4	5	3	5
Innovation	2	5	1	2
Economical	7	1	5	3
Manufactureability	7	2	5	4
Multi-Terrain Capable	10	3	2	3
Weighted Sum		204	208	235
Relative Rating		3	2	1

5.1.2 Frame Selection

On the first row of Table 12 is DOM 1" diameter .120" wall tube, which is the specified tube by which any other selected tube must exceed the bending stiffness and strength. The DOM 1.25" diameter .083" wall tube on row 2 is the optimal DOM tube available from IMS, as it weighs less and has a higher bending stiffness and strength than the required values. Row 3 on Table 12 is the lightest weight option available to build the roll cage and is available from IMS. However, it is made of 4130 Chromoly, which is an ideal material to TIG weld. TIG welding saves weight, but is more difficult than MIG welding and requires longer lead and manufacturing times. The last 4130 option is the most likely pick of the Chromoly materials. The 4130 1.375" Diameter .065" wall tube is the second lightest option while maintaining the highest bending strength and second highest bending stiffness. The larger diameter will also provide a larger surface area for tabs welded to the frame. This is significant because it allows for more strength on item mounts, which can be seen as a common failure in previous Baja competitions. Lastly, the DOM 1.375” diameter .065” wall tube compares well with the same size in 4130; however, it is not available at IMS. By choosing a DOM tube, the team would be able to MIG weld the frame which would save time and money. The team will be looking into other steel providers to compare pricing and sizing options. Thus far a final decision is unable to be made, it will be decided upon once the available sizes and timeline of the designs are determined.

Table 12: Frame Materials

Material	Diameter (in.)	Thickness (in.)	Weight (lb/ft)	Bending Stiffness (lb*in^2)	Bending Strength (lb*in)	IMS Available
DOM	1	0.123	1.128	583306	2397	Y
DOM	1.25	0.083	1.034	854862	2649	Y
DOM	1.375	0.065	0.909	917227	2532	N
4130	1.25	0.065	0.823	668324	2594	Y
4130	1.375	0.065	0.909	895918	3146	Y

5.1.3 Suspension Selection

5.1.3.1 Front Suspension

For the front end, heavy emphasis was placed on increasing reliability (Through decreasing the number of pivot points), simplicity of analysis, and feasibility of making the design work. For wheel alignment, importance was placed on staying within tolerances as well as allowing negative camber in bump.

Due to high durability, as well as its simplicity of design and analysis, the Twin I Beam was selected. Although this is not a typical design utilized in competition, its simplicity and design time fits exactly what the team is looking for. Additionally, the use of a swing set steering systems allows for greater design freedom in steering ratios and forces, which would be locked by using a control arm system with a rack and pinion. Packaging concerns of the system can be mitigated through the use of the rack and pinion swing set steering system. Un-sprung weight can also be minimized with careful component design. The front suspension selection table is seen in Table 13.

Table 13: Front Suspension Selection Chart

Criteria	Twin I Beam	Control Arm - Equal Length	Control Arm - SLA	Advantage
Vertical Wheel Travel, Front (in)	Offers easy shock mounuting to acheve 10+ inches	Will require beefy lower arm for shock mounting	Will require beefy lower arm for shock mounting	Twin I Beam
Travel Ratio, Front (up/total) (%)	Ratio close to 60% minimizes camber change	Acceptable	Acceptable	None
Track Width (in)	Requires relatively wider track width	Allows for narrow track width	Allows for narrow track width	Control Arm Variants
Caster Change	Long radius arms can achieve target value	None	Fully customizable	SLA
Camber Change	Can achieve target, negative camber in bump	None	Fully customizable - can allow negative camber in bump	SLA
Bump Steer (deg)	Requires swing set steering for zero	Some - Can be accounted for	Some - difficult to analyze	None
Steering Ratio (deg/deg)	Swingset allows full customizable ratios	Depends on rack and pinion	Depends on rack and pinion	TIB
Height, Roll Center, Front (in)	Easier analysis	Complex dynamic/geometric analysis	Complex dynamic/geometric analysis	TIB
Roll Stiffness (lb-ft/rad)	Radius arm allows for packaging of sway bars	Requires too much shock angle	Requires too much shock angle	TIB
Total Unsprung Weight (lb)	Heavier, but smart design minimizes	Lightweight	Lightweight	Control Arm Variants
Durability	Extremely durable, minimal suspension pivots	Many suspension pivots - prone to breaking	Many suspension pivots - prone to breaking	TIB
Performance	Good in rough sections, minimize slip in turns	Standard - determined by shock tuning	Standard - geometry becomes game of compromise	TIB and Equal Length
Manufacturability	Creative design minimizes difficulty	Requires excellent measurement and welding	Requires excellent measurement and welding	TIB
Innovation	Innovative	Not innovative	Can be innovative through modified geometry	TIB

5.1.3.2 Rear Suspension

What drives rear suspension design is ensuring that geometry aligns with rear drivetrain design in order to achieve maximum suspension travel and ground clearance. Wheel alignment is not as much of a design concern for the rear suspension, but toe adjustability will allow for the correction of oversteer and understeer. Since there will be more weight on the rear wheels, durability becomes a large factor as well.

The analysis of the rear suspension designs considered led to the team's selection of the Trailing Arm with Camber Arms design. This design is popular among SAE Baja teams and high end industry competitors. It allows for maximum travel by easily matching drivetrain geometry while retaining the high strength of trailing arm designs. Though four rod ends will be used per wheel, they are loaded in tension and compression rather than in bending like a control arm design. In terms of performance, the use of a sway bar is easily packaged, and the rearwards movement in bump travel allows for a smoother ride. Fabrication concerns are still a factor, but can be mitigated by detailed computer modeling as well as rapid prototyping prior to manufacturing. The rear suspension selection matrix can be seen in Table 14.

Table 14: Rear Suspension Selection Chart

Engineering Requirement	Triangulated Trailing Arms	Control Arm - Equal Length	Trailing Arm - Camber Arms	Advantage
Vertical Wheel Travel, Rear (in)	Limited by packaging and axle geometry	Achieves target values	Achieves target values	Control Arm and TA/CA
Travel Ratio, Rear (up/total) (%)	Limited by total travel	Acceptable	Acceptable	Control Arm and TA/CA
Track Width (in)	Acceptable	Acceptable - allows minimization	Acceptable - allows minimization	Control Arm and TA/CA
Camber Change	Unpredictable, large changes possible	None	None	Control Arm and TA/CA
Toe (in)	No adjustability	Some adjustability	Maximim adjustibility	TA/CA
Height, Roll Center, Rear (in)	Difficult to analyze	Complex dynamic/geometric analysis	Complex dynamic/geometric analysis	none
Roll Stiffness (lb-ft/rad)	No advantage	No advantage	Allows for easy packaging of sway bars	TA/CA
Drive Axle Plunge (in)	Will exist, be difficult to design for	None if designed correctly	None if designed correctly	Control Arm and TA/CA
Drive Axle Max Angle (deg)	Will limit wheel travel	Can account for	Long trailing arm minimizes longitudinal displacements	Control Arm
Total Unsprung Weight (lb)	Standard	Lightweight	Slightly heavier	Control Arm
Durability	Extremely durable, if axle geometry is matched	Many suspension pivots - prone to breaking	Extremely durable - correct use of rod ends	TA/CA
Performance	Limited travel creates rough ride	Standard - determined by shock tuning	Best shock mounting and tendency to "follow" bump	TA/CA
Manufacturability	Requires excellent measurement and welding	Requires excellent measurement and welding	Difficult fabrication of trailing arm	TIB
Innovation	Standard	Standard	Innovative - New Trend (winning cars)	TIB

5.2 Design Description

5.2.1 Drivetrain Analysis

The Drivetrain system will involve an individual mounting frame that will be directly welded to the vehicle's Frame. This individual frame will have the engine mounting assembly, followed by the transaxle assembly with a skid plate mounted to the rear. Intermittent to the engine and transaxle will be the CVT, which couples the two components. This allows the drivetrain package to be flexible in its location on the vehicle's frame, while keeping the assembly as low as possible to help optimize suspension and keep a low center of mass. In addition to this, the packaging allows for the system to stay compact, and relatively small, which is essential in the effect for the overall vehicle.

The full Drivetrain Assembly can be seen in Figures C-3 and C-4 in Appendix C.

5.2.1.1 Wheel Outputs and Component Selection Analysis

Analysis Process

A MATLAB code was written to calculate the necessary torque and velocities because the engage rpms, the CVT ratios, and the transaxle ratio vary depending on the CVT and transaxle picked. The code then gives torque outputs at different steps along the drivetrain system to determine required intermediate shafts and

CV axles. The code also tells the user the max velocity and the required torques to move the vehicle. The user can then select the desired CVT and transaxle to have an ideal system to compete in all events. To simplify the calculations, assumptions were made that there is full traction between the tires and the ground surface, the CVT belt does not slip, the CVT Clutch driver engages exactly at the specified engine RPMs and the weight is evenly distributed throughout the car.

Equations Utilized

Gear Reduction Equation

This equation calculates the reduction in RPMs (R) from the engine engagement of the CVT (Engage or Dis) divided by the product of the CVT ratio (X) and the Transaxle Ratio (Y). This equation is used for the engagement RPM of the CVT and the max RPM of the engine.

$$Rpm_{LOW} = \frac{RPM_{Engage}}{X * Y}$$

Velocity Equation

This Equation calculates the Velocity from the RPMs. This is done by a unit conversion from RPMs to Miles per hour (Mph). The circumference (C) is multiplied by the conversion from inches to miles (1.5783E-5), the conversion from minutes to hours (60), and the RPMs (R) to the tire from the gear reduction equation.

$$V = 60 * 1.5783e - 5 * C * R$$

Torque Output

This Equation calculates the torque from the horsepower output of the engine at different RPMs (Engage of CVT and Max RPMs), which are found from the manufacturer. The torque (T) is found by dividing the product of the Horsepower (Hp) and a constant (63025) by the RPMs (R). This gives the torque in pound inches, which is then converted to lb-ft.

$$T = \frac{63025 * Hp}{R}$$

Torque Required for Motion

This equation calculates the torque required (TR) at the tires to move the vehicle from a static state. To find the torque, the weight is distributed per tire (W/4) and half of the diameter of the tire (d/2). This finds torque in pound inches, which is then converted to lb-ft.

$$TR = \frac{D}{2} * \frac{W}{4} * \frac{1}{12}$$

Code

The code requires 4 user inputs: The engage rpms of the CVT, the low CVT ratio, the High CVT ratio and the Transaxle ratio. The code will then output different torque values at specified locations and velocities. This code was tested for three different CVTs and two different transaxles. The three CVTs tested are the Comet 44 Series, Comet 780 series, and a Comet 790 series. The two transaxles tested are the Dana H-12 FNR and the SNPT M5101B.

The code can be seen in Appendix A.

Results

After seeing the results of the CVTs and transaxles ran through the code, the CVT series 780 and the Dana H-12 FNR has the closest torque output and max velocity to get the system to the target values. The projected max velocity is 35MPH, which is above the target value. The max torque output is 602 lb-ft, which will allow the team to accelerate quicker in tighter turns. Therefore, these will be the selected components for the system, as seen in Figures 16 & 17.

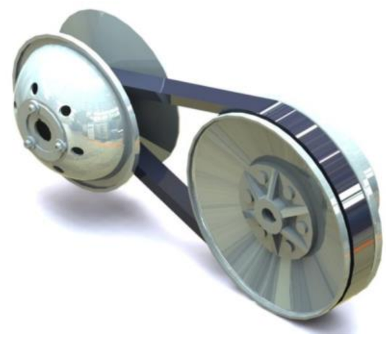

Figure 16: Comet 780 Series CVT

Figure 17: Dana H-12 FNR Transaxle

5.2.1.2 Engine Mounting Design and Analysis

Calculation Assumptions and Driving Equations

Because this analysis is looking at the system response of a vibrational system, calculations relied heavily on equations from Rao's Mechanical Vibrations 6th Edition [11]. The first step to analyze the system is to define exactly what type of vibration is occurring. Because the engine is a rotating unbalanced mass that is linked to a semi flexible plate and bushing assembly, the system was modeled as a vibration isolation system with a flexible foundation. The equivalent figure of the system can be seen in figure 1.

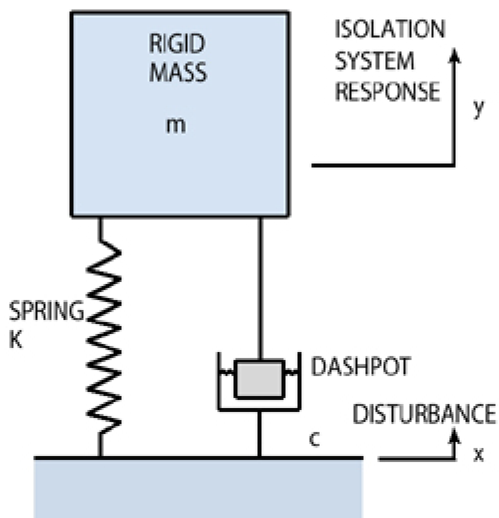

Figure 18: Vibration Isolation System w/ Flexible Foundation

The generated equations of motion for the system are as follows:

$$m_1\ddot{x}_1 + k(x_1 - x_2) = F_0 \cos(\omega t)$$
$$m_2\ddot{x}_2 - k(x_1 - x_2) = -x_2 Z(\omega)$$

Where m_1 is the mass of the engine and top mounting plate and m_2 is the mass of the 4 bushings and bottom frame plate. The displacements of the system x_1 and x_2 are the engine and bushing displacements, respectively. F_0 is the rotating amplitude force caused by the engine. This force is modeled as $F_0 = m_1 e\omega^2$.

In this equation, e is the rotating unbalance eccentricity of the engine. Lastly, in the equations of motion, k is the equivalent spring coefficient of the system as $k = 4k_b$ where k_b is the spring coefficient of a single bushing.
A list of variables, definitions, units, and any assumptions can be seen in Table 15:

Table 15: Vibration Analysis Variables

Variable	Definition	Units	Assumption
m_1	Engine Mass	lb	70
m_2	Bushing and Base Plate Mass	lb	10
x_1	Engine Displacement	in.	-
x_2	Bushing Displacement	in.	-
N	Engine Rotational Speed	RPM	[1750; 3800]
ω	Engine Firing Frequency	rad/s	-
e	Engine Eccentricity	in.	0.25
k_b	Bushing Spring Constant	lb/ft.	-
k	Equivalent System Spring Constant	lb/ft.	-
R	Force Reduction Percentage	%	-
ω_n	Bushing Natural Frequency	rad/s	-
r	Frequency Ratio	:	-
T_f	Transmitted Force Percentage	%	-
F_0	Engine Amplitude Force	lbf	-
c	Damping Constant from Dashpot	lb/(ft.-s^2)	0

Masses were all assumed based on known dimensions and material types. The rotational speeds represent the idle and max RPM that the engine can achieve. The eccentricity is estimated high to be conservative. Due to the bushings isolating the vibrations, there is no damping required, thus c = 0. This is because any damping in the system will have a negative effect on the bushings, causing the bushings to not operate as desired.
Another major assumption during calculations is that the engine's frequency acts as a 1-D harmonic motion. This allows the system to be analyzed with equations, rather than testing the engine. Therefore, $x_j = X_j \cos(\omega t)$ where X_j represents the mode shape amplitude of displacement seen in the system. The Z term of the second equation of motion is the mechanical impedance of the system, which is the overall displacement due to the forcing frequency.

