Cowboy Motorsports

Spring 2017 Report

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Problem Statement

To design and build a cost effective, reliable, and innovative frame, steering system, and suspension system for the Oklahoma State University Quarter Scale tractor team. The design will take into account the team's budget, timeline, and resources for the 2016-2017 competition.

Figure 1. Team picture from the 2015-2016 competition

Introduction

Each year ASABE holds the international quarter scale tractor student design competition in Peoria, Illinois. This competition is designed to give students an opportunity to take a project from concept to finished product. The competition is made up of several parts. Each portion is assigned a maximum possible point value as seen in **Error! Reference source not found.** below.

Design Judging is an interactive portion of the competition where teams present their design's attributes in the particular category to the panel of judges. The judges may then ask questions for further details or provide comments for development of the team's next model. The design judging portion is made up of the following six categories; manufacturability, serviceability, ergonomics, safety, test and development, and sound judging. Each category is worth 70 points and are judged by professional engineers, technicians, or operators from industry.

Technical inspection is the pass or fail portion of the competition. All teams are required to pass a full technical inspection prior to participating in practice pulls or competing in any Performance Competition. This process is broken into two independent portions: Initial Weigh-in and a Detailed Technical Inspection. Technical inspection verifies compliance with the rules set forth by the competition committee. Operator safety and weight limit are the main focus of this inspection. The initial weigh-in will receive a 100 point bonus for starting and operating under its own power, having all shielding in place as best as possible, being on time to the scheduled tech time slot, and completing the inspection in under 24 hours from the end of the assigned time slot.

The pull performance event is comprised of a multi-stage tractor pull using a progressive sled. Points are gained by the number of feet the sled is pulled by the respective tractor. Each team will be allowed one scored pull in three separate heats.

The Maneuverability Course Event is held to encourage consideration for maneuverability in tractor design. The team(s) with the lowest number of overall 'course demarcations' will receive a maximum of 100 points (course demarcations indicate number of direction changes, distance traveled, and number of collisions with cones).

The Durability Event is conducted on an oval course setup on the pulling track that consists of bumps and loose sand. The bumps are no taller than 2.5 inches and set up in a random array to be determined at competition. The loose sand has a depth of approximately 6 inches. Teams will be required to tow a 4-wheel cart weighing up to 2000 lbs (with approximately 0% tongue weight) through the entire course. The cart attaches to the rear hitch of the tractor. Laps are 250 +/- 50 feet in length.

Points allotted to teams for sound level are based on the sound decibel level recorded during the team's first attempt in the sound level Tech Inspection station. The team with the lowest value below the required 91 decibel will receive the full 70 points. Other teams will receive points on a scale from 91 decibel to the lowest level, with allowed points weighted more heavily toward the lowest decibel value (i.e. this will not be a linear scale). No points will be awarded if the sound technical inspection is not passed during the first attempt.

Table 1. Competition points break down

Design Report	500 pts
Team Presentations	500 pts
Design Judging	420 pts
Technical Inspection	Pass/Fail
Tractor Pulls	600 pts
Maneuverability	100 pts
Durability Event	200 pts
Initial Weigh in	100 pts

The proposed project redesigns the main frame, support structures, suspension, and steering of the ¼ scale pulling tractor for the 2017 international competition. The basis of the project is to increase competitiveness of the tractor by increasing functionality of the frame, suspension, and steering. This is achieved by providing a product that makes use of CAD programs to model and test the product. By doing so, the design will have the added benefit of a seamless assembly while optimizing the use of materials required for the product.

Impact

This project is purely of the mechanical nature and part of a larger team design project for the ASABE International Quarter Scale Design Competition. It provides teams with insight into engineering in industry. The team must go through the engineering process and design solutions to address the challenges set forth by the competition. The competition rewards teams that design products with manufacturability, serviceability, ergonomics, and safety in mind. The project allows students to have a hands on experience with taking a concept all the way to production.

Competition Requirements

Our client requires us to follow the 2017 International ¼ Scale Tractor Student Design Competition Rules. These rules provide guidance on how the tractor can be designed and built. One of the two requirements it sets for the frame is that the tractor cannot be longer than 96 inches when measured from the center of the rear axle to the farthest part forward. The other requirement is that it has to be fully customized. This means a frame cannot be a modified frame from a similar vehicle. It must be designed by the team specifically for the ¼ scale tractor. Steering must be achieved with the front tires. Articulated tractors, tricycle front ends, and skid type steering are also against the rules. All steering components must use grade 5 or M8.8 fasteners and locking nuts with a minimum of two threads showing. The suspension falls under the same fastener rules as the steering components.

