SAE Mini Baja: Suspension and Steering

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Team 19

Project Proposal

Document

Submitted towards partial fulfillment of the requirements for
Mechanical Engineering Design I – Fall 2013

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Memorandum

To: Dr. John Tester

From: Team 19; SAE Mini Baja Suspension Design Team; Benjamin Bastidos, Victor Cabilan, Jeramie Goodwin, William Mitchell, Eli Wexler

CC:

Date: 12/13/2013

Subject: Final Suspension & Steering Design and Expected Final Cost

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The SAE Mini Baja design challenge tests teams to design every component for a competitive vehicle in a rough, off road environment. The originally very large team split into three smaller teams, each in charge of designing the frame, suspension/steering or drivetrain for the vehicle. This memo is regarding the suspension and steering design aspect of the vehicle as well as the expected cost of the suspension components.

Final Design

The final concepts that have been selected for the suspension design follow the main objectives of being light, durable while also remaining as cheap as possible. The front suspension follows a simple double a-arm design while the rear suspension is a 3-link, semi-trailing arm with links. Material choice is planned to be made from AISI 4130 Steel following the frame team and their choice to use the same material. The final concept for the steering is an off the shelf rack and pinion, with tie rods made of a smaller diameter 4130 Steel in order to have an acceptable strength.

Expected cost

The expected cost of the front suspension components, including brakes, is $1440.33. The rear suspension component total cost is $1067.67. The steering components, including materials of tie rods, is the lowest at $324.60. These costs are merely the costs based off what we calculated needing, there are cheaper ways to get some suspension parts that could drastically cut costs.

Sincerely,
Team 19

Nomenclature

$Fi = Force\ of\ impact$
$Fs = Weight$
$h = Drop\ Height$
$Ks = Spring\ Constant$
$V = Velocity$
$M = Mass$
$T = Time$

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Abstract

The steering and suspension systems are mandatory components of a vehicle considering that they are what control the interaction between the driver and the environment. Without these components the vehicle would not be able to suitably interact with the environment. In the case of building an off-road vehicle, the right materials and the correct suspension and steering geometry shall be used to traverse and navigate rugged terrain with the most ease possible. When designing and selecting parts for the suspension and steering the goal is to obtain parts with optimal strength and durability while being able to obtain them at a reasonable price. To achieve this goal, during the design process decisions were made by decision matrices, engineering analysis and cost analysis. The decision process consisted of choosing various designs and parts for the steering and suspension and deciding on the most efficient design by analysis of the pros and cons of each design. Each pro and con were weighted by importance so that a design suiting the project needs would be selected. After selecting designs, the designs were then put through analytic engineering simulations to calculate if the designs would be accurate not by characteristics but by structural analysis. The parts were then cost calculated to reassure that the parts list were within the budget of the project. Even in the worst case scenario where sponsorships would not be available the MSRP prices of each part are calculated and totaled.

Chapter 1: Introduction

The process of designing suspension and steering components for the Collegiate Series SAE BAJA Competition, the main goals were to design a system with an improved steering system and improved suspension components while keeping the system lightweight. The client, SAE International, has requested that all competing collegiate teams produce a safe and potentially marketable vehicle for competitive use in El Paso, Texas that must adhere to the SAE Collegiate Design Series [1] rulebook for the 2014 season. The project advisor has requested that the Mini Baja team design a vehicle that is a major improvement over the previous 2009 vehicle, with emphasis on the reducing the steering radius.

Multiple suspension components were analyzed to produce an improved front and rear system. The front suspension will consist of an independent front a-arms and the rear will consist of a 3-link trailing arm design. The decision was conducted on the fact that a-arm suspension is generally used for the front because the geometry keeps the camber on the wheels static and this is an adequate suspension to have for handling rugged terrain. The steering is kept simple by using a rack and pinion design with 180 degrees of freedom on the steering wheel for quick response from the driver. The operating environment of the competition is in El Paso Texas where there will be four tests for the vehicle to traverse. The tests will consist of a speed test, a rock climb, maneuverability (obstacle course) and an endurance test which will consist of all of the other tests put together. The suspension design is very crucial for all tests but mostly for the rock climb considering that the rock climb will take the suspension to the maximum stresses. The rock climb will also create large stresses on the tie rods and steering system so each system will be analyzed for strength with values from simulating the stresses on the parts during a rock climb. For the other tests, the components were analyzed dynamically with separate forces and stresses.

Table 1 - Steering Quality Function Deployment

Engineering Requirements for given design								
Customer Needs	**Weight**	**Y.S.**	**Caster Angle**	**Ackerman Angle**	**Turning Radius**	**Cost**	**Bolt Shear**	**Width**
1. Lightweight	10					3	1	
2. Maneuverability	10		9	9	9			9
3. Relatively inexpensive	6	9				9	3	
4. Stable/safe	9		9	9	3			9
5. Must be durable	8	9				9	3	
6. Transportable	8				3			3
	Raw score	126	171	171	141	156	52	195
	Relative Weight	12%	17%	17%	14%	15%	5%	19%
	Unit of Measure	psi	degrees	degrees	ft	$	psi	lb

Table 2 - Suspension Quality Function Deployment

Engineering Requirements for given design								
Customer Needs	**Weight**	**Ground Clearance**	**Suspension Travel**	**Y.S.**	**Stiffness**	**Spring Rate**	**Cost**	**Weight**
1. Lightweight	10					3	3	9

2. Maneuverability	10	9	9		3	9	3	9
3.Relatively inexpensive	6		1				9	
4. Must be safe	7	3	1	9	3		1	
5. Must be durable	8			9	9		3	
6. Transportable	8	3	3					3
	Raw score	135	127	135	123	120	145	204
	Rel. Weight	14%	13%	14%	12%	12%	15%	21%
	Units	in	in	in	lb	lb/in	$	ft

Chapter 2: Concept Generation and Selection

The suspension and steering systems were open to much debate early in the design process, with three concepts between the front suspension, rear suspension and steering systems[1]. While all the designs considered had high points, the selections ultimately were chosen because of their simplicity, durability and the satisfaction of objectives.

I: Front Suspension

Ia. Front Suspension Concept 1: Twin Trailing Arm Design

Starting with front suspension, the three concepts were double a-arms, twin I-beam and a front twin trailing arm in the style of an early Volkswagen suspension. The twin trailing arm (seen in *Figure 1*) is a durable design that has the advantage of moving away from stuck obstacles in a backwards arc toward the rear of the vehicle. Twin trailing arms are also very heavy, bulky and lack any camber change throughout their travel, making them not ideal for a front suspension system.

[1] The double a-arm design is used for the front and rear suspension

Figure 1 - Front twin trailing arm suspension design [2]

Ib. Front Suspension Concept 2: Twin I-beam

Another considered design for the front suspension was the twin I-beam (seen *in Figure 2*). This design is very simple because it is just solid beams that pivot on the opposite sides of the wheel it is controlling, making it very durable with lots of travel. The design suffers from being very heavy though and with radical camber and caster change, making it moderately difficult to control though the entirety of travel.

Figure 2 - Twin I-beam front suspension [3]

Ic. Front Suspension Concept 3: Double A-arm:

The final design considered was for the front and also the rear suspension, the double a-arm (seen in *Figure 3*), and is very common with the SAE Mini Baja. The double a-arm was chosen to be the suspension design for the front because of the camber control throughout the entirety of the travel, making the vehicle easier to control over obstacles. This design is weaker than previously discussed designs but for the needs of the vehicle it should have more than enough strength if designed correctly. This design was not chosen for the rear because camber control is not as high of a priority

Figure 3 - Double A-Arm suspension [4]

II: Rear Suspension

IIa. Rear Suspension Concept 1: Rear Semi-Trailing Arms

This suspension concept utilizes independent lever arms pivoting from one or two points on the frame and continuing at an angle back to the CV axles for drive. The semi-trailing arm design has the advantage of being durable and strong while also being very simple to design in a desired amount of travel and static camber. Unfortunately though, the amount of camber hardly changes throughout the travel of this design, letting it suffer from some stability and traction problems at the extremes of travel. Though, with those disadvantages in mind, a much larger vehicle might experience more dangerous consequences than anything in the SAE Mini Baja competition.

Figure 4 - Rear Semi-trailing Arm [5]

IIb. Rear Suspension Concept 2: Rear Live Axle (solid axle)

This suspension concept utilizes a front independent suspension with a live rear axle. Live axles get their name from the fact that the whole axle moves whenever either wheel hits a bump. Live axles are simpler, tougher, and more durable than independent suspension systems. They also allow for increased articulation which is beneficial for rock crawling. The tradeoff is that live axles are heavier, increase the un-sprung weight, and do not allow the wheels to independently follow the contours of a rough road.

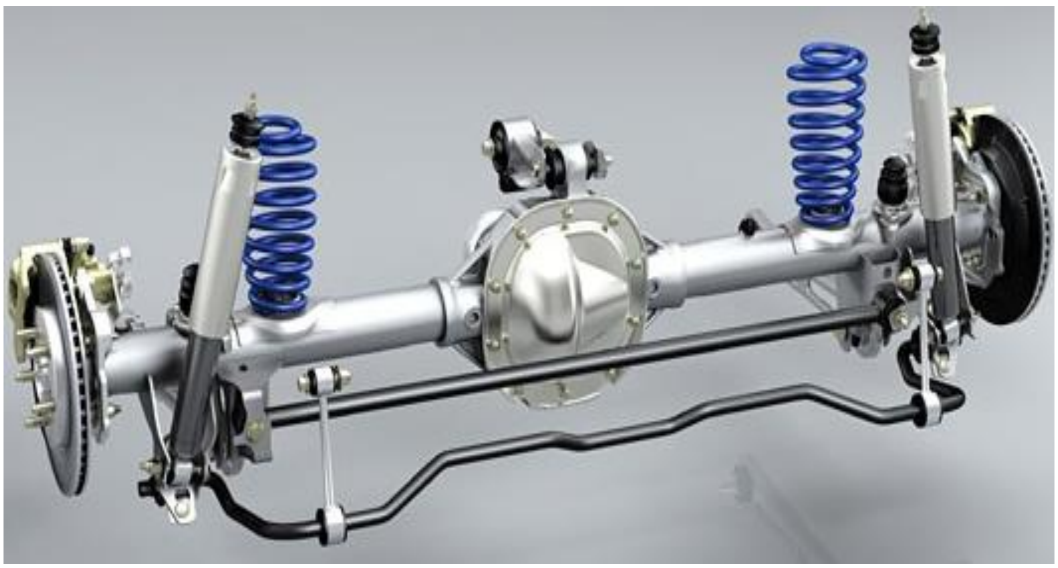


Figure 5 - Solid Rear Axle Design [6]

III: Steering Designs

IIIa. Steering Concept 1: Rack and Pinion Steering

The Rack and Pinion Type of steering will consist of a gear that is driven by the steering column and a gear rack that will mesh with the steering column gear. The rack is then connected to the tie rods that are connected to the hubs in a way where if they are pulled or pushed by the tie rods, the wheels will turn in the direction driven by the steering wheel.

The types of rack and pinion steering that are available are the spur gear type and the helical gear type. The difference between the two is the angle that the teeth of the gear make with the face of the gear, where the teeth on the spur gear are always 90 degrees with the face of the gear and the helical have an angle less than 90 degrees to the face of the gear. The difference in performance with the two are that the helical type has a smoother gear mesh while the spur type has a rough gear engagement. Although the drawback of a smoother mesh is a thrust load to the steering column that is created by the helix angle on the helical rack and pinion type.

If there is a problem in design where the gear ratio and type cause a problem with meshing then the right design that will be used will be the helical rack and pinion type. As for rack and pinion being compared to other types of steering, the response that rack and pinion produces is great but the amount of stress put on the driver can be taken into account according to the gear ratio that is used for the system. This factor will also count on whether the system is mechanically or hydraulically driven.

Figure 6: Simple Rack and Pinion with Spur Gear [7]

IIIb. Steering Concept 2: Pitman Arm

A pitman arm steering system consists of a box that converts the steering wheel input into a lever arm output. This Pitman Arm lever controls a track rod. Depending on the variation of this design the track rod is in some way connected to the tie rods that directly control the wheels to steer. The advantage of the Pitman arm system is that it is simple robust, and provides a mechanical advantage to the driver. For these reasons Pitman Arms are common on jeeps and other off-road vehicles. The disadvantages of the Pitman Arm system are that they have a "dead spot" allowing the steering wheel to turn before the wheels. With the advent of modern power steering systems that give the same mechanical advantage without the dead spot the Pitman Arms are falling out of favor.

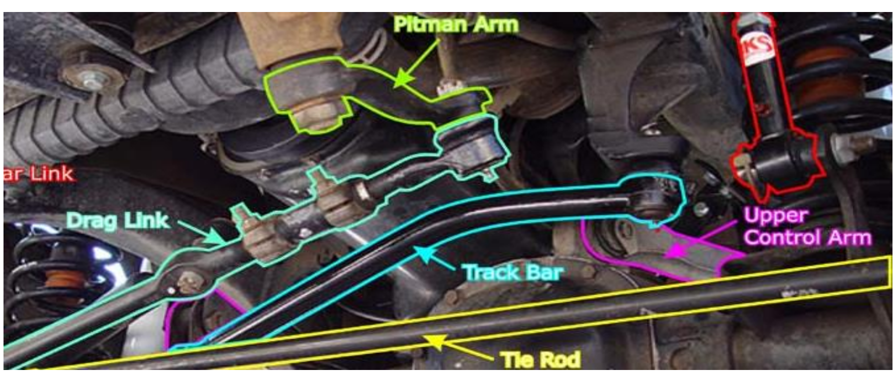


Figure 7: Pitman Arm Steering Assembly [8]

IIIc. Steering Concept 3: Steer by wire

Steer by wire systems are becoming more common as the price of computing power falls. In theory they can be simpler than traditional steering systems. They can save weight by using electrical controls instead of mechanical linkages. They allow for more advanced forms of Electronic Traction and Stability control. However because of the importance of steering the electrical connections need to be very secure along with the programs to control them. Also in the event of anything breaking they need to be very well grounded to allow for welding repairs in the field.

Steer by wire can be any type of steering system type with the intermediate step between the driver and the wheels being an electronic response device. The interaction with the steering wheel by the driver, later drives an electric motor that will drive the rack and pinion. With this type of steering system the advantages are corrections that can be made to the steering and the ease on the driver since an electrical motor will be driving the wheels instead of a person.

IV: Concept Selection

Based on the customer and team needs, several criteria were identified as the most important focuses of the design. These criteria are shown in Table 3:

Table 3 - Requirements Definitions and Weight

Requirements	Definition	Weight
Simplicity of build	The build must be easy to build with the equipment and materials available to the team	0.20
Reliability	The design must be reliable in a racing environment	0.30
Weight	The design must relatively light i.e. low un-sprung weight	0.30
Cost	The cost of the design and build must be affordable and cost effective	0.20

These criteria were identified from the project need statement as well as the customer's requests. It is important that the designs be simple to build with the limited equipment available to the team. The designs must also be reliable, the vehicle will be used in an off road race environment so the parts must be able to handle varying terrain and events. Thirdly the weight of each design must be relatively low. A higher weight in the suspension and steering systems could affect the vehicle's performance during competition by increasing the power to weight ratio as well as increase costs. Finally the designs must be relatively cheap to purchase from off the shelves if necessary.

Using these requirements decision matrices were formed for the front and rear suspension systems as well as the steering system. The decision matrices were used to help the team in deciding which design would be most beneficial. The weights were assigned to each of the aspects the team felt were most important, the higher the weight the more important the requirement. After the criteria were weighted, the designs were rated 1-5 (1 being the worst and

5 being the best) for each criteria and then the weighted amounts were summed. The decision matrices are shown in Tables 2-4:

Table 4 - Suspension Decision Matrix (Front)

Requirements	A-arm	Equal I-beam	Solid Axle
Simplicity (0.20)	4	4	5
Reliability (0.30)	4	4	5
Weight (0.30)	3	3	1
Cost (0.20)	4	2	2
Totals	3.7	3.2	3.2

Table 5 - Suspension Decision Matrix (Rear)

Requirements	A-arm	Solid Axle	Trailing Arms
Simplicity (0.20)	3	4	4
Reliability (0.30)	3	5	3
Weight (0.30)	4	1	4
Cost (0.20)	4	2	4
Totals	3.5	3.0	3.7

Table 6 - Steering Decision Matrix

Requirements	Rack & Pinion	Pitman Arm	Steer by Wire
Simplicity (0.20)	5	4	2
Reliability (0.30)	4	5	2
Weight (0.30)	4	3	3
Cost (0.20)	4	3	1
Totals	4.2	3.8	2.1

These designs scored the highest in their respective decision matrices, and fulfill the design requirements created by the customer and team. Based on the team's requirements the decision matrices confirm that the most beneficial designs are:

- Front Suspension: independent, double a-arms
- Rear Suspension: trailing arms
- Steering system: rack and pinion

V: Team Designs (Initial and Final)

Based on the decision matrices above the team decided to research and design a few different designs that fall into the rack and pinion, double a-arm, and trailing arm categories. The following section will describe each initial design with the aid of SolidWorks and other visuals. For the suspension systems the team had to work around the frame and drivetrain teams' designs for efficient meshing when the final production begins. For the steering system, two rack and pinion designs were used: the first design is a helical rack and pinion, and the second is a spur rack pinion. Once these initial designs were created, they were evaluated and redesigned for better efficiency and meshing with the other Mini Baja components.

Va. Front Suspension

The final design for the front suspension is a short long arm double A-arm design. This design allows for a good range of travel, durable, and is cheap to manufacture. The final design geometry is laid out in *Figure 8*. This Figure shows the overall geometry of the suspension based on the constraints of frame width of 20", a spindle with 8" between the eyes of the uniballs, and a need to keep the overall width under 64". With the goal of keeping the scrub radius to zero and having a kingpin angle of no more than 15 degrees the final a-arm dimensions are 12.125" for the upper arm and 14.5" for the lower arm.

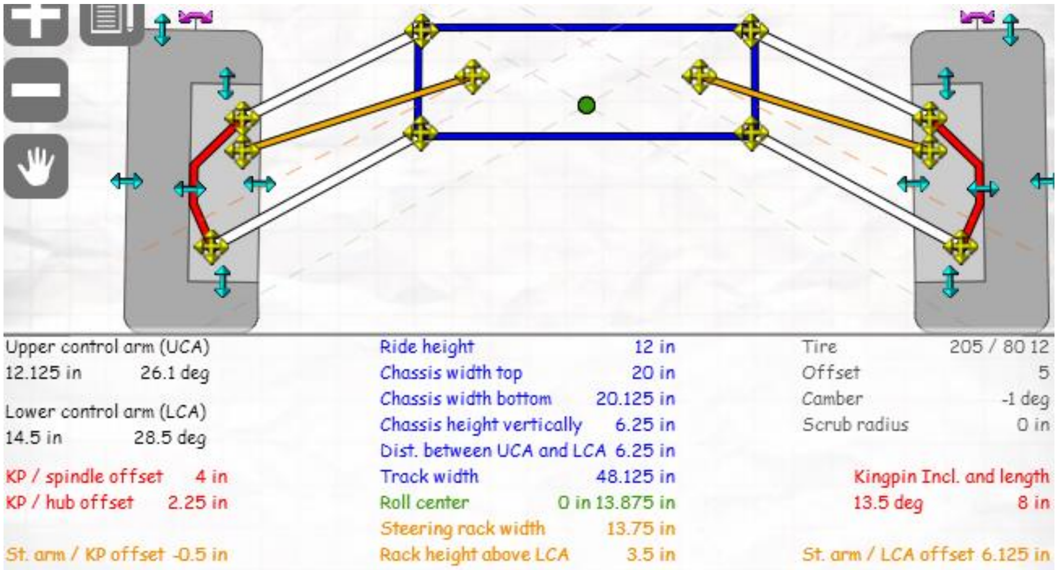


Figure 8 - Front suspension geometry [9]

Figure 9 shows the suspension geometry at full compression. It shows the wheel loses 4.5 degrees of camber at a maximum of 7" upward travel. This is beneficial to maximize the wheels contact area with the ground under hard cornering. It also shows that the steering control tie rod not interfering with the travel.

