2017 Bearcats Baja SAE – Front Suspension

A Baccalaureate thesis submitted to the
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by

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2017 Bearcats Baja SAE – Front Suspension

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ABSTRACT

Baja SAE is a collegiate design competition in which seniors build and race a mini baja car. The cars are powered by a ten horsepower Briggs and Stratton engine that is standard for every team. SAE Baja also fulfills many students' senior capstone project. Baja SAE takes students from the design phase all the way through manufacturing and assembly. My task for the 2017 team was to design a new front suspension that will fit the previous year's frame.

INTRODUCTION

The front suspension of the car is designed to absorb the initial impact from obstacles and make the ride as smooth as possible for the operator. The front suspension maintains the contact between the tires and the Earth while going over these obstacles and terrain. There are a variety of obstacles we will face such as boulders, fallen trees, deep mud, and tight turns. The suspension must have enough travel to clear these obstacles successfully and be durable enough to withstand the subsequent impacts. This report will show the reasoning behind my design and my calculations to reinforce it.

Problem Statement: "Design and build a light weight and durable front suspension for the 2017 car that will allow the car to complete all the dynamic events and endurance race with minimal problems."

RESEARCH

Baja SAE rules directly related to front suspension (1)

B1.1.2 Maximum Vehicle Dimensions
Width: 162 cm (64 in) at the widest point with the wheels pointing forward at static ride height.
Length: Unrestricted, see note below.

NOTE: Teams should keep in mind that Baja SAE courses are designed for vehicles with the maximum dimensions of 162 cm (64 in) width by 274 cm (108 in) length.

B1.2 All-Terrain Capability
B1.2.1 The vehicle must be capable of safe operation over rough land terrain including obstructions such as rocks, sand jumps, logs, steep inclines, mud and shallow water in any or all combinations and in any type of weather including rain, snow and ice.

B1.2.2 The vehicle must have adequate ground clearance and traction

CUSTOMER REQUIREMENTS

(For the front and rear suspension)

Track Width Center to Center: Front 52" (2014 51")
Track Width Center to Center: Rear 50" (2014 50")
Wheelbase: 70" (2014 70")

Ground Clearance – Front: 14" (2014 14")
Ground Clearance – Rear 12"
Ground Clearance – Belly pan 12"

Suspension Travel – Front: 11" (2014 10.7")
Suspension Travel – Rear: 11"

DESIGN RESEARCH

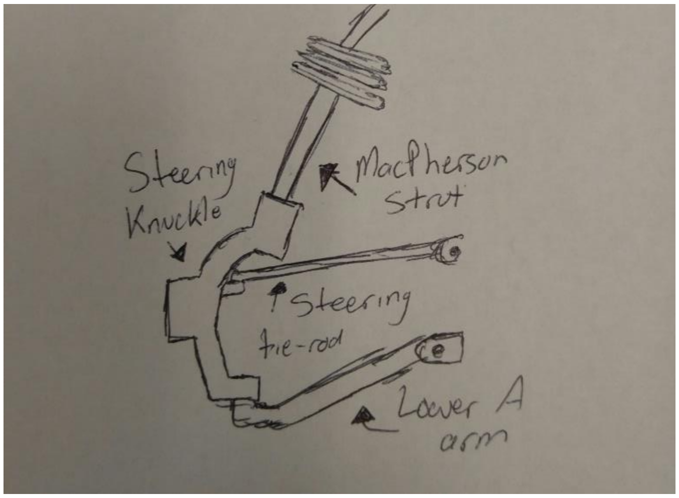

Figure 1 MacPherson Strut Suspension

MacPherson Suspension

Advantages:
1. Fewer components
2. Less unsprung weight
3. Ease of design

4. Low cost
5. Minimal vibration

Disadvantages:
1. Camber changes under load
2. Limited travel
3. Reduced handling

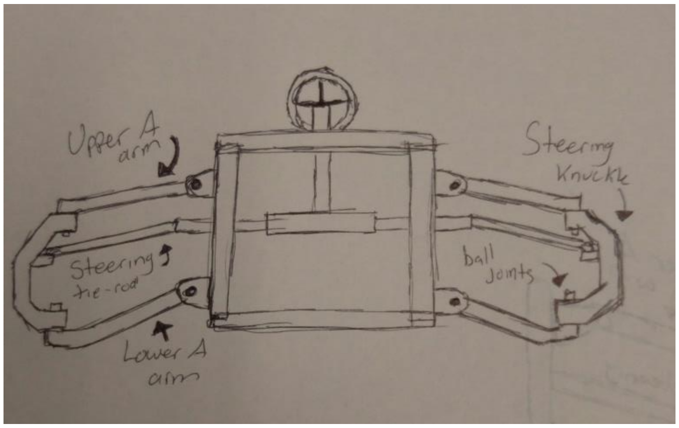

Figure 2 Double Wishbone Suspension

Double Wishbone Suspension

Advantages:
1. Large range for travel
2. Design freedom
3. Increased stability
4. Performs well under stress

Disadvantages:
1. Complex design due to many geometries
2. Higher cost

DESIGN

After going through the pros and cons of each design, it was decided to use a double wishbone suspension. It would provide the car with the most travel and be robust enough to handle the rough terrain and obstacles it would face. The frame was designed by a previous team who intended to use a double wishbone design front suspension. Redesigning for a MacPherson system would be very complicated. This contradicts one of the main advantages of a MacPherson system.

INITIAL DESIGN

After looking at previous teams' successful designs, designing for a few initial measurements proved to be advantageous. A few important characteristics of suspension are camber, caster, kingpin axis, and scrub radius.

Camber

Camber is the angle between the vertical axis of the vehicle and the vehicle axis of the tire as viewed from the front. Camber has a large effect on the slippage of the vehicle in turns. For off-road vehicles, negative camber is preferred. When making a turn, the body rolls in the opposite way of the turn. This puts extra force on the outside suspension components and causes them to compress farther. Negative camber then becomes zero camber, causing the tire to sit flush on the ground, thus enhancing grip. This system was designed with the static camber to be around -1 degrees, which is similar to the 2014 car. One can expect to see around 9 degrees change over the travel of the suspension. This is due to roll and will be addressed later.

Caster

Caster is the angle between the steering axis and vertical axis of the tire viewed from the side. Positive caster creates a moment about the tire that causes it to return to straight in the direction of movement. Caster can often be seen in the double digits, however this system is designed to be around +6 degrees. With a double A-arm design, the caster should not change over the travel of the suspension.

Kingpin Axis

Kingpin inclination, also known by kingpin axis (KPA) or steering axis, is the angle between the vertical axis and the imaginary axis that goes through the upper and lower ball joints. In modern cars and offroad vehicles, the KPA is fixed and cannot be changed. This is due to the knuckle being cast with fixed points for mounting the upper and lower ball joints. The knuckle from last year's team was chosen as it has new bearings and has not seen the track. The knuckle is from a 2005 Polaris Predator 500 and has a fixed KPA of 18 degrees.

Scrub Radius

Scrub radius is the distance from the kingpin axis to the center contact patch of the tire where the two points would theoretically meet the ground. Scrub radius can be best seen from the front of the car. Scrub causes a loss of speed when going over rough terrain and landing jumps. Scrub also causes understeer, which is turning less than expected. This should be minimized to reduce parasitic losses.

Shock Mount

Another important component to design for is where the shock will mount on the lower a-arm. It has been shown that the lower control arms experience the greatest amount of force of all the suspension components. By mounting the shock to the lower arm, you can also extend the lower arm farther out promoting a good kingpin axis and increasing front track width. In this design, the lower A-arm will receive strong material as well as the shock mount.