Calculation Flow and Results

Estimate Frequency Ratio:

With a known RPM, the estimated firing frequency (ω) of a 4-stroke single cylinder engine [12] is

$$\omega = \frac{\pi N}{30}$$

In order to isolate the vibration of the engine from the frame, the frequency ratio (r) is calculated as

$$r = \frac{\omega}{\omega_n}$$

After the frequency ratio is known, it can then be used to estimate the force transmissibility (T_f) of the system at the given N values.

$$T_f = \frac{F_t}{F} = \frac{1}{r^2 - 1}$$

These equations and assumptions are then put into MATLAB [Please reference Appendix A for the code] over the range of engine RPM's and plotted to yield results seen in Figure 19:

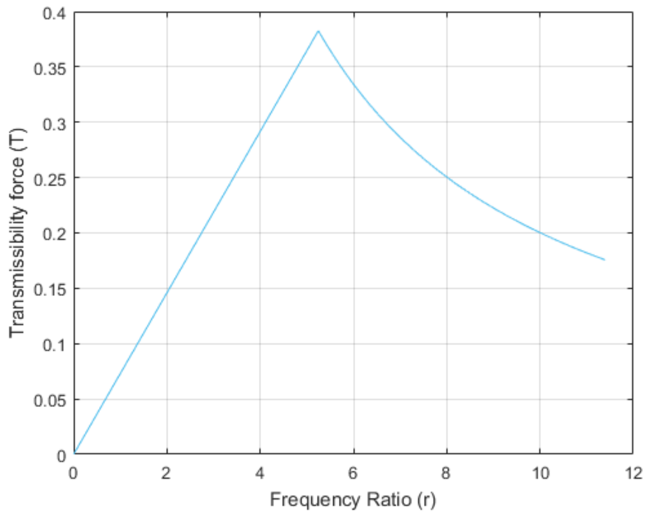

Figure 19: Transmissibility for Different Frequency Ratios

The results show that an optimal range for frequency ratios is $0 < r < 2 \text{ or } 10 < r < 12$

Transmissibility Equation:

With the equations of motion known, the system's natural frequencies can be modeled as the roots of the coefficients in the equations of motion. The coefficients ($Z(\omega)$) are put into matrix form as

$$Z(\omega) = \begin{vmatrix} (k - m_1\omega^2) & -k \\ -k & (k - m_2\omega^2) \end{vmatrix}$$

This results to:

$$\omega_1^2 = 0 \qquad \omega_2^2 = \frac{(m_1 + m_2)k}{m_1 m_2}$$

The first frequency represents the rigid body motion of the system, which is irrelevant to the isolation analysis. After solving the mechanical impedance of the system using matrix math and vibration concepts [11, 12], the modes are represented as:

$$X_1 = \frac{[k + Z(\omega)]F_0}{[Z(\omega)(k - m_1\omega^2) - km_1\omega^2]}$$

$$X_2 = \frac{kF_0}{[Z(\omega)(k - m_1\omega^2) - km_1\omega^2]}$$

By definition, the transmitted force (F_t) is equal to $X_2 Z(\omega)$.

$$\therefore F_t = \frac{kZ(\omega)F_0}{[Z(\omega)(k - m_1\omega^2) - km_1\omega^2]}$$

Transmissibility (T_f) of the isolators (Bushings) can then be defined as

$$T_f = \frac{F_t}{F_0} = \frac{kZ(\omega)}{[Z(\omega)(k - m_1\omega^2) - km_1\omega^2]}$$

The Spring constant of the system (k) can then be solved as

$$k = \frac{T_f Z(\omega) m_1 \omega^2}{Z(\omega) + T_f m_1 \omega^2 - T_f Z(\omega)}$$

By utilizing this equation in MATLAB, the optimal (average) solution for the spring rate of the system is

$$k = 9.65 \frac{kip}{ft.}$$
$$\therefore k_b = \frac{k}{4} \geq 2.413 \frac{kip}{ft.}$$

Engine Mounting Plate

The dimensions from base plate of the engine can be found on Briggs and Stratton's website [13] for a Model 20 engine. The plate will have 1.5-inch clearance from all edges to ensure a proper fit to the engine with the bushings. Therefore, the plate will be (11" X 7" X .25") Aluminum. Aluminum 1060 is the selected material because this will provide ample strength in the plate, while maintaining a relatively light weight for the vehicle.

Bushing Selection

With a system requirement of at least $2.413 \frac{kip}{ft.}$, bushings can now be looked at from various websites available. Due to limited resources, a bushing manufacturer with in depth spec sheets was selected as the vendor for the required bushings. Barry Controls [14] manufactures a low profile vibration isolator (Bushing) that can withstand the loads from the engine. Particularly, Barry Controls' Model 633A-200 Series has a spring constant of $2.53 \frac{kip}{ft.}$. The dimensions of the bushing are also such that they can fit within the base plate mounting assembly dimensions (11" X 7"). The model selected can be seen on the next page.

Barry Controls Model 633A-200 [14]:

Table 16: Bushing Specs

NATURAL FREQUENCY	8 Hertz
TRANSMISSIBILITY AT RESONANCE	8:1
RESILIENT ELEMENT	Neoprene
STANDARD MATERIALS	Cold Rolled Steel or Stainless Steel
WEIGHT	633A-60 through 130 = 7 oz. 633A-200 and 260 = 16 oz.

Figure 20: Model 633A

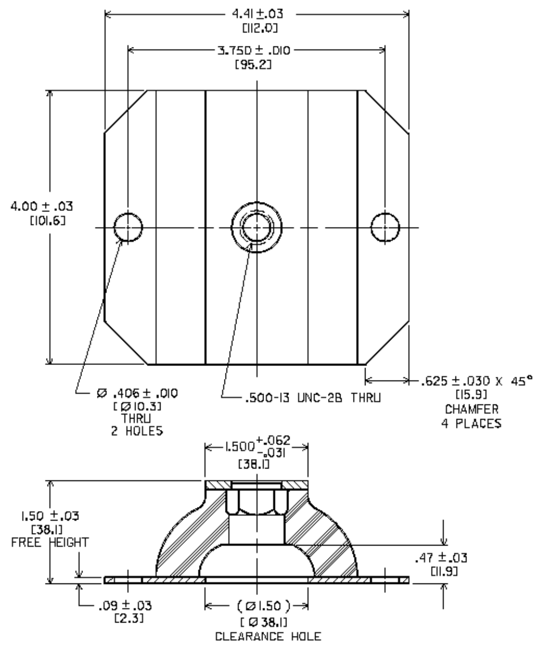

Figure 21: Model 633A Drawing

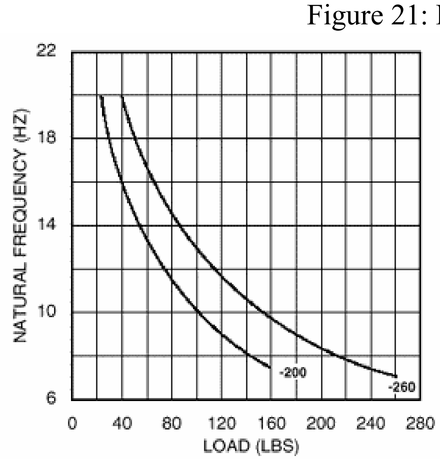

Figure 22: Model 633A Load to Frequency Curve

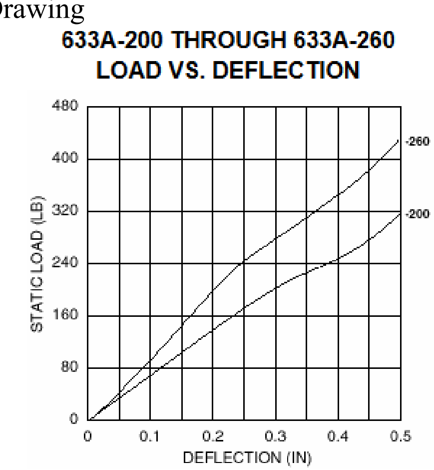

Figure 23: Model 633A Load - Deflection Curve

Influence on Project

The selection of the 633A bushing will have a bigger impact on the overall weight of the vehicle than desired; however, in order to optimize the operation of the isolation system, a higher bushing weight is required, as seen in the calculation equations. The weight of the 633A-200 model is 1 pound. This gives an overall weight of 4 pounds added to the system. That being said, these bushings will have a noticeable effect on limiting any resonance and RPM change effects of the engine to the rest of the vehicle. This is extremely important due to the potential of the vibrations of the engine to shear and compress many components throughout the vehicle (Frame, Suspension, Hardware). Now that the bushings and mounting plates are designated. The next design step for the Drivetrain team is to select correct hardware for the system based on engineering design standards set by SAE. In addition to hardware design, the Drivetrain packaging assembly needs to be created.

SolidWorks Model

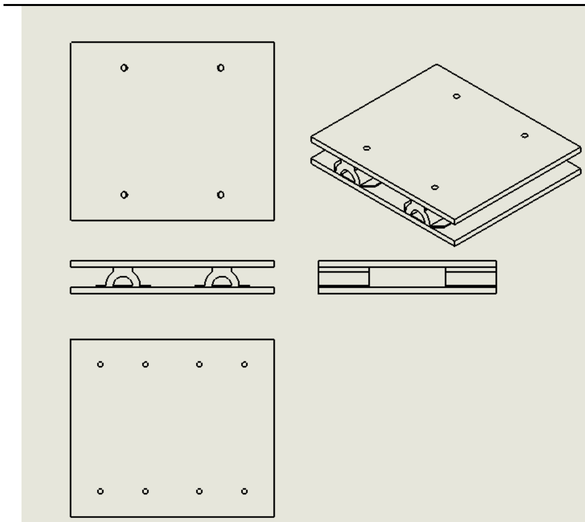

Figure 24: Engine Mount Assembly Drawing

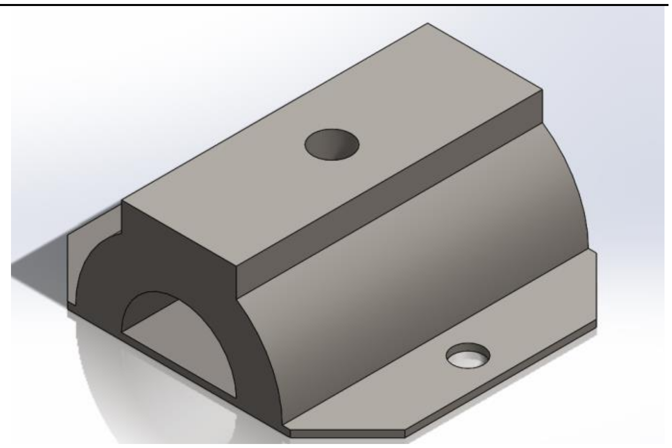

Figure 25: Model 633A (Missing Damper)

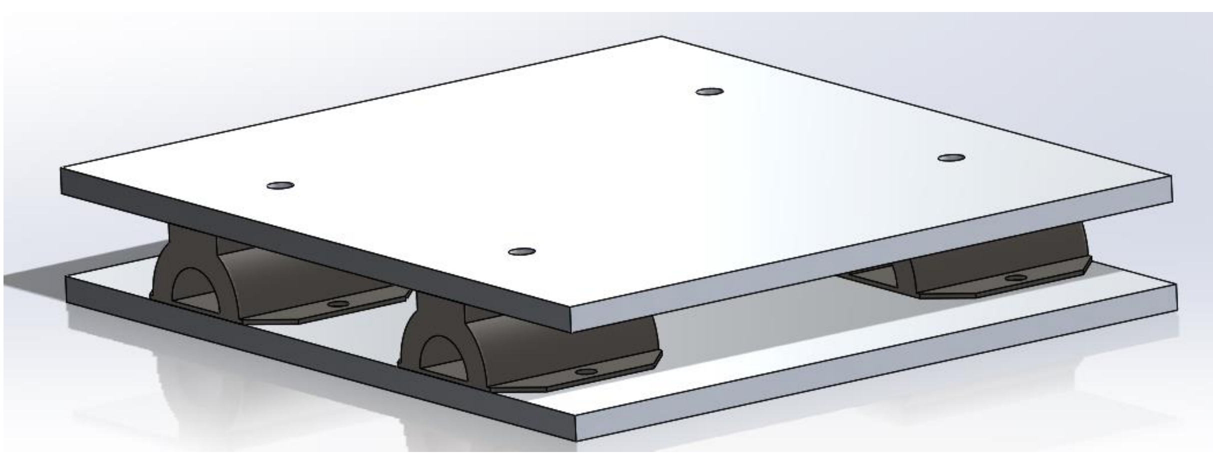

Figure 26: Engine Mount Assembly

All dimensions are mentioned in the paper. The Bushing dimensions are seen in the schematics.
***NOTE** The neoprene isolator component of the bushing is not integrated into the assembly. (Reference

Bushing Schematic)

5.2.2 Frame Analysis

Designs that Drive Calculations

Tube Selection

The frame of the SAE Baja car is specified to be made from a circular steel thin wall tubing. Competition rules regulate the primary tubes and must meet or exceed a bending stiffness and strength of 1" OD x .120" wall tube with a carbon content of at least 0.18% and may not have a wall thickness of less than 0.065" or an outer diameter of less than 1". The secondary members must have at least an outer diameter of at least 1" as well and a wall thickness of 0.035" at minimum.

Drawn over Mandrel vs. 4130 Chromoly

The frame tubing must be made of steel and teams at competition will generally use one of two types of materials. Drawn over Mandrel (DOM) and 4130 Chromoly tubing each fit the material requirements with the correctly selected tube dimensions. Though the materials have a very similar density and are assumed to have the same Modulus of elasticity of 205 GPa, their tensile yield strength and manufacturability greatly differentiate. The tensile yield strength was found to be 370 MPa for DOM tube and 460 MPa for 4130 Chromoly. This variance in Yield Strength gives the 4130 a much larger bending strength when comparing to the same size tube of DOM. When manufacturing the vehicle, DOM tube has the option of being MIG or TIG welded while 4130 Chromoly must be TIG welded and heat treated. MIG welding although it adds additional weight to the frame, can be done over a shorter period of time without needing the welding expertise that TIG requires. In the event that a member must be welded at competition, MIG welding to repair would be much preferred over having to TIG weld a material.

Bending Stiffness and Strength

Competition rules define the bending stiffness proportional to the equation: EI where 'E' represents the Modulus of elasticity and 'I' represents the second moment of area for the structural cross section. The bending strength of the material is given by: $S_y I/c$ where S_y represents the yield strength and 'c' represents the distance from the neutral axis to extreme fiber. From the baseline material in competition rules we can calculate that the chosen primary material must meet or exceed a bending stiffness of 583,306 lb-in^2 and a bending strength of 2,397 lb-in.

Final Tube Selection and Purchasing

After many iterations involving different material, tube outer diameter and wall thickness, three optimal choices were left. Of those three the DOM 1.375" OD x .065" wall tube was the team's favorite for being light weight, having a high bending stiffness and a large diameter to provide more surface area to tabs welded on. Although the material is not available through Industrial Metal Supply who has previously sponsored the project, several other companies were found to supply the desired primary and secondary member tubing. Of those other companies, Totten Tubes was the only provider with a warehouse in Arizona. The team has yet to reach out to Totten Tubes as we are finalizing the frame to estimate overall tube length needed and will be in hopes to acquire another local sponsor as Industrial Metal Supply has in the past. We have gathered rough estimates from online companies to compare pricing when we do reach out to Totten Tubes. Alro metals can provide 72 feed to primary and secondary material for $206 and $114 respectfully with $100 in shipping costs for a total of $420. Another estimate from Online Metals who only carried the primary member tubing would charge $234 before shipping for the same 72 feet. Glendale Steel Supply is another local provider in Arizona however they only carry the desired material for the primary members.

Frame Design

Nose

In designing the front section of the frame, the team did benchmarking on previous competition winners including University of Michigan and Cornell University. Both of these teams lacked a "nose" section of frame (Defined by points G and E of the official SAE Baja rules). It was noted that these teams were still able to fit all of their required systems (I.e. brakes and suspension) within the reduced front end. The team decided to create a design that lacked a nose based on this benchmarking with the fact that no nose would mean less tube required for the frame and thus less weight.