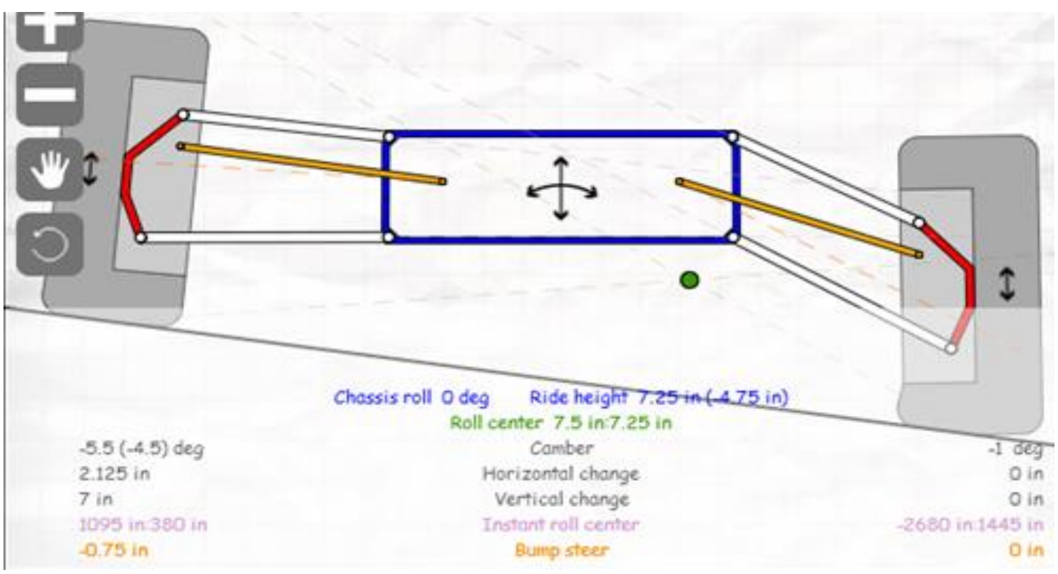


Figure 9 - Front suspension at full compression

Figure 10 shows the suspension geometry at full droop. It shows a modest camber gain of .6 degrees at the full drop of 4". It also shows that the steering tie rod has clearance. This figure shows a 3d cad model of the front suspension as it mounts to the frame.

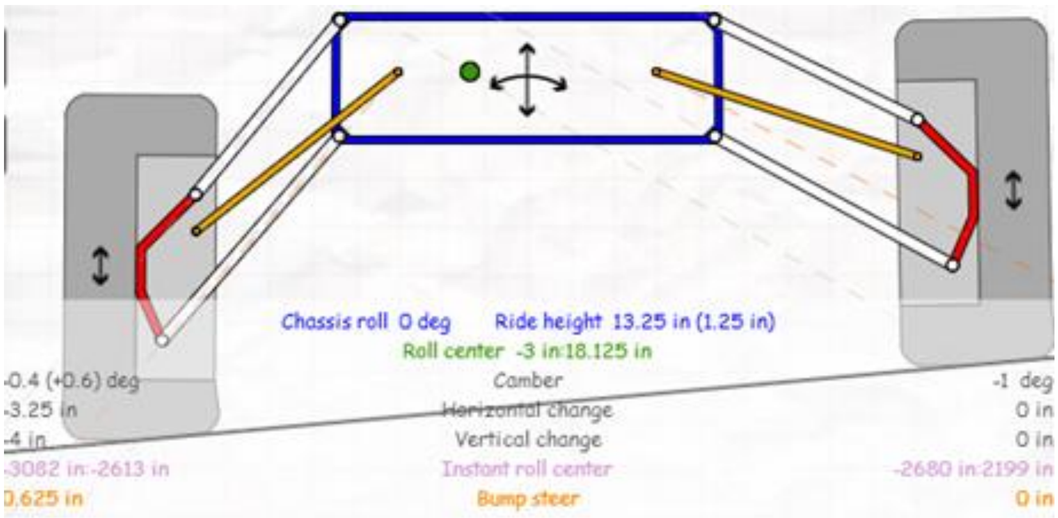


Figure 10 - Front suspension full droop

Once the geometry for the front a-arms was set, a 3d SolidWorks model was created for visual inspection, mating to the frame, and engineering FEA analysis. *Figure 11* and *Figure 12* show the completed assembly of the SolidWorks a-arm model and the final frame design with a basic mock-up of a front shock.

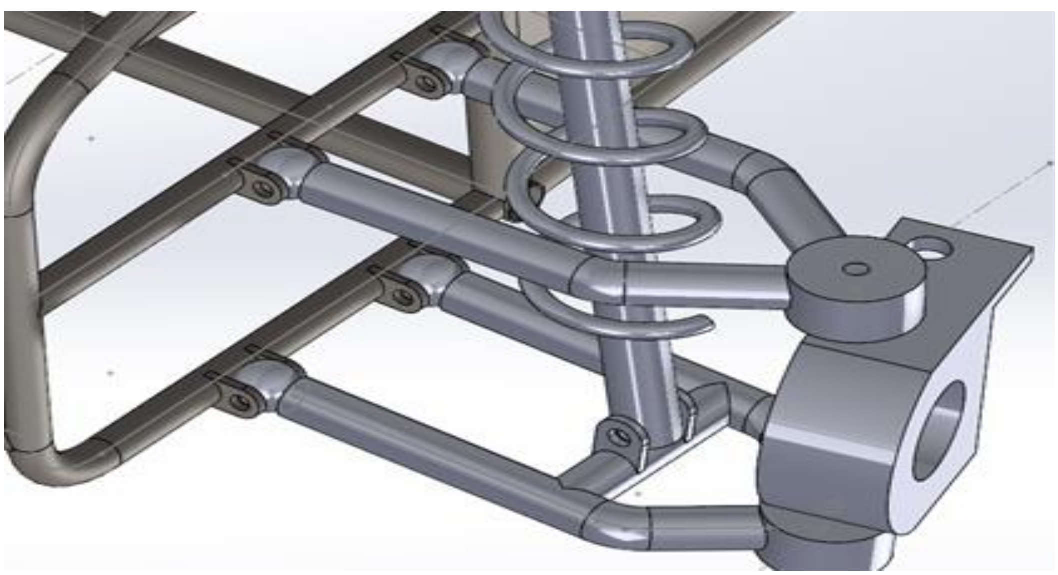


Figure 11 - Front Suspension Geometry (iso-view)

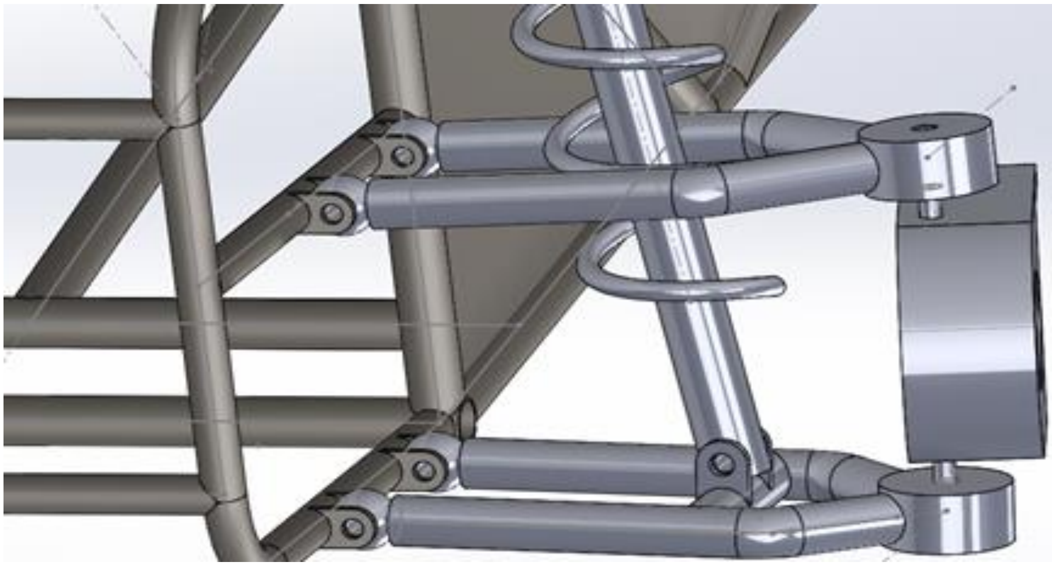


Figure 12 – Front suspension geometry (front view)

Vb. Rear Suspension

Several iterations were conceptualized during the design process for the rear suspension. All of the designs considered for the rear suspension were of independent nature, meaning each wheel is allowed to move throughout its travel independently of each other. The team felt this was an extremely desired design characteristic to have in an off-road racing environment. Each possible design flaws and attributes were analyzed and compared with one another with the use of design matrices and thorough research. In the end the entire SAE Mini Baja Team decided that a three link rear trailing arm adaptation would be the best design route for the following reasons: ease of build, simple geometry, weight, allowable travel, and this design (along with other multilink designs) are becoming the industry standard. The team thought this would be an important consideration to take into account because not only is the Mini Baja vehicle supposed to be competitive, it may be seen by ATV/UTV industry leaders as a possible marketable vehicle.

Figure 13 shows the initial design for the rear trailing arm. It is made of AISI 4130 chromoly steel with a 2in OD and 0.063 wall thickness. Because this was an initial design, it was overbuilt to handle any expected forces during competition and testing.

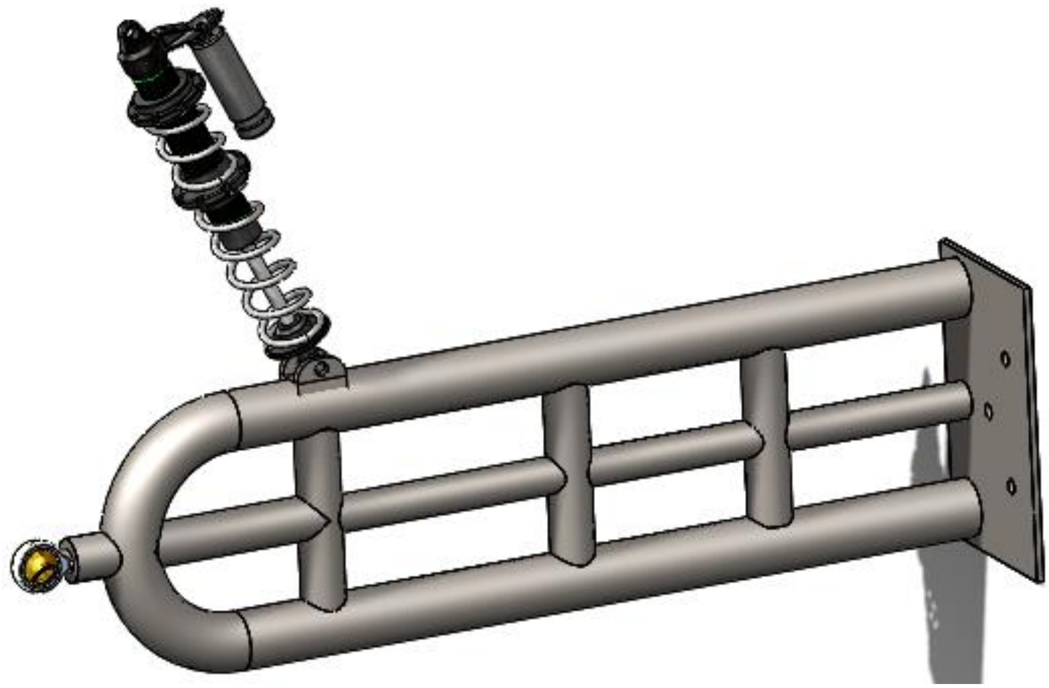
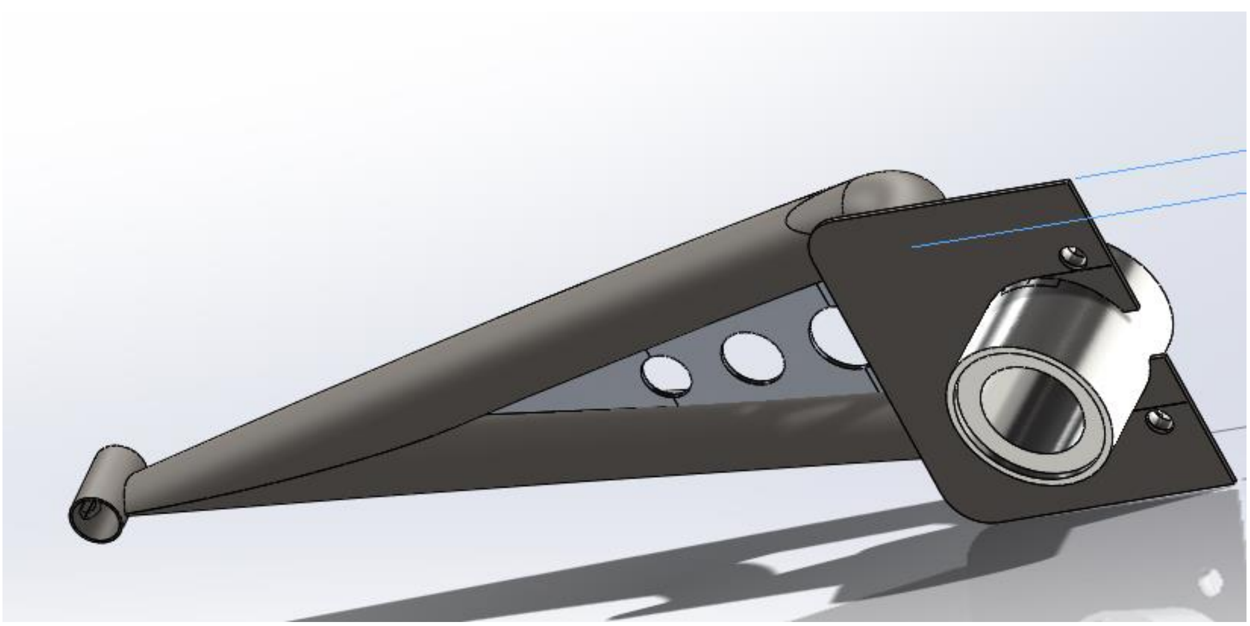


Figure 13 – Rear trailing arm geometry

Due to high component weight, unnecessary tube dimensions, and an unattractive, impractical design the rear trailing arm was redesigned to meet the desired strength, weight, and cost, limitations. *Figure 14* shows the final rear trailing arm design assembly.



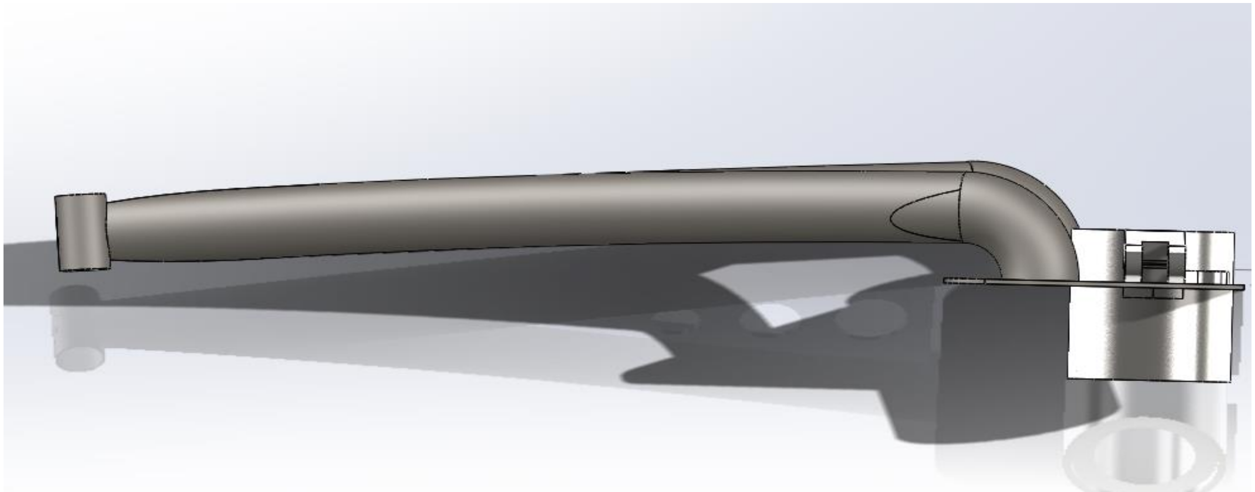


Figure 14 – Front and top view of trailing arm

Further research was done on rear trailing arm designs in commercially produced recreational vehicles and other collegiate SAE Baja teams. The final design used benchmark data to from the Polaris RZR XP 900 UTV model, shown in *Appendix Figure C2* [9], and other schools for the final design. The trailing arm dimensions are shown in *Table 7* below.

Table 7 - Rear Trailing Arm Dimensions

Rear Trailing Arm	**Dimensions**			
	L (in)	**OD(in)**	**Wall Thickness (in)**	**θ (degrees)**
	26.00	1.50	0.063	3.00

Many of the components that will be used in the front and rear suspension systems will be off-the-shelf parts, used and new. As stated above, the final trailing arm design resembles the Polaris RZR XP 900 for assembly. The team plans to use many Polaris ATV parts for the final assembly, *Figure 15* is a rough depiction of the rear upright for the rear trailing arm and linkages that will connect to the frame. The rear upright is made of a cast steel, and the linkages are made of 1.0 in OD solid stock AISI 1018 steel. The drawing sand dimensions are shown in *Appendix A.*

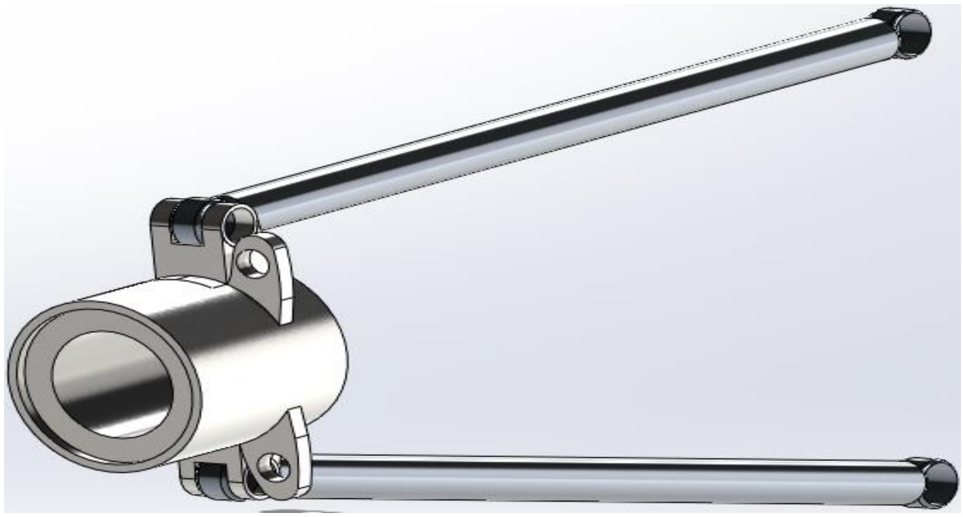


Figure 15 – Rear upright with linkages

Vc. Steering (Tie Rod)

When designing the steering for the mini Baja the team chose the spur rack and pinion design because it was the simplest design and can easily be easily obtained from local shops or junk yards. The concepts analyzed from the steering system are the gears on the rack and pinion and the tie rod. The tie trod design can be seen in *figure 16* below, where Heim joints are connected to a solid AISI 4130 Steel rod for strength when turning. The rod will be connected to the hub at one Heim joint and the other Heim Joint will be connected to the rack. The reason for the Heim joints are to provide movement as the A arm suspension moves up and down. The Heim joints will provide an optimal amount of movement but not so much as to change the suspension or steering geometry or to interact with any of the frame or suspension parts. The tie rod is easily manufacture-able as a part because it is a simple cylinder. The part is also readily available in all of the motorsports world, the only problem with buying the part would be adjusting the length of the tie rod to reach between the hub and the end of the rack.