First Glance

Since this frame was from the previous year, the design process could begin much earlier than in the past. While the previous team's suspension was still mounted, the overall measurements needed for the new suspension design were attained. These measurements were beneficial because the travel and ride height was comparable to the previously stated goals. At first glance, the suspension arms looked weaker compared to the older cars. After further investigation it came to light that the front of the car was narrower than that of the previous cars. Thus, in order to achieve the desired track width, the arms needed to be longer. The material also ended up being thicker than previous cars, which allowed for the smaller diameter tubing that made the A-arms look thin. Finally, the upper shock mount location was neither visually appealing, nor strong. To alleviate this problem, the plan was to add a bent frame tube for the shock mount.

DESIGN

With having a tight budget, part of the task for this year was to make use of what was available in the shop. This meant going through bins and drawers and buckets to minimize the spending on manufactured and purchased parts. Since the previous team's car was never in running shape, this request seemed a little easier. There was enough tubing material left over to be able to manufacture the main portions of a lower A-arm and two upper A-arms. Since the ball joints were never used, they were able to be pressed out and saved. The ball joint cups, which house the ball joints, were cut off and salvaged as well. The wheel hubs and knuckles that the previous team used recently had the bearings replaced and never saw action. This meant a knuckle was already chosen for the design. Finally, the new suspension was designed to use the shocks selected by the previous team. This task was complex because the desire was to minimize the force on the arms while maximizing the travel and ride height. Limiting the shock options to only the previous team's purchase meant being creative and innovative. With tubing, knuckles, and shocks selected, a strong basis was already chosen for the design. The challenge was going to be getting the stresses manageable while maximizing performance.

With the new budget restriction requirements, the revised design goals are as follows:
Track Width Center to Center: 52"
Ground Clearance – Nose 14 ¾ "
Ground Clearance – Belly pan 12"
Suspension Travel – 9"

Lower arm

The first iterations of the lower arms were based off the 2013 Cincinnati car. They were true to the wishbone shape and used the dimensions obtained from the current car's frame. The UC baja team has learned a lot throughout the years in respect of what works in regards to adjusting camber and caster of the suspension. With that knowledge, the lower suspension was designed to be rigid with no adjustment. Any and all adjustments would have to come from the upper arm. Figure three shows what this lower arm looked like.

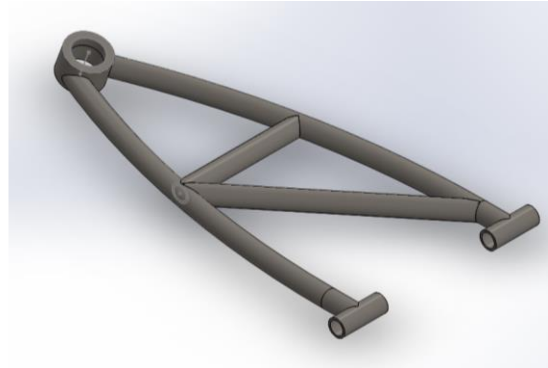

Figure 3: Initial Lower Arm

Upper Arm

The first iterations of the upper arms were also based off the 2013 Cincinnati car. Their shape was also a wishbone and included adjustment for camber and caster. Since the upper arm sees significantly less force than the lower arm, the use of threaded inserts and heim joints were acceptable. These joints give the arm some flexibility and allow the camber and caster to be adjusted to desired numbers. Figure 4 shows the first iteration of the upper arm.

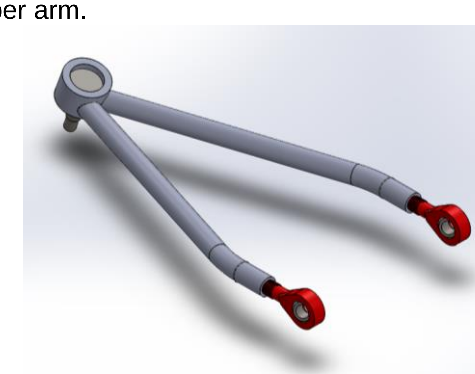

Figure 4: Initial Upper Arm

Improvements

Both of the arm designs proved to have sufficient strength when put under stress by the forces calculated in Solidworks (more details on this later). However, when it came to manufacturing, there was another obstacle. The available machine shop did not have the ability to bend the complex wishbone shapes. With the request to spend as little money as possible, it was back to the drawing board.

The other option here was to do textbook A-arms. They would only consist of straight pieces which could be manufactured in-house with the available tools. The arms had to withstand the same forces that were previously calculated. This was much more difficult than expected. The sharp edges of the arms had to be placed precisely in the right location so there would not be any unnecessary stress risers on the tubes.

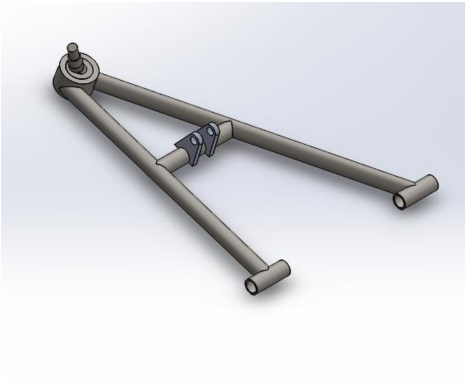

Figure 5: Optimized Lower Arm

Figure 6: Optimized Upper Arm

Figures 5 and 6 show the modified lower and upper arms respectively. By using the A shape design, the extra cross arm was eliminated. This design also reduced the overall weight as well as material usage. Therefore, the A shape was stronger and subsequently a better choice than the wishbone design.

Calculations

The suspension arms were tested in Solidworks using the calculations laid out in this section. Two scenarios were deemed "worst case" from which the digital simulation was based. This type of testing said that the parts will withstand the test given, but would fail if they experience more than is tested.

The scenarios were as follows:
Scenario one: Vertical Drop from 5 feet in which the car landed evenly on both tires
Scenario two: Collision with a buried tree travelling 25 mph (36.66 fps)

Scenario One: 5 foot Drop

The first thing that was needed was the velocity of the car as it hit the ground. One of the fundamental equations of constant acceleration was used for this calculation.

$V^2 = V_o^2 + 2a\Delta h$

Since the car was dropped, one can assume the original velocity to be zero, thus eliminating it from the equation.

$V^2 = 2gh$

Solving the equation for velocity and inputting the calculated values:

$V = sqrt(2gh) = sqrt(2 * 32.2ft/s^2 * 5ft)$

$V = 17.94ft/s$

Now that the velocity before the car hit the ground is known, the kinetic energy could be found.

$KE = (1/2)mV^2$

The total goal weight of the car was 525lbs. This is a measurement of force, not mass. The weight was needed to find the mass used in the kinetic energy equation.

$F = ma ==> m = F/a$

$m = 525lbf/32.2ft/s^2 = 16.3\ slugs$

Now that the mass was known, one could solve for kinetic energy. The kinetic energy was needed to find the average force from the falling car.

$KE = \left(\frac{1}{2}\right)(16.3\ slugs)(17.94ft/s)^2 = 2625ft * lbf$

Next, the decompression distance was used to solve for the average force. The decompression distance is the distance the kinetic energy can be slowed down in. In this case, after the travel of the shocks was utilized by the wheels, the car could no longer compress any more. Therefore:

$$d_d = wheel\ travel(T) = 9in * \frac{1ft}{12in} = .75ft$$

Using kinetic energy and decompression distance, the average force could be calculated.

$f_{avg} = \frac{KE}{T} = \frac{2625ft * lbf}{.75ft} = 3500lbf$

Assuming the landing was divided perfectly between both wheels, the force would be halved resulting in 1750lbf on one side of the suspension. Transferring this force to the shock tabs means it was halved again with 875lbf going into each tab. This can be seen in figure 7.