Under Seat Member

The floor of the cockpit is composed of a set of members called the "Under seat members." The team brainstormed a couple different designs with the most promising geometries displayed below in Figure 27. The left layout incorporates a cross design that was both proposed by SAE Baja rules and was found in NAU's old Baja car. The right layout features a "Kite" layout that has become favored after communicating with the Suspension team. Suspension is planning on integrating a radius arm and trailing arm converging at approximately the same point in the middle of the frame. The kite layout would allow for multiple beams to converge on a single point where that suspension arm stress concentration would be. Finite element analysis (FEA) iterations detailed in later sections hope to validate what geometry is needed for the under seat members.

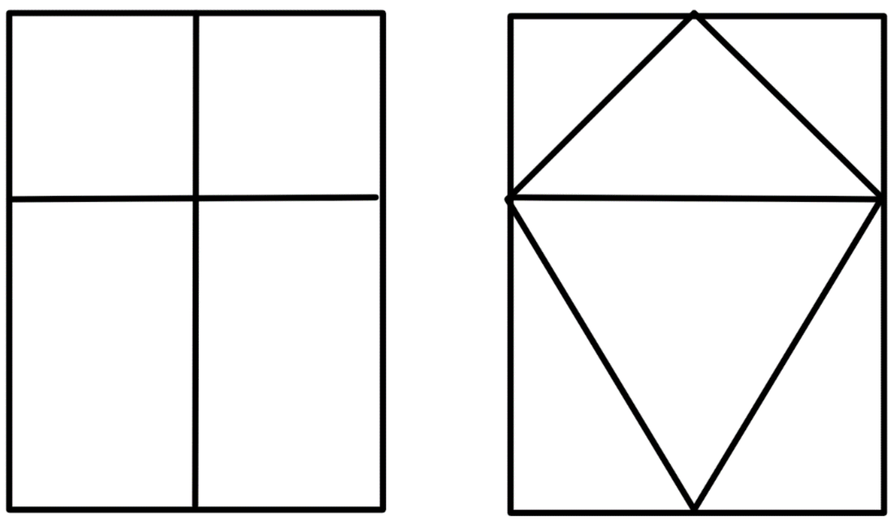

Figure 27: Under Seat Member Layouts

Rear Roll Hoop (RRH)

The rear roll hoop composes a structural panel located behind the driver, or the end of the cockpit. Research of various Baja teams revealed that a wide range of RRH designs; however, the three designs in Figure 28 are the most reoccurring. The left design is proposed by the SAE Baja rules while the middle has the simplest geometry and the right was used by Michigan and Cornell University's team. Currently, the team plans on using the left design since it allows for the frame to bow out where the side impact members (SIM) will connect creating a wider "Tub" for easier access to and from the vehicle. The chosen geometry also uses less tube and therefore weighs less than the rightmost design.

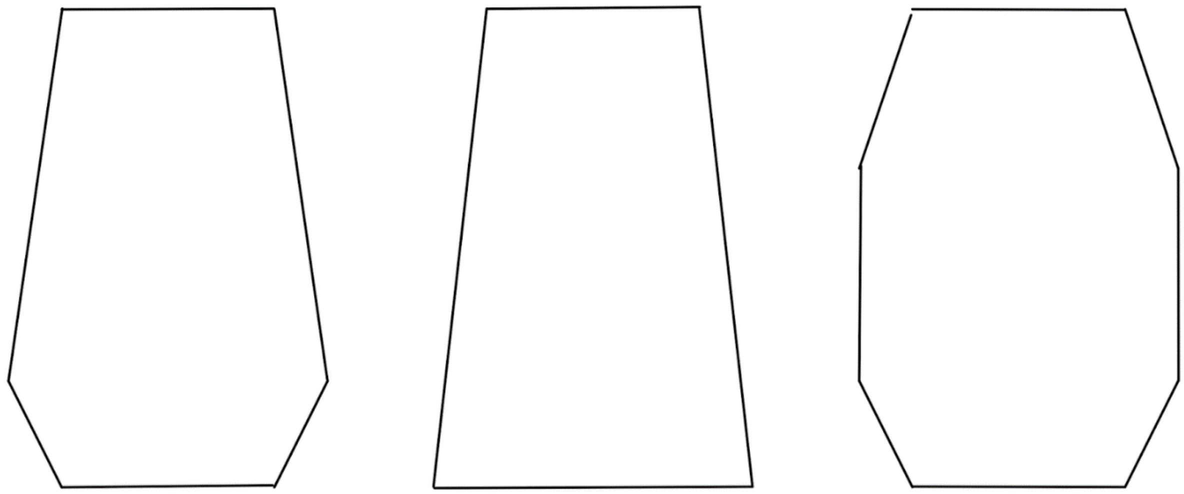

Figure 28: RRH Design Options

Fore/Aft Bracing Members
These members are located behind the rear roll hoop to hold the drivetrain and rear suspension components. These are specified as secondary members; however, FEA testing will decide whether the loading forces of the suspension will exceed the capability of the weaker material. The lateral positioning of these members have not been finalized as they are dependent of the transaxle width and final positioning. The overall length of these members have been closely estimated as the suspension team closes in on the desired length of the trailing arm.

Side Impact Member, Lower Frame Side Member and Diagonals
The side impact members (SIMs) will define the top of the “Tub” of the cockpit and run from the RRH to the front of the vehicle. Initially the team planned on straight SIMs but benchmarking showed that many teams incorporated SIM members with one or multiple bends. The team analyzed this idea and deduced that the most likely reason for such a design was to widen the tub of the cockpit for easier exit of the vehicle. Our SIM members will incorporate a bend to widen the tub and will be angled in such a fashion that the bend will be the lowest point of tub. SIM members must be between 8 and 14-inches above the bottom of the driver's seat. Having them oriented in such a way that a low point is created at mid-thigh of a seated driver will make it that much easier for the driver to exit the vehicle in under the five seconds required.

The SIM members are reinforced by some secondary member diagonals which transfer force from the SIMs to the lower frame side members (LFS). These diagonals are oriented so that they point up and converge at the bend in the SIM. The team considered inverting these diagonals to point downward and reinforce the node for the radius and trailing arms but that would have defeated the purpose of the bend in the SIM member since they would have constrained the inside of the cockpit tub. The FEA detailed later incorporates iterations of the frame with and without a vertical support that would run perpendicular to the LFS from the suspension arm node and would meet the SIM member at the bend. The possible addition of this member would allow the team to keep the diagonals oriented pointing up while reinforcing the suspension node on the LFS.

Gussets and Maximum Length
As specified in the competition rules the maximum length of any straight member must not exceed 40”. As the team approached the final model of the frame, it was realized that two members may surpass the

maximum length. One concern is that the Roll Hoop Overhead Members (RHO) passes longitudinally above to the left and right side of the driver. If the design is unable to keep this length within regulation, a gusset will need to be added. If the RHO is within the correct length a FEA analysis will be done with and without the gusset to evaluate the additional strength provided by the gusset at the cost of the additional weight.

The other members showing a possibility of extending longer than 40” are the Front Bracing Members (FBM). These extend from the front of the RHO down to the nose of the car. If these members are longer than regulation the nose of the car will need to angle upwards, which would increase the difficulty in manufacturing the frame and may cause conflicts with the Suspension team.

Finite Element Analysis

The model of the frame has been made with SolidWorks program, which is the program the team will be utilizing for the FEA analysis. This analysis allows the team to apply a force, similar to what can be expected during the most extreme of conditions, to the frame. The results will indicate where the greatest stress, strain and displacement occur. With this, the team is able to iterate different designs to optimize the strongest and most durable frame.

Preliminary Results

The testing of an older frame design was done to demonstrate how FEA works and learn about the results from the simulation. For consistency purposes a force of 1000N was applied in each simulation. The loading, fixture and meshing of the part was applied to best replicate actual forces foreseen in testing and competition situations.

When testing a front end collision, the top and bottom of the RRH were fixed into place as the force was applied to the front lateral members as well as the base of the RRH where the radius arm would attach. The maximum displacement in this testing was 0.5mm from the 1000N force applied. The results of the FEA can be seen in Figures 29 through 32 below.

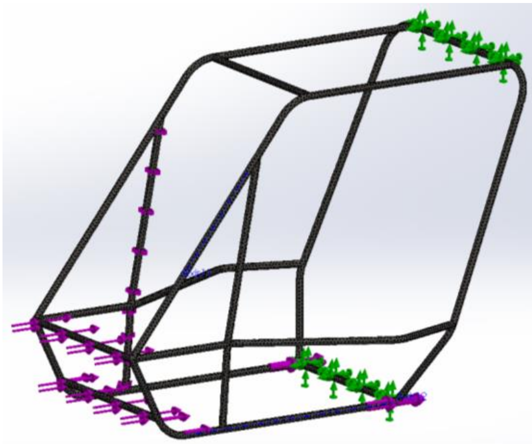

Figure 29: Frame Mesh

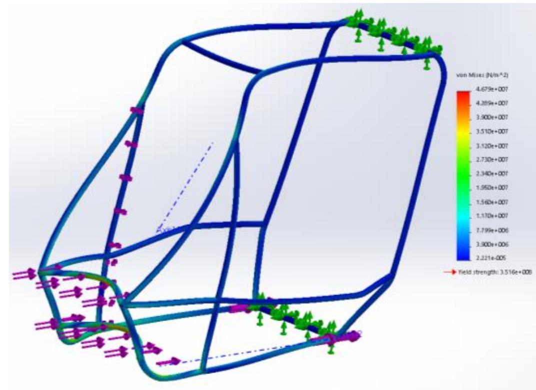

Figure 30: Von Mises Failure

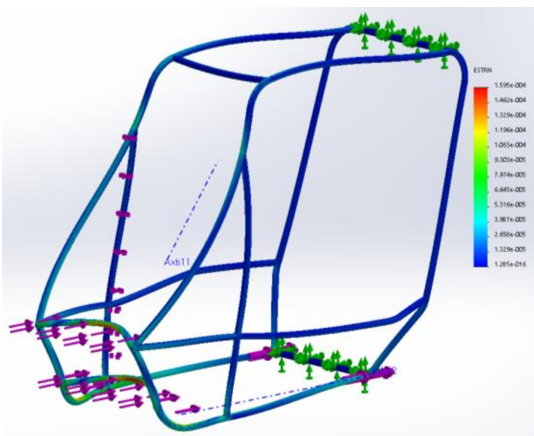

Figure 31: Experienced Strain

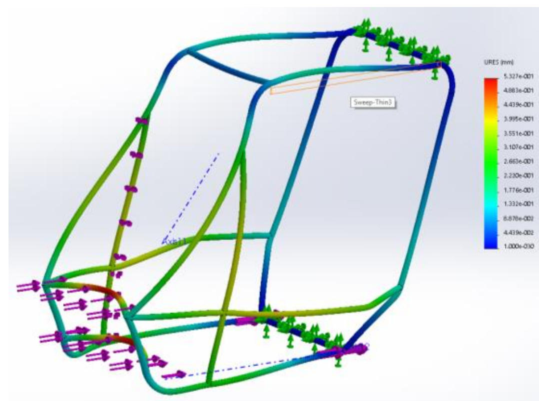

Figure 32: Displacement

Rear collision testing of the frame was done choosing force points that the rear FAB members would be in contact with. Figure 33 shows the highest stress occurring at the bends in the SIM member. The resulting maximum deformation from this analysis was 1.24mm as seen in Figure 34.

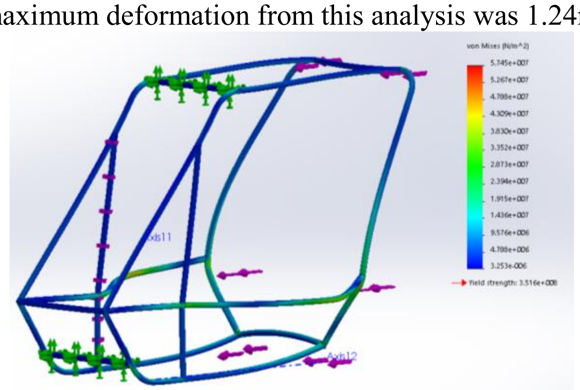

Figure 33: Rear Collision Failure

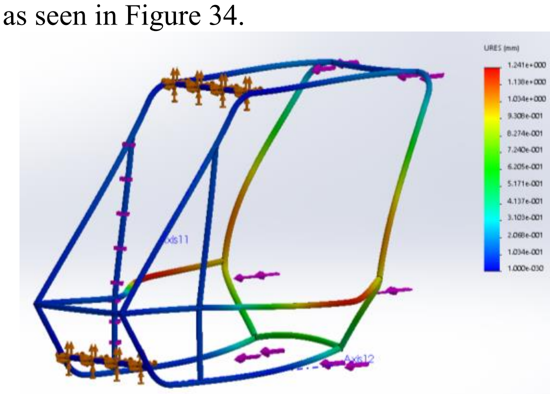

Figure 34: Displacement for Rear Collision

Side impact loading on the car was done at both the SIM and LFS members. With a maximum displacement of 9mm shown in Figure 35 it is obvious where the deflection would take place at. The Von Mises stresses in Figure 36 illustrate the maximum occurrence and states the yield strength that SolidWorks assumes.

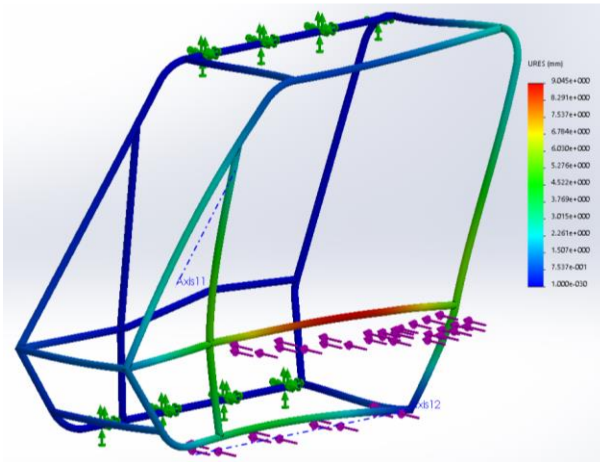

Figure 35: Side Impact Displacement

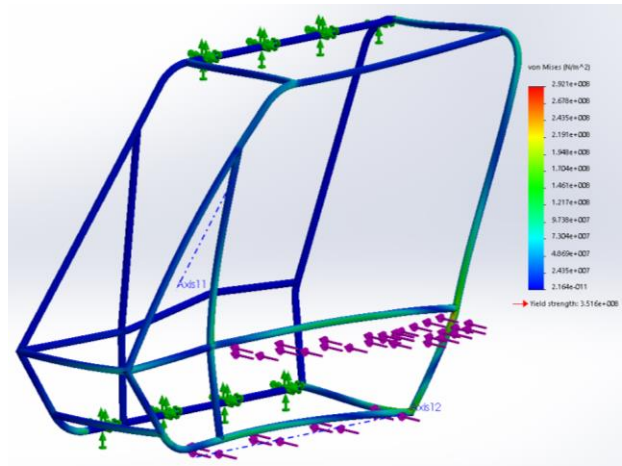

Figure 36: Side Impact Failure

Lastly the top loading analysis is done to predict a roll over situation. The maximum displacement in this simulation was larger than any of the other tests at 9.7mm in the center of the RHO gathered from Figure 37. In Figure 38 it is noticed that higher stresses occur at the FBM to RHO joint and the joints above the RRH.