Figure 16 - Tie rod with heim joints

Vd.Steering(Rack &Pinion)

After deciding on choosing a rack and pinion design for our steering system, the team began to calculate what the minimum radius would be for the pinion and what the stresses would be on that minimum size pinion. These stresses and geometric measurements were calculated using MATLAB code and making several assumption. For example, we assumed a non-crowned pinion, not operation at high temperatures (temperatures above 300 degrees Celsius). The following stresses and measurements for the pinion are as follows:

Table 8: Rack and Pinion Stresses and Geometry

	Teeth Number	Face Width (in.)	Bending Stress (kpsi)	Radii for Pitch Circle (in)	Radii for Base Circle (in)	Adden. (in.)	Dedden (in)
pinion	20	0.74	0.04 - 3.9	0.787	.739	0.078	0.098
rack	40	0.74	-	inf	inf	0.078	0.098

As for the geometry for the rack, the team decided that the steering column would turn a maximum of 360 degrees to the left and right. This information would be used to decide how many teeth the rack would have. The teeth number on the rack was calculated by multiplying the pinion teeth number by 2, given that the rack would have to have the equivalent of two full rotations of the pinion. This number ended up coming out to 40 teeth for the rack. In addition, the team was able to calculate a rack length of 9inches. The team then took this information and began to calculate a turning radius for the vehicle and came out with a number of 12 feet. Since this turning radius is not at the range that the team wanted to be (which was to be less than 75% of 15 feet), the team decided to increase the rack size to 14 inches in order to create a smaller turning radius for the vehicle. The following figures show the current rack and pinion selected for the team's competition vehicle.

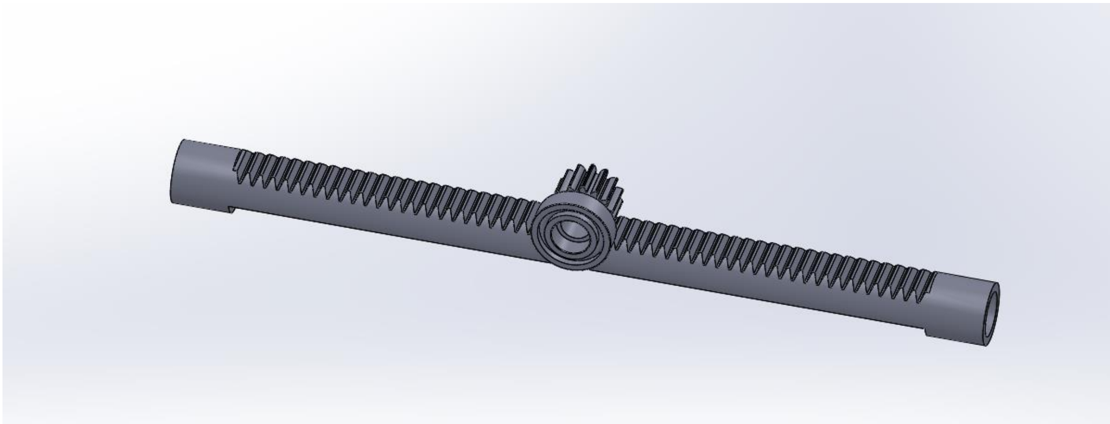


Figure 17: Rack and Pinion Interior

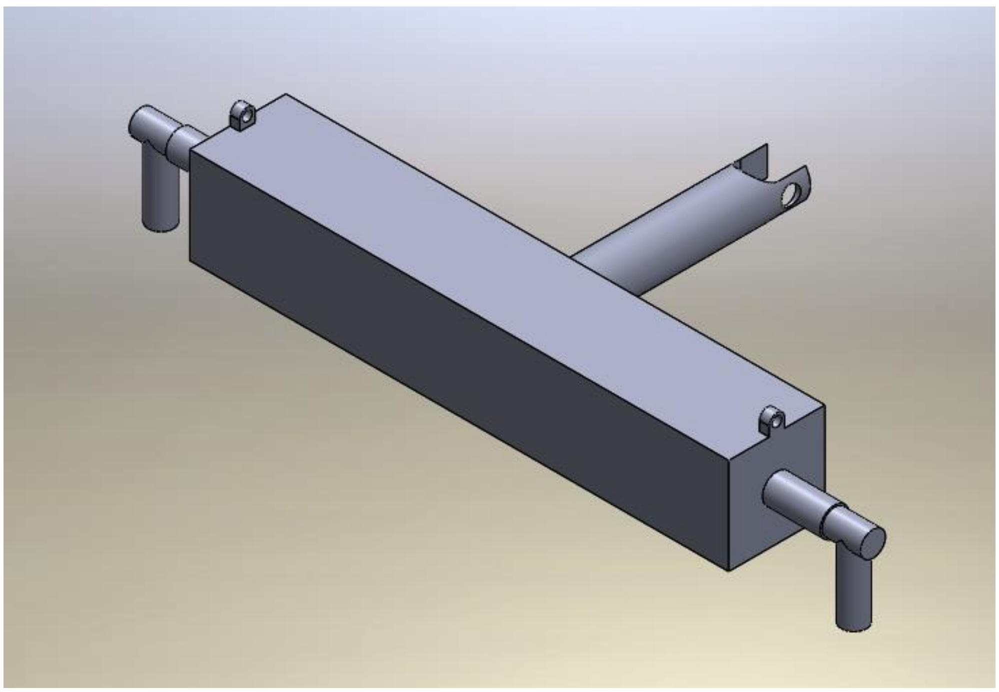


Figure 18: Rack and Pinion Enclosed

With these calculated stresses and geometric measurements, the team will have values to reference from when buying a “off shelf” rack and pinion steering system.

Chapter 3: Engineering Analysis

After deciding on the corresponding designs for the front suspension, rear suspension, and the steering the team began to conduct FEA calculation analysis using SolidWorks. This was in order to see where the higher stresses were located on each design. Knowing this information, the team would be able to create a tested design without building a physical model and use the teams limited project funding. The analysis was completed for the front suspension members and components, the rear suspension members, rear upright, and steering system components.

I: Material Selection

The material selection process for the Mini Baja vehicle was a team decision. The Mini Baja SAE rulebook states that the frame must be made of AISI 1018 steel or another steel with equivalent bending strength, bending stiffness, and minimum wall thickness of 0.062 in. Through research, the frame team found that AISI 4130 was the best choice. This particular steel is widely used in racing applications outside of the SAE collegiate Competitions and in some aerospace applications. In order to cut cost in the suspension team's budget, leftover 4130 tubing, with a 1.5 in OD and 0.063 in wall thickness from the frame build will be used for the suspension members. *Table 8* shows AISI 1018 and 4130 steel material properties.

Table 9 - 1018 vs. 4130 Steel

Material	Properties					
	Sy (ksi)	Ts (ksi)	E (ksi)	ρ (lb/in^3)	G (ksi)	ν
AISI 1018 (CD)	54	64	29000	0.284	11600	0.292
AISI 4130 (normalized)	63	97	29700	0.284	11600	0.292

II: Tie Rod Analysis

When analyzing the steering components of the SAE Baja vehicle, the most important factors are the yield strength of the tie rod and the steering column. Other factors that are dependent on these specific parts of the system are how the gearing and the torque from either the driver or the environment affect the system intermediately. By analyzing the steering system with different situations in MATLAB (*figure 18)*, it could be seen that the force put on the tie rod by the driver and the vehicles environment affects the need for a larger cross sectional area of the tie rod. The placement of the tie rod on the back of the hubs will also affect the steering system because the force exerted on the tie rod will be greater if the tie rod is connected farther from the point of rotation on the hubs of the vehicle.

Assumptions made are that the material that will be used for the tie rods will be chromoly (AISI 4130 Steel) and that the steering column be rigid because the torsional yield (shear stress on outer radius) of the steering column would be negligible compared to the yield stress in the tie rod. The reason for using chromoly as a base material is because most pre manufactured tie rods are made of this common steel so during post analysis, choosing a tie rod for the vehicle will come with ease. The properties of the chromoly used for the simulated analysis is given in *Figure 18*. The results generally show that as the forces exerted by the tie rod and the environment increase so does the need of a greater radius on the tie rod.

A more detailed factor to take into account in analyzing the steering system is the gearing of the rack and pinion and it is directly correlated to the forces that are exerted on the tie rod and the steering column.

During the analysis of the tie rod by the use of SolidWorks, it can be seen that with a rod made of AISI 4130 Steel, a tie rod with a very small radius can be used. But due to the available heim joints that will be able to take the forces of the hub and the steering wheel, the threads would have to be larger than the minimal tie rod radius. The deformation can be seen for the rod at 3000 lbf in the *Figure 17*. At 3000 lbf the maximum deformation of the rod is 0.13mm. The forces on the tie rod will not be this great and the tie rod design is centered on available heim joints so the tie rod will not yield before the heim joints by the forces applied.

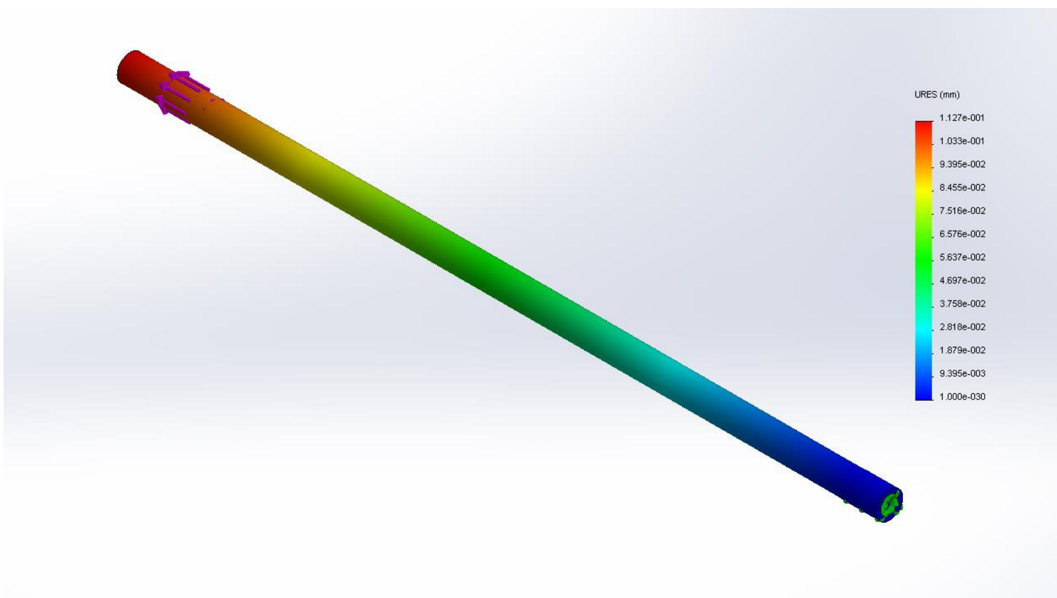


Figure 19 - Tie rod axial deformation at 3000 lbf

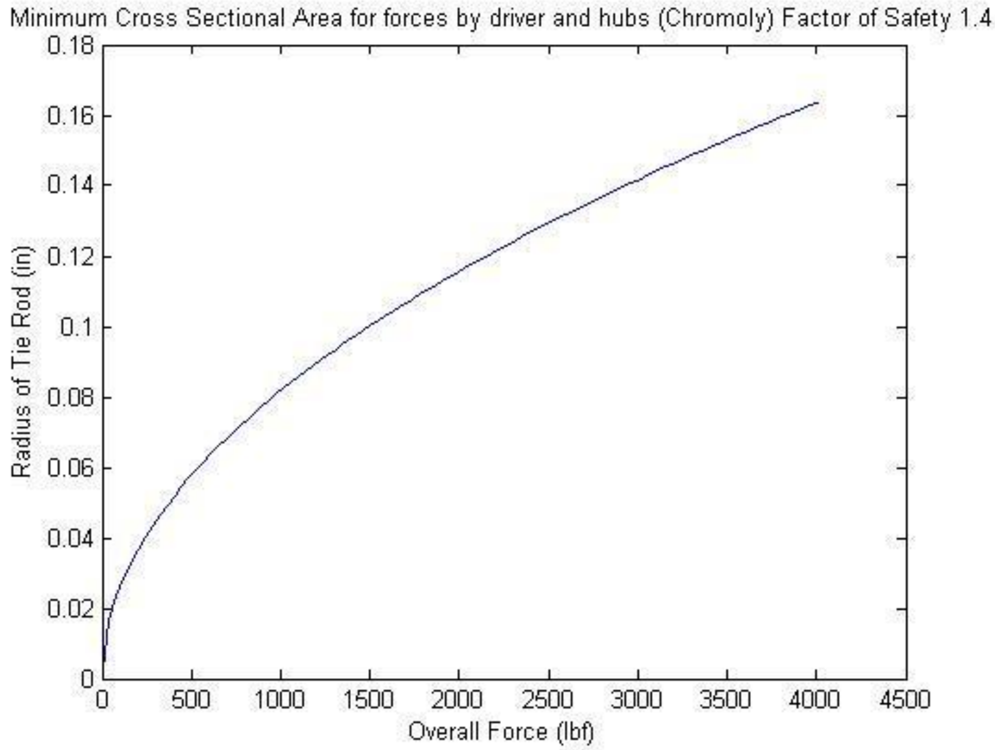


Figure 20 - Varying Force with Required Tie Rod Radius

III: Upper and Lower A-arm Analysis

The following calculations were done to determine the maximum force that the Mini Baja is likely to encounter. *Equation 1* [10] calculates the maximum force from a vertical drop onto one wheel.

Drop Test Assumptions:

1. $Fi\ =\ Force\ of\ impact$
2. $Fs = 500\ lb\ Weight$
3. $h =\ 6\ ft\ Drop\ Height$
4. $Ks =\ 160\ lbin\ \ (using\ shocks\ from\ Polaris\ RZR\ 570)$

$$Fi =\ Fs\ +\ ((Fs)\ 2\ +\ 2\ x\ K\ x\ 12\ x\ Fs\ x\ h)1/2 \qquad (1)$$

$$Fi = 1022.53\ lbf$$

Based on this calculated force the expected maximum force that will be exerted on each arm will be about 511 lbs. The following figures show the arms in SolidWorks FEA analysis under a 700 lbf load. This is a 40% higher load than the arms are expected to experience under the worst case scenario. *Figure 16* shows the Von Mises stress FEA analysis result using the above assumptions and force calculated in *Equation 1*.

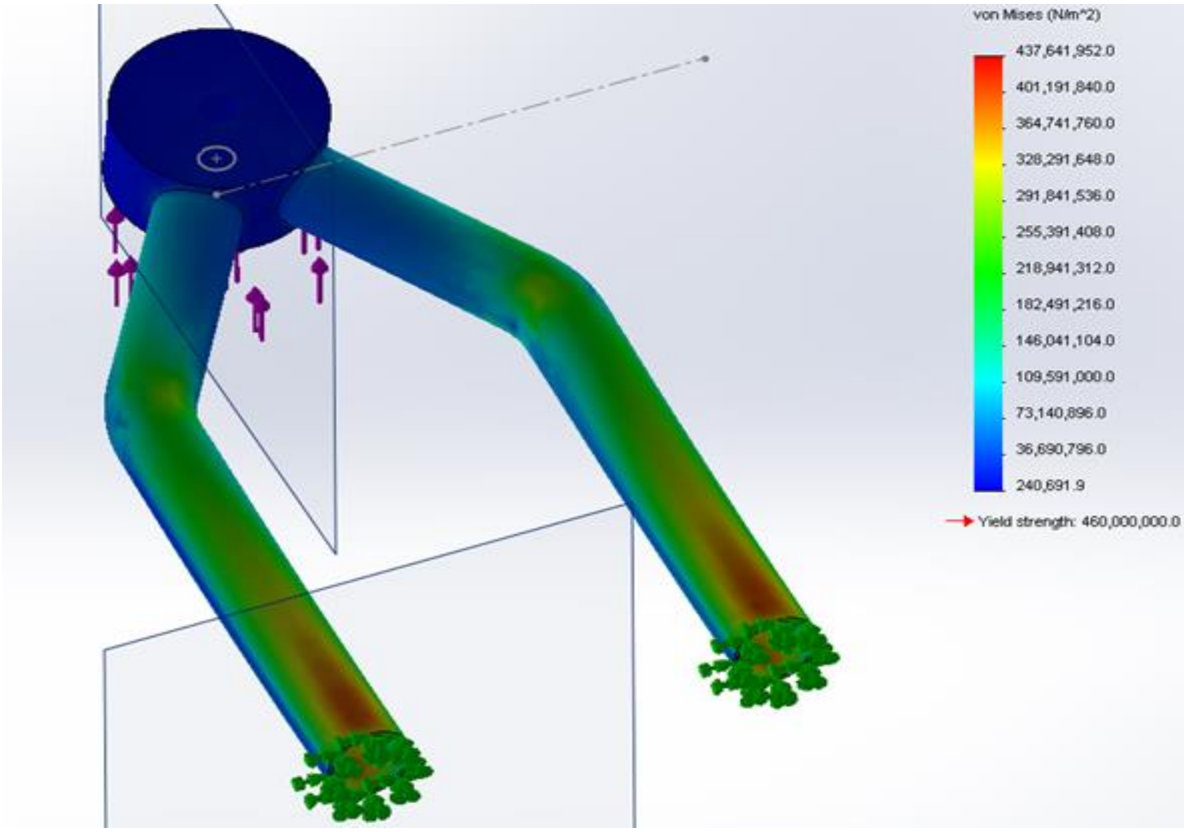


Figure 21 - FEA of upper a-arm (bottom view)

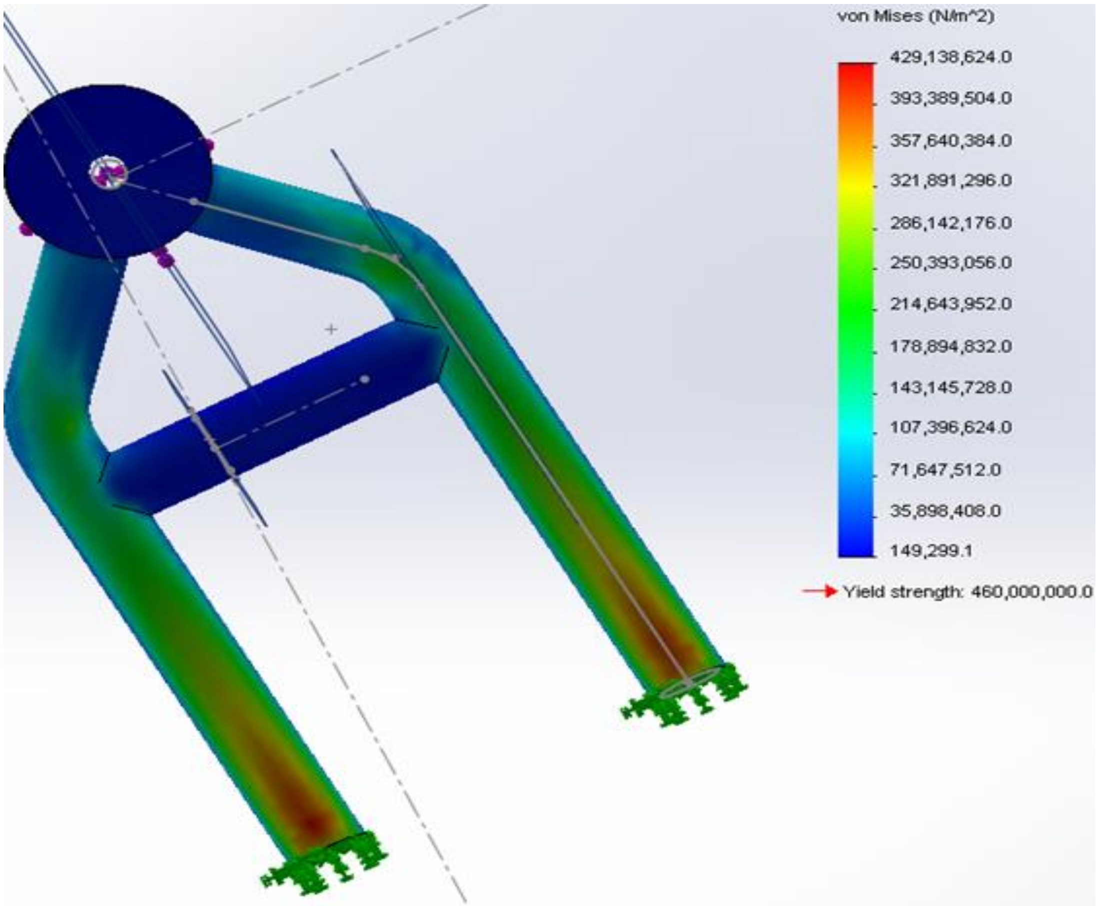


Figure 22 - FEA of lower a-arm (bottom)

The following calculations using *Equation 2* [10] are of the worst case scenario of a front collision. This is based on the theoretical top speed that the Mini Baja can achieve. This calculation is of the Baja running into a wall at full speed. Any impact forces that the Mini Baja would experience during normal operation would only be a fraction of this force.