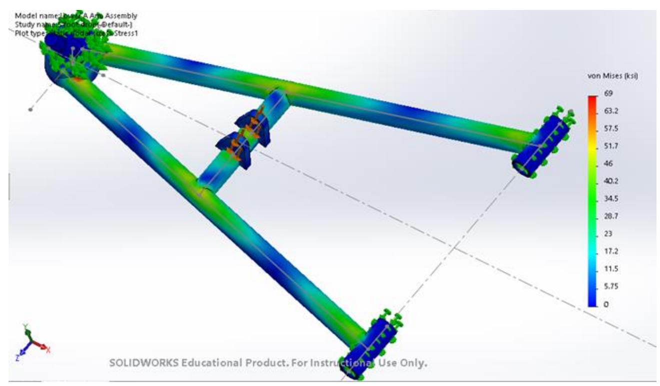

Figure 7: Lower A Arm 5 feet Drop Loading

The maximum stress seen here was 69ksi which was below the yield strength of the material, 70ksi. The maximum areas of stress occured at weld joints and tube connections. The filler rod used would have 80ksi strength giving a worst case scenario safety factor of 1.16. This means that should the car drop farther than 5 feet, the suspension would most likely fail.

Scenario Two: Impact with a Tree at 25 MPH

This scenario's loading conditions were based on data contained in The Motor Insurance Repair Research Centre (2). The data showed impulse time and G forces caused by crashes ranging from 11-32 mph.

The only thing needed to calculate here was the force using Newton's second law, f=ma. Mass was calculated in scenario one which determined a value for the G force from the data center. Plugging in these values resulted in:

$$f = ma = 16.3\,slugs * 4g's * 32.2ft/s^2 = 2100lbf$$

The force applied in Solidworks was slightly lower at 2000 pounds. Contacting a tree at this velocity was highly unlikely so the rounding to 2000 pounds would not make a significant difference.

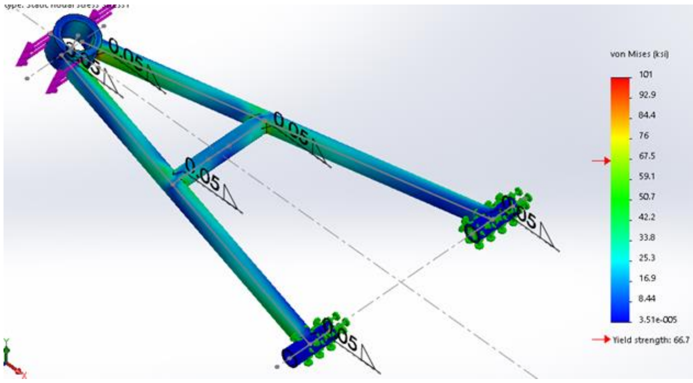

Figure 8: Tree Contact at 25 MPH

Figure 8 shows the resulting max stress of 101ksi. Using a feature called isoclipping, a selected stress level could be entered and only those areas would appear in color. Figure 9 shows isoclipping with a value of 70ksi, the yield strength of the material, entered.

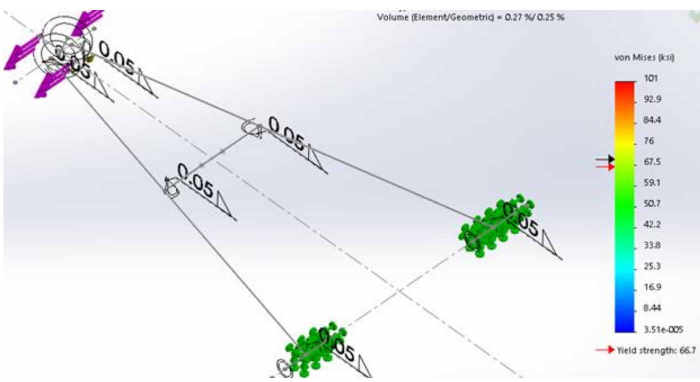

Figure 9: Isoclipping of 70ksi

As seen in Figure 9, only .25% of the arm experienced stress greater than 70ksi. Further isoclipping showed that only 3% of the arm experienced stress levels above 50ksi. The locations showed were all at points of intersections of tubes. In reality, these single points do not exist. The computers used to run FEA cannot handle running a finer mesh to get more accurate values and eliminate these impossible points. In reality, the stress level was much lower.

Taking the max stress value experienced at 50ksi, a safety factor of 1.4 was obtained.

More forces taken into consideration were the reaction forces to the mounting hardware. Figure 10 shows the reaction forces to the mounting points after the 25 mph collision with a tree.

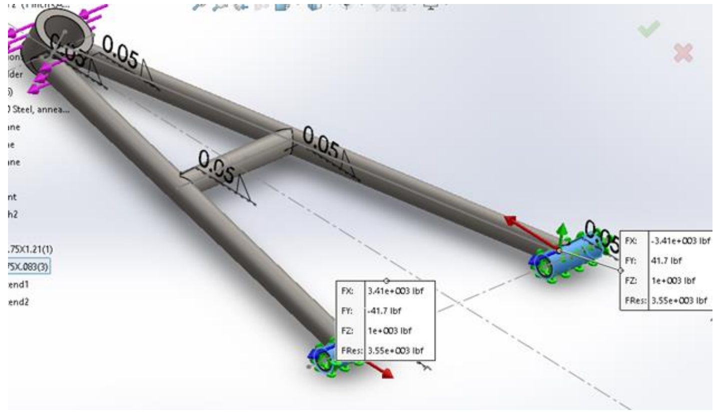

Figure 10: Reaction Forces on Hardware

The reaction force was shown to be 3410 lbf. Rounding to 3500 lbf, the shear forces on the mounting hardware was calculated. The equations were as follows:

$$\tau = \frac{F}{2A}$$

This equation was used since the bolt was mounted in double shear. Calculating the area of the expected m10 bolt being used:

$$\tau = 3500lbf/(2 * .1217in^2)$$

$$\tau = 14{,}375psi = 14.4ksi$$

The shear stress rating for grade 8 bolts was 80 ksi. This leaves a safety factor of 5.56.

SHOCK CHOICE

The shocks were chosen solely off cost reduction. Shocks are generally the most expensive part of a suspension setup. This is perhaps due to shocks being the most important part of the suspension. Without dampening, ball joints and control arms would be breaking repeatedly because there would be no suspension effect at all.

Since the previous year's team did not compete, their shocks were reused. They were only loaded to test for bump steer and travel. The shocks had a travel of 4.5 inches, which allowed for 9 inches of travel with a motion ratio of .5. The arms were designed with this motion ratio in mind. The shocks also came at a weight around 2 pounds which helped keep the weight of the system to a minimum.

MANUFACTURING

All of the parts for the front suspension system that needed to be fabricated were built in-house. This was completed in the North Lab machine shop on the University of Cincinnati's Victory Parkway (VP) campus. The shop consisted of many horizontal and vertical milling machines as well as several lathes and drill presses. These would prove very useful when it came to fabricating parts.

All of the tube sections for the front suspension were straight sections. In the past, members have designed wishbone arms that have slight bends to help with force dispersal. However, neither the team nor the VP lab had the means to achieve this. With cost savings constantly in the picture, straight arms were decided upon so outsourced work did not have to be done.

The material chosen was purchased by the prior team. There was enough material to build a front suspension system with only one more six foot section needing to be ordered for the fabrication of spares. This material was 4130 chromoly steel with a yield strength of 73ksi. These characteristics were input into Solidworks for the analysis which proved this material was perfect for the front suspension system.

The tubes were measured and cut to rough length using a tape measure and angle grinder. The final lengths were obtained by using a tube notcher as well as a flap disc to ensure parts mated closely for welding. This allowed fabrication within +/- .015 inches. The tolerance was made up by using threaded heim joints that could effectively extend or shorten the tubes. A picture of the notching setup can be seen in figure 11.

Figure 11: Tube notching setup

After notching all of the tubes, they needed to be welded. A welding jig was created so the process could be repeated quickly and accurately. The weld jig was able to be used for the upper and lower arms as well as the rear suspension trailing arm. The jig consisted of a quarter inch steel plate with drilled and tapped holes to secure block with machined profiles of the tubes. A wedge was also used to obtain the correct angle of the ball joint cup. A picture of the jig can be seen in figure 12.