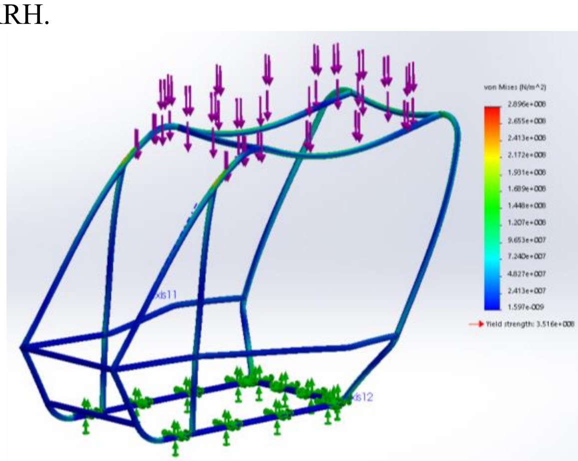

Figure 37: Top Loading Failure

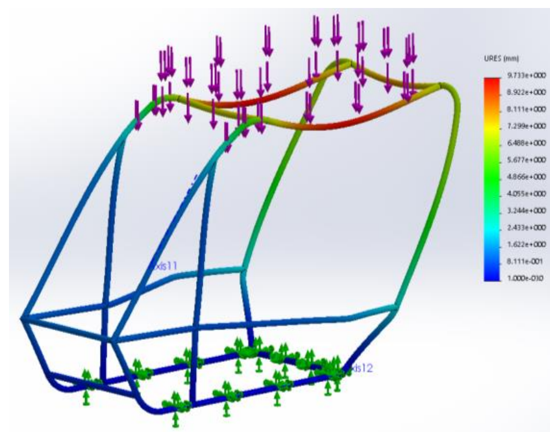

Figure 38: Top Loading Displacement

5.2.3 Front Suspension Analysis

Frame Mounting and Fixed Values

Front suspension geometry is determined by analyzing the locations of the pivot points and wheels relative to the chassis. Table 17 lists variables and lengths that are either constraints, choices, or assumptions. These values will be explained as they appear in the design report.

Table 17: Suspension Geometry

Variable	Description	Value	Note
l1	LFS lower frame rail width	22"	As of current frame model
l2	Long distance, front bar to rear mount	40"	As of current frame model
Bend	Bend radius	4.5"	Available Tooling
TW	Target TW	60"	Design goal, limited to 64" by rules
Yu	Taget Uptravel	6"	Design Goal
Yd	Target Downtravel	4"	Design Goal
Lrh	Target frame height at RH	14"	Design Goal
lwms	Distance between WMS	54"	TW-2*((wt-ww)/2+wwo)
wt	Width of tire	7"	Tire size: 22x7 - 10
ww	Width of wheel	5"	Wheel size: 10x5
wwo	Width of wheel outside WMS	2"	Wheel Offset: 3+2
dt	Diameter of tire	22"	Tire size: 22x7 - 10
rta	Adjusted tire radius	10"	Accounts for Deformation
Y	Wheel Travel	Y = 0 @ RH	Uptravel Negative, Downtravel Positive

A Twin-I Beam suspension system consists of an I-beam locating the wheel laterally, and a Radius Arm locating the wheel longitudinally. Both of these links pivot at the frame and are fixed at the wheel upright. Mounting locations on the frame are to be placed along the lower frame side (LFS) members. These members are aligned longitudinally, and make the lowest plane of the chassis. The projected height to these frame rails is 14”. They terminate at the front lower lateral member (LLM) at the front of the chassis.

As seen in Figure 39, both I-beam and Radius Arm mounts are located at the opposite corner 2” below the bottom of the LFS members for clearance, resulting in a pivot height of 12” to be used later. The value l1, lower frame rail width, defines the I-beam mounts as 22” apart. The value l2 is the longitudinal distance between I-beam and Radius Arm pivots on the same side, and is 40”. Note that I-beam mounts may be staggered axially for arm clearance.

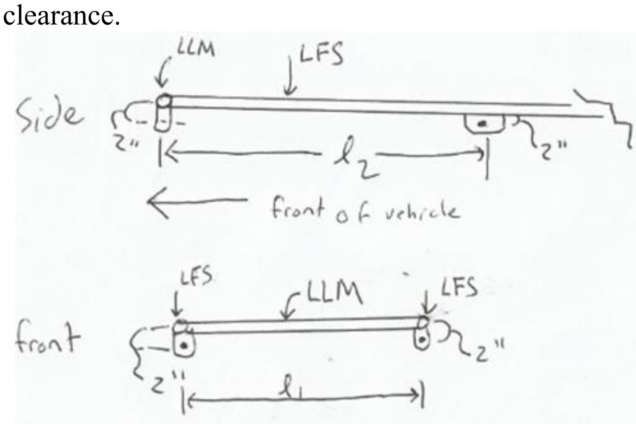

Figure 39: I-beam and Radius Arm Mounts

Suspension Geometry

In order to develop a dynamic model of the system, a geometric model must be created. Suspension links are simulated as simple straight radii connecting the frame mounting point to the point where the wheel axis intersects the wheel mounting surface plane. The distance between wheel mounting surfaces will be used as the limiting width value.

Based on the dimensions to this point, the geometric model is shown in Figures 40, 41 and 42. Point A is the I-beam pivot, Point B is the Radius Arm pivot, and point C is the wheel location point as defined in the previous paragraph. Simple geometric analysis was used to find values shown in Figures 40, 41 and 42, which are tabulated in Table 18.

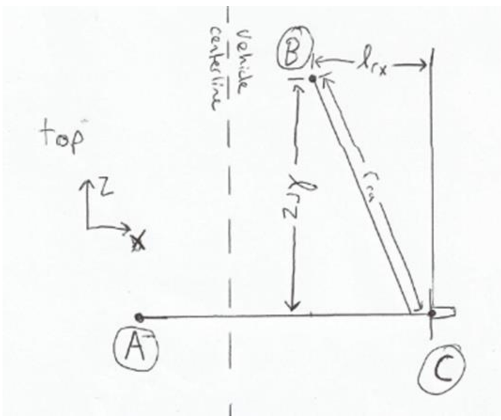

Figure 40: Dimensioning A

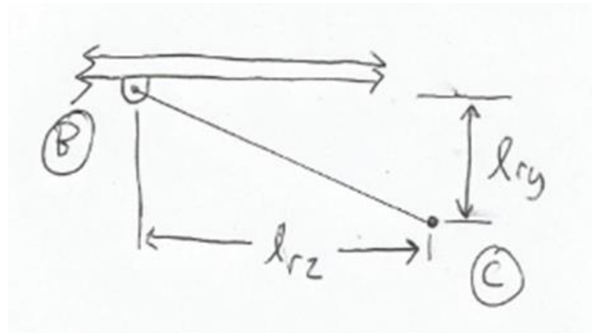

Figure 41: Dimensioning B

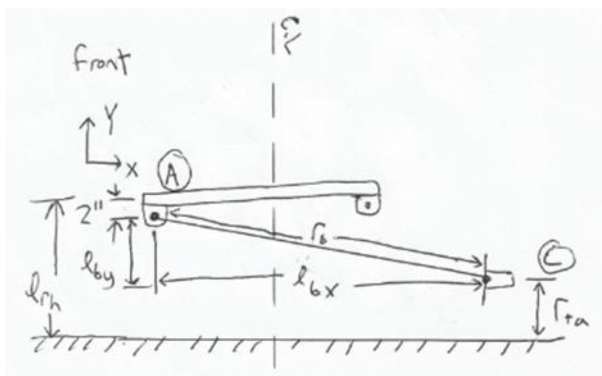

Figure 42: Dimensioning C

The final arm geometry is tabulated in Table 18 and the proposed front and side views are Figures 43 and 44, respectively.

Table 18: Final Arm Geometry

Variable	Description	Value	Note
hm	Height of pivot	12"	lrh-2"
lbx	Width of beam, horizontal	38"	lwms-(lwms-l1)/l2
lby	Drop of beam, vertical	2"	hm-rta
rb	Dynamic radius of beam	38"	magnitude (lbx,lby)
lrx	Width change RA	12"	lbx-l1-4"
lry	RA Drop	2"	hm-rta
lrz	Long. Length of RA	40"	l2
rra	Dynamic Radius RA	41.81"	magnitude(lrx,lry,lrz)

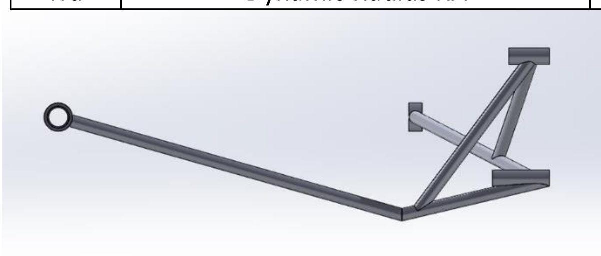

Figure 43:Arm Front View

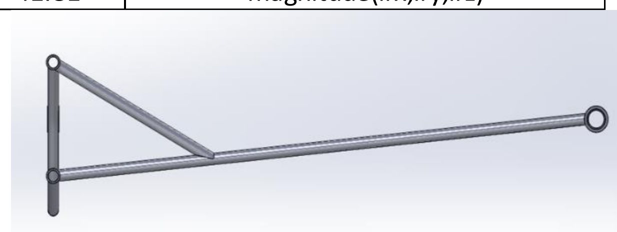

Figure 44: Arm Side View

Dynamics of Front Suspension

The dynamics of this front suspension primarily consists of changes in two angles with wheel travel. These are the camber and caster angles. Camber is the angle between the amount the top of the wheel is leaned into or away from the body and the vertical, and is noted here as alpha. Caster is the angle between the steering axis of the wheel and the vertical, and is noted as beta.

Dynamic models of the system are shown in Figures 45 and 46. Due to the simplicity of the system, camber change is related only to the position of the I-beam, and caster change is related only to the position of the radius arm. Theta is used to define position of the I-beam, then is corrected to find alpha. The same is true of phi and beta. Vertical travel is defined as the projection of the end movement of the arm onto the vertical, and defined as Y. The equations developed for this system are as follows:

$$\theta(Y) = Y\frac{1}{\cos\theta_0 * r_b} + \theta_0 \qquad \theta_0 = \tan^{-1}\frac{l_{by}}{l_{bx}}$$

$$\alpha(Y) = \theta(Y) + \alpha_0 - \theta_0 \qquad \emptyset_0 = \tan^{-1}\frac{l_{ry}}{l_{rz}}$$

$$\emptyset(Y) = Y\frac{1}{\cos\emptyset_0 * r_{ra}} + \emptyset_0 \qquad \alpha \equiv \text{Camber Angle}$$

$$\beta(Y) = \beta_0 - \emptyset(Y) + \emptyset_0 \qquad \beta \equiv \text{Caster Angle}$$

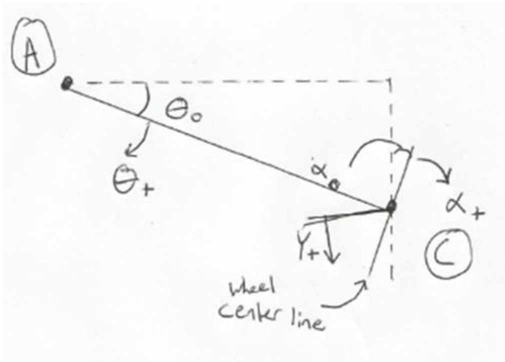

Figure 45: Dynamic Model A

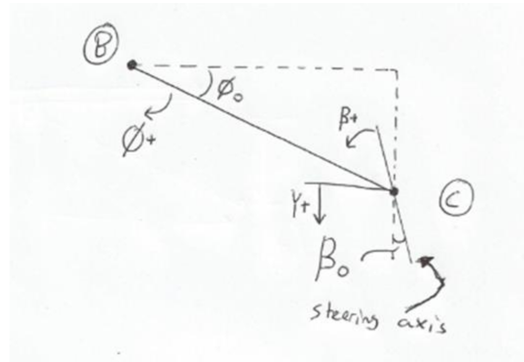

Figure 46: Dynamic Model B

Ride height values of caster and camber are required to find the same as a function of wheel travel. A MATLAB computer code was used in an iterative process to select these initial values, and is attached as Appendix A at the end of this document. Decisions were made primarily based on the bump travel (-Y) portion of the charts, due to the unloaded nature of the system in droop. 2* of camber was chosen at ride height, in order to slightly reduce negative camber at full bump. A relatively small value of 4* of caster at ride height was selected, in order to minimize steering effort on flat ground and allow for higher caster values and self-correction at full bump.

The final camber and caster curves over wheel travel are shown in Figure 47, respectively. Note that Y ranges from -8" to 8" which are greater than the targeted bump and droop travel values.

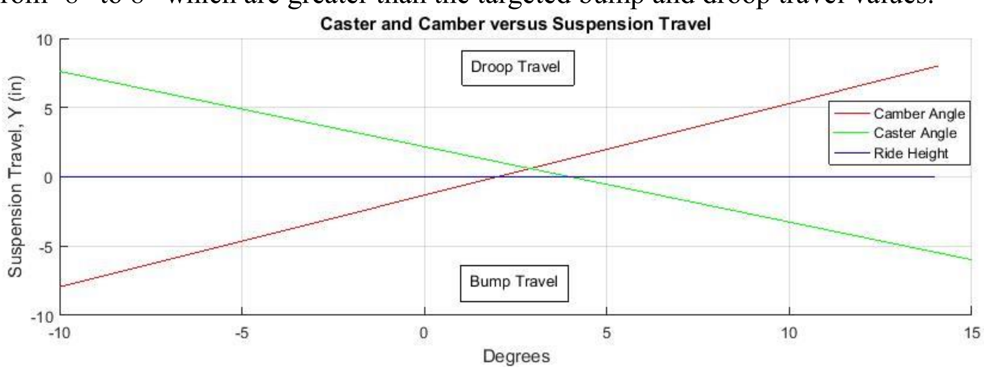

Figure 47: Camber - Caster Curves over Wheel Travel

Steering Geometry

The swing set steering system to accompany the Twin-I Beam suspension system can be evaluated symbolically. Figure 48 shows the layout of the swing set from the front view, with notation of lengths and directions. The direction of motion of the rack should be the same as that of a typical rear-steer direct rack and pinion, because the switch to front steer and the directional change of the swing set will offset. Table 19 lists the values involved in the steering system.

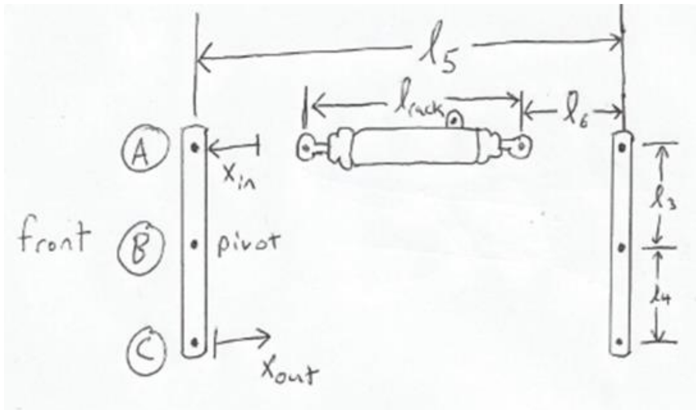

Figure 48: Swing Set Steering Model

Table 19: Swing Set Steering Analysis Values

Variable	Description	Value	Note
l3	Distance from point A to point B	4"	Arbitrary
l4	Distance from point B to point C	4.4"	Relate l4/l3
l5	Distance between pivots (B)	22"	Determined by mounting location
l6	Distance between rack eye and A	4"	.5*(l5-lrack)
lrack	Length of rack, eye to eye	14"	QDS Rack
l4/l3	Required ratio for chosen rack	1.0996	xout/xin
Thetak	Desired degrees of turn at spindle	45deg	Standard
rk	Radius of steering knuckle	3.5"	Approximated from similar parts
xout	Required lateral tie rod displacement	2.75"	Thetak*rk
xin	Rack - lateral displacement, center to lock	2.5"	QDS Rack
N	Rack - number of turns, center to lock	0.75	QDS Rack
sr	Steering ratio deg@SW/deg@spindle	6:1	N*2*pi/Thetak

Note that Xin and N are selected through rack choice, and that steering ratio, (l4/l3), and l6 are dependent upon rack choice. A MATLAB code was created to analyze the system and report dependent values based on Xin and N inputs. This code is attached in Appendix A. The results of this code for the racks considered are found in Table 20 below. These values will be analyzed in the dynamic section.