Front Impact Assumptions:

1. $Max\ speed\ is \sim 35MPH = 51.33Ft/s$
2. $M = 500lb/32.2 = 15.53slug$
3. $T = .2s$

$$F_{impact} = M(V/T_{impact}) \tag{2}$$

$$F_{impact} = 15.53(51.33/.2) = 3985.77lbf$$

Based on this calculations, and distributing the load between the 4 arms, the maximum force that each arm would experience is 1000 lbf. The following SolidWorks FEA images show the effects of a 1000 lbf frontal load. *Figure 9* shows the FEA results of the frontal impact test done in SolidWorks Simulation Xpress.

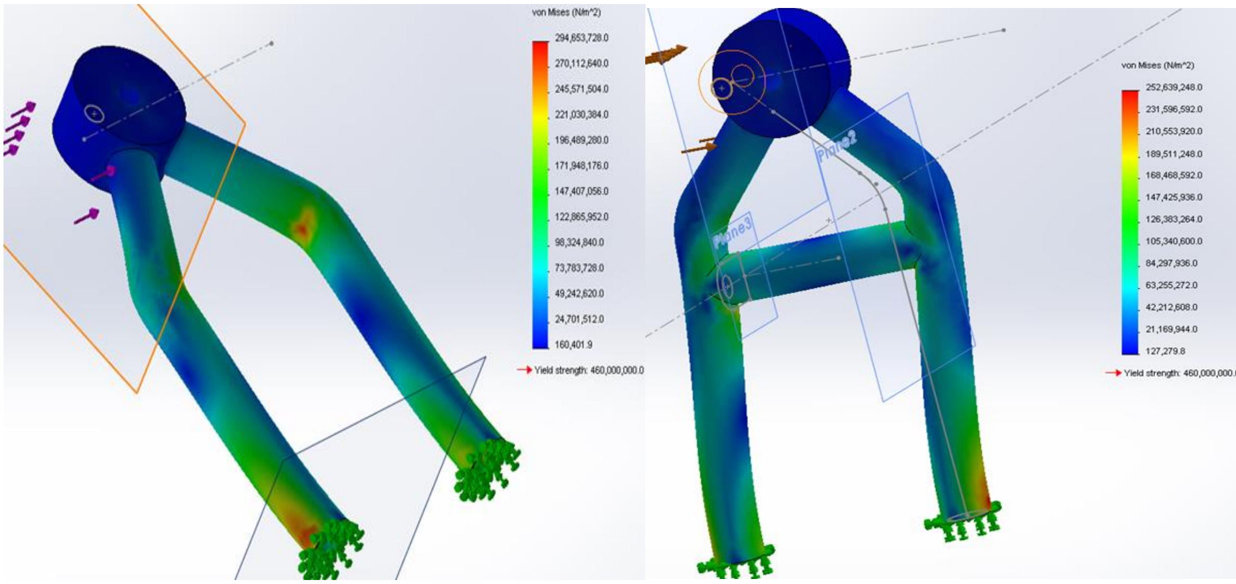


Figure 23 - FEA of upper a-arm (top and front view)

IV: Trailing Arm and Components Analysis

The following calculations were performed on the trailing arm member:

1. Impact test form a 6 ft drop
2. Side impact test
3. In order to design a reliable rear suspension design, the team had done extensive research in the areas of material strength, suspension member geometry, and other collegiate SAE Mini Baja designs. The first step was choosing the best material for the job.

The first analysis done on the rear suspension member was the drop test. Using the same assumptions in *Equation 1*, the maximum force that will be exerted on each arm will be about 511 lbs. *Figure 22* and *Figure 23* show the von Mises stresses and displacement results for an extreme value of 6000 lbf and a factor of safety of 3.5. The 6 ft drop assumption came from a maximum drop height that the team expects during competition most likely

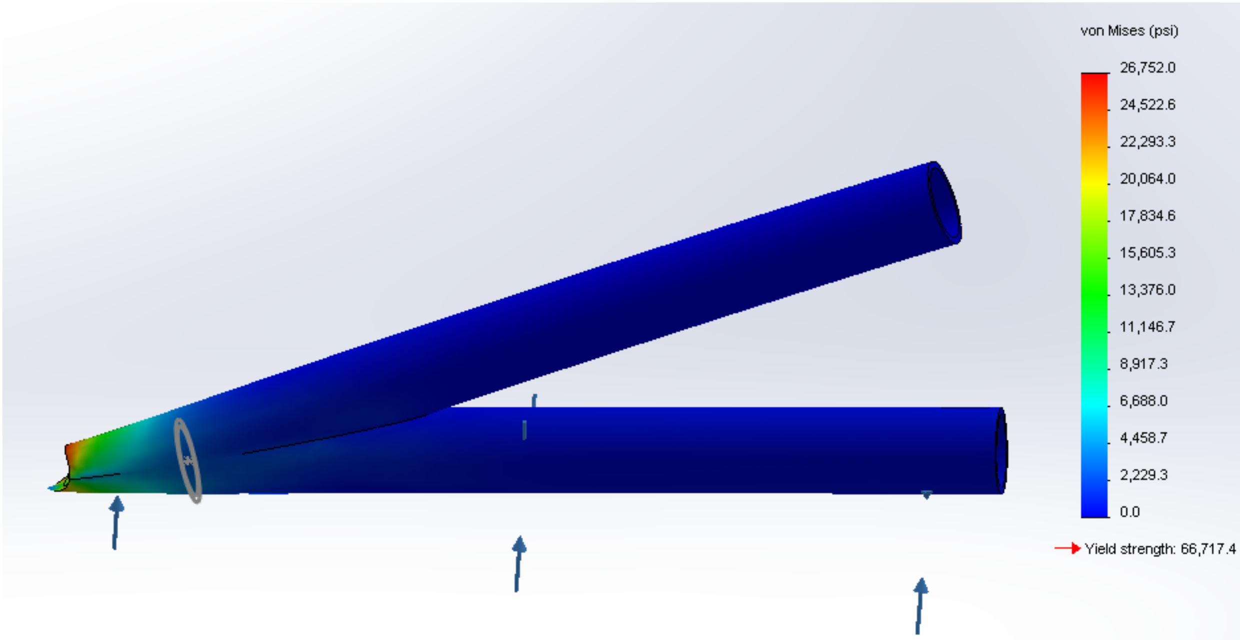


Figure 24 – Von Mises stress trailing arm bottom impact at 6600 lbf FS = 3.5

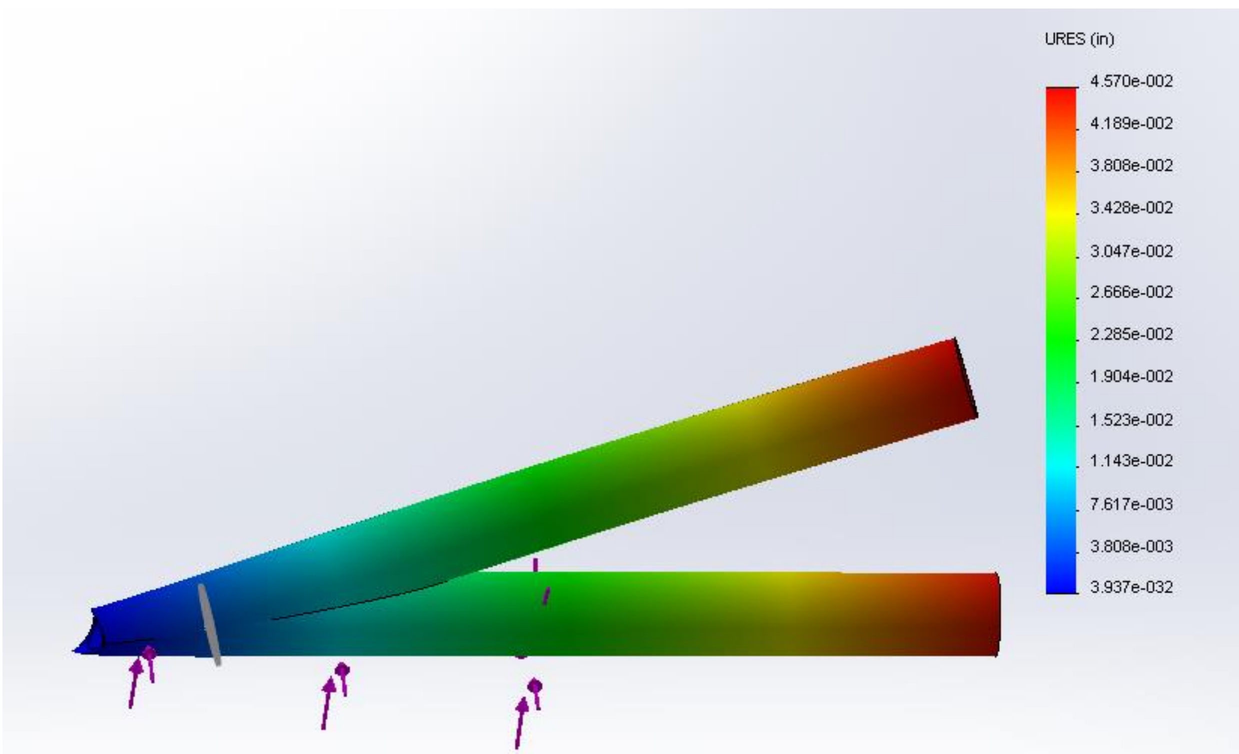


Figure 25 – Displacement trailing arm bottom impact at 6600 lbf FS = 3.5

V: Rear Upright Analysis

Many of the parts used in the suspension design will be off the shelf parts; such as shocks hubs, uprights, and other critical parts. Because of this design constraint, team decided to model the rear trailing arms after a well-known, ATV design. The rear upright shown in *Figure 24* is the displacement result of an FEA side impact test used in the a-arm analysis and *Equation 2*. Using the calculated force as a benchmark, the rear upright was tested at several different forces that ranged from probable to outlandish for this type of application. The analysis used an impact force of 10,000 lbf.

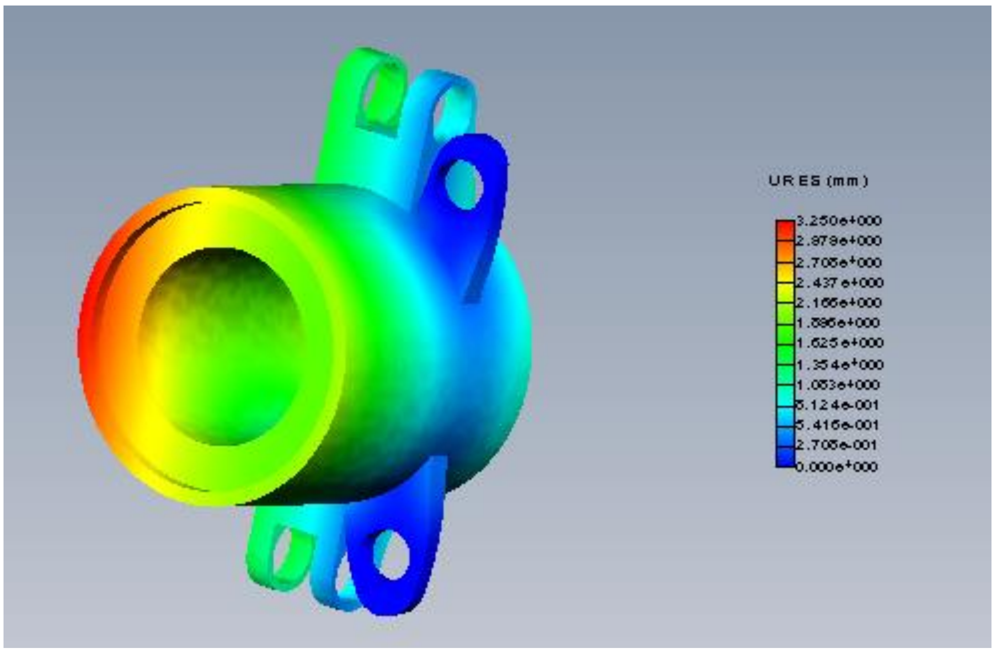


Figure 26 – FEA Rear upright, side impact 10,000 lbf, displacement

Figure 25 shows the FEA analysis of the same side impact test. The figure depicts the von Mises Stresses that the member will experience at 10,000 lbf. The results show that the greatest amou8nt of stress occurs at the mounting points, which is to be expected as its cross section are is much smaller than that of the main upright body.

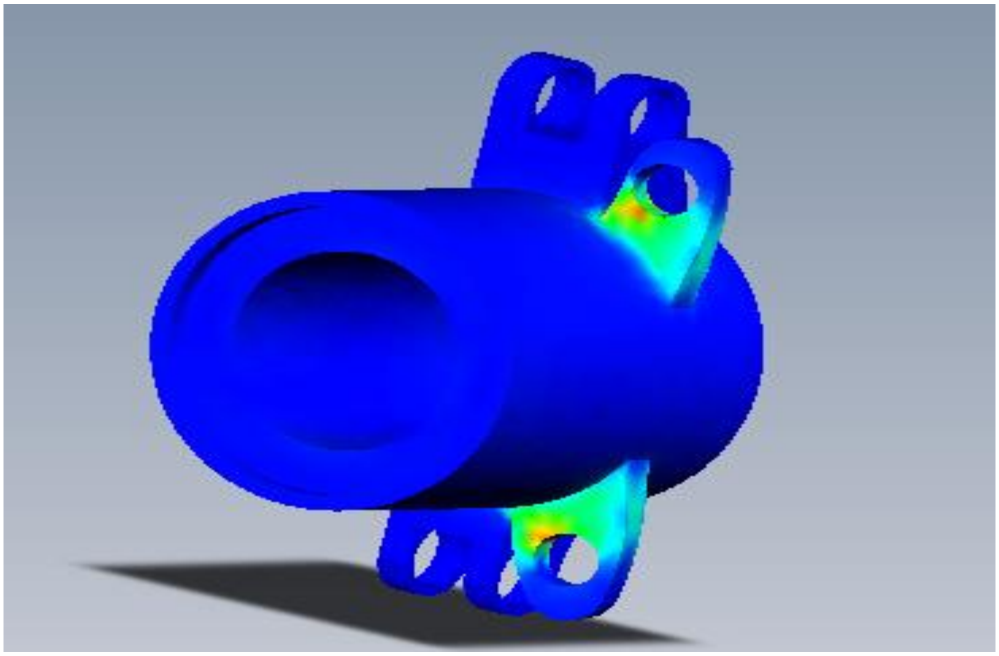


Figure 27 - Rear upright, side impact 10,000 lbf, Von Mises Stress

Chapter 4: Cost Analysis

The total cost of the front and rear suspension, steering, and brakes has been calculated based on assembled parts list of the respective systems. The parts lists and the included parts can be found in appendix B. Most of the parts are sourced from Polaris due to their generous sponsorship program. The following table shows the cost of the respective systems at full retail and at sponsorship prices.

Table 10 - Cost Analysis of Suspension and Steering Systems

	Front Suspension and Brakes	Rear Suspension	Steering	Total Cost
Full Retail Price	\$2,529.33	\$1,868.14	\$649.20	\$5,046.67
Polaris Sponsorship Price	\$1,440.33	\$1,067.67	\$324.60	\$2,832.60

Chapter 5: Conclusion

In order to fulfill requirements set by our customer, Dr. John Tester, and SAE Mini Baja Competition, the suspension and steering systems were designed to be lightweight, strong and overall competitive. Through several designs, for the front suspension, rear suspension and the steering systems, the very best were picked out and were modeled once chosen. Double a-arms were chosen for the front in order to have the front wheels keep contact with the surface for as much of the time as possible. Semi-trailing arms with lateral links were chosen for the rear due to their inherent strength and design simplicity. Rack and pinion is the chosen steering system because of the simplicity and lack of multiple mechanical components, which reduces the risk of a catastrophic failure during competition. Using a material choice of 4130 Steel, unanimously, for structural members all finite element analysis lead to promising results for all designed components and operating within a factor of safety of 3.5. After analysis was finished with models were finalized, costs were determined to be 2832.60 once sponsorship is acquired.