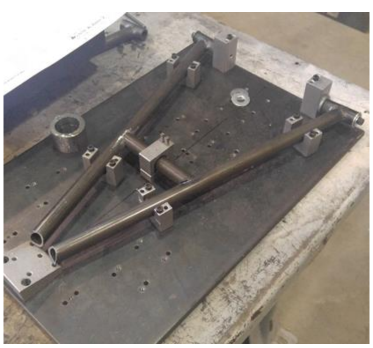

Figure 12: Welding Jig

One of the major manufacturing challenges was creating the ball joint cups. These cups attached to the end of the control arms and housed the ball joints. The joints allowed the suspension to travel and maintain correct geometry. The ball joints were designed to be press fit and retained by an external snap ring. Special attention was required to measure and machine this tight tolerance.

A boring bar with carbide inserts was used on a manual lathe in combination with a dial indicator to keep tolerances within +/- .0005 inches. Outside diameter (OD) micrometers were used to measure the outside dimension of the ball joint. The inside of the ball joint cup was measured using telescoping gauges that were then read by OD micrometers.

Another challenging component that needed fabrication was the upper shock mount. The frame was not well equipped to mount the shock the way it was designed. A joint of 4 welded tubes was the only place available for mounting the shock. A tube was designed to span the welded joint and attach on either side of it. This still allowed for the force vector to intersect the welded joint while simplifying the shock mounting. The tube was bent on the in-house tube bender and notched using the same processes as the arms. Figure 13 shows the completed shock tube with the shock mounted to it.

Figure 13: Upper shock tube

The shock tabs seen in figure 13, as well as the tabs to mount the arms to the frame, were all manufactured on the CNC plasma cutter at the VP lab. Solidworks has an output file type that was compatible with the plasma cutter. Two-dimensional drawings were imported into the machine and the tabs could be cut precisely to the drawing.

The final part manufactured was the threaded inserts. These inserts were welded to the upper control arms to allow for camber and caster adjustments. They could also correct any variance in tolerance as previously mentioned. They were turned to the correct OD on a manual lathe out of a solid piece of bar stock. A hole was drilled to the appropriate tapping size. The tap was then started on the lathe to ensure the thread was perfectly in the center. The turning process can be seen in figure 14.

Figure 14: Threaded insert lathe setup

RESULTS AND PROOF OF DESIGN

After manufacturing, painting, and final assembly, the final weight came out to be 54 pounds. This is 4 pounds over the goal but still 4 pounds under the previous team's weight. This was not detrimental to the project but did increase the overall weight of the car and driver. Going back and applying the heavier loads did not change much in the failure analysis. With the safety factors in place, the arms could still handle the stresses expected.

The car had a ride height/ground clearance of 14 inches. This was measured to the front bar. Just behind the tires, the belly pan height was nearly 12 inches. The track width came out to be 52 inches, which was the design goal. The full 9 inches of travel was not seen because a jump large enough to compress the shocks completely was not encountered. However, the shock did compress to 3.5 inches, meaning 7 of the 9 inches of tire travel was used. Over this distance, toe gain was minimal at 1 degree per 9 inches of travel. This meant the bump steer was minimal and complete control of the car could be maintained at all times.

The car was tested for about 7 hours total. 5 hours of this was at a local off-road park, Haspin Acres. They have around 600 acres of unpaved, ungroomed trails that are perfect for mimicking and exceeding conditions seen in competition. The team went after 2 days of rain in 45 degree weather. This gave sloppy conditions that caused the car to slip and slide and also gain weight in the form of mud. Throughout the testing no front suspension parts were broken. The car was not tested at loading conditions outlined in this report because, due to time and minimal spare parts, breakage was not wise. Figure 15 shows the car after this testing.

The remaining two hours was testing at VP campus. The car was driven on asphalt with concrete obstacles such as parking blocks and stairs. The biggest test was jumping off of a three-foot stair set. The car repeatedly jumped off this without damaging anything. The suspension performed as expected.

Figure 15: Complete car after testing

CONCLUSION

The customer requirements and product objectives of this car were met. Even though the team did not get to go to the SAE Baja competition, this car should do very well at competition. The way it performed against an older car was a clear indication that this car would do well. Through experience in off-road racing, worst case scenario testing conditions were realistic and not uncommon. This allowed for a thorough and well thought out design that would prove itself in competition. Hopefully next year, the car can be raced and put to the test.

Big thanks to the sponsors and Nick in the VP machine shop.

RECOMMENDATIONS FOR FUTURE BAJA SAE TEAMS

As the 2017 year comes to a close, many things have been learned about suspension geometry and handling. An old recommendation that has still not been answered is anti-dive geometry. When the car is under braking, weight shift causes clearance issues. The ground clearance was increased from previous years, but presidential duties kept from further researching and implementing anti-dive. Braking into obstacles is always common and would definitely be useful on this car.

CONTACT

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WORKS CITED

1. **International, SAE.** 2017 Collegiate Design Series SAE Baja Rules. [Online]
2. **Astrid Linder, Matthew Avery, Maria Krafft, Anders Kullgren.** *CHANGE OF VELOCITY AND PULSE CHARACTERISTICS IN REAR IMPACTS: REAL WORLD AND VEHICLE TESTS DATA.* [pdf] Thatcham : The Motor Insurance Repair Research Centre.

APPENDIX A

DEFINITIONS, ACRONYMS, ABBREVIATIONS

‘,ft: Foot

“: Inches

d_d: Decompression distance

f: Force

g: Gravitational constant

h: Height

KE: Kinetic energy

lb: Pound

lbf: Pound force

lbm: Pound mass

m: Mass

s: Second

Slugs: English unit of mass

FPS: feet per second

MPH: miles per hour

psi: pounds per square Inch

ksi: Kips per square inch

τ: Shear Stress

APPENDIX B

HOUSE OF QUALITY

	Customer Requirements	Importance wt.	Engineering Requirements (units)														Customer Satisfaction Rating (0.00 - 1.00)			
			Suspension Travel (in)	Ground Clearance (in)	Yield Strenght (ksi)	Complex Design (high/med/low)	Budget ($)	Weight (lbs)												
			1	2	3	4	5	6	7	8	9	10	11	12	13	14	CP	2016	2014	2013
1	Clears Obstacles Easily	0.25	9	9				3										n/a	0.8	0.9
2	Rides Smooth	0.15	3	3	1			3										n/a	0.9	0.9
3	Durable	0.25		1	9		1	1										n/a	0.9	0.7
4	Easy to maintain	0.15				9												n/a	0.7	0.8
5	Low cost replacement	0.20			3	1	9											n/a	0.7	0.7
6																				
7																				
8																				
9																				
10																				
	Total Importance	1.00	2.7	3	3	1.6	2.05	1.45												
	Engineering requirement importance																			

Performance	Current Product	1	2	3	4	5	6	7	8	9	10	11	12	13	14
	2016 Car	9	14	70	med	1500	39.6								
	2014 Car	10.7	14	70	med	2000	58								
	2013 Car	10.4	14	100.2	med	2500	54								
	New Product Targets	11	14	70	med	1300	45								

APPENDIX C

CUSTOMER SURVEY

In order to better cater to our customers, we would like feedback on a front suspension system. The goal of the survey is to determine which parameters are most important so we can focus on improving them.

How important is each feature to you for the design of an off-road front suspension? Please circle the appropriate answer.
1 = low importance 5 = high importance

Suspension Travel	1 2 3 4 5 N/A
Ground Clearance	1 2 3 4 5 N/A
Durability	1 2 3 4 5 N/A
Maintenance	1 2 3 4 5 N/A
Replacement parts cost	1 2 3 4 5 N/A
Ride comfort	1 2 3 4 5 N/A

How satisfied are you with the current off-road front suspension? Please circle the appropriate answer.
1 = very unsatisfied 5 = very satisfied

Suspension Travel	1 2 3 4 5 N/A
Ground Clearance	1 2 3 4 5 N/A
Durability	1 2 3 4 5 N/A
Maintenance	1 2 3 4 5 N/A
Replacement parts cost	1 2 3 4 5 N/A
Ride comfort	1 2 3 4 5 N/A

How much would you be willing to pay for a complete off-road front suspension?