Table 20: Rack Design Options

Rack	N (Turns)	xin	lrack	l4/l3	sr
C42-336	0.5625	1.5	8.5	1.8326	4.5
C42-339	0.875	2.3125	11.25	1.1887	7
C42-340	0.875	2.3125	11.25	1.1887	7
C42-341	0.875	2.3125	11.25	1.1887	7
C42-344	1.125	2.3125	11.25	1.1887	9
C42-348	1.375	2.3125	11.25	1.1887	11
Deser Karts	0.75	1.125	11	2.4435	6
EMPI 3157	0.75	1.25	14-3/16	2.1991	6
QDS	0.75	2.5	14	1.0996	6

Bump Steer

The geometry of the system relative to the chassis was determined by bump steer considerations. Bump steer occurs when the axis of the tie rod does not pass through the instant center of the arm, drawn in the x-y plane (where z is the longitudinal axis). In the case of Twin-I Beam suspensions, the instant center (in the x-y plane) is always located at the pivot of the I-beam. Therefore, in order to eliminate bump steer, the pivot of the tie rod must be located at the same point as the I-beam pivot in the same plane.
This defines the mounting location – when the wheels are straight, point C must be 22" (l1) apart, and 12" (hm) from the ground. A front view, which illustrates the location of tie rod pivots, is in Figure 49

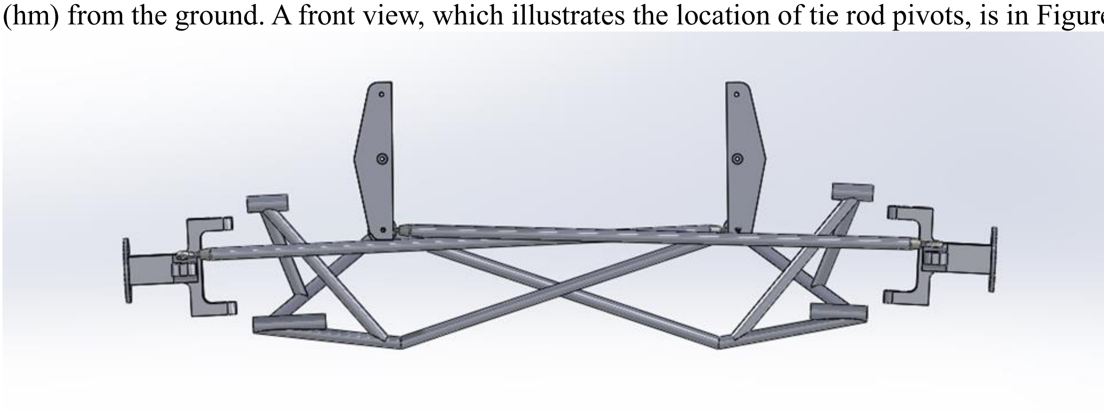

Figure 49: Tie Rod and Pivot Positions

Dynamic Steering Considerations

After analyzing bump steer, the remaining dynamic considerations for the steering system are to minimize steering effort while minimizing steering ratios. Steering effort is controlled by many variables including caster, trail, kingpin inclination, knuckle radius, tire choice, and tire pressure, among others. As of this point, steering effort has already been considered in caster angles as mentioned earlier in this report, and by the selection of wheel offset to reduce trail. Other values such as kingpin inclination and knuckle radius are determined by spindle choice.
Rack choice effects steering ratio as well as (l4/l3). Minimizing steering ratio is a design goal, but can have undesirable effects on steering effort. (l4/l3) ratios determine the mechanical advantage of the tie rod over the rack. This can be seen in Figure 50 with the associated equation. The actual swinger design is pictured in Figure 51

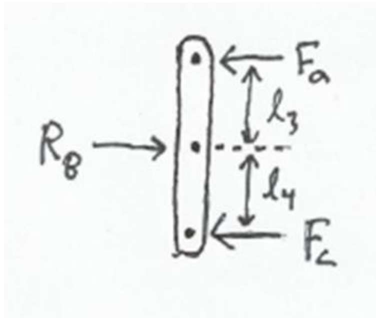

Figure 50: Swing Arm

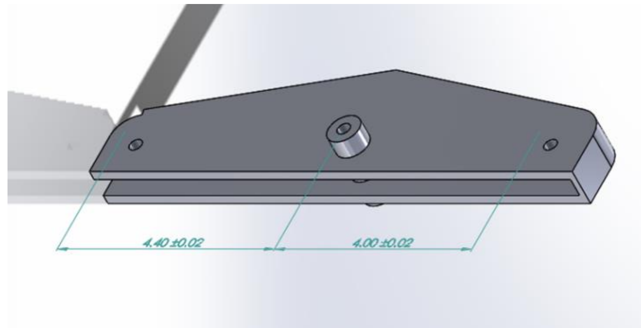

Figure 51: Swing Arm Model

The lateral displacement of the tie rod required by steering angle and knuckle radius is 2.75”. This is larger than any rack displacement available, meaning the tie rod will necessarily have a mechanical advantage over the rack. Comparison of the rack and pinions available has led to the selection of the QDS rack, which has an (l4/l3) of about 1.1 (the lowest available) while maintaining a relatively low steering ratio of 6:1. An initial value of l3 = 4” was chosen, pending packaging considerations.

5.2.4 Rear Suspension Analysis

Set-Up and Assumptions

A MATLAB code [Reference Appendix A] was used to plot the geometry of the frame including the rear differential output locations; that was then treated as a reference to establish the suspension geometry. Two types of plots were created in 2-D, two side views portraying the trailing arm (Reference Figures A-1 and A-2 in Appendix A) and two rear views illustrating the camber arms and CV axles (Reference Figures A-3 and A-4 in Appendix A). The trailing arm length is assumed to be between 36 and 40 inches to keep the wheel base near the team goal of 80 inches. Initially, the camber arms were assumed to be equal length. This was to keep the upper and lower arm at the same radius, keeping the hub constantly perpendicular to the ground. CV axles were assumed to be connected between the differential outputs and the hub. The max CV angle was assumed to be 45 degrees because this is a common maximum angle for CV joints. The code allows the user to input the trailing arm length, the lower camber arm length, and the vertical distance between the camber arms, which will be the hub length. Each input can be varied to determine advantages and disadvantages of different lengths. This will help the team determine proper CV axles, camber arm lengths, and a trailing arm length as well as make any changes in the future if needed.

Results

After running the code, it was acknowledged that the mounting point of each upper camber arm changes in the x-direction due to the uprights being slanted on the frame. In turn, the code outputs a calculated upper control arm length to keep the radius constant along rotation. It also outputs the total vertical movement of each member and the change in x-direction with respect to the trailing arm. The total track width and the ride height are currently being added into the program and are not yet functional. Due to constant changes in the geometry of the frame and other assumptions as the team finalizes designs, a final length for camber arms and CV axles was not chosen. Instead, the code was run with various lengths to determine a range of best options. An example of a successful result can be seen in Appendix A.

Conclusion

The lower camber arm needs to be a minimum of 13 inches to achieve 10 inches of vertical travel with maximum CV angle of 45 degrees, but will most likely be much longer than this to avoid reaching max angles. Assuming a trailing arm length of 36 inches is valid because it only changes a maximum of half an inch in the x-direction, which does not harm the CV joints. Anything longer than 36 inches will only decrease the change in x. Ride height and track width are still being added to the code and will be a part of the final report. Tire widths and diameters as well as frame dimensions need to be finalized before final components can be calculated and chosen.

6 Proposed Design

After completing the Bill of Materials and updating the budget, the team is still within the provided funds. Expenses are reaching the budget limit and will require the team to reach out to a local restaurant or industry for a sponsorship, which the team has already been discussing with the SAE Club. Now that the BOM and budget is more detailed (As seen in Table B-3 of Appendix B), the team can proceed with writing the budget proposal. In addition, we are on track to start ordering parts and materials before the end of the semester. This will allow the team to start building over winter break or the first day spring semester begins.

Building will take place in the NAU machine shop (Building 98C) and will provide the necessary tools for assembling parts. The team will also be providing consumable materials (Please Reference Table B-3 in Appendix B) such as welding wire, anti-seize, etc. for the manufacturing process.

The implementation process beginning over winter break will first involve solidifying the final design. This includes a full assembly down to nuts and bolts, along with correct tolerances and tolerance stacking. Once the design has GD&T and is ready to purchase components. Vendors for components of the design can be seen in detail in the BOM. There is an overall expected lead time of 1 month to receive all parts. Therefore, the team will begin purchasing in early December. Once all of the components are received, the team is now ready to begin manufacturing.

The first component of the vehicle to be fabricated is the frame. This will involve projecting a 1:1 image of the CAD file to a flat board for reference will bending the tubing to within tolerances. Once all of the tube is bent to spec, the team will begin to weld the tubing together as designed. In addition to this, all welding tickets will be created and sent to SAE for weld approvals. Once the frame is complete, the team will then begin the fabrication and assembly of the suspension.

During the fabrication and integration of the other sub-assemblies, the Frame Team will be installing the necessary accessories and hardware for the vehicle. This includes the tow points, safety harnesses, kill switches, pedals, plating, paint, stickers, etc..

Suspension will be a combination of in-house manufactured components coupled with prefabbed components as well. This will involve milling, lathing, and utilizing the CNC machine to cut the material as desired to spec. Once the components are received and fabricated, the suspension can now be assembled. Once the suspension is assembled, the team will integrate it into the frame.

Once the suspension is integrated into the frame, the drivetrain packaging will be assembled and integrated. The framing for the drivetrain packaging would have been completed during the frame fabrication period. Once the framing and base plates are fabricated, the assembly can be easily coupled through prefabricated components (Engine, Transaxle, CVT, Isolators). From there, the entire vehicle assembly is now complete.

The final step in the design process is to test and fine tune the final build up to competition. This will be done by pushing the vehicle to its absolute limits, and studying any failure modes that may occur.

The Gantt Chart for the projected implementation procedure can be referenced as Figure C-1 in Appendix C

The full assemblies can be seen below:

Drivetrain

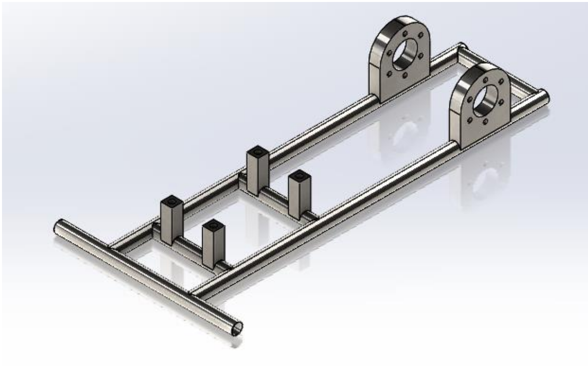

Figure 52: Drivetrain Framing

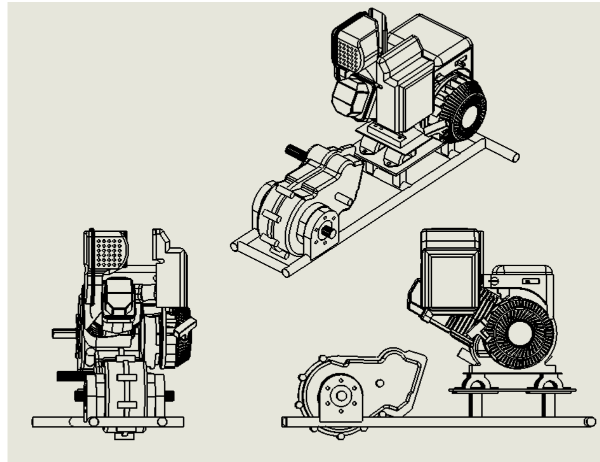

Figure 53: Drivetrain Package Drawing

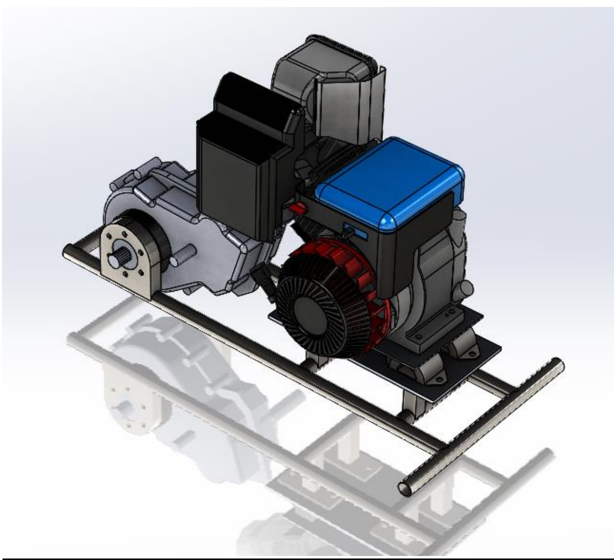

Figure 54: ISO View of Drivetrain Package

Figure 55: Lower View of Drivetrain Package

***NOTE** The CVT is not in the CAD packages shown; however, the dimensions are such that the clutch and driver of the selected CVT will properly couple to the output and input shafts of the engine and transaxle, respectively to keep the required tension in the belt