References

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[2] MOFOCO, VW Complete Recondition King Pin & Link Pin Front Axle Beam Beetle & GHIA 49-65," http://www.mofoco.com/item/VW_COMPLETE_RECONDITIONED_KING_PIN_and_LINK_PIN_FRONT_AXLE_BEAM_BEETLE_and_GHIA_49_65/599/c156

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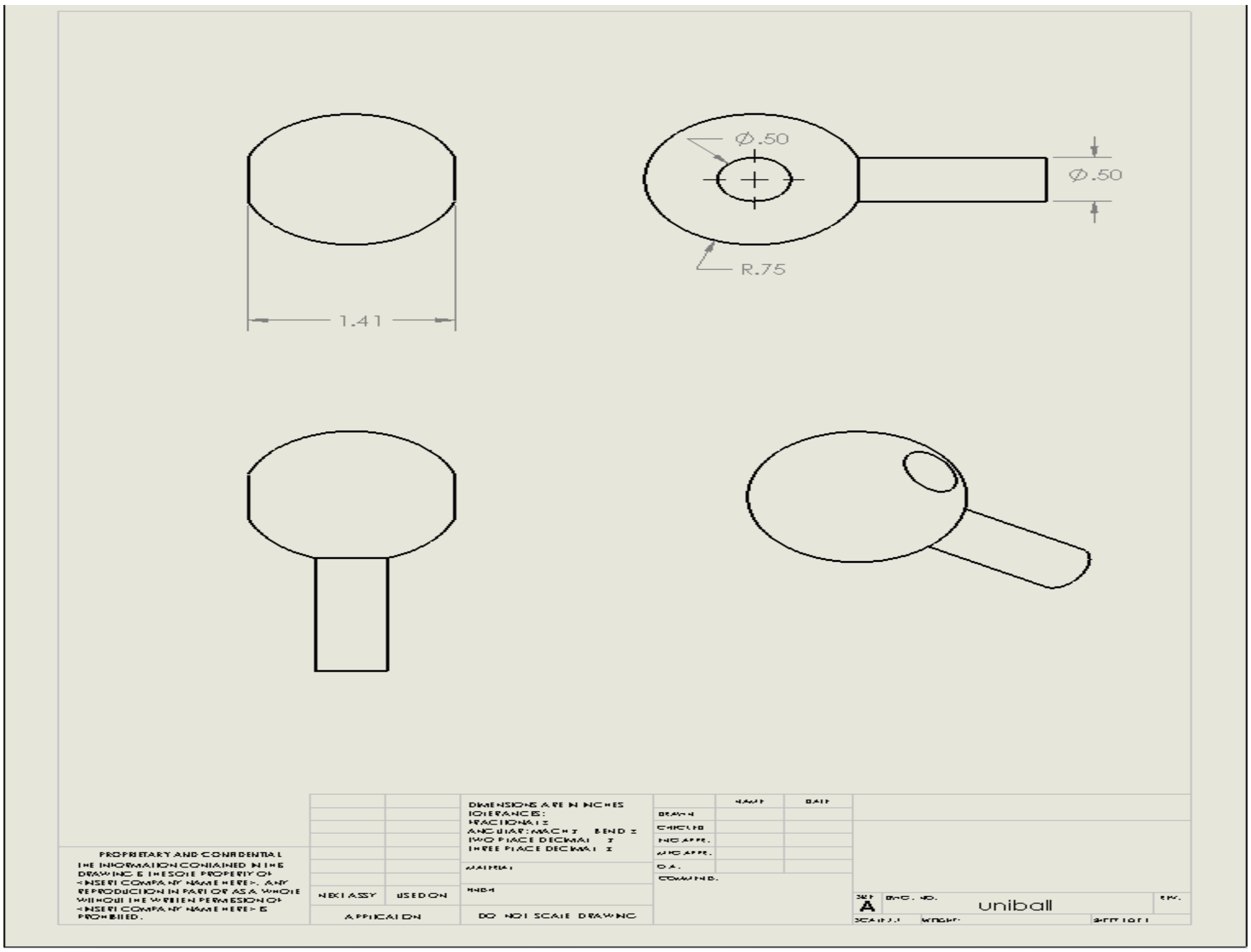
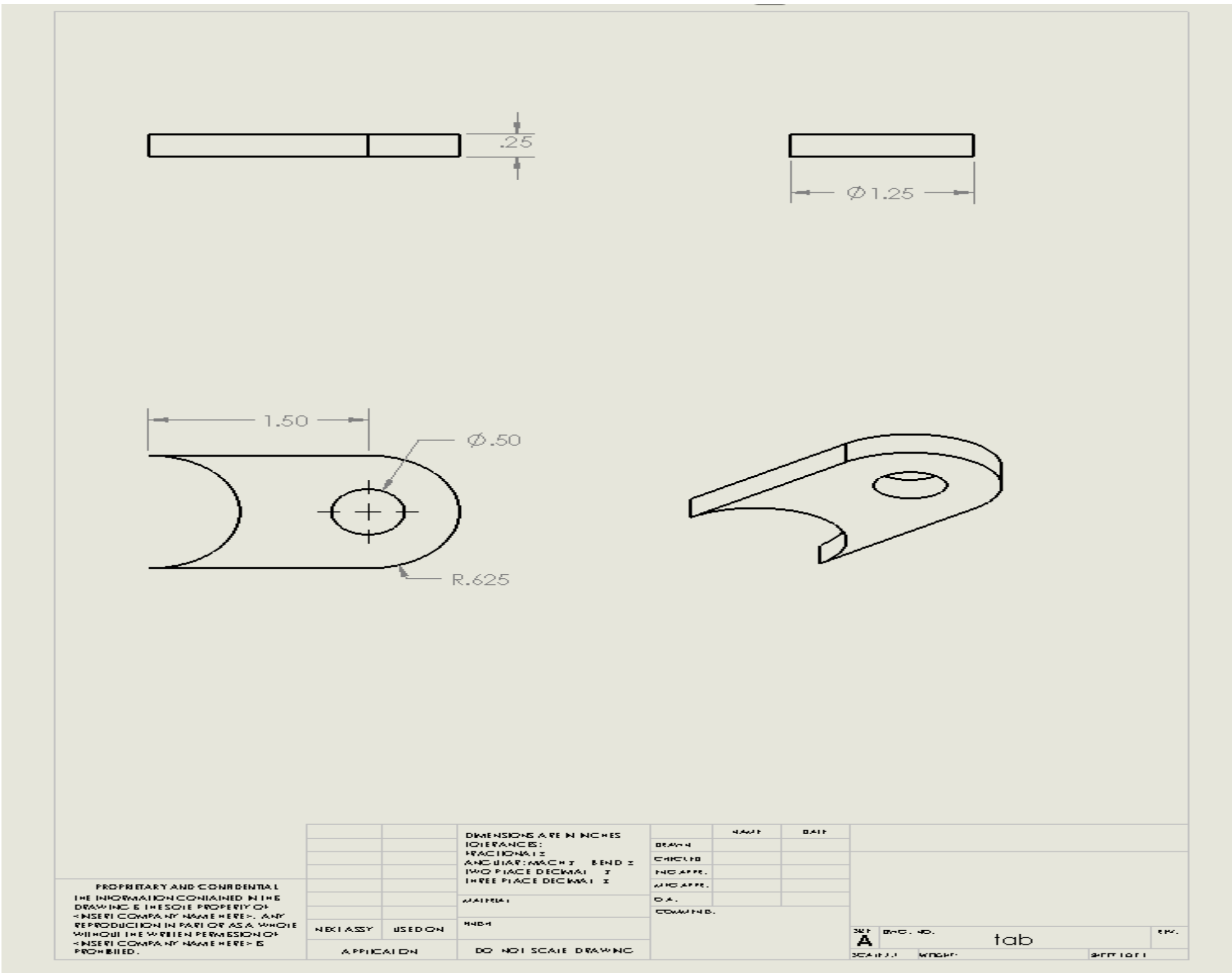
[10] Polaris RZ XP 900, "Suspension Design," http://www.utvguide.net/polaris_rzr_900_xp.htm, 2010

[11] Kenedi, Pacheco, Vieira, Jorge, Danninger, "Analysis of the Transmissibility of the Rear Suspension of a Mini-Baja Vehicle," Warrendale, Pennsylvania, ISBN 0-7680-1109-4
844085-9:101-110, 2002.

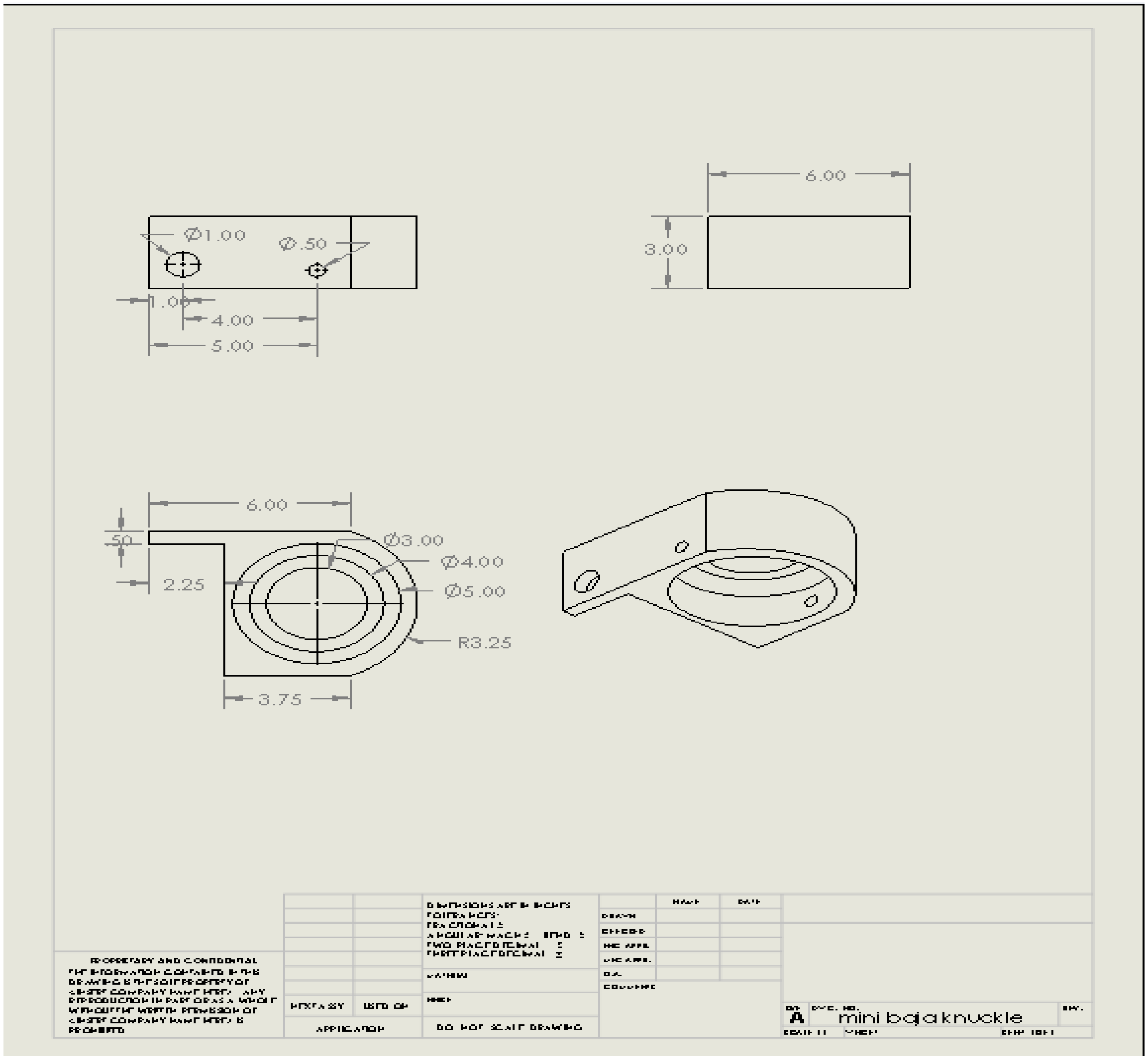
Appendix A: Engineering Drawings



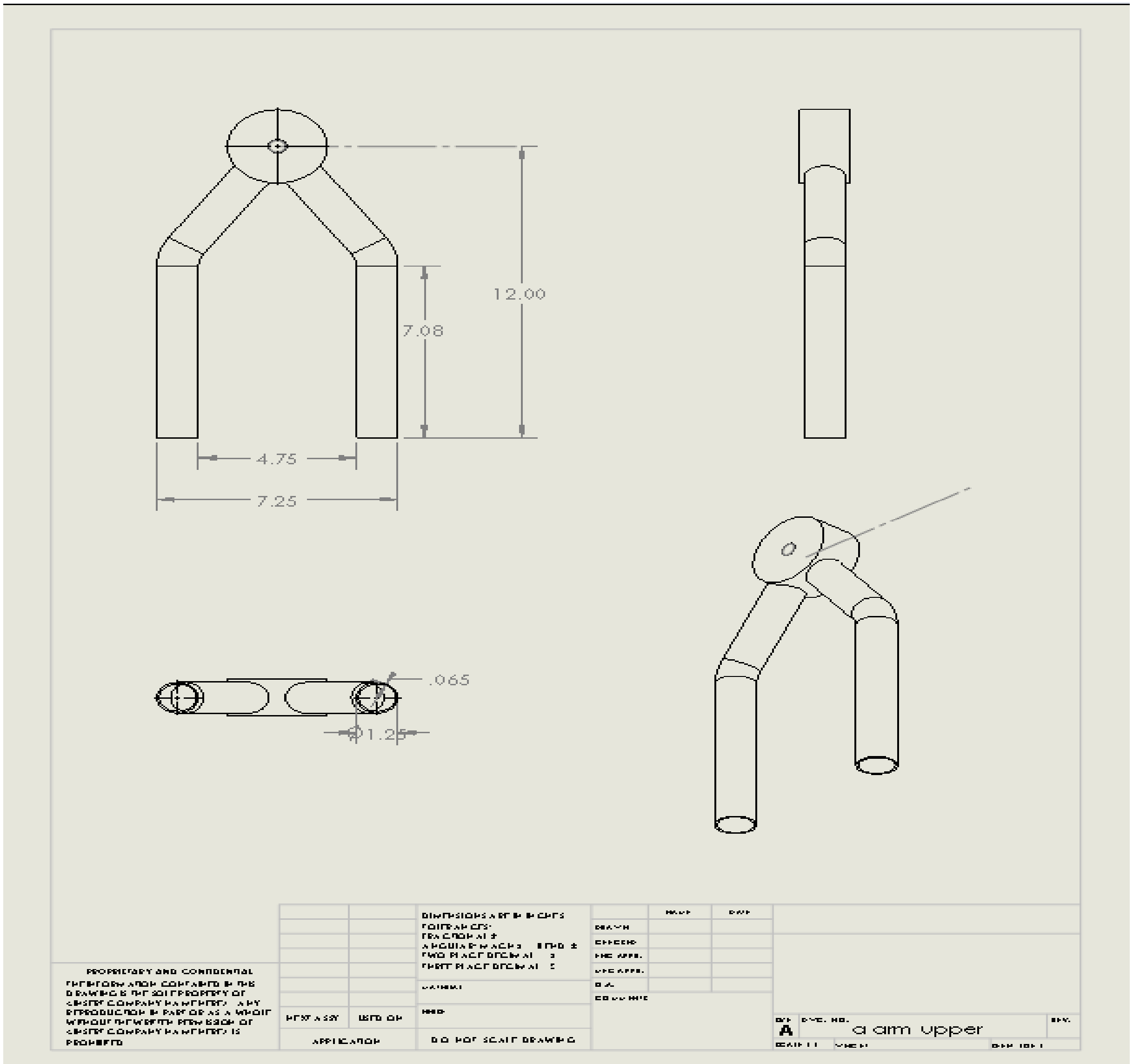
Appendix Figure A1 - Front suspension uniball design



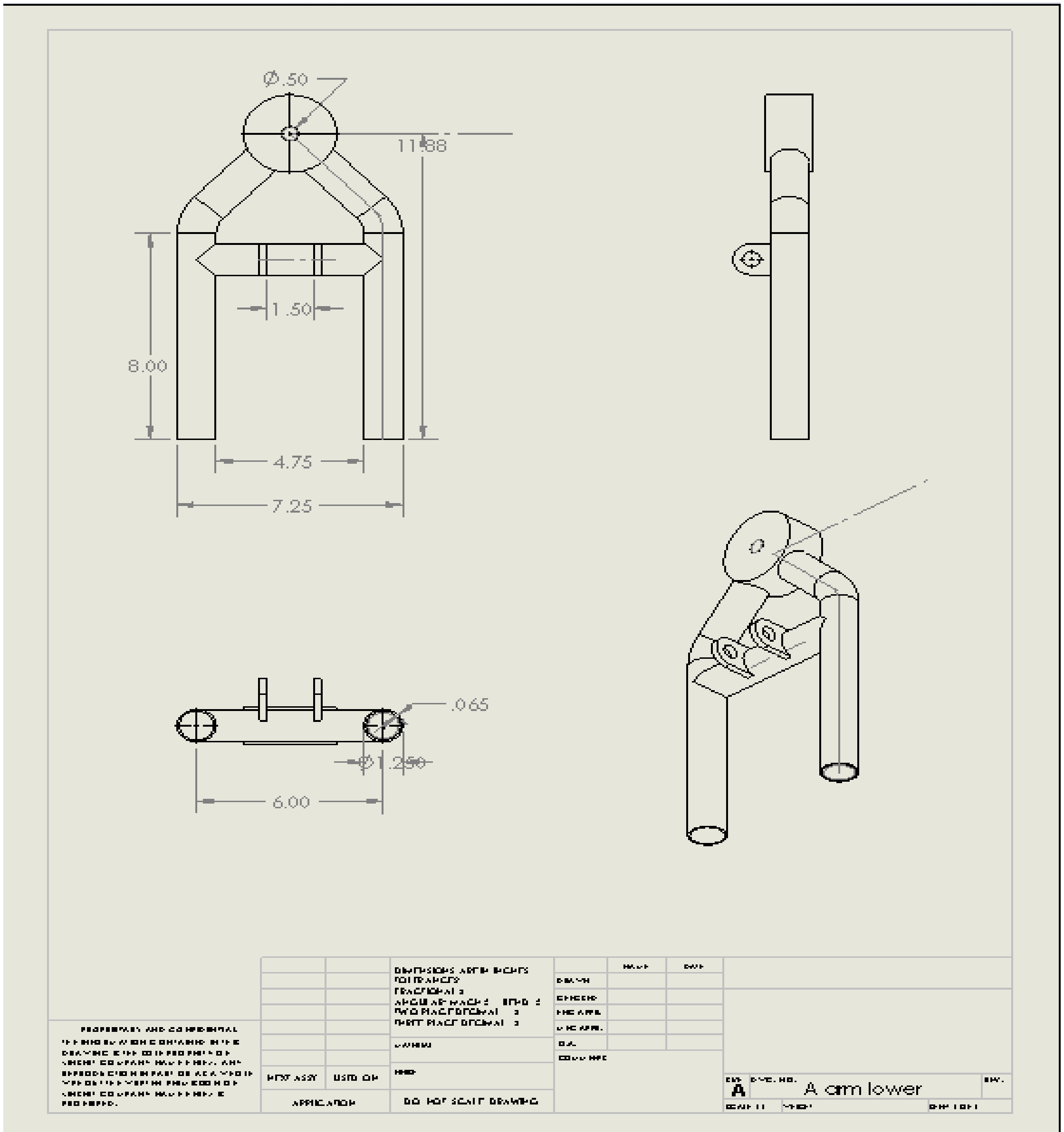
Appendix Figure A2 - Front suspension frame mounting bracket



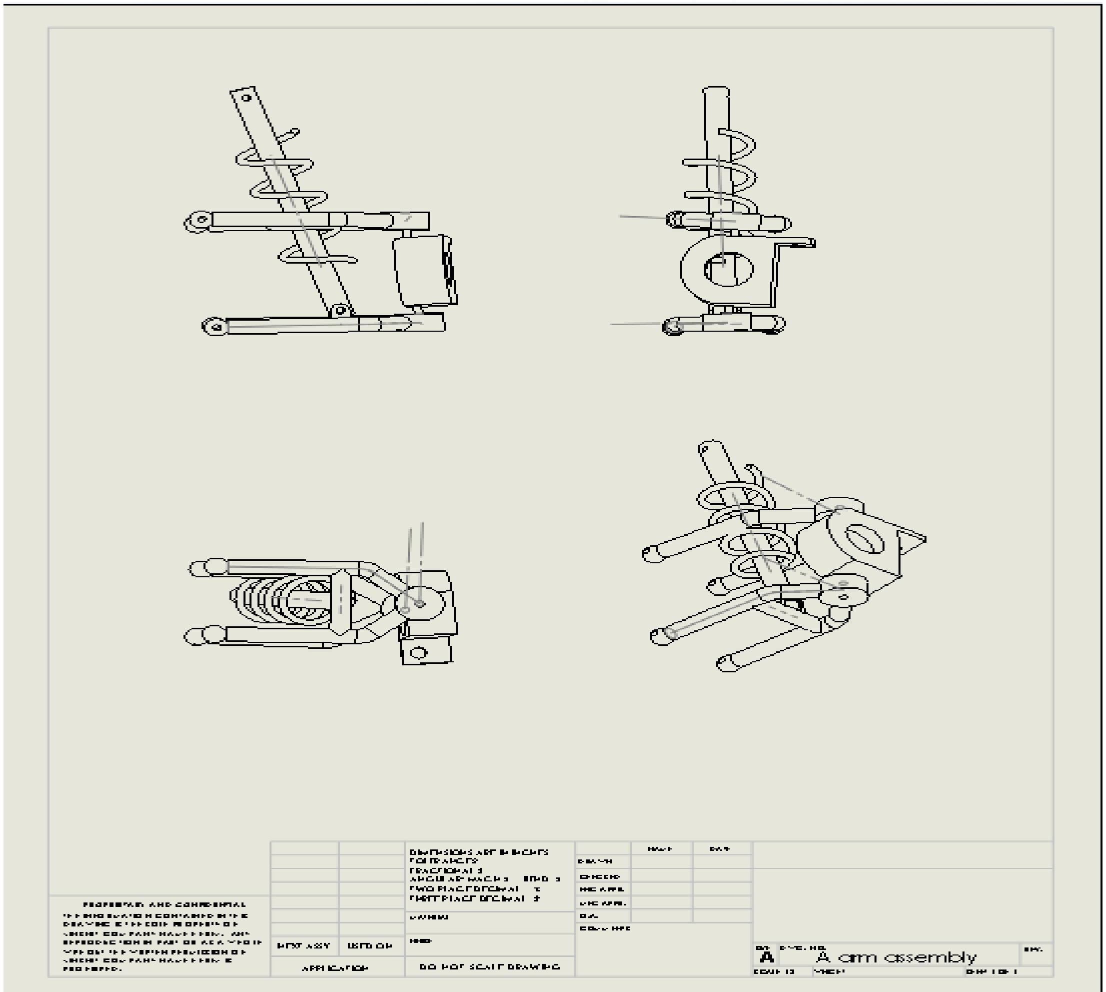
Appendix Figure A3 - Front suspension Mini Baja knuckle