$50-$100 $100-$200 $200-$500 $500-$1000 $1000-$2000

APPENDIX D

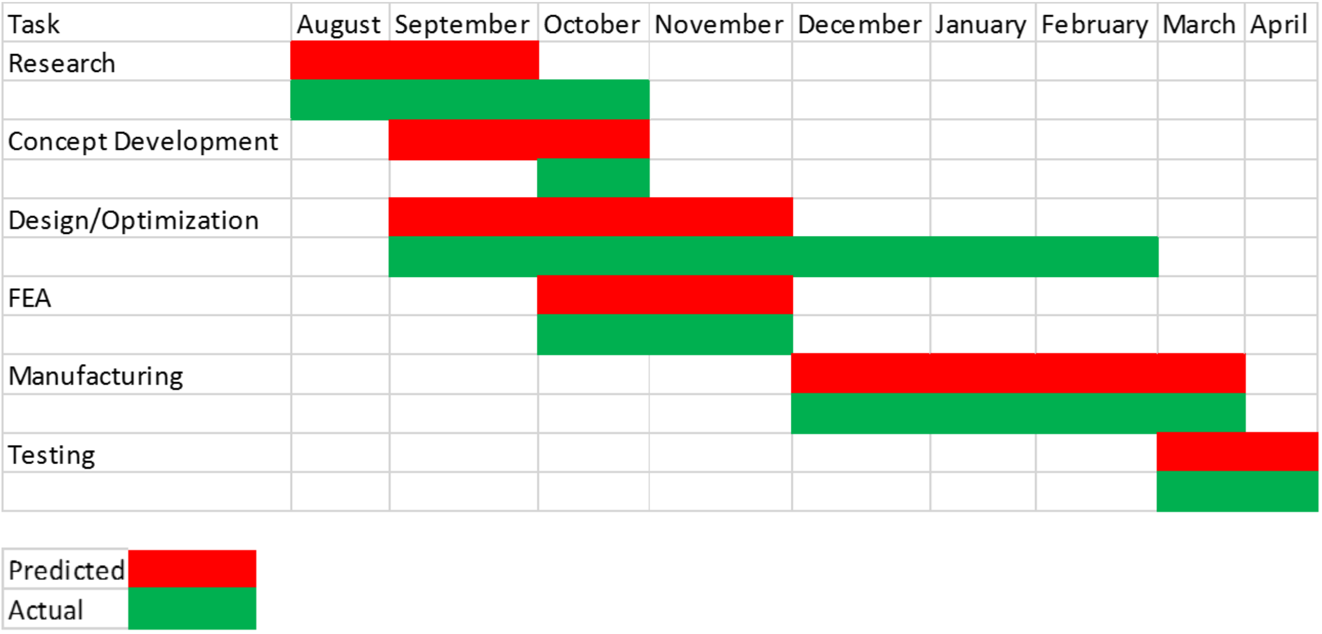
BUDGET

Front Suspension Parts	
Part	Cost
Tap for Heims	16.59
Delrin for bushings	14.75
Heim Joints	127.50
Heim Hex nuts	7.53
Lower A Arm bolts	13.30
Mounting Nuts	8.91
Mounting Washers	4.36
Steel Tubing for Arms	75.00
Shock Nuts	9.37
Shock Washers	3.23
Shock Bolts	9.77
Upper A Arm Bolts	10.24
Hub Spacer	12.00
Total	312.55