Frame

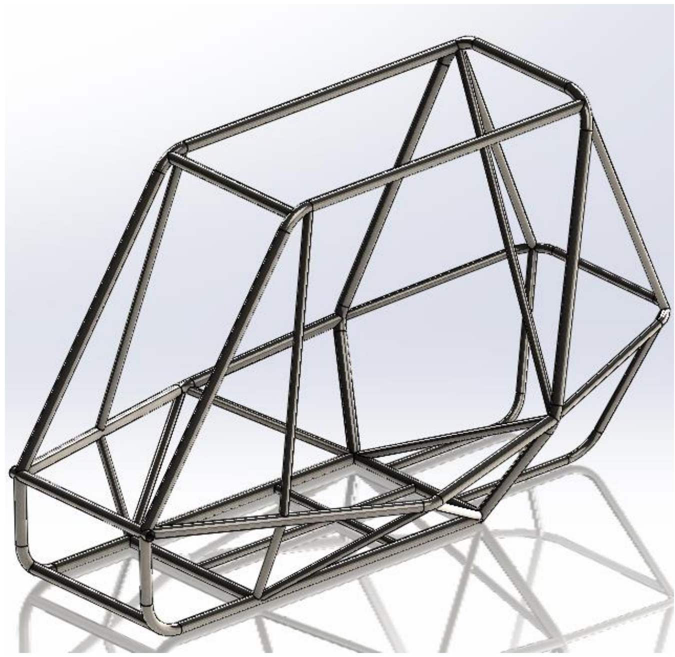

Figure 56: Frame Assembly

Suspension

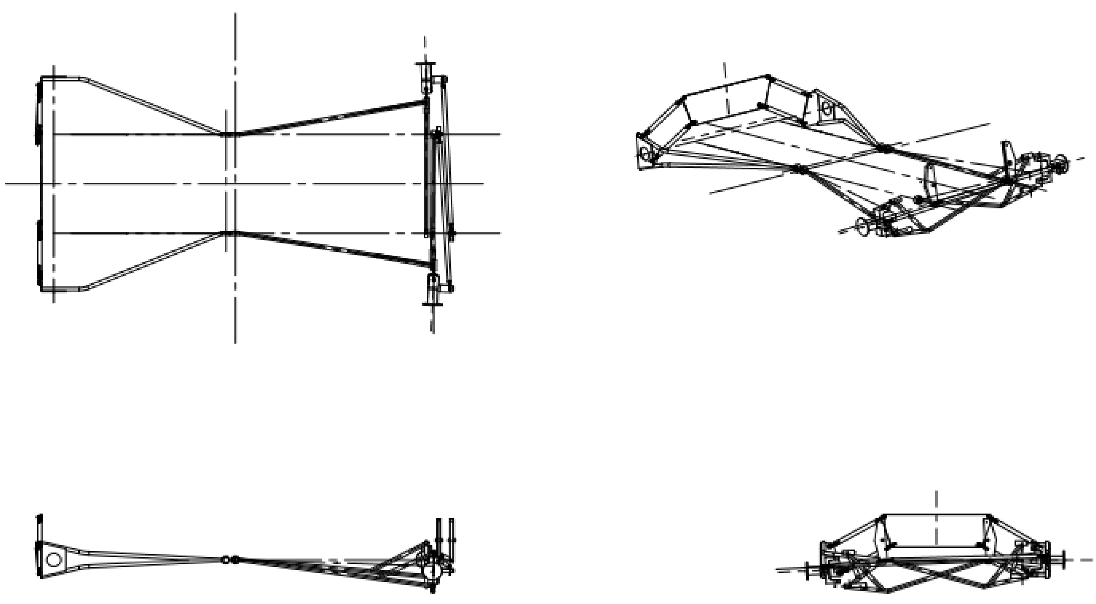

Figure 57: Full Suspension Package

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APPENDICES

Appendix A – Individual Analysis Codes

Drivetrain Code

```
clc
clear
v= 30; % mph
d = 22;% tire diameter in "
C = d * pi(); % circumference
hp = 5.5; %hp at 2000 rpms
Hp = 10;%10 horsepower at 3800 RPMs
W = 700; %weight of car
w = W/4; %weight on each tire
engage = input(' engage rpms ');
dis = input(' Disengage rpms ');
X = input( ' low CVT ratio ');
x = input( ' high CVT ratio ');
Y = input( ' Transaxle ratio ');
r = engage/(X*Y);
Velocityl = C*1.5783e-5*60*r;
R = dis/(x*Y);
Velocityh = C*1.5783e-5*60*R;
dif = (v - Velocityh)/v*100;
%torque out of engine at engage rpms in LB*ft
TL1= hp*63025/(engage*12);
%torque out of engine at dis rpms LB*ft
TH1= Hp*63025/(dis*12);
%torque out of the CVT at engage RPMs in LB*ft
TL2= hp*63025/(engage/X*12);
%torque out of CVT at dis rpms LB*ft
TH2= Hp*63025/(dis/x*12);
%torque out of transaxle at engage RPMs in LB*ft
TL3= hp*63025/(r*12);
%torque out of transaxle at dis rpms LB*Ft
TH3= Hp*63025/(R*12);
%torque required to move on flat fround in LB*ft
TR = d/2*w/12;
%torque requird to go up a 30 degree incline
Goffset = W*sind(30);
TH = d/2*Goffset/12;
fprintf('Max velocity achieved is %4.2f \nPercent difference from 30mph is %4.2f\n',Velocityh,dif)
fprintf('Torque required to move the vehicle on flat ground is %4.2f LB*ft\n',TR)
fprintf('Torque required to move the vehicle on a 30 degree slope is %4.2f LB*ft\n',TH)
fprintf('Low RPM Torque out of engine is %4.2f LB*ft\nHigh RPM torque out of engine is %4.2f LB*ft\n',TL1,TH1)
fprintf('Low RPM Torque out of CVT is %4.2f LB*ft\nHigh RPM torque out of CVT is %4.2f LB*ft\n',TL2,TH2)
fprintf('Low RPM Torque out of transaxle is %5.2f LB*ft\nHigh RPM torque out of transaxle is %5.2f LB*ft\n',TL3,TH3)
```

Outputs

Run 1: Comet 44 Series with Dana H-12 FNR

engage rpms 1900
Disengage rpms 3800
low CVT ratio 2.43
high CVT ratio 1
Transaxle ratio 10.15
Max velocity achieved is 24.50
Percent difference from 30mph is 18.32
Torque required to move the vehicle on flat ground is 160.42 LB*ft
Torque required to move the vehicle on a 30 degree slope is 320.83 LB*ft
Low RPM Torque out of engine is 15.20 LB*ft
High RPM torque out of engine is 13.82 LB*ft
Low RPM Torque out of CVT is 36.94 LB*ft
High RPM torque out of CVT is 13.82 LB*ft
Low RPM Torque out of transaxle is 374.98 LB*ft
High RPM torque out of transaxle is 140.29 LB*ft

Run 2: CVT 780 Series with Dana H-12 FNR

engage rpms 1800
Disengage rpms 3800
low CVT ratio 3.7
high CVT ratio .7
Transaxle ratio 10.15
Max velocity achieved is 35.01
Percent difference from 30mph is -16.68
Torque required to move the vehicle on flat ground is 160.42 LB*ft
Torque required to move the vehicle on a 30 degree slope is 320.83 LB*ft
Low RPM Torque out of engine is 16.05 LB*ft
High RPM torque out of engine is 13.82 LB*ft
Low RPM Torque out of CVT is 59.38 LB*ft
High RPM torque out of CVT is 9.67 LB*ft
Low RPM Torque out of transaxle is 602.68 LB*ft
High RPM torque out of transaxle is 98.20 LB*ft

Run 3: CVT 790 Series with Dana H-12 FNR

engage rpms 1800
Disengage rpms 3800
low CVT ratio 3.3
high CVT ratio .5
Transaxle ratio 10.15
Max velocity achieved is 49.01
Percent difference from 30mph is -63.36
Torque required to move the vehicle on flat ground is 160.42 LB*ft
Torque required to move the vehicle on a 30 degree slope is 320.83 LB*ft
Low RPM Torque out of engine is 16.05 LB*ft
High RPM torque out of engine is 13.82 LB*ft
Low RPM Torque out of CVT is 52.96 LB*ft
High RPM torque out of CVT is 6.91 LB*ft
Low RPM Torque out of transaxle is 537.53 LB*ft
High RPM torque out of transaxle is 70.14 LB*ft

Run 4: Comet 44 Series with SNPT M5101B

engage rpms 1900

Disengage rpms 3800
low CVT ratio 2.43
high CVT ratio 1
Transaxle ratio 2.47
Max velocity achieved is 100.69
Percent difference from 30mph is -235.64
Torque required to move the vehicle on flat ground is 160.42 LB*ft
Torque required to move the vehicle on a 30 degree slope is 320.83 LB*ft
Low RPM Torque out of engine is 15.20 LB*ft
High RPM torque out of engine is 13.82 LB*ft
Low RPM Torque out of CVT is 36.94 LB*ft
High RPM torque out of CVT is 13.82 LB*ft
Low RPM Torque out of transaxle is 91.25 LB*ft
High RPM torque out of transaxle is 34.14 LB*ft

Run 5: CVT 780 Series with SNPT M5101B
engage rpms 1800
Disengage rpms 3800
low CVT ratio 3.7
high CVT ratio .7
Transaxle ratio 2.47
Max velocity achieved is 143.85
Percent difference from 30mph is -379.49
Torque required to move the vehicle on flat ground is 160.42 LB*ft
Torque required to move the vehicle on a 30 degree slope is 320.83 LB*ft
Low RPM Torque out of engine is 16.05 LB*ft
High RPM torque out of engine is 13.82 LB*ft
Low RPM Torque out of CVT is 59.38 LB*ft
High RPM torque out of CVT is 9.67 LB*ft
Low RPM Torque out of transaxle is 146.66 LB*ft
High RPM torque out of transaxle is 23.90 LB*ft

Run 6: CVT 790 Series with SNPT M5101B
engage rpms 1800
Disengage rpms 3800
low CVT ratio 3.3
high CVT ratio .5
Transaxle ratio 2.47
Max velocity achieved is 201.39
Percent difference from 30mph is -571.29
Torque required to move the vehicle on flat ground is 160.42 LB*ft
Torque required to move the vehicle on a 30 degree slope is 320.83 LB*ft
Low RPM Torque out of engine is 16.05 LB*ft
High RPM torque out of engine is 13.82 LB*ft
Low RPM Torque out of CVT is 52.96 LB*ft
High RPM torque out of CVT is 6.91 LB*ft
Low RPM Torque out of transaxle is 130.81 LB*ft
High RPM torque out of transaxle is 17.07 LB*ft

```
clear
clc

for i = 1750:3800
```

```
    w(i) = (i-1)/30;
    T(i) = sqrt((1+(2*w(i))^2)/((1-w(i)^2)^2+2*w(i)^2));
    plot(w,T)
    hold on; grid on
end
xlabel('Frequency Ratio (r)')
ylabel('Transmissibility force (T)')
```

Suspension Code

```
%Jake Hinkle
%NAU SAE BAJA 2016
%FRONT SUSPENSION CALCS


%Set
Y = [-8:0.2:8];
RHX = [-10:.3:14];
RHY = zeros(1,81);
alpha = zeros(1,81);
beta = zeros(1,81);

%Camber
rb = 38;                    %dynamic radius of beam, inches
theta0 = 3;                 %deg
alpha0 = input('Enter camber at RH in degrees: ');
theta = (Y./(cosd(theta0)*rb)) + (theta0*pi/180);
theta = rad2deg(theta);


%Caster
rra = 41.81;                 %dynamic radius of RA, inches
phi0 = 2.85;                 %deg
beta0 = input('Enter caster at RH in degrees: ');
phi = (Y./(cosd(rra)*rra)) + (phi0*pi/180);
phi = rad2deg(phi);


%Plots of Alignment
for i = 1:81
    alpha(i) = theta(i) + alpha0 -theta0;
    beta(i) = beta0 - phi(i) + phi0;
end
grid on
hold
plot(alpha,Y,'r',beta,Y,'g',RHX,RHY,'b');
```

```
%Jake Hinkle
%NAU SAE BAJA 2016
%Steering System code for rack and pinion selection


thetak = 0.7854;                 %desired turning angle at spindle
rk = 3.5;                        %knuckle radius in inches (apprx)
xout = thetak*rk;                %output displacement required in inches
xin = input('Input rack displacement from center to lock in inches: ');
14to13_ratio = xout/xin;         %required 14/13 value
N = input('Input number of turns from center to lock in inches: ');
steering_ratio = N*(2*pi)/thetak; %define (deg at steering wheel/deg at tire)

%Results
display(xout);                   %fpr 45 deg steering at spindle
display(14to13_ratio);           %Lever arms of swinger(in/in)
display(steering_ratio);         %deg/deg(SW/tire)
```

```
%Code: Rear Suspension Geometry Overview
```

```
%Created by: Brody Beebe
%NAU Lumberjacks Baja
%------------------------------------------------------------------------
%Assumptions:
% Trailing arm length is between 36 and 40 inches
% Desired vertical travel up is 6 inches
% Desired vertical travel down is 4 inches

clc; close all; clear all;
Ta=input('trailing arm length?\n'); %Ta = trailing arm length (assume 36")
LCA=input('Lower camber arm length?\n'); %CA = Camber arm lengths
S=input('desired vertical length between camber arms\n'); %S= vertical distance
between camber arms
DH = 2.12; %height to center of output on differential


Tr=23; %tire diameter
figure(1)
%frame geometry left side view
LB=38; UB=10;
LVB=11.32;
plot([0 LB],[0 0],'ro-');hold on;
plot([0 4.1],[0 LVB],'ro-');hold on;
plot([4.1 46],[LVB 15.33],'ro-');hold on;
plot([46 LB],[15.33 0],'ro-');hold on;
plot([-5 60], [-14 -14],'ko-'); hold on;
axis([-50 50 -50 50]);

title('Left Side View of Trailing Arm')
ylabel('Vertical (in.)')
xlabel('Horizontal (in.)')
%-----------------------------------------------------------------------
%Rear suspension geometry left side view
x=[];
y=[];
X=[];
Y=[];
for th=-atan(4/Ta):((atan(6/Ta))-(-atan(4/Ta)))/5:atan(6/Ta)  %Set min and max theta
based off of desired vertical travel
    figure(1);
    %frame geometry left side view
    LB=38; UB=10;
    LVB=11.32;
    plot([0 LB],[0 0],'ro-');hold on;
    plot([0 4.1],[0 LVB],'ro-');hold on;
    plot([4.1 46],[LVB 15.33],'ro-');hold on;
    plot([46 LB],[15.33 0],'ro-');hold on;
    axis equal;
    %Rear suspension geometry left side view
    plot([0 Ta*cos(th)],[0 Ta*sin(th)],'bo-');hold on;
    x(end+1)=Ta*cos(th);
    y(end+1)=Ta*sin(th);
    for th1=0:0.0873:2*pi
        X(end+1)=(Tr/2)*cos(th1)+x(end);
        Y(end+1)=(Tr/2)*sin(th1)+y(end);

    end
    plot(X,Y,'g-');
    pause(0.1);


end
```

```
figure(2);
plot(x,y,'b-'); hold on;
plot([0 LB],[0 0],'ro-');hold on;
plot([0 4.1],[0 LVB],'ro-');hold on;
plot([4.1 46],[LVB 15.33],'ro-');hold on;
plot([46 LB],[15.33 0],'ro-');hold on;
axis equal
title('Left Side View of Hub Path at end of Trailing Arm ')
ylabel('Vertical (in.)')
xlabel('Horizontal (in.)')
%----------------------------------------------------------------------
%frame geometry rear view
figure(3)
LBR=22;
plot([0 LBR],[0 0],'ro-');hold on;
plot([0 -6.64],[0 11.32],'ro-');hold on;
plot([-6.64 .5],[11.32 15.33],'ro-');hold on;
plot([.5 21.5],[15.33 15.33],'ro-');hold on;
plot([21.5 28.64],[15.33 11.32],'ro-');hold on;
plot([28.64 22],[11.32 0],'ro-');hold on;
plot([4.75 .5],[0 15.33],'ro-');hold on;
plot([17.25 21.5],[0 15.33],'ro-');hold on;
plot([6.06 6.06],[1.62 2.62],'ro-'); hold on;
plot([15.94 15.94],[1.62 2.62],'ro-'); hold on;
plot([-20 40],[-14 -14],'ko-'); hold on;
axis equal
title('Rear View of Frame and Suspension')
ylabel('Vertical (in.)')
xlabel('Horizontal (in.)')
th4=74.505;  % theta calculated from frame geometry
CAD=S/(tand(th4)); %CAD = Camber Arm Distance from eachother, dx
%--------------------------------------------------------------------------
%Camber Arms and CV axle geometry right side
CA1x=[];
CA1y=[];
CA2x=[];
CA2y=[];
CVRx=[];
CVRy=[];
for th2=-22.5*(pi/180):pi/12:22.5*(pi/180)
    figure(3)
    CA1x(end+1)=LCA*cos(th2)+17.25;
    CA1y(end+1)=LCA*sin(th2);
    plot([17.25 LCA*cos(th2)+17.25],[0 LCA*sin(th2)],'bo-'); hold on
    CA2x(end+1)=LCA*cos(th2)+17.25;
    CA2y(end+1)=LCA*sin(th2)+S;
    plot([17.25+CAD LCA*cos(th2)+17.25],[S LCA*sin(th2)+S],'bo-'); hold on
    CVRx(end+1) = CA1x(end);
    CVRy(end+1)=CA1y(end)+(S/2);
    plot([15.94 LCA*cos(th2)+17.25],[DH LCA*sin(th2)+(S/2)],'go-'); hold on
    UCA=sqrt((LCA*cos(th2)-CAD)^2+(LCA*sin(th2))^2); %Upper control arm calculation
end
%------------------------------------------------------------------------------------
-----------------
%Camber Arms and CV axle geometry left side
CA3x=[];
CA3y=[];
CA4x=[];
CA4y=[];
CVLx=[];
```

```
CVLy=[];

for th5=157.5*(pi/180):pi/12:202.5*(pi/180)
    figure(3)
    CA3x(end+1)=LCA*cos(th5)+4.75;
    CA3y(end+1)=LCA*sin(th5);
    plot([4.75 LCA*cos(th5)+4.75],[0 LCA*sin(th5)],'bo-'); hold on
    CA4x(end+1)=LCA*cos(th5)+(4.75);
    CA4y(end+1)=LCA*sin(th5)+S;
    plot([4.75-CAD LCA*cos(th5)+(4.75)],[S LCA*sin(th5)+S],'bo-'); hold on
    CVLx(end+1)=CA3x(end);
    CVLy(end+1)=CA3y(end)+(S/2);
    plot([6.06 LCA*cos(th5)+4.75],[DH LCA*sin(th5)+(S/2)],'go-'); hold on
end

figure(4)
axis equal
plot(CA1x,CA1y,'b.-'); hold on
plot(CA2x,CA2y,'r.-'); hold on
plot(CA3x,CA3y,'b.-'); hold on
plot(CA4x,CA4y,'r.-'); hold on
plot(CVLx,CVLy,'g.-'); hold on
plot(CVRx,CVRy,'g.-'); hold on

title('Rear View of Hub path at end of Camber Arms and CV')
ylabel('Vertical (in.)')
xlabel('Horizontal (in.)')

%find max vertical travel of hub/camber arms
Max_pos_dy=0;
for i=1:length(CA1y)
    if CA1y(i)>Max_pos_dy
        Max_pos_dy=CA1y(i);
    end
end
Max_neg_dy=0;
for i=1:length(CA1y)
    if CA1y(i)<Max_neg_dy
        Max_neg_dy=CA1y(i);
    end
end
Max_pos_dy2=0;
for i=1:length(CA2y)
    if CA2y(i)>Max_pos_dy2
        Max_pos_dy2=CA2y(i);
    end
end
Max_neg_dy2=10;
for i=1:length(CA2y)
    if CA2y(i)<Max_neg_dy2
        Max_neg_dy2=CA2y(i);
    end
end
Max_pos_dy3=0;
for i=1:length(CVRy)
    if CVRy(i)>Max_pos_dy3
        Max_pos_dy3=CVRy(i);
    end
end
Max_neg_dy3=0;
for i=1:length(CVRy)
    if CVRy(i)<Max_neg_dy3
```

```
        Max_neg_dy3=CVRy(i);
    end
end

figure(5)
plot([0 LBR],[0 0],'ro-');hold on;
plot([0 -6.64],[0 11.32],'ro-');hold on;
plot([-6.64 .5],[11.32 15.33],'ro-');hold on;
plot([.5 21.5],[15.33 15.33],'ro-');hold on;
plot([21.5 28.64],[15.33 11.32],'ro-');hold on;
plot([28.64 22],[11.32 0],'ro-');hold on;
plot([4.75 .5],[0 15.33],'ro-');hold on;
plot([17.25 21.5],[0 15.33],'ro-');hold on;
plot([6.06 6.06],[1.62 2.62],'ro-'); hold on;
plot([15.94 15.94],[1.62 2.62],'ro-'); hold on;
plot([-20 40],[-14 -14],'ko-'); hold on;
axis equal
title('Rear View of Frame and Suspension at ride height')
ylabel('Vertical (in.)')
xlabel('Horizontal (in.)')
th4=74.505;  % theta calculated from frame geometry
CAD=S/(tand(th4)); %CAD = Camber Arm Distance from eachother, dx
%--------------------------------------------------------------------------
%Camber Arms and CV axle geometry right side
CA6x=[];
CA6y=[];
CA7x=[];
CA7y=[];
CVR1x=[];
CVR1y=[];
for th8=-22.5*(pi/180):pi/12:22.5*(pi/180)
    figure(3)
    CA6x(end+1)=LCA*cos(th8)+17.25;
    CA6y(end+1)=LCA*sin(th8);
    plot([17.25 LCA*cos(th8)+17.25],[0 LCA*sin(th8)],'bo-'); hold on
    CA7x(end+1)=LCA*cos(th8)+17.25;
    CA7y(end+1)=LCA*sin(th8)+S;
    plot([17.25+CAD LCA*cos(th8)+17.25],[S LCA*sin(th8)+S],'bo-'); hold on
    CVR1x(end+1) = CA6x(end);
    CVR1y(end+1)=CA6y(end)+(S/2);
    plot([15.94 LCA*cos(th2)+17.25],[DH LCA*sin(th2)+(S/2)],'go-'); hold on
    UCA1=sqrt((LCA*cos(th2)-CAD)^2+(LCA*sin(th2))^2); %Upper control arm calculation
ride height
end
%------------------------------------------------------------------------------------
-----------------
%Camber Arms and CV axle geometry left side
CA8x=[];
CA8y=[];
CA9x=[];
CA9y=[];
CVL1x=[];
CVL1y=[];

for th5=157.5*(pi/180):pi/12:202.5*(pi/180)
    figure(3)
    CA8x(end+1)=LCA*cos(th5)+4.75;
    CA8y(end+1)=LCA*sin(th5);
    plot([4.75 LCA*cos(th5)+4.75],[0 LCA*sin(th5)],'bo-'); hold on
    CA9x(end+1)=LCA*cos(th5)+(4.75);
    CA9y(end+1)=LCA*sin(th5)+S;
    plot([4.75-CAD LCA*cos(th5)+(4.75)],[S LCA*sin(th5)+S],'bo-'); hold on
```

```
    CVL1x(end+1)=CA8x(end);
    CVL1y(end+1)=CA8y(end)+(S/2);
    plot([6.06 LCA*cos(th5)+4.75],[DH LCA*sin(th5)+(S/2)],'go-'); hold on
end

fprintf('-------------------------------------------------------\n');
fprintf('upper control arm length = %1.3f\n',UCA);
dx=x(3)-x(end);
fprintf('Trailing arm change in x = %0.3f\n',dx);
Total_dy=Max_pos_dy-Max_neg_dy;
fprintf('Lower Camber Arm total change in y = %1.3f\n',Total_dy);
Total_dy2=Max_pos_dy2-Max_neg_dy2;
fprintf('Upper Camber Arm total change in y = %1.3f\n',Total_dy2);
Total_dy3=Max_pos_dy3-Max_neg_dy3;
fprintf('CV axle total change in y = %1.3f\n',Total_dy3);
```