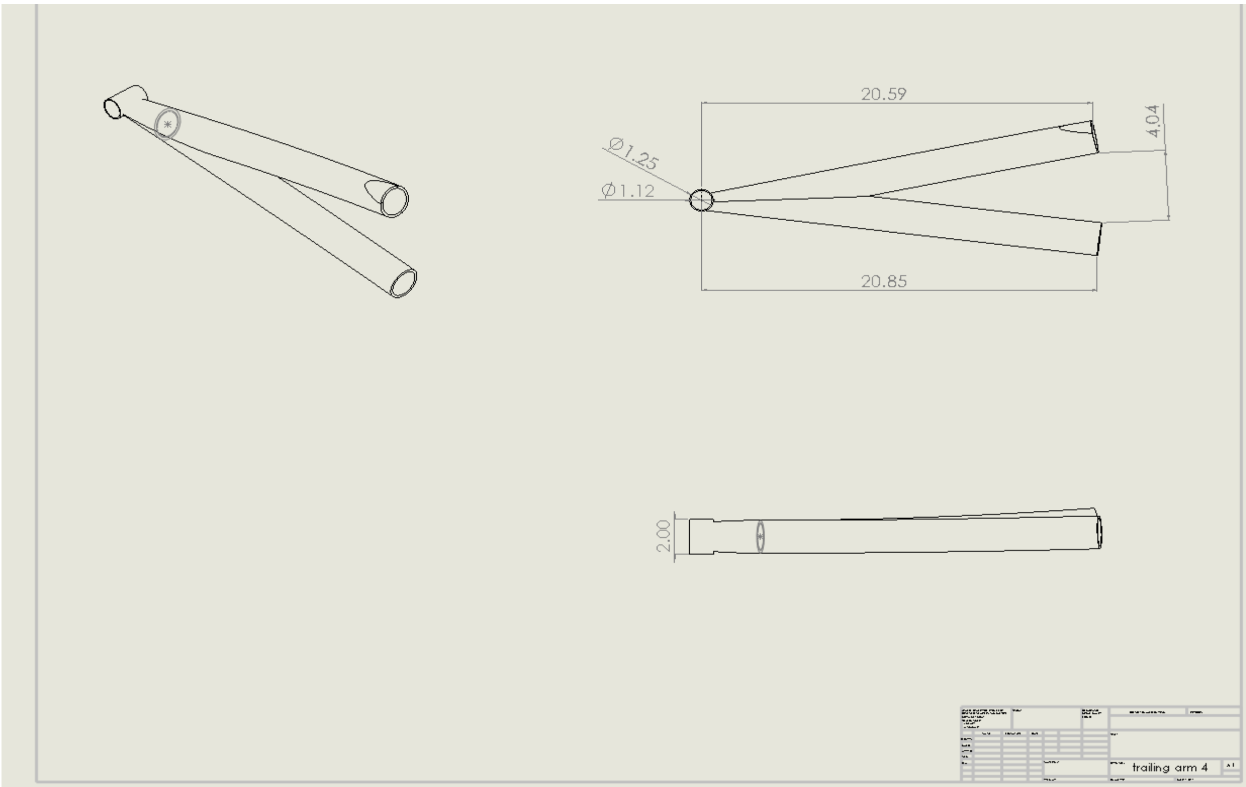
Appendix Figure A4 - Upper a-arm member



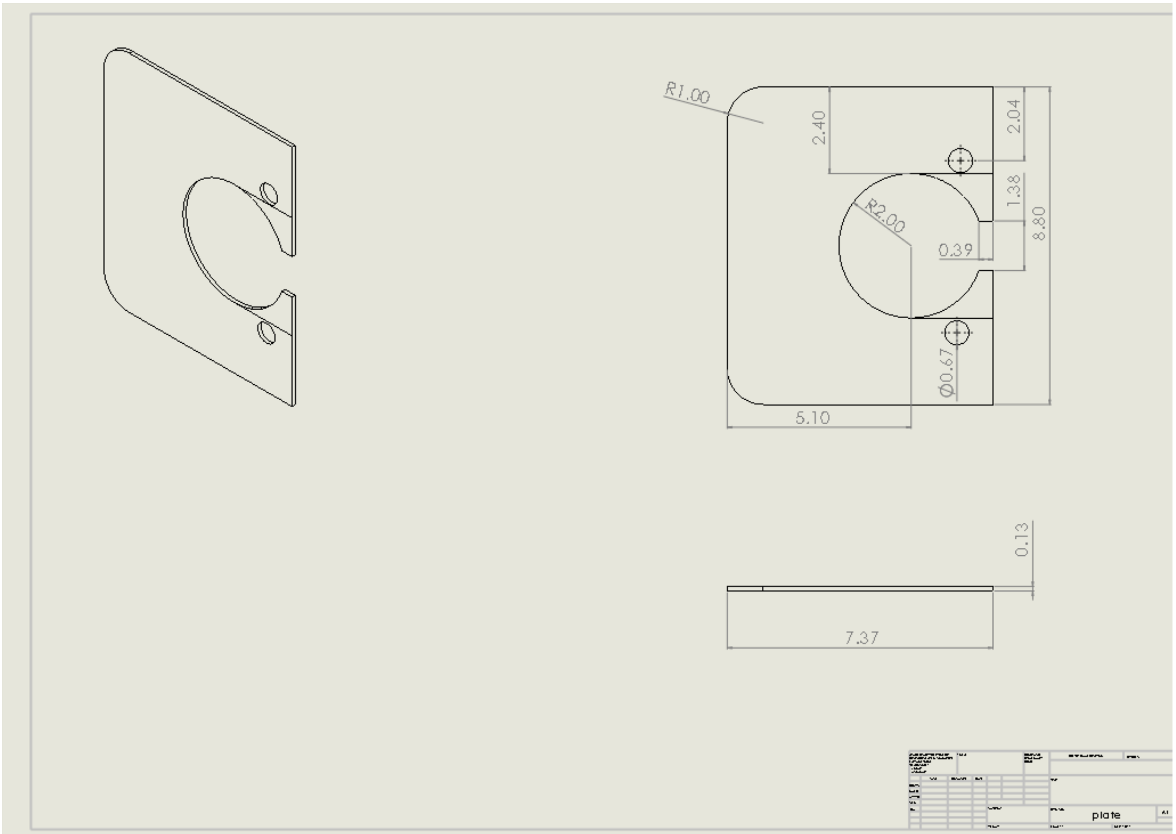
Appendix Figure A5 - Lower a-arm member



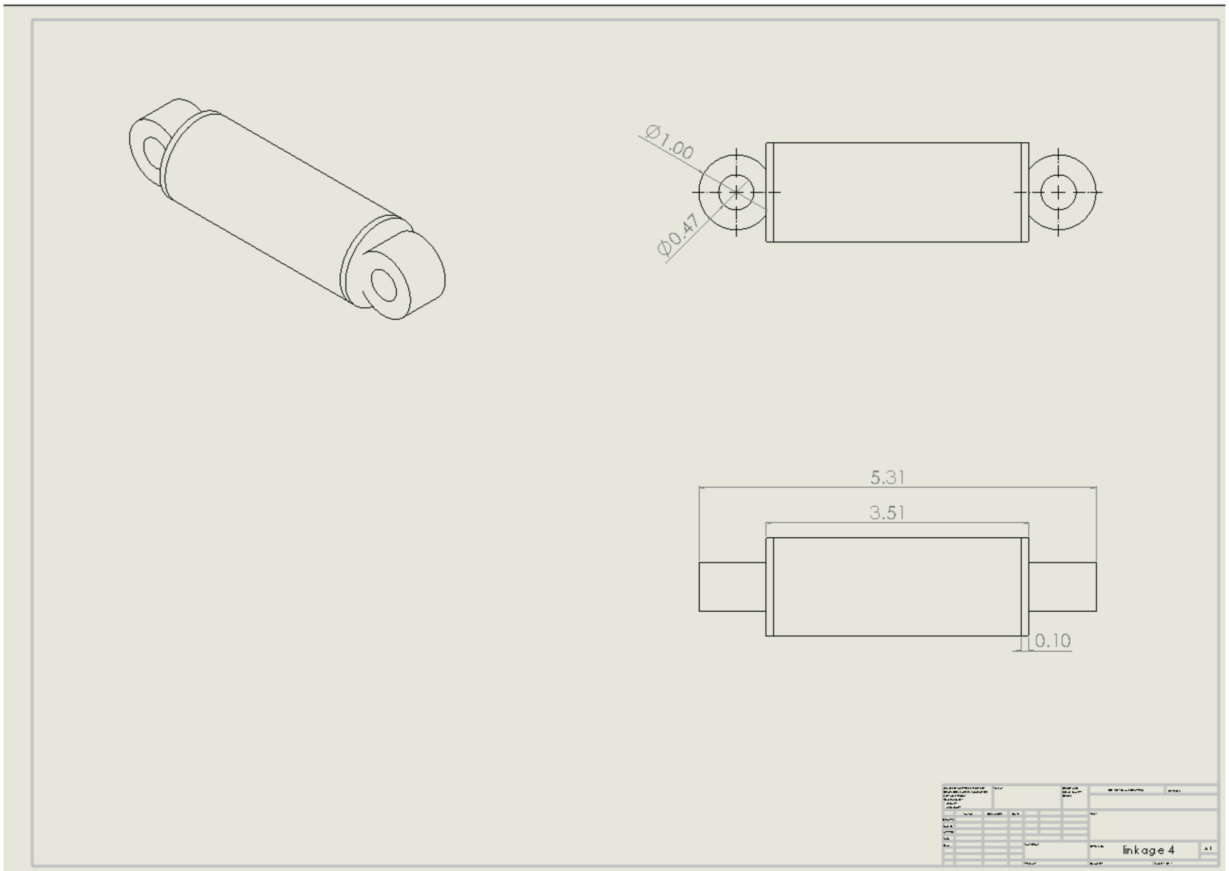
Appendix Figure A6 - Front a-arm assembly



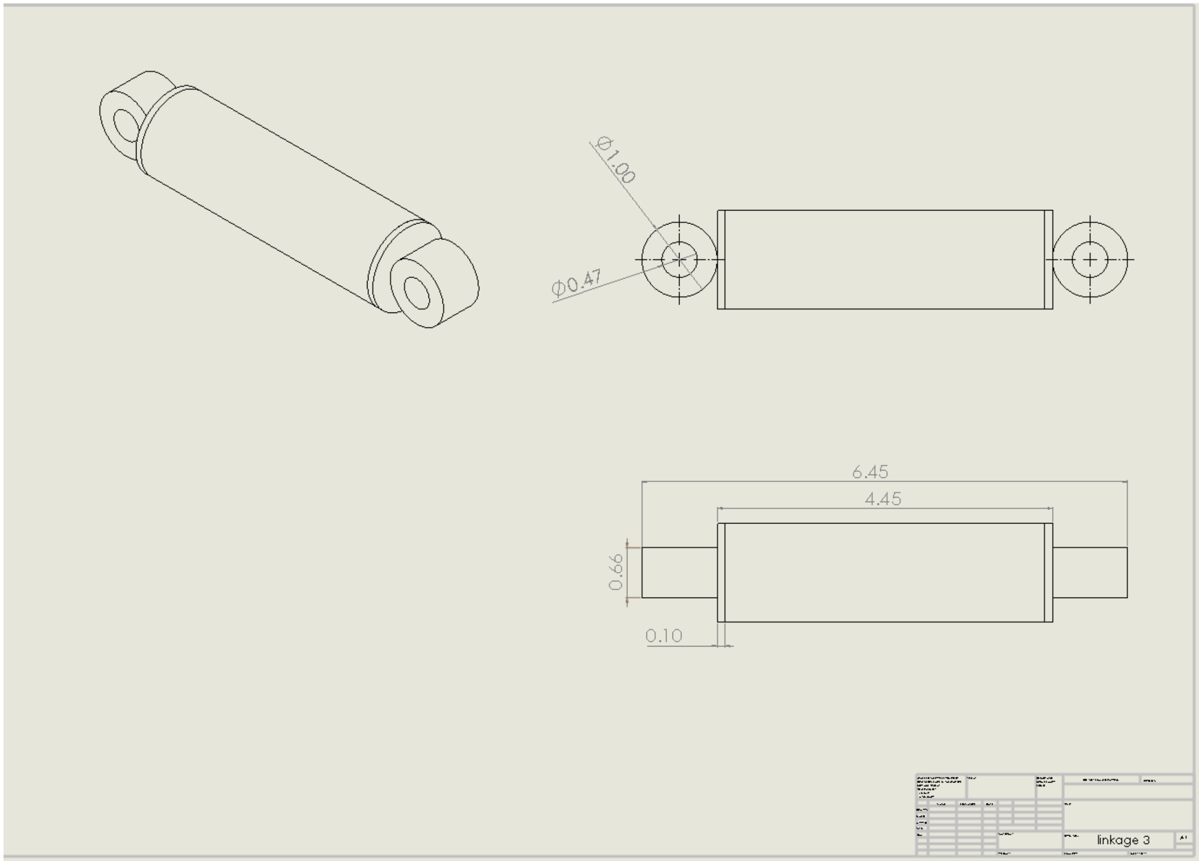
Appendix Figure A7 - Trailing arm members



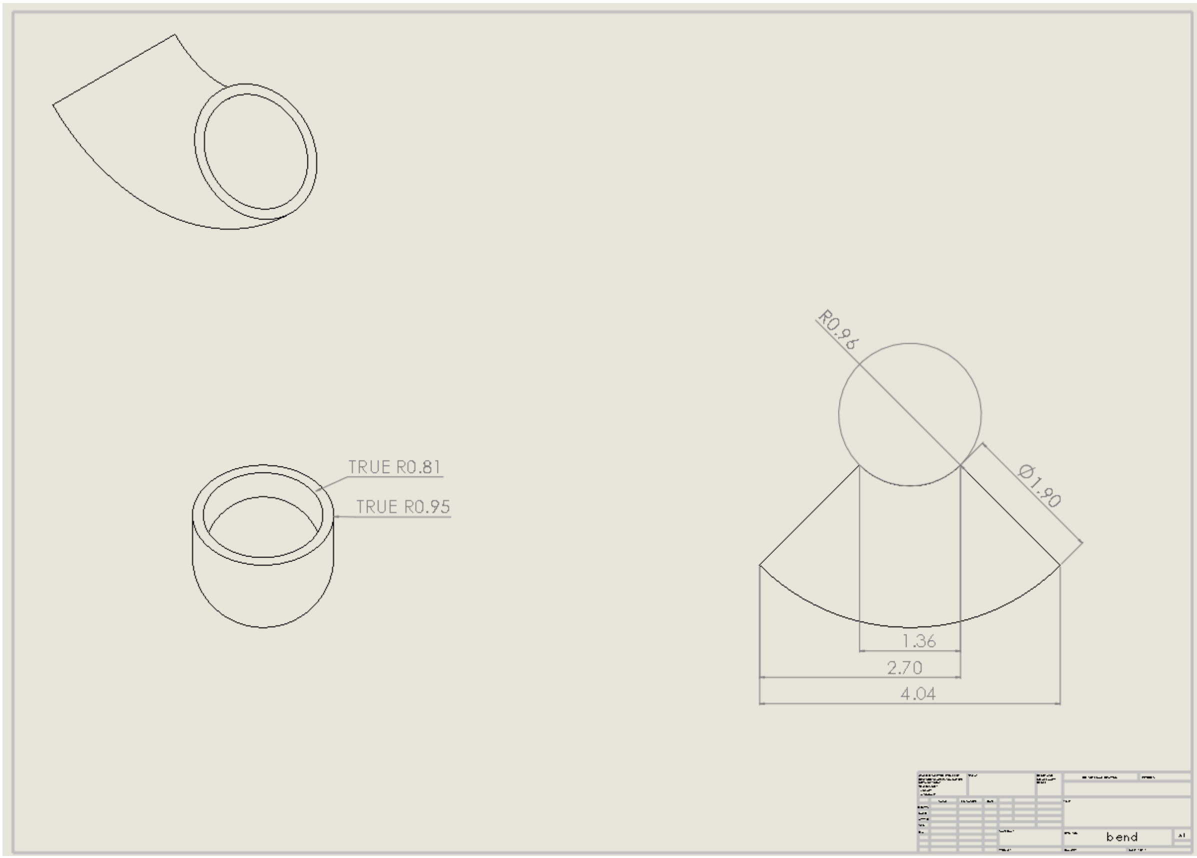
Appendix Figure A8 - Rear upright, hub plate