Client Requirements

The entire design needs to take into consideration what design features the judges look at. This includes manufacturability, serviceability, ergonomics, weight, cost, and strength. Each of these areas have points associated with them for the competition. All the events throughout the competition require the use of the frame, steering, and suspension. Therefore, the effectiveness of these components in each event need to be considered throughout the design process. The overall goal is to score as many points as possible throughout the entire competition.

Client requests

- Frame
  - 100% welded frame
  - Support structures that are incorporated with mounting brackets
  - Reduce weight from previous model
  - Display university and club name
- Steering
  - Reduce force needed to turn steering wheel
  - Improve alignment of steering components
  - Improve steering geometry
  - Improve adjustability
- Suspension
  - Incorporate an adjustable ride height
  - Improve damping of impact stresses applied to the tractor

Design Concepts

Factors of Steering and Handling

Steering is defined by the alignment of the tires and the geometry of the wheel base. Using the parameters of camber, caster, toe, steering axis inclination, included angle, scrub radius, and Ackermann steering geometry a vehicle's steering system can be tuned for the best performance based on the challenge at hand. The bulk of these parameters can be grouped into the category of wheel alignment, which by definition is the complex system of angles and adjustment of suspension components (Auto Dimensions Inc., 2016).

Camber is defined as the angle of the wheel, which is measured in degrees off of the true vertical plane. This angle can limit traction and act as a direct influence on toe angle (Auto Dimensions Inc., 2016). The angle of camber is largely determined by suspension travel and the type of control arm. Caster is the angle at which the steering knuckle pivots and can affect the straight line tracking of a vehicle. A positive angle results in difficult steering and steering wheel kick as the tire impacts obstacles. A negative angle causes difficulties maintaining a straight line (Auto Dimensions Inc., 2016). Toe is defined as the angle of the tires in respect to the centerline of a vehicle. For most rear wheel drive vehicles the toe is set positive to provide better straight line tracking. On the other hand front wheel drive vehicles are typically set negative to compensate for the forward movement of suspension (Auto Dimensions Inc., 2016). The steering axis inclination, included angle, and scrub radius are all affected by the camber, caster, and toe of a vehicle.

Ackermann steering geometry is simply defined as the two steering wheels pivoting at the ends of an axle beam at different angles so that the lines drawn through their stub-axles converge at a single point in-line with the rear axle (The Ackermann Principle as Applied to Steering, 2016). The idea behind Ackermann geometry is that the inner tire travels a shorter distance than the outer tire as is demonstrated in Figure 2.

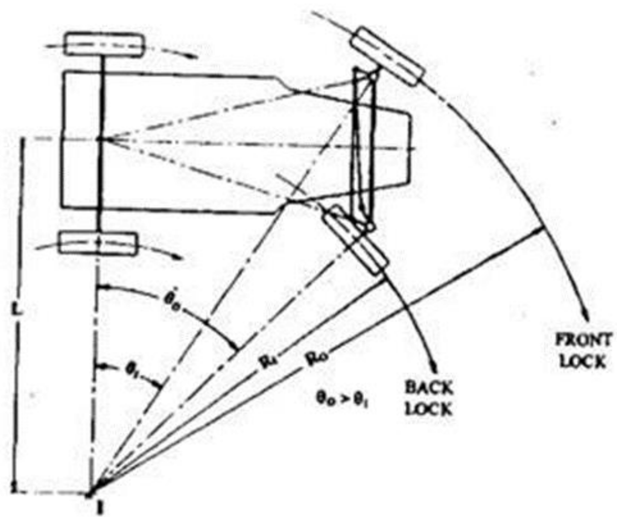


Figure 2. Ackermann steering geometry, from *The Ackermann Principle as Applied to Steering*

This particular steering setup is advantageous over a parallel steering system because Ackermann geometry keeps the two steering tires from fighting against each other during turns. In a parallel system, the two tire paths want to intersect as shown in Figure 3. This forces the tires to push against each other and causes unpredictable steering.