<u>Suspension Results</u>

INPUT:
trailing arm length?
36
Lower camber arm length?
15
desired vertical length between camber arms
5
OUTPUT:
----
upper control arm length = 13.730
Trailing arm change in x = 0.490
Lower Camber Arm total change in y = 11.481
Upper Camber Arm total change in y = 11.481
CV axle total change in y = 11.481

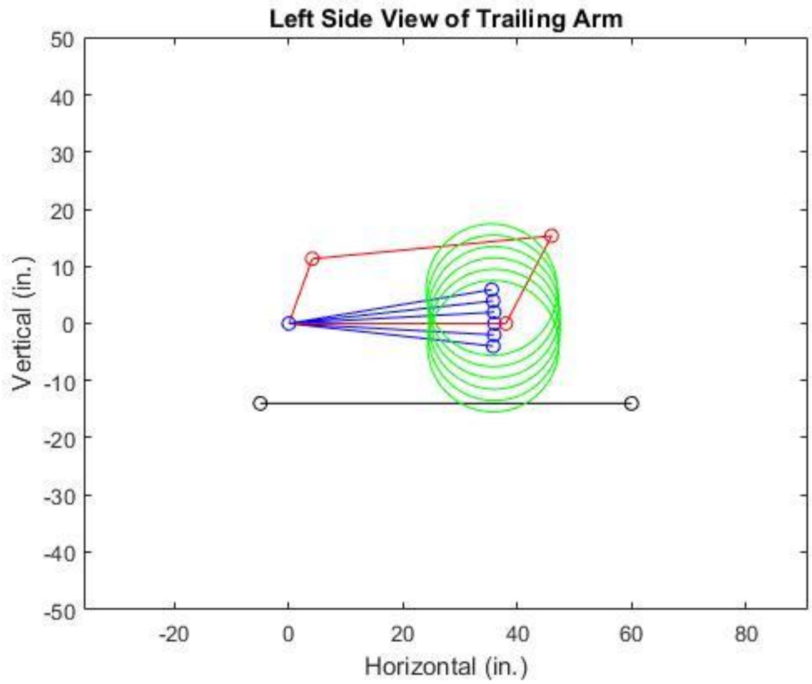


Figure A- 1: In this plot the blue lines represent the trailing arm movement, the red lines represent the frame, the green circles represent the tire, and the black line underneath represents the ground.

Figure A- 2: The blue line represents the path of the spindle attached to the end of the trailing arm, while the frame is shown in red.

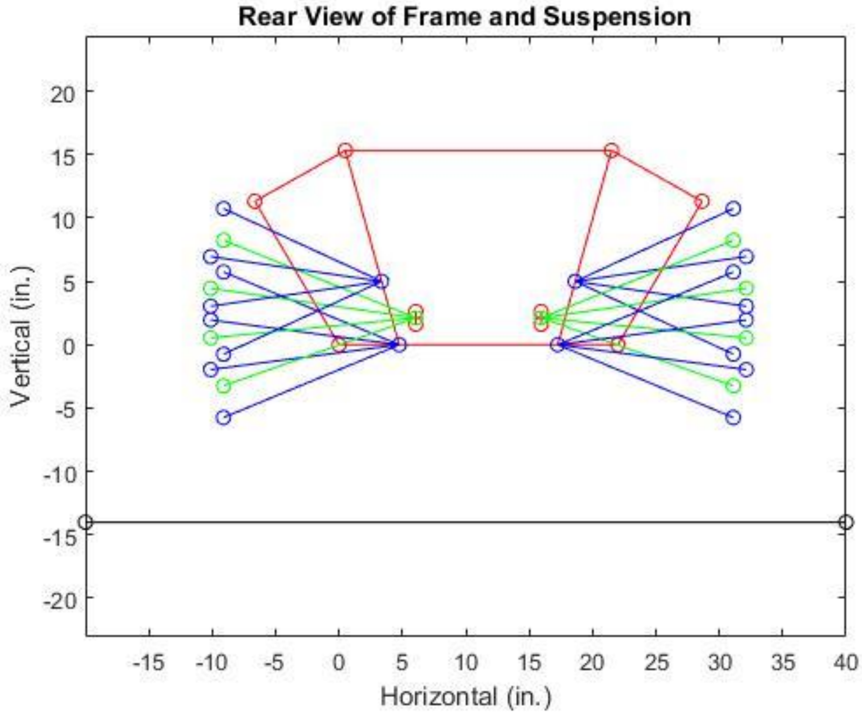


Figure A- 3: Blue lines represent the camber arm movement, green lines show CV axle movement, and red lines illustrate the frame

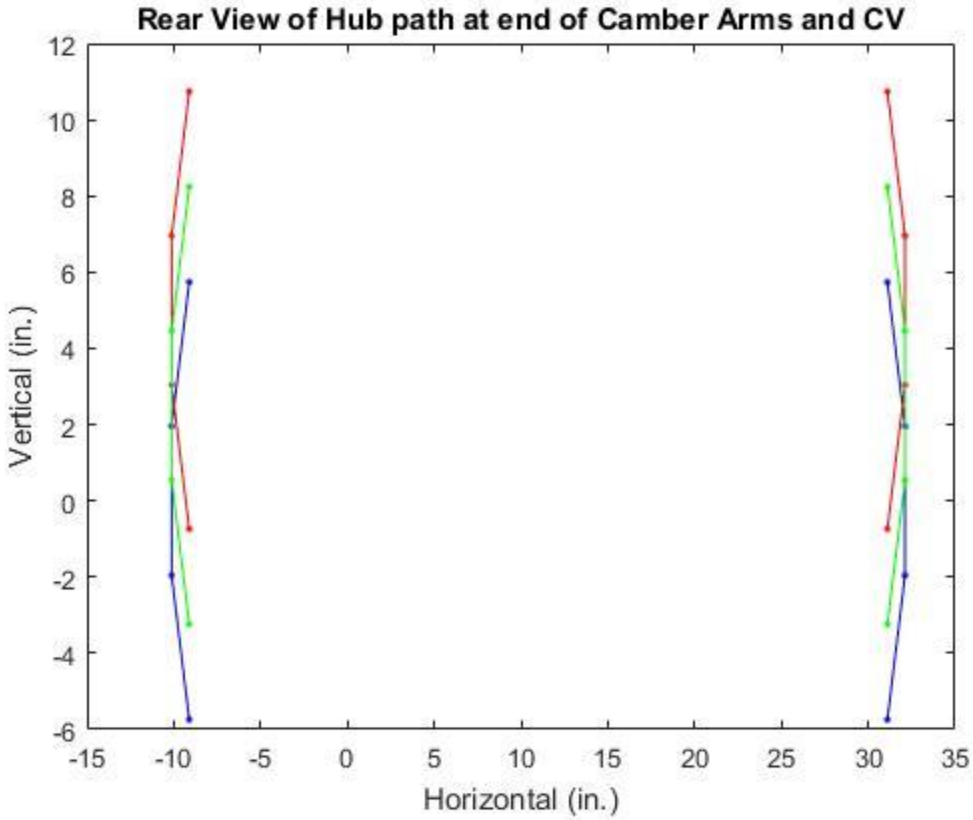


Figure A- 4: Red, green, and blue lines represent the path of the upper camber arm, CV axle, and lower camber arm, respectively.

Appendix B – BOM & Budget Analysis

Table B- 1: Funds and Expenses

FUNDS	
SAE	
Previous Teams	$5,297.41
Gore	$4,917.05
Capstone	
Class Funds	TBD
Initial Total	$10,214.46
Total After Expenses	$1,444.93
EXPENSES	
Registration	$1,250.00
Frame	
Frame Material	$420.00
Electrical (Kill Switch, Wiring, etc.)	$40.00
Outer Covers, Skid Plates, etc.	$30.00
$35/Labor Hour	
Suspension/Steering	
Fox Air Shocks	$1,117.50
Spindles	$100.00
Rod Ends	$300.00
Ball Joints	$80.00
Bearings	$40.00
Wheels & Tires	$431.70
Material (4130 steel)	$30.00
Steering Wheel	$5.00
Miscellaneous (bushings, fasteners, etc.)	$150.00
$35/Labor Hour	
Drivetrain	
Engine Order	$250.00
CVT Driver (780 series)	$210.00
CVT Driven (780 series 32D degree)	$40.00
CVT Belt	$45.00
CVT Accessories	$60.00
Gearbox (Dana H-12 FNR)	$1,295.00
Miscellaneous (bushings, fasteners, etc.)	$40.00
Kill Switch	$35.00
$35/Labor Hour	
Safety	
5-Point Harness	$145.00
Helmet Support	$35.00
Hand Constraints	$38.00
Travel	
Van/Trailer Rental	$200.00
Gas, Food, Etc.	$600.00
Hotel Rooms	$1,000.00
Total Excluding Labor	$7,664.20
Total Including Labor	$7,769.20