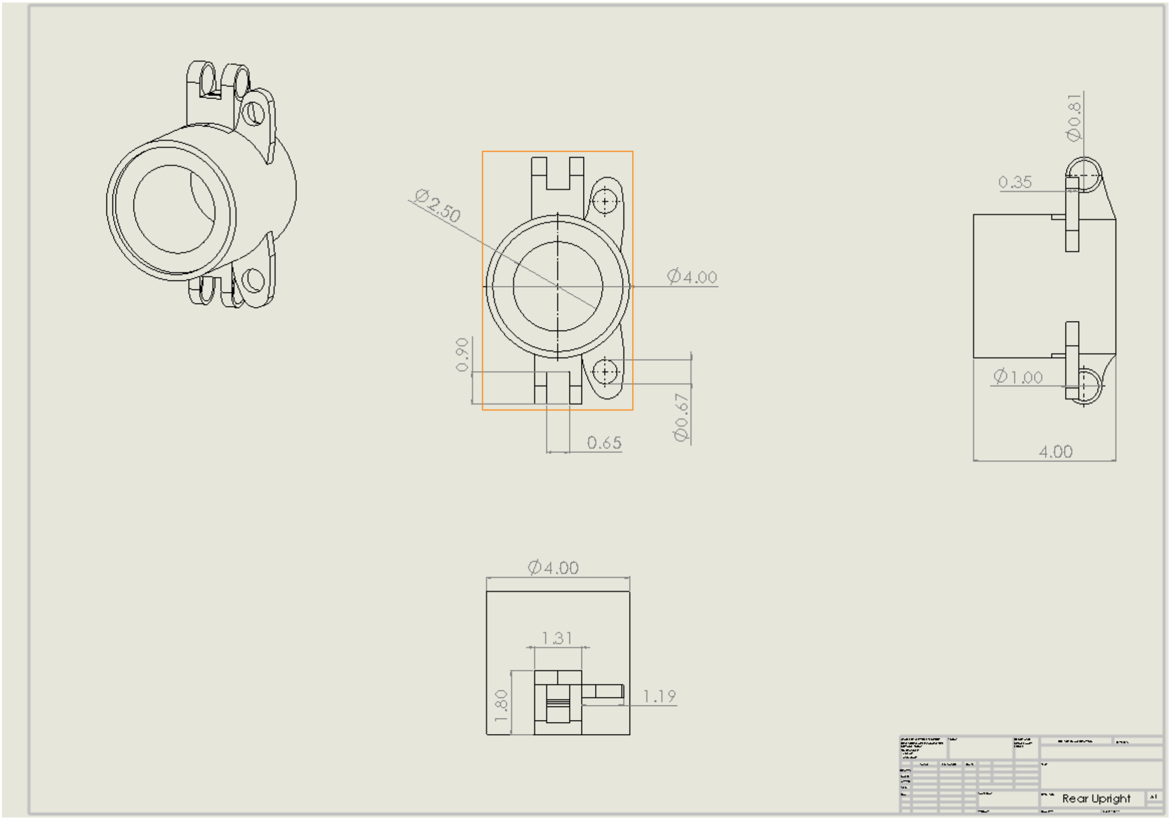
Appendix Figure A9 - Linkage bottom



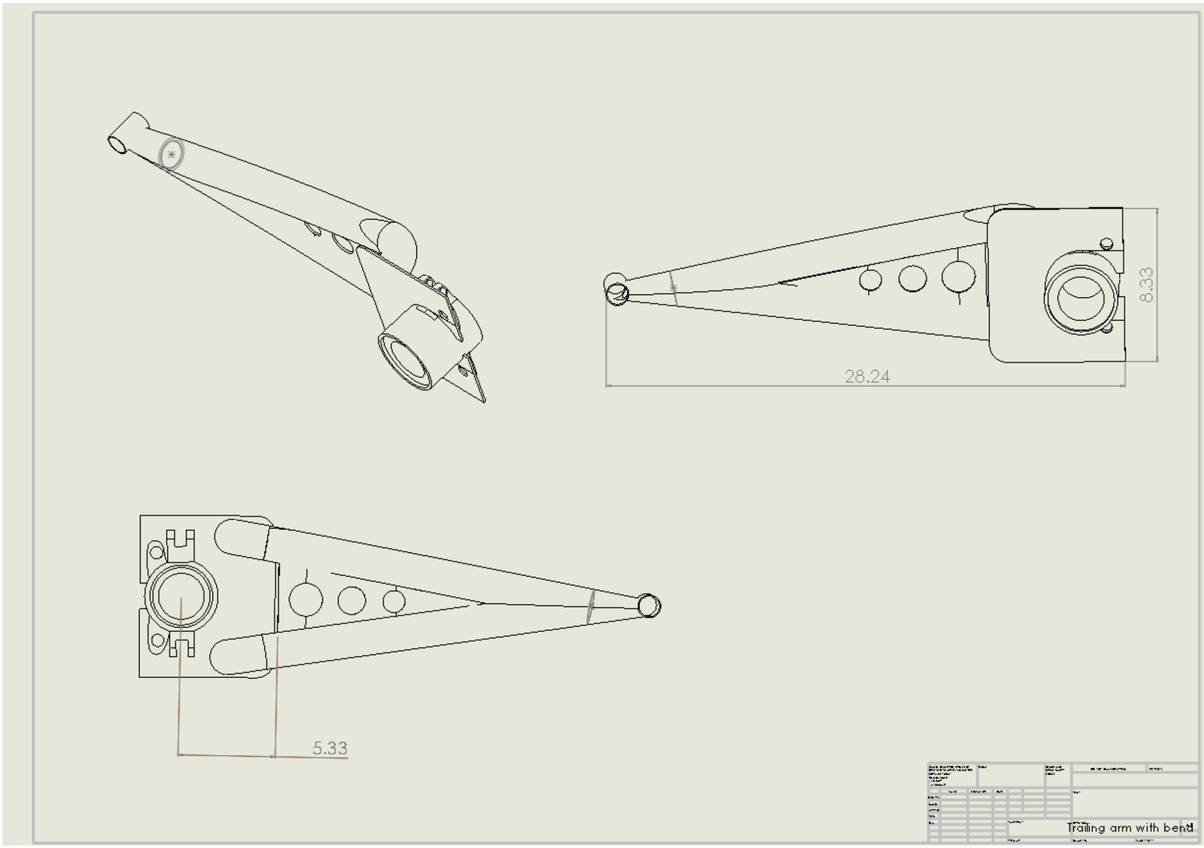
Appendix Figure A10 - Linkage top



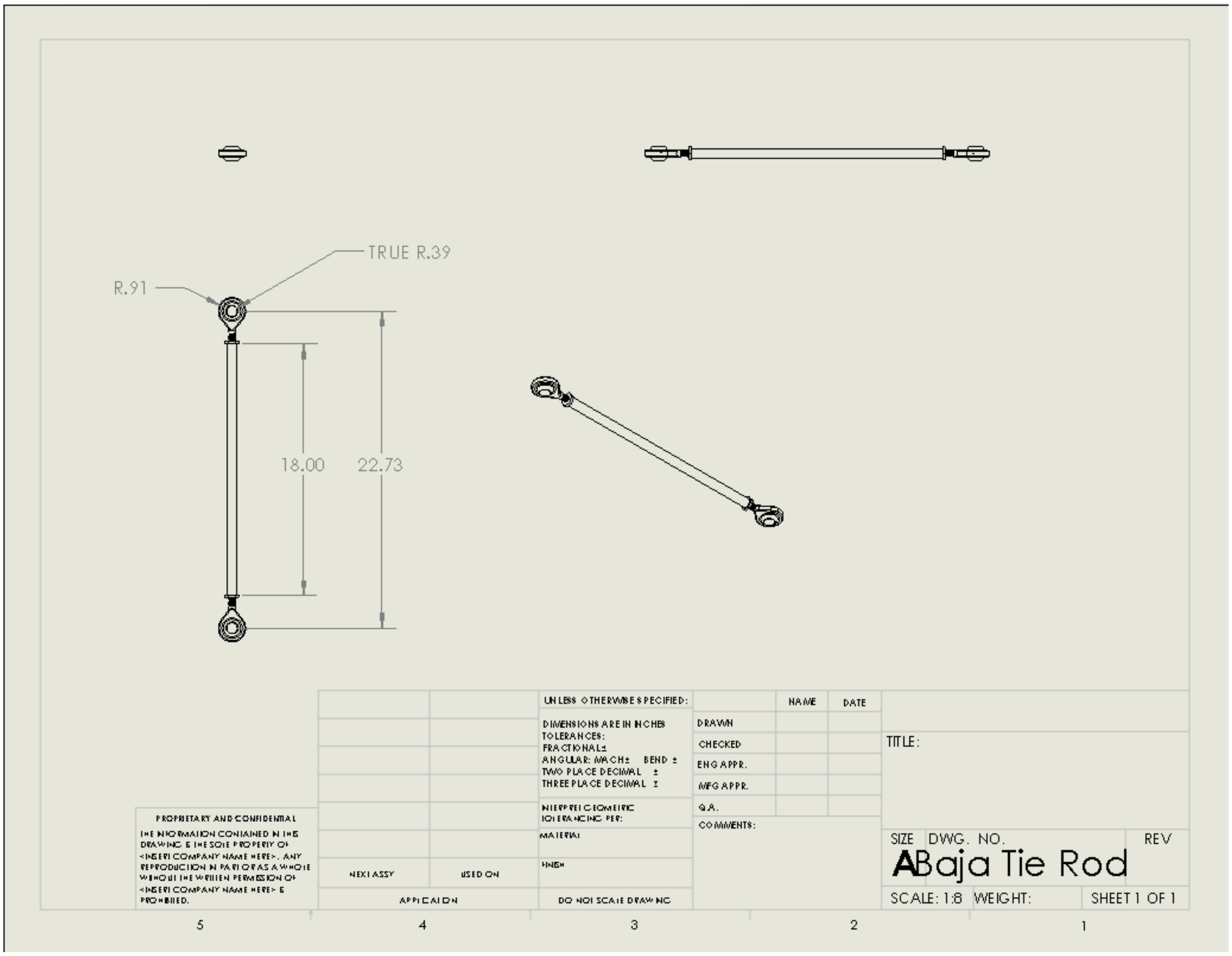
Appendix Figure A11 - Tube bend to upright plate



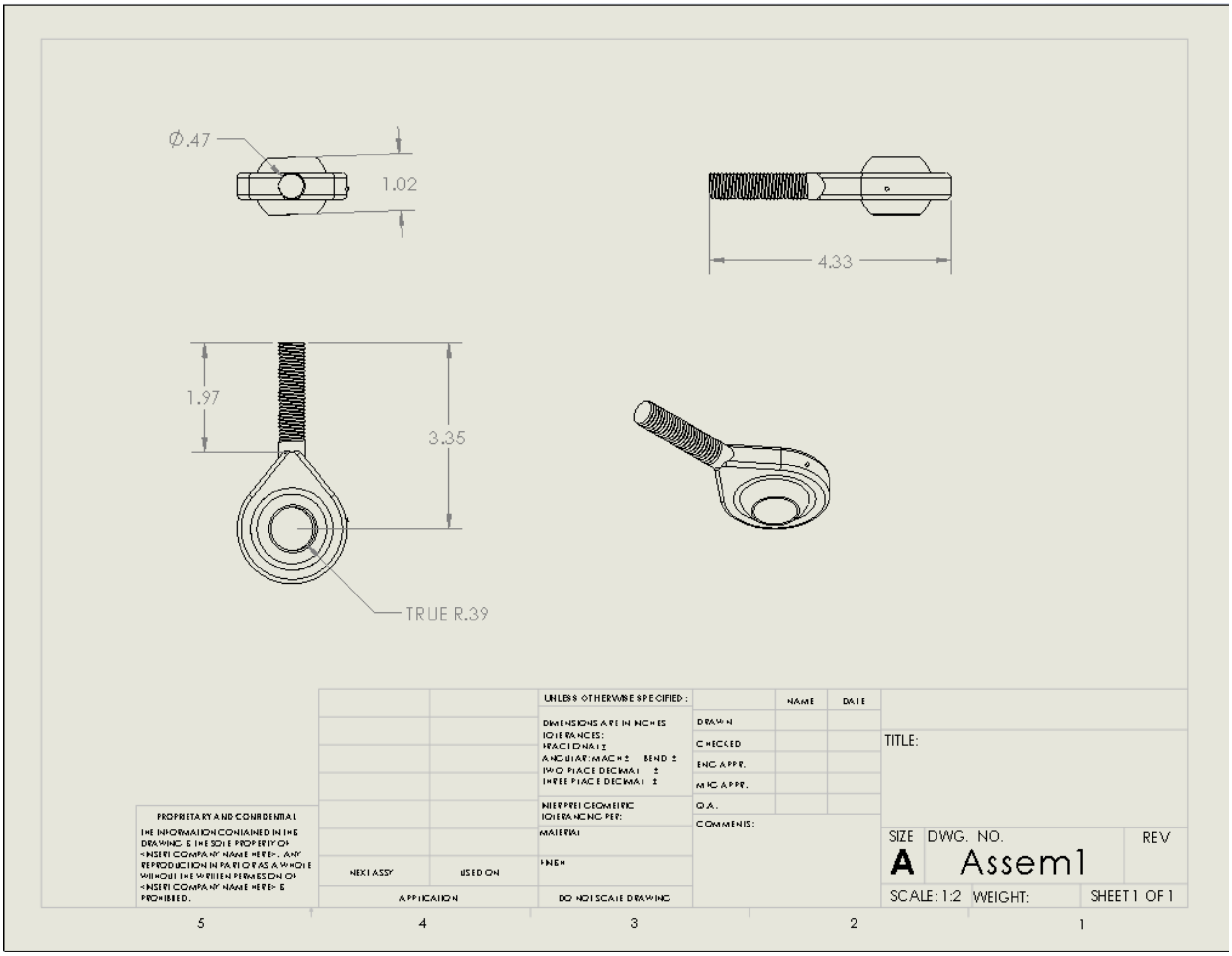
Appendix Figure A12 - Rear upright



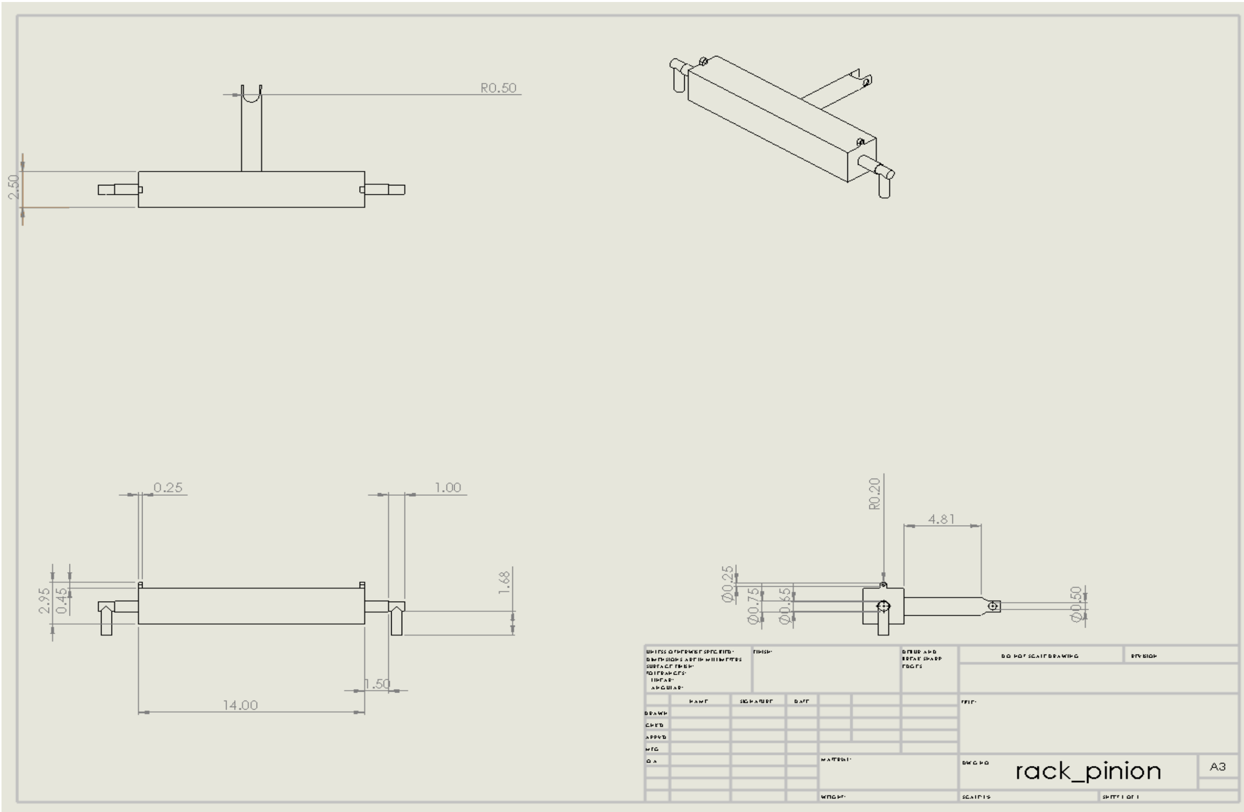
Appendix Figure A13 - Trailing arm assembly



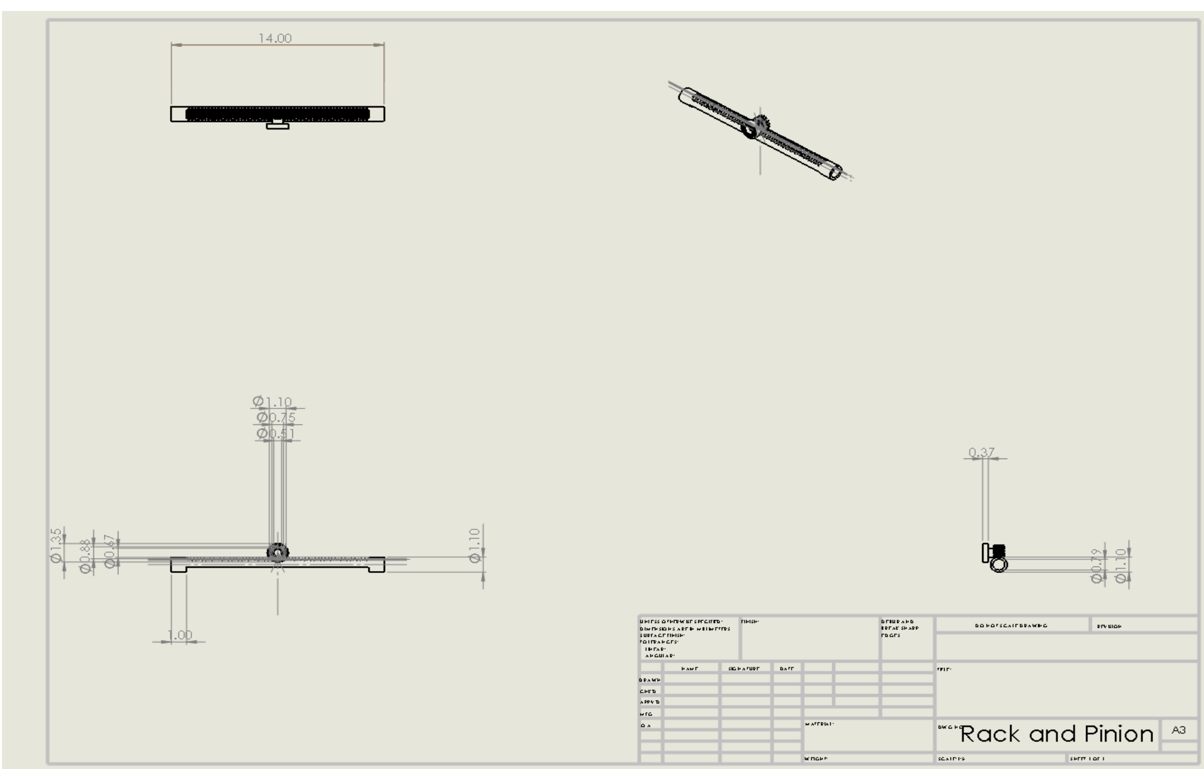
Appendix Figure A14 – Tie Rod Assembly



Appendix Figure A15 – Heim Joint Assembly



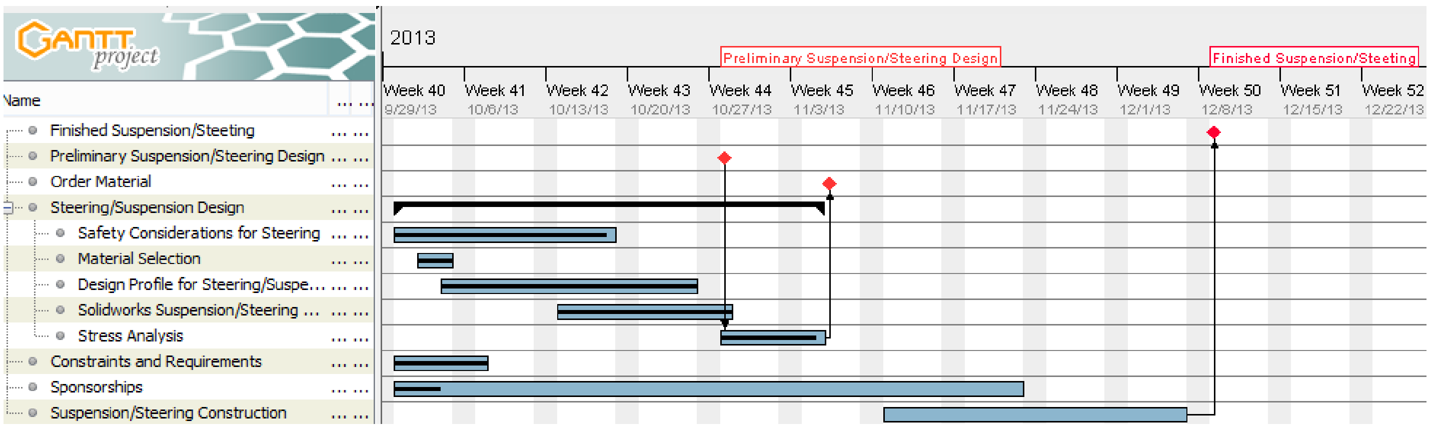
Appendix Figure A16 – Rack and Pinion Enclosed



Appendix Figure A17 – Rack and Pinion

Appendix B: Project Planning

Include the final developments in the Gantt chart and include a new Gantt chart for Spring 2014 semester



Appendix Figure B1 - Fall 2013 semester Gantt Chart

Appendix Figure B1 - Spring 2014 semester Gantt Chart

Table Appendix B2 - Front Suspension and Brakes Cost

Part Number	Description	Qty.	Price Each	Total	Retail Price	Retail Total
7175576	Decal-Hood Top	2	$0.00	$0.00	$0.00	$0.00
7078465	Decal-Polaris Star	2	$0.00	$0.00	$0.00	$0.00
7547237	NUT, FLANGED [7AD]	8	$1.19	$9.52	$2.38	$19.04
7547309	NUT, WHEEL [EAK]	8	$1.16	$9.28	$2.32	$18.56
1525017	VALVE, RIM	2	$1.35	$2.70	$2.70	$5.40
7547337	NUT, CASTLE	2	$0.69	$1.37	$1.37	$2.74
7661404	PIN, COTTER	2	$0.69	$1.37	$1.37	$2.74
7555796	WASHER, CONE	4	$2.11	$8.42	$4.21	$16.84