Budgeted	$2000.00
Spent	$312.55
Remaining	$1687.45

APPENDIX E

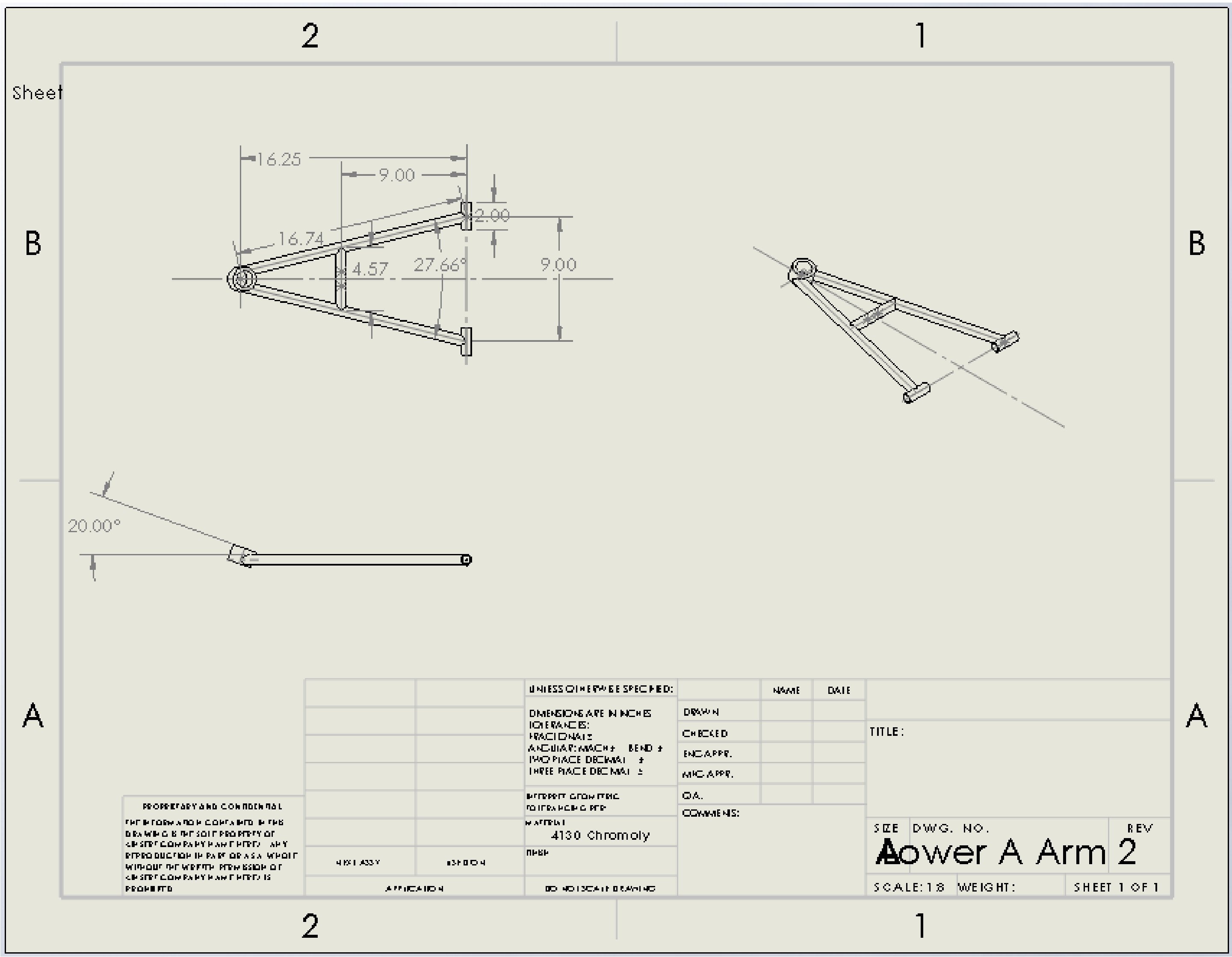
TIMELINE



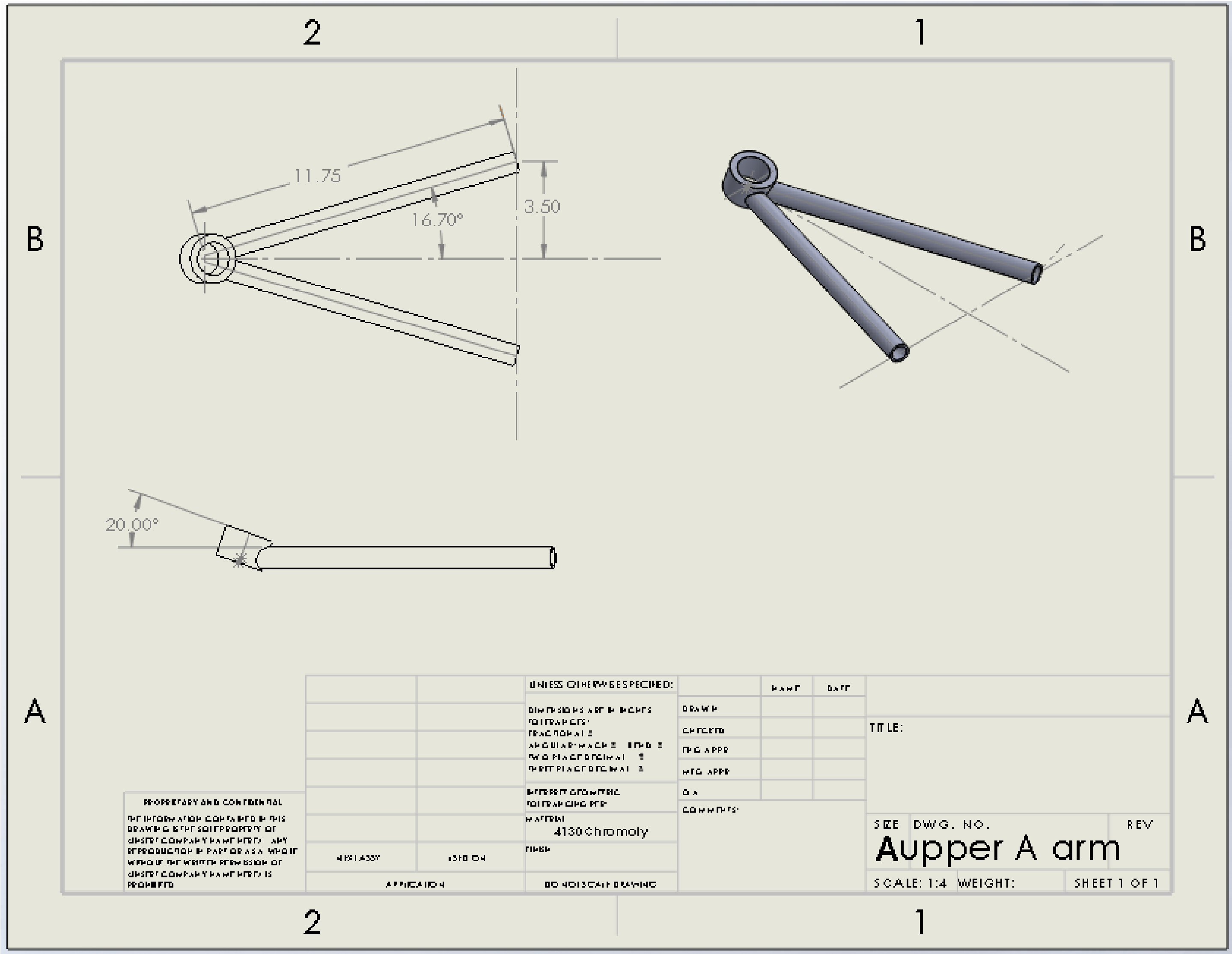
APPENDIX F

MANUFACTURING DRAWINGS

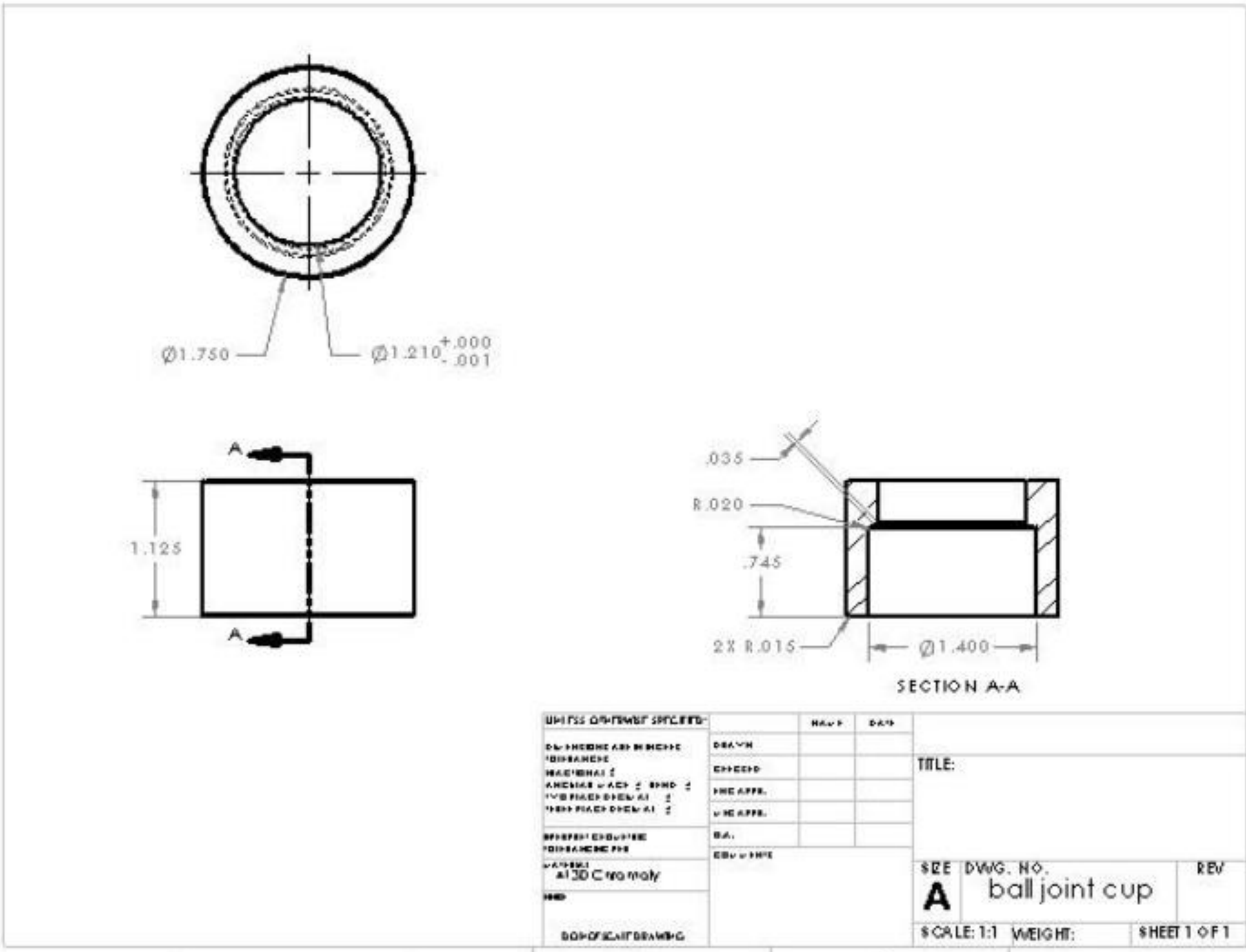
LOWER CONTROL ARM



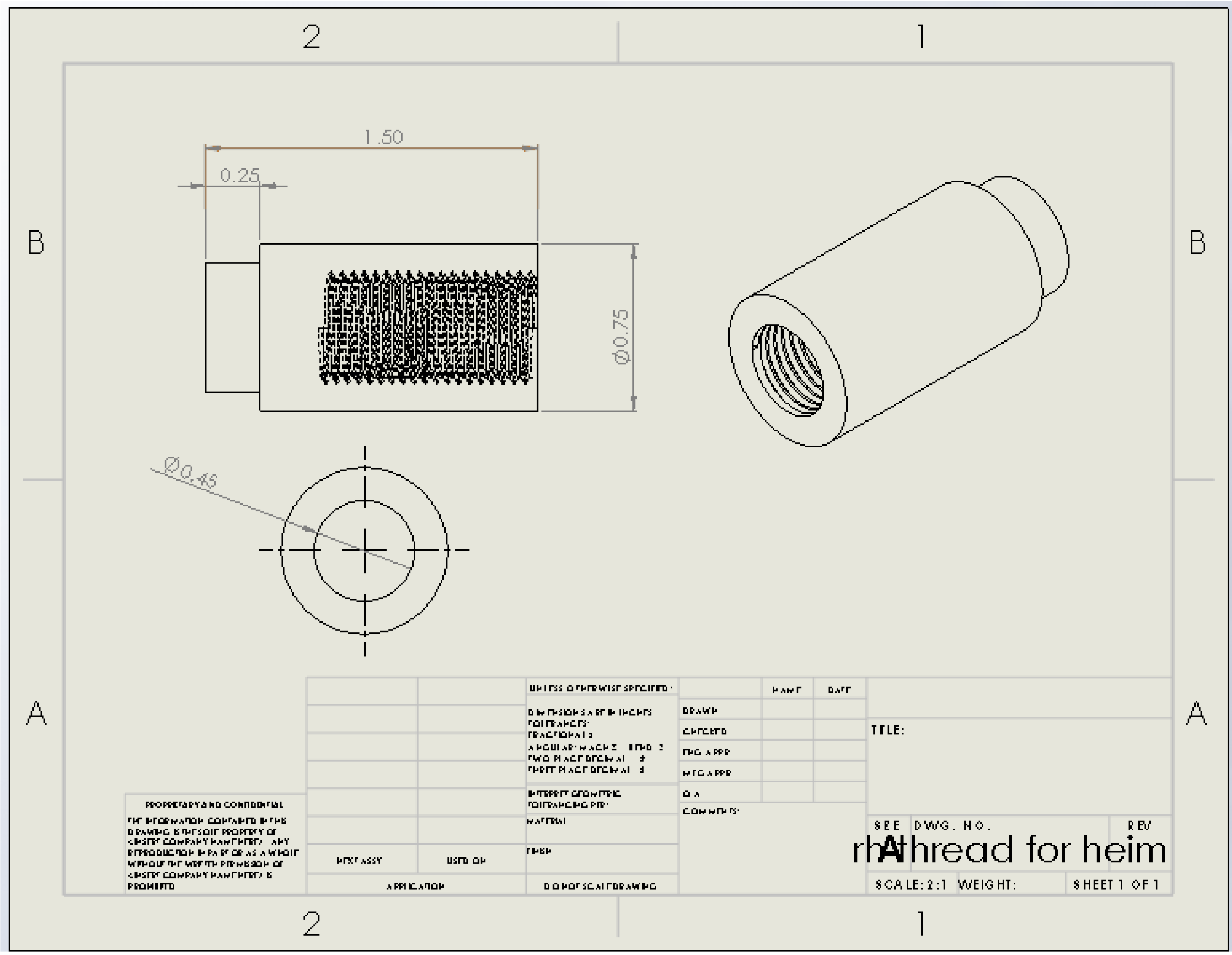