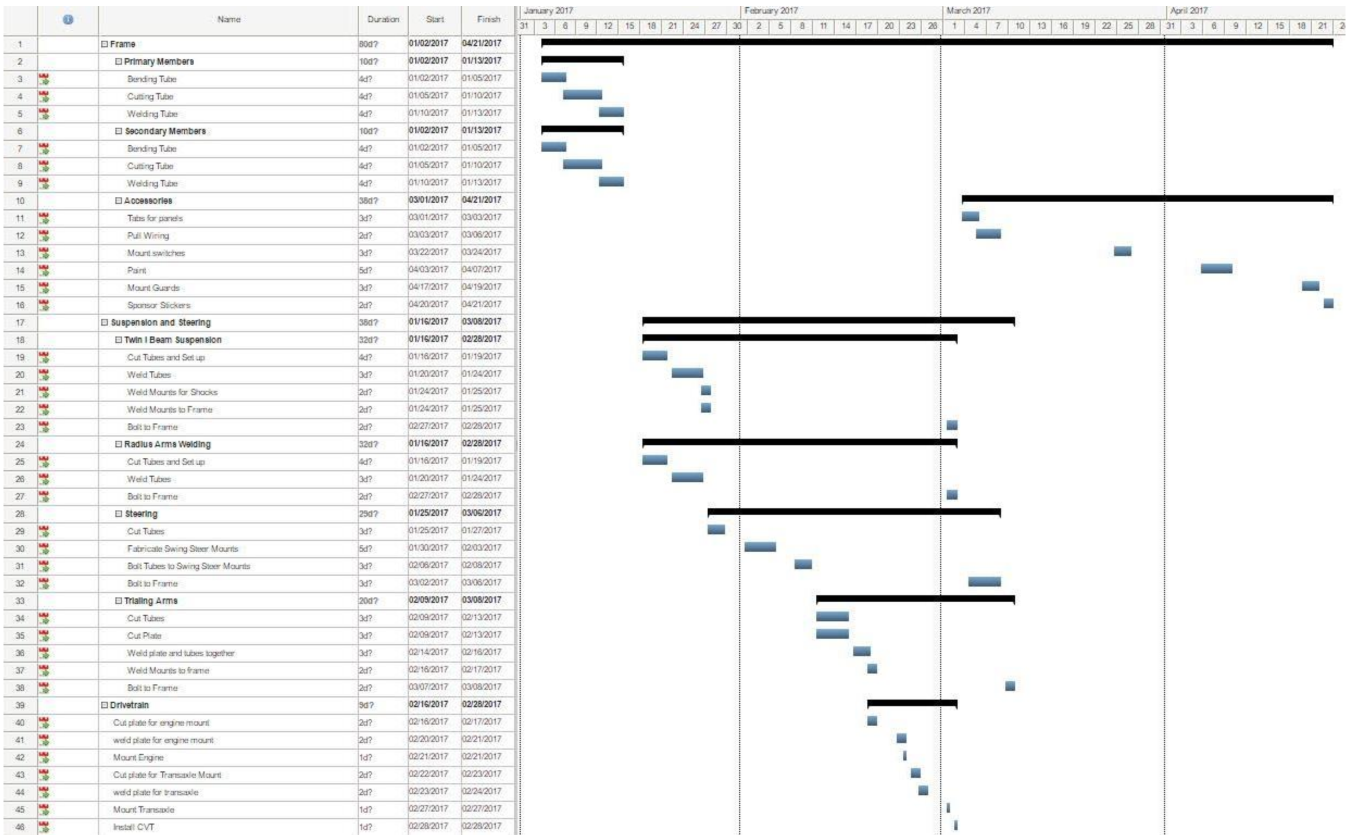
Table B- 2: Post Design Expenses

UPDATED EXPENSES	
Registration	$1,250.00
Frame	$558.21
Drivetrain	$1,900.00
Suspension/Steering	$2,993.32
Electrical (Kill Switch, Wiring, etc.)	$50.00
Safety	$218.00
Travel	
Van/Trailer Rental	$200.00
Gas, Food, Etc.	$600.00
Hotel Rooms	$1,000.00
Total	$8,769.53

Table B- 3: Detailed BOM

Item	Category	Description	Vendor Part Number	Purchasing	Fabricating	Vendor	Quantity	Material Cost	Labor Cost	Extended Material Cost	Extended Labor Cost	Extended Total	
1	Engine	Model 20 - 10HP (Donated)	19L232-0054 G1	x	[redacted]	Briggs & Stratton	1	$0.00	$250.00	$0.00	$250.00	$250.00	Drivetrain Sum
2	CVT Driver	780 series		x	[redacted]	QDS	1	$210.00	$0.00	$210.00	$0.00	$210.00	$1,900.00
3	CVT Driven	780 series 32D degree		x	[redacted]	QDS	1	$40.00	$0.00	$40.00	$0.00	$40.00	
4	CVT Belt			x	[redacted]	QDS	1	$45.00	$0.00	$45.00	$0.00	$45.00	
5	CVT Accessories			x	[redacted]	QDS	1	$60.00	$0.00	$60.00	$0.00	$60.00	
6	Gearbox (Dana H-12 FNR)	Reduction transaxle (10.15:1)	012AS109-3	x	[redacted]	QDS	1	$1,295.00	$0.00	$1,295.00	$0.00	$1,295.00	
7	Frame Material	DOM 1.375" OD x .065"	N/A	x	[redacted]	Alro Metals	1	$420.00	$0.00	$420.00	$0.00	$420.00	Frame Sum
8	Skid Plate Material	24" x 24" - .125" 6061-T6 Aluminum	N/A	x	[redacted]	IMS	1	$47.63	$0.00	$47.63	$0.00	$47.63	$558.21
9	Skid Plate Material	36" x 48" - .02" 6061-T6 Aluminum	N/A	x	[redacted]	IMS	1	$58.96	$0.00	$58.96	$0.00	$58.96	
10	Skid Plate Material	36" x 24" - .125" 6061-T6 Aluminum	N/A	x	[redacted]	IMS	1	$31.62	$0.00	$31.62	$0.00	$31.62	
11	Fox Air Shocks (pair)	Float 3 - 16.2" X 4.5"	830-10- 301	x	[redacted]	Fox Shox	2	$558.75	$0.00	$1,117.50	$0.00	$1,117.50	Steering/Suspension Sum
12	Spindle Ball Joints	4130 Chomoly Heim Ball Joint	N/A	x	[redacted]	American Star	2	$36.95	$0.00	$73.90	$0.00	$73.90	$2,926.50
13	Steering Rod Ends	5/16" (left hand threads)	4475T102	x	[redacted]	Mcmaster-carr	3	$13.37	$0.00	$40.11	$0.00	$40.11	
14	Steering Rod Ends	5/16" (right hand threads)	4475T101	x	[redacted]	Mcmaster-carr	3	$13.37	$0.00	$40.11	$0.00	$40.11	
15	Steering Tube End Weld Nuts	5/16" - 24 thread (right hand thread)	94640A110	x	[redacted]	Mcmaster-carr	3	$5.30	$0.00	$15.90	$0.00	$15.90	
16	Steering Tube End Weld Nuts	5/16" - 24 thread (left hand thread)	94640A114	x	[redacted]	Mcmaster-carr	3	$5.30	$0.00	$15.90	$0.00	$15.90	
17	Steering Jam Nuts	5/16" -24 Grade 8 steel (pack of 50)	93839A810	x	[redacted]	Mcmaster-carr	1	$7.52	$0.00	$7.52	$0.00	$7.52	
18	Steering Bearings	Aurora HAB-10T	AU-HAB-10T	x	[redacted]	RoadRaceParts	6	$36.99	$0.00	$221.94	$0.00	$221.94	
19	Suspension Material	3/4"x.065" A513 DOM (5 feet)		x	[redacted]	IMS	1	$9.96	$0.00	$9.96	$0.00	$9.96	
20	Suspension Material	7/8"x.058" A513 DOM (3 feet)		x	[redacted]	IMS	1	$11.61	$0.00	$11.61	$0.00	$11.61	
21	Suspension Material	3/4"x.095" A513 DOM (30 feet)		x	[redacted]	IMS	1	$60.00	$0.00	$60.00	$0.00	$60.00	
22	Suspension Material	3/16"-36"x36" Mild Steel Plate		x	[redacted]	IMS	1	91.06	$0.00	$91.06	$0.00	$91.06	
23	Knuckle	Front left	1.31179E+11	x	[redacted]	Rocky Mountain ATV	1	$129.30	$0.00	$129.30	$0.00	$129.30	
24	Knuckle	Front right	1.31179E+11	x	[redacted]	Rocky Mountain ATV	1	$129.30	$0.00	$129.30	$0.00	$129.30	
25	Castle Nut	12mm	1.31179E+11	x	[redacted]	Rocky Mountain ATV	4	$4.09	$0.00	$16.36	$0.00	$16.36	
26	Washer	Plain, 12mm	1.31179E+11	x	[redacted]	Rocky Mountain ATV	4	$0.66	$0.00	$2.64	$0.00	$2.64	
27	Caliper Sub-Assy.	left hand	1.31179E+11	x	[redacted]	Rocky Mountain ATV	1	$136.39	$0.00	$136.39	$0.00	$136.39	
28	Caliper Sub-Assy.	right hand	1.31179E+11	x	[redacted]	Rocky Mountain ATV	1	$132.29	$0.00	$132.29	$0.00	$132.29	
29	Bolt, Disk	hardware for caliper	1.31179E+11	x	[redacted]	Rocky Mountain ATV	6	$4.19	$0.00	$25.14	$0.00	$25.14	
30	Disk, Front Br.	brake disk	1.31179E+11	x	[redacted]	Rocky Mountain ATV	2	$67.17	$0.00	$134.34	$0.00	$134.34	
31	Collar, Wheelside		1.31179E+11	x	[redacted]	Rocky Mountain ATV	2	$4.67	$0.00	$9.34	$0.00	$9.34	
32	Axle Nut	14mm	1.31179E+11	x	[redacted]	Rocky Mountain ATV	2	$10.01	$0.00	$20.02	$0.00	$20.02	
33	Split Pin	3.0x25	1.31179E+11	x	[redacted]	Rocky Mountain ATV	2	$0.47	$0.00	$0.94	$0.00	$0.94	
34	Hub Sub-Assy.	Front	1.31179E+11	x	[redacted]	Rocky Mountain ATV	2	$140.98	$0.00	$281.96	$0.00	$281.96	
35	Camber Arm Rod End	7/16"- 20 (Right hand)	4475T131	x	[redacted]	Mcmaster-carr	4	$18.63	$0.00	$74.52	$0.00	$74.52	
36	Camber Arm Rod End	7/16"-20 (Left hand)	4475T132	x	[redacted]	Mcmaster-carr	4	$18.63	$0.00	$74.52	$0.00	$74.52	
37	Tube Weld Nut	7/8" 4130 Alloy (Right hand)	94640A155	x	[redacted]	Mcmaster-carr	4	$5.42	$0.00	$21.68	$0.00	$21.68	
38	Tube Weld Nut	7/8" 4130 Alloy (Left hand)	94640A159	x	[redacted]	Mcmaster-carr	4	$5.42	$0.00	$21.68	$0.00	$21.68	
39	Camber Arm Jam Nuts	7/16"-20 Grade 8 Steel (pack of 25)	93839A820	x	[redacted]	Mcmaster-carr	1	$10.57	$0.00	$10.57	$0.00	$10.57	
40	Rack and Pionion	14" w/ U-joint and stub shaft	QDS111-14	x	[redacted]	QDS	1	$165.00	$0.00	$165.00	$0.00	$165.00	
41	Steering Wheel	12" diameter	1805	x	[redacted]	QDS	1	$35.00	$0.00	$35.00	$0.00	$35.00	
42	Steering Shaft & Hub Kit	34" shaft	1868-34	x	[redacted]	QDS	1	$29.95	$0.00	$29.95	$0.00	$29.95	
43	Steering Wheel Cap	Cap	1877	x	[redacted]	QDS	1	$9.50	$0.00	$9.50	$0.00	$9.50	
44	Wheels	Douglas A5 Wheel (10"x 5")	1462480011	x	[redacted]	Rocky Mountain	4	$54.99	$0.00	$219.96	$0.00	$219.96	
45	Tires (Front)	ITP Holeshot XCR (21"x7"-10")	1087450004	x	[redacted]	Rocky Mountain	2	$49.99	$0.00	$99.98	$0.00	$99.98	Wheel/Tire Sum
46	Tires (Rear)	ITP Mud Lite AT (22"x8"-10")	1087540004	x	[redacted]	Rocky Mountain	2	$55.88	$0.00	$111.76	$0.00	$111.76	$431.70
47	5-Point Harness	3" 5-point Lap and Shoulder Harness	332710.LG	x	[redacted]	QDS	1	$145.00	$0.00	$145.00	$0.00	$145.00	Safety Sum
48	Helmet Support	Blue	40003	x	[redacted]	QDS	1	$35.00	$0.00	$35.00	$0.00	$35.00	$218.00
49	Arm Constraints (2)	Blue	605000	x	[redacted]	QDS	1	$38.00	$0.00	$38.00	$0.00	$38.00	
50	Kill Switch	Baja Approved (QDS-171)	QDS-171	x	[redacted]	QDS	1	$35.00	$0.00	$35.00	$0.00	$35.00	
51	Brake Parts Cleaner	Consumable	72408	x	[redacted]	Oreillys Auto Parts	6	$4.29	$0.00	$25.74	$0.00	$25.74	
52	Permatex Anti-Sieze Lubricant	Consumable	80078	x	[redacted]	Oreillys Auto Parts	1	$7.49	$0.00	$7.49	$0.00	$7.49	
53	Masterpro Marine Grease	Consumable	90380	x	[redacted]	Oreillys Auto Parts	1	4.99	$0.00	$4.99	$0.00	$4.99	
54	Permatex Thread Locker Red	Consumable	27200	x	[redacted]	Oreillys Auto Parts	1	$12.99	$0.00	$12.99	$0.00	$12.99	
55	Permatex Ultra Black RTV	Consumable	82180	x	[redacted]	Oreillys Auto Parts	1	7.49	$0.00	$7.49	$0.00	$7.49	

Appendix C – Implementation Gantt Chart



	Name	Duration	Start	Finish
1	**Frame**	80d?	**01/02/2017**	**04/21/2017**
2	**Primary Members**	10d?	**01/02/2017**	**01/13/2017**
3	Bending Tube	4d?	01/02/2017	01/05/2017
4	Cutting Tube	4d?	01/05/2017	01/10/2017
5	Welding Tube	4d?	01/10/2017	01/13/2017
6	**Secondary Members**	10d?	**01/02/2017**	**01/13/2017**
7	Bending Tube	4d?	01/02/2017	01/05/2017
8	Cutting Tube	4d?	01/05/2017	01/10/2017
9	Welding Tube	4d?	01/10/2017	01/13/2017
10	**Accessories**	38d?	**03/01/2017**	**04/21/2017**
11	Tabs for panels	3d?	03/01/2017	03/03/2017
12	Pull Wiring	2d?	03/03/2017	03/06/2017
13	Mount switches	3d?	03/22/2017	03/24/2017
14	Paint	5d?	04/03/2017	04/07/2017
15	Mount Guards	3d?	04/17/2017	04/19/2017
16	Sponsor Stickers	2d?	04/20/2017	04/21/2017
17	**Suspension and Steering**	38d?	**01/16/2017**	**03/08/2017**
18	**Twin I Beam Suspension**	32d?	**01/16/2017**	**02/28/2017**
19	Cut Tubes and Set up	4d?	01/16/2017	01/19/2017
20	Weld Tubes	3d?	01/20/2017	01/24/2017
21	Weld Mounts for Shocks	2d?	01/24/2017	01/25/2017
22	Weld Mounts to Frame	2d?	01/24/2017	01/25/2017
23	Bolt to Frame	2d?	02/27/2017	02/28/2017
24	**Radius Arms Welding**	32d?	**01/16/2017**	**02/28/2017**
25	Cut Tubes and Set up	4d?	01/16/2017	01/19/2017
26	Weld Tubes	3d?	01/20/2017	01/24/2017
27	Bolt to Frame	2d?	02/27/2017	02/28/2017
28	**Steering**	29d?	**01/25/2017**	**03/06/2017**
29	Cut Tubes	3d?	01/25/2017	01/27/2017
30	Fabricate Swing Steer Mounts	5d?	01/30/2017	02/03/2017
31	Bolt Tubes to Swing Steer Mounts	3d?	02/06/2017	02/08/2017
32	Bolt to Frame	3d?	03/02/2017	03/06/2017
33	**Trailing Arms**	20d?	**02/09/2017**	**03/08/2017**
34	Cut Tubes	3d?	02/09/2017	02/13/2017
35	Cut Plate	3d?	02/09/2017	02/13/2017
36	Weld plate and tubes together	3d?	02/14/2017	02/16/2017
37	Weld Mounts to frame	2d?	02/16/2017	02/17/2017
38	Bolt to Frame	2d?	03/07/2017	03/08/2017
39	**Drivetrain**	9d?	**02/16/2017**	**02/28/2017**
40	Cut plate for engine mount	2d?	02/16/2017	02/17/2017
41	weld plate for engine mount	2d?	02/20/2017	02/21/2017
42	Mount Engine	1d?	02/21/2017	02/21/2017
43	Cut plate for Transaxle Mount	2d?	02/22/2017	02/23/2017
44	weld plate for transaxle	2d?	02/23/2017	02/24/2017
45	Mount Transaxle	1d?	02/27/2017	02/27/2017
46	Install CVT	1d?	02/28/2017	02/28/2017

Figure C- 1: Implementation Gantt Chart