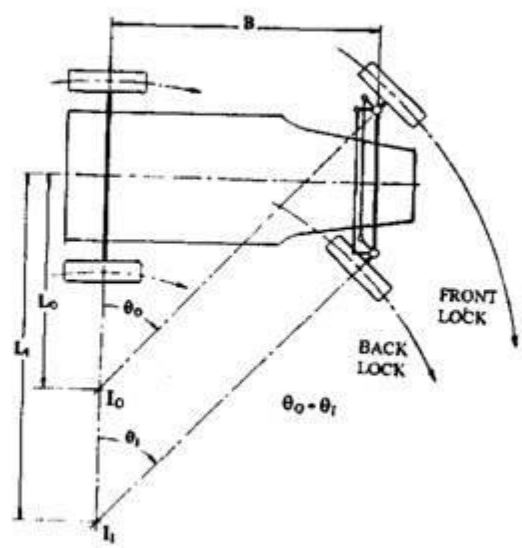


Figure 3. Parallel set steering geometry, from *The Ackermann Principle as Applied to Steering*

Steering Methods and Systems

There are a variety of steering systems used in industry. The most common are rack and pinion, steering box, power assisted steering (both hydraulic and electric), and electronically controlled steering.

As shown in Figure 4, the rack and pinon system makes use of a small pinion gear located at the bottom of the steering shaft. It is seated in a housing that contains a row of teeth. This system very simply changes the rotational movement of the steering wheel into lateral movement that is used to move the tires (How the steering system works, 2016). Smaller vehicles and equipment, like go karts and riding lawn mowers, often use a rack and pinion system. This type of system is best used on non-driven tires and can be more difficult to steer. This system is optimal for small machines due to its compact size and simplicity. The greatest drawback is that the fully manual steering can be cumbersome when the contact surface of the tire is increased.

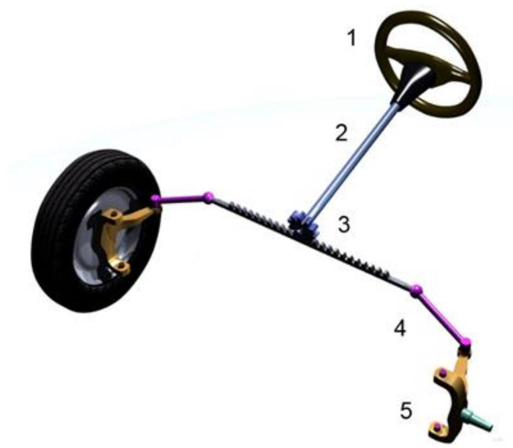


Figure 4. General rack and pinion system in modern vehicles

The steering box system is a bulkier version of the rack and pinion that makes use of a worm gear which controls a lever arm known as the pitman arm. The movement of the arm then controls a mechanical linkage that then steers the tires of the vehicle. This provides a less precise method of steering and more potential for wear (How the steering system works, 2016). The steering geometry is controlled by a drag link and tie rod that connects the hub assembly to the pitman arm. This sees most of its application in off-road vehicles and many rear wheel drive vehicles.

Power assisted steering is less its own system and more of an addition to the previous two. Using a hydraulic or an electrical system, the torque generated by the driver on the steering wheel is amplified in the steering box or rack to ease the steering. This method is widely used in the automotive and agricultural industries today. Having a mechanical system in place if damage occurs to the hydraulics or electronics is an important safety feature. The steering may become cumbersome, but the operator can still maintain control of the vehicle (How the steering system works, 2016). Electronically controlled steering is most commonly found in large ships, airplanes, and modern cars. This method strictly uses an electronic system to control actuators and motors to control the steering. This results in a very quick and light steering that can only be operated while electrical power is being supplied to the system.

Basic Frame Design

The frame is the main supporting structure of a motor vehicle. It is used to mount components and bare the weight of the machine. It needs to be strong enough to support the vehicle, but small enough to be economical. The manufacturers have to take into account where forces will be applied, how large they are, and how they can be spread throughout the frame to avoid overloading one area.

Frame Types

When looking at car and full size tractor frames, many of them are made up of large rectangular steel tubing, shown in Figure 5. Rectangular tubing is used because of its load bearing capacity. The webbing on both sides of the top and bottom flanges enable it to support forces and moments enacted on it, while keeping it at a reasonable size when compared to the overall machine. Its shape allows it to spread out the stress and torque applied to it. This type of shape is a good starting point because it provides the rigidity and durability needed for these machines.