UPPER CONTROL ARM



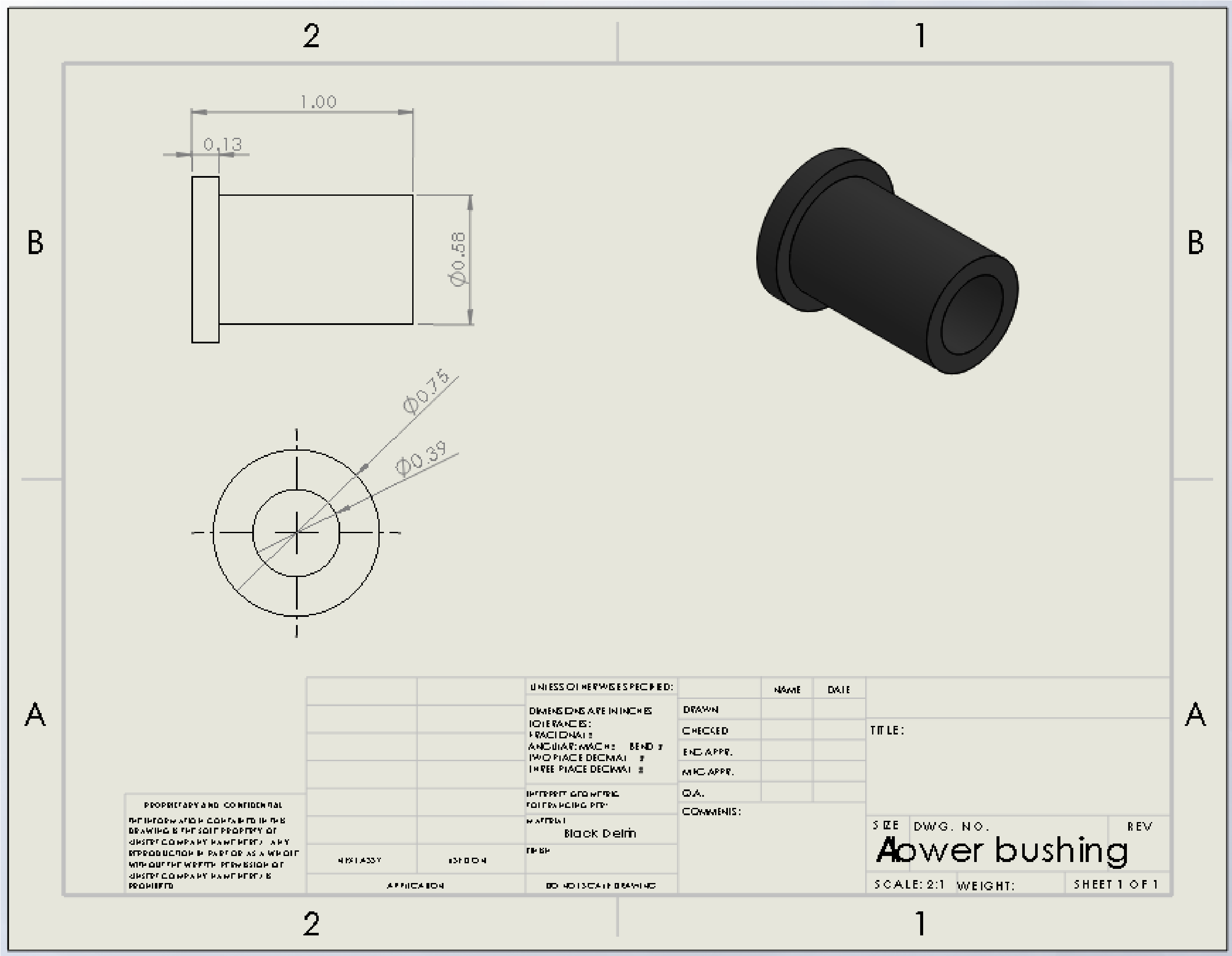
BALL JOINT CUP



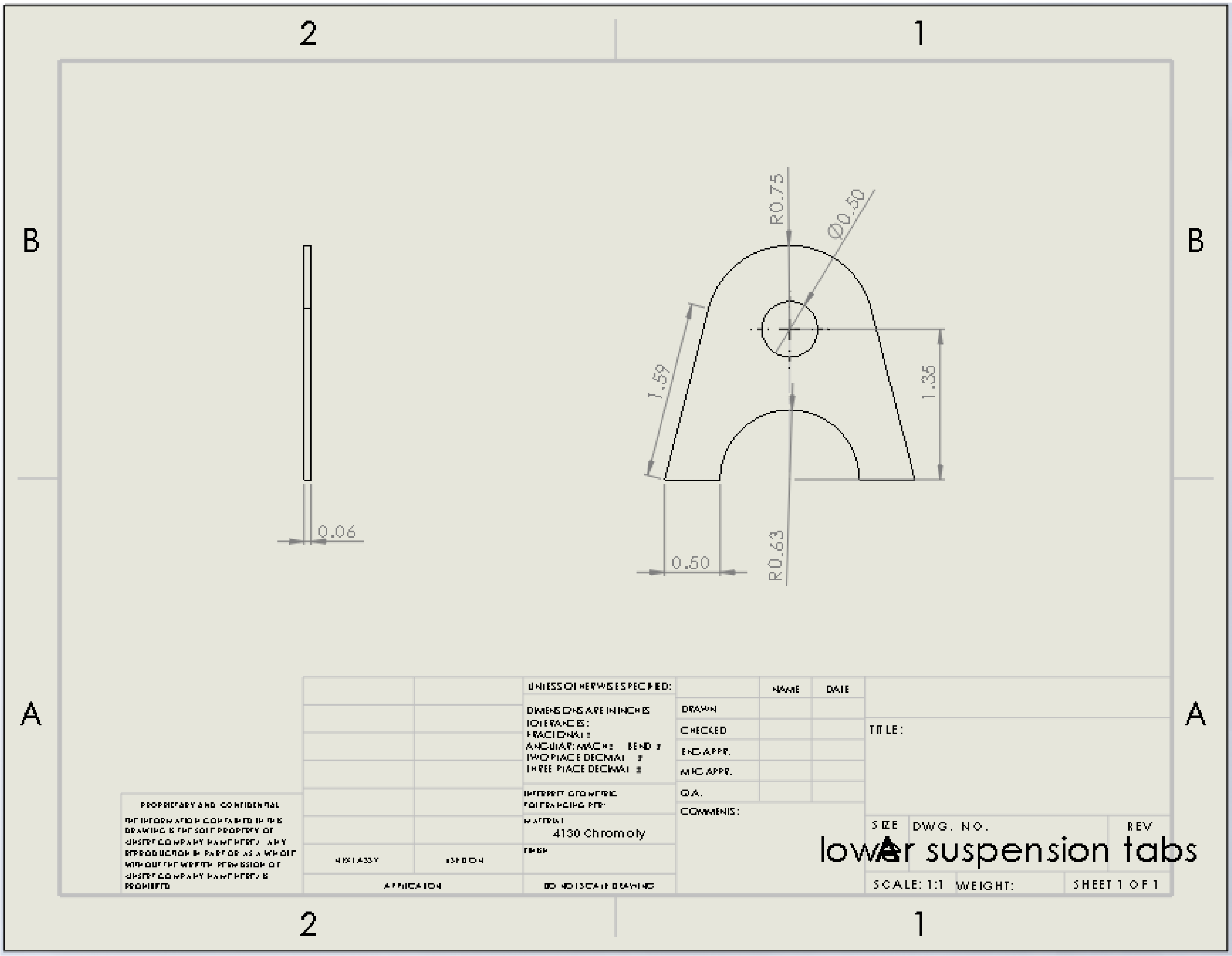
THREADED INSERT



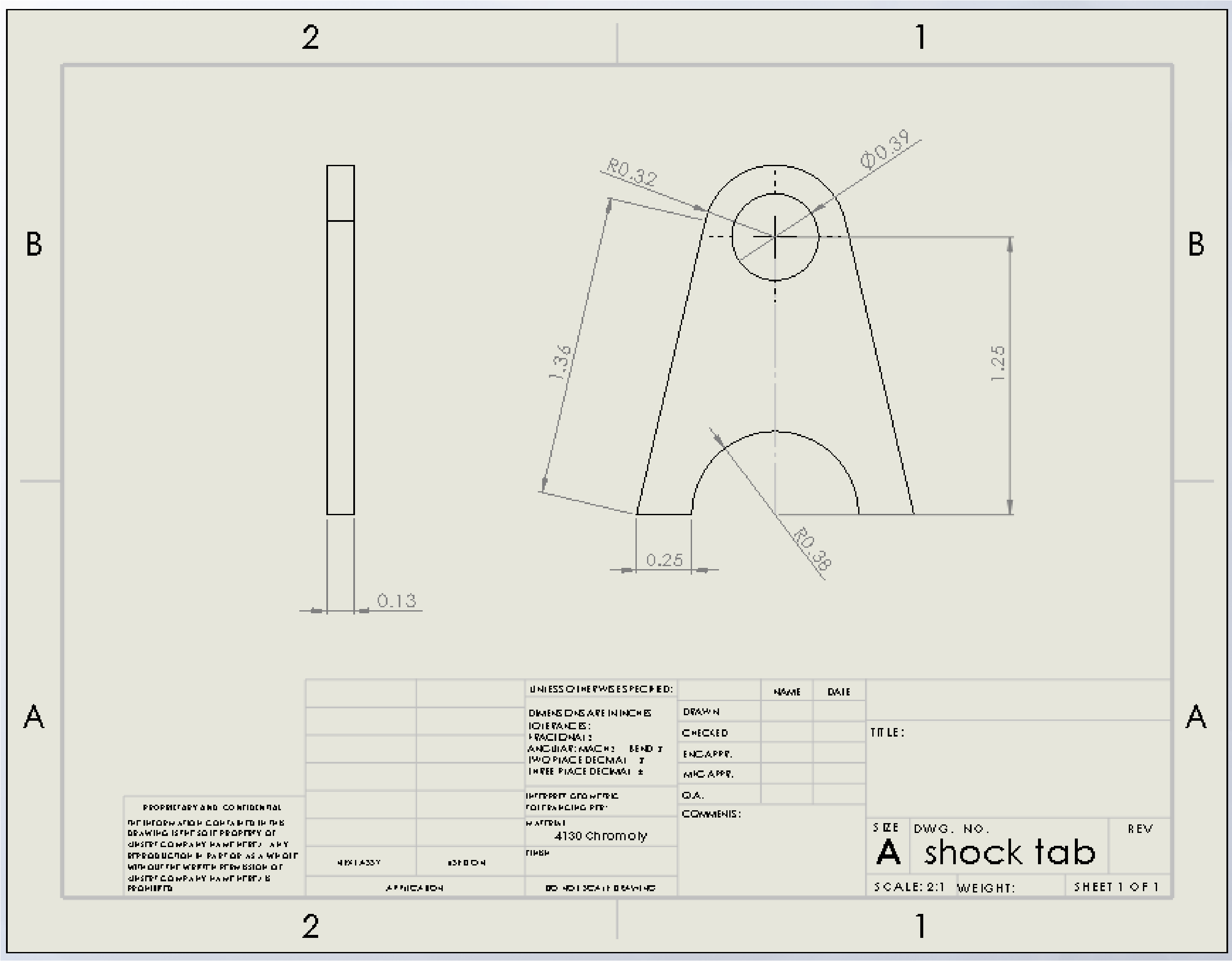
LOWER CONTROL ARM BUSHING



LOWER AND UPPER MOUNTING TABS



SHOCK TABS



R0.32
Ø0.39
1.36
1.25
0.25
R0.38
0.13
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
4130 Chromoly
FINISH
NEXT ASSY
USED ON
APPLICATION
DO NOT SCALE DRAWING
NAME
DATE
DRAWN
CHECKED
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
TITLE:
SIZE
A
DWG. NO.
shock tab
REV
SCALE: 2:1
WEIGHT:
SHEET 1 OF 1
PROPRIETARY AND CONFIDENTIAL