5137219	HUB, WHEEL, FRONT	2	$22.00	$43.99	$43.99	$87.98
7518654	STUD	8	$1.10	$8.76	$2.19	$17.52
5250068	DISC, BRAKE, FRONT	2	$25.50	$50.99	$50.99	$101.98
7710440	RING, RETAINING	2	$4.08	$8.16	$8.16	$16.32
3514699	BEARING, BALL, SEALED	2	$15.00	$29.99	$29.99	$59.98
5135443	CARRIER, BEARING, RH	1	$55.00	$55.00	$109.99	$109.99
5135442	CARRIER, BEARING, LH	1	$55.00	$55.00	$109.99	$109.99
1911529	ASM., CALIPER, FRONT, LH	1	$107.50	$107.50	$214.99	$214.99
1911530	ASM., CALIPER, FRONT, RH	1	$107.50	$107.50	$214.99	$214.99
7518760	BOLT, FLANGED	4	$2.19	$8.76	$4.38	$17.52
7518558	BOLT, FLANGED	4	$0.82	$3.28	$1.64	$6.56
7661140	PIN, CLIP	1	$0.50	$0.50	$1.00	$1.00
7661843	PIN, CLEVIS	1	$1.02	$1.02	$2.04	$2.04
2204458	KIT, SERVICE, MASTER CYLINDER, TANDEM	1	$110.00	$110.00	$219.99	$219.99
7547332	NUT, FLANGE, NYLOK	2	$0.50	$1.00	$1.00	$2.00
7043422	Fox Podium x rzr 800 front	2	$270.00	$540.00	$449.99	$899.98
7043574-589	main spring 160#/in	2	$19.00	$38.00	$45.99	$91.98
7043227-589	tender spring 60#/in	2	$16.00	$32.00	$39.99	$79.98
5436643	spring spacer	2	$1.00	$2.00	$1.98	$3.96
5630580	spring retainer	2	$2.00	$4.00	$2.54	$5.08
59915K276	Mcmaster carr Heim 1/2-20	4	$30.06	$120.24	$30.06	$120.24
	Synergy 1" uniball cup	4	$20.00	$80.00	$20.00	$80.00
		Total		$1,440.33		$2,529.39

Table Appendix B3 - Rear Suspension

Part Number	Description	Qty.	Price Each	Total	Retail Price	Retail Total
7547237	NUT, FLANGED [7AD]	8	$1.19	$9.52	$2.38	$19.04
7547309	NUT, WHEEL [EAK]	8	$1.16	$9.28	$2.32	$18.56
1525017	VALVE, RIM	2	$1.35	$2.70	$2.70	$5.40
7547337	NUT, CASTLE	2	$0.69	$1.37	$1.37	$2.74
7661404	PIN, COTTER	2	$0.69	$1.37	$1.37	$2.74
7555796	WASHER, CONE	4	$2.11	$8.42	$4.21	$16.84
7518654	STUD	8	$1.10	$8.76	$2.19	$17.52
7710440	RING, RETAINING	2	$4.08	$8.16	$8.16	$16.32
5137278	HUB, WHEEL, REAR	2	$32.50	$64.99	$64.99	$129.98
7518378	STUD	8	$0.52	$4.12	$1.03	$8.24
7518978	SCREW	8	$0.60	$4.76	$1.19	$9.52
3514699	BEARING, BALL, SEALED	2	$52.00	$103.99	$103.99	$207.98
5137863	CARRIER, BEARING, WHEEL, RH	1	$52.00	$52.00	$103.99	$103.99
5137862	CARRIER, BEARING, WHEEL, LH	1	$52.00	$52.00	$103.99	$103.99
59915K276	Mcmaster carr Heim 1/2-20	4	$30.06	$120.24	$30.06	$120.24
7043419	Fox Podium x rzr 800 rear	2	$270.00	$540.00	$449.99	$899.98
7043573-293	main spring 210#/in	2	$36.00	$72.00	$89.99	$179.98
5630580	spring retainer	2	$2.00	$4.00	$2.54	$5.08

		Total		$1,067.67		$1,868.14

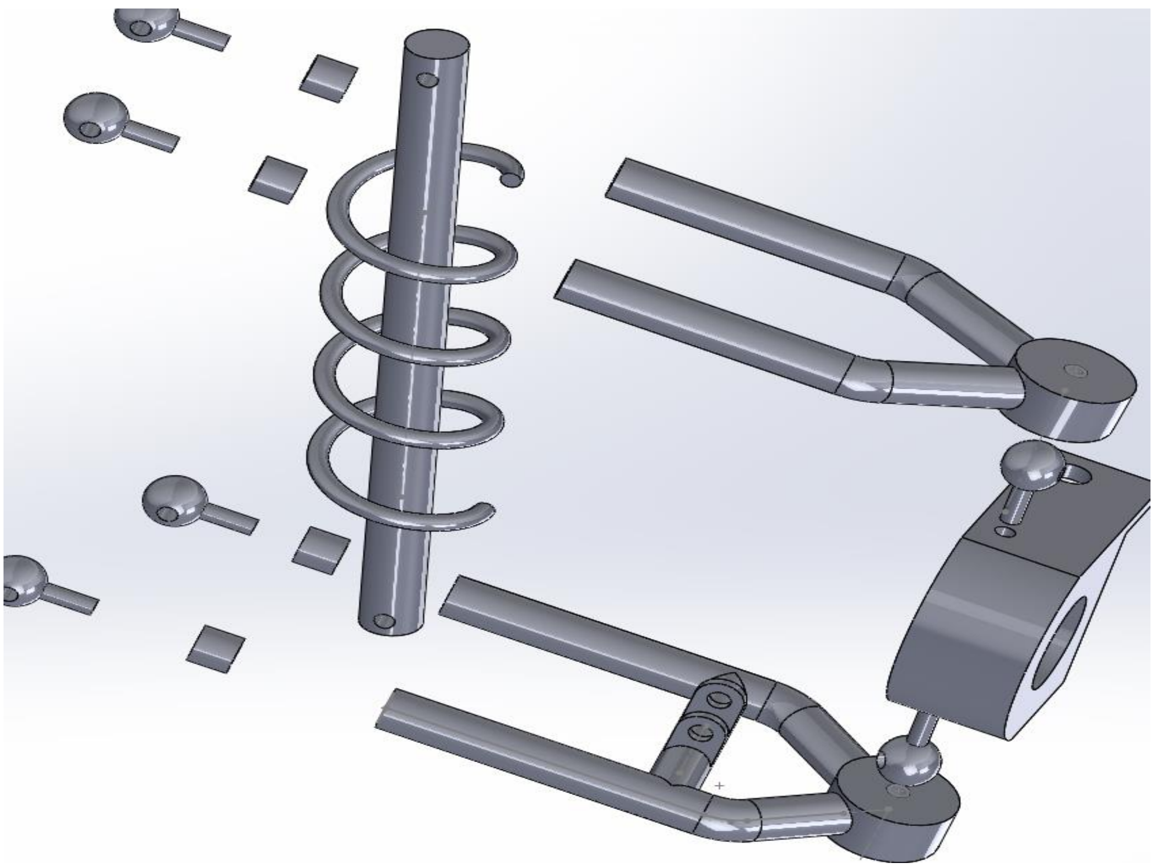
Table Appendix C3 - Rear Suspension

Part Number	Description	Qty.	Price Each	Total	Retail Price	Retail Total
5411920	BOOT, SEAL, ROD END	2	3.995	7.99	7.99	15.98
7061054	ROD END	2	19.495	38.99	38.99	77.98
7547028	NUT, JAM	2	0.865	1.73	1.73	3.46
-------	TIE ROD, INNER	2	0	0		0
7080978	CLAMP, BOOT, SMALL	2	1.025	2.05	2.05	4.1
-------	BOOT, PASSENGER, SIDE RACK	1	0	0		0
7080979	CLAMP, BOOT, LARGE	2	2.57	5.14	5.14	10.28
7515382	BOLT	3	0.865	2.595	1.73	5.19
7517827	SCREW	1	1.485	1.485	2.97	2.97
7547177	NUT	2	1.19	2.38	2.38	4.76
7512371	BOLT	2	0.685	1.37	1.37	2.74
1823465	ASM., GEAR BOX, STEERING [INCL. 2-7,23]	1	178.995	178.995	357.99	357.99
7556063	WASHER, WAVE	2	1.535	3.07	3.07	6.14
7661203	PIN, COTTER	2	0.5	1	1	2
1542766	SHAFT, STEERING, UPPER/LOWER	1	66.115	66.115	132.23	132.23
7547385	NUT	1	0.5	0.5	1	1
7542324	NUT	5	0.5	2.5	1	5

-------	BOOT, DRIVER SIDE, RACK	1	0	0		0
7517909	BOLT	2	2.565	5.13	5.13	10.26
7556099	WASHER	4	0.515	2.06	1.03	4.12
7558402	WASHER, FLAT	3	0.5	1.5	1	3
[59915K276](59915K276)	Mcmaster carr Heim 1/2-20	2	$30.06	$60.12	$30.06	$60.12
			Total	324.6		649.2

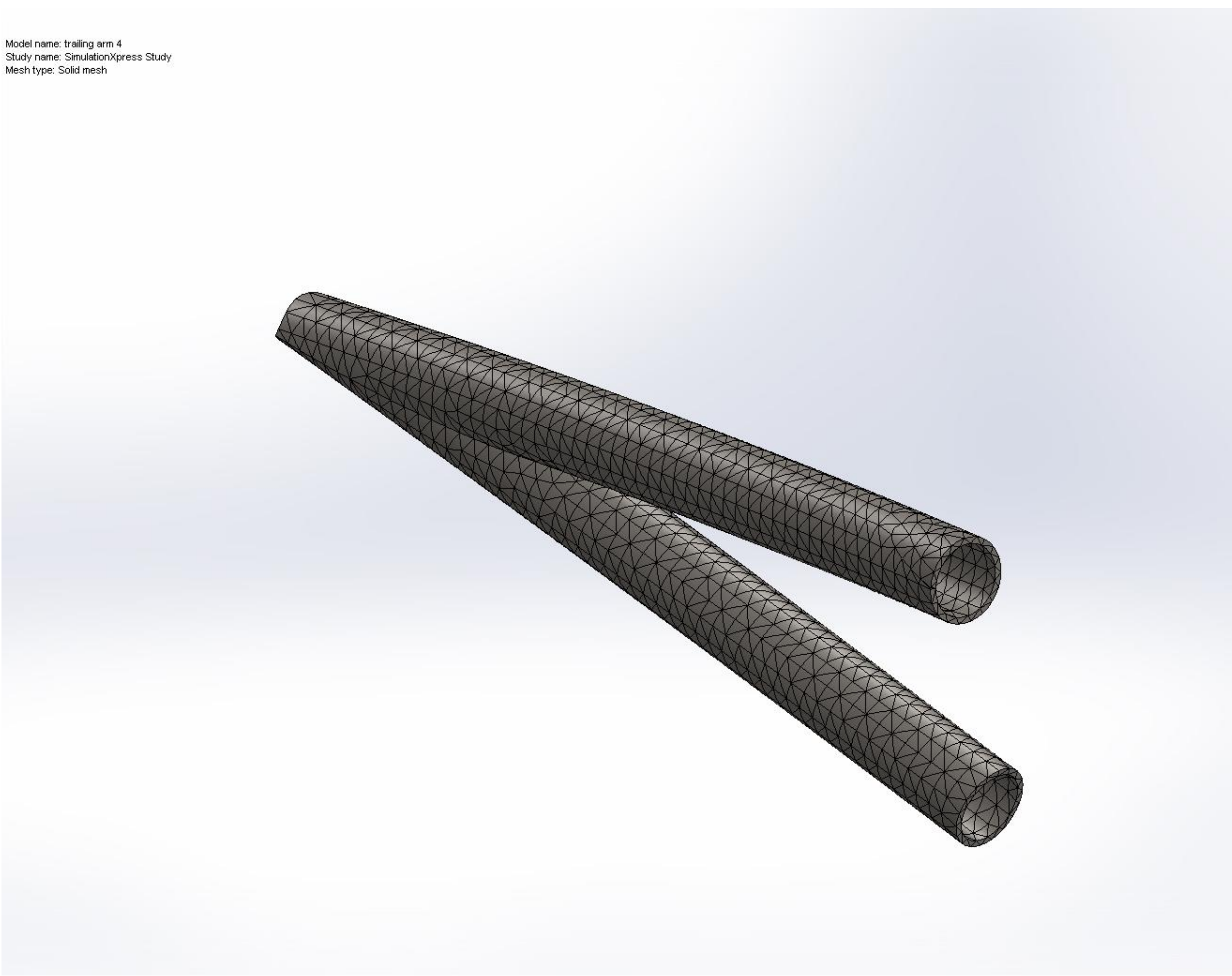
Appendix C:

Include any additional material that is not directly part of the report here



Appendix Figure C1 – Double a-arm assembly exploded view

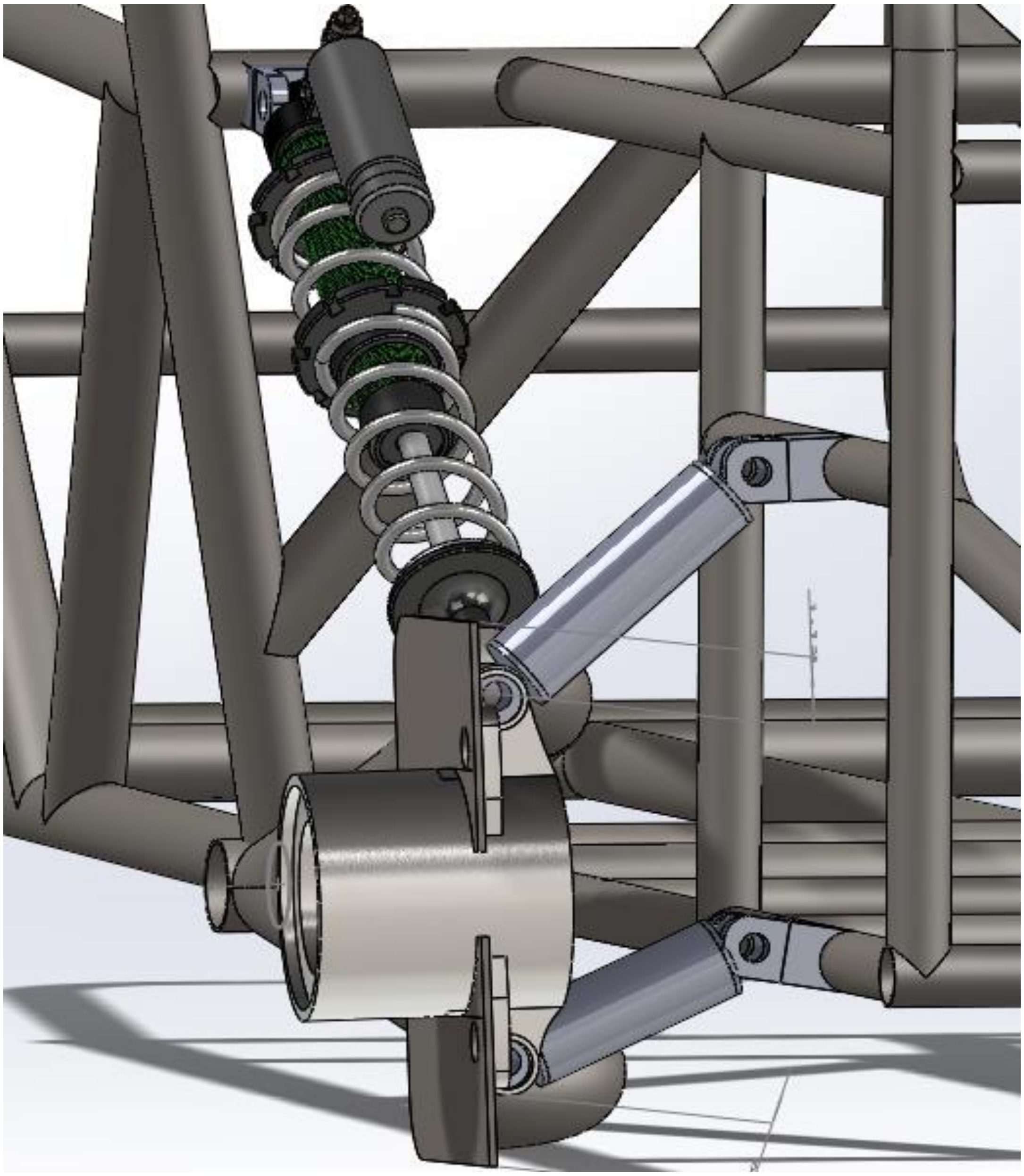
Appendix Figure C2 – Polaris RZR XP 900 rear trailing arm



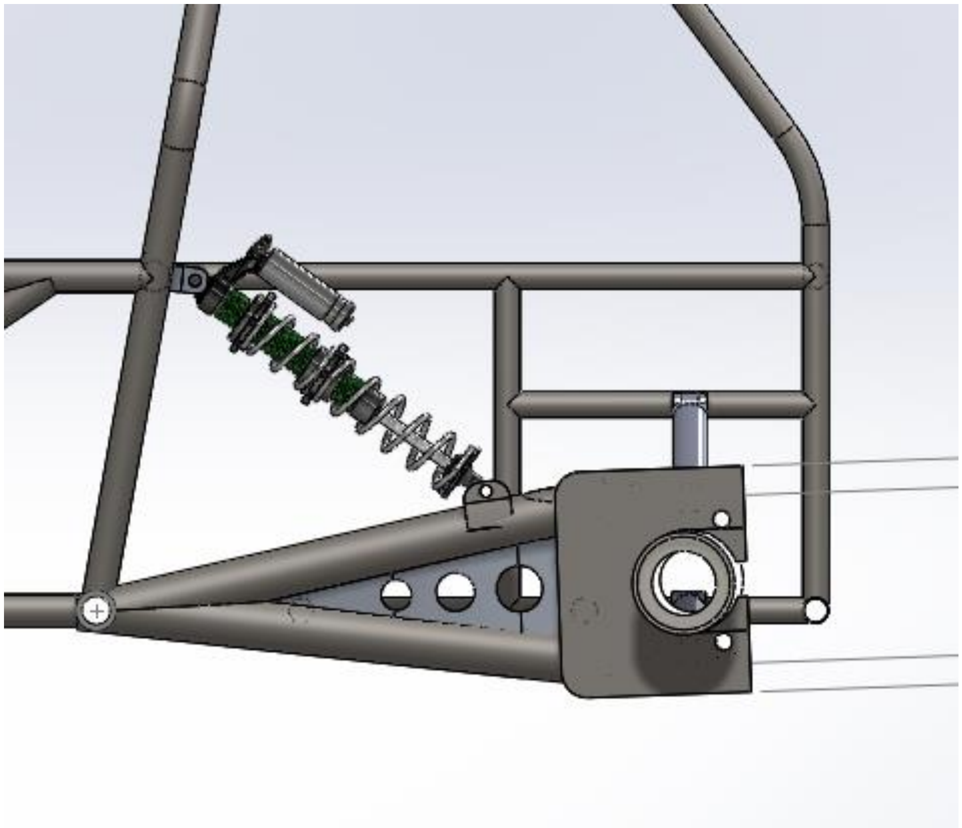
Name	Type	Min	Max
Stress	VON: von Mises Stress	0.00593253 psi	26752 psi

Name	Type	Min	Max
		Node: 10011	Node: 19
Displacement	URES: Resultant Displacement	0 in Node: 1	0.0457006 in Node: 2349

Appendix Figure C3 – Trailing arm FEA meshing



Appendix Figure C4 – Rear suspension frame assembly (linkage focus)



Appendix Figure C5 – Rear suspension frame assembly (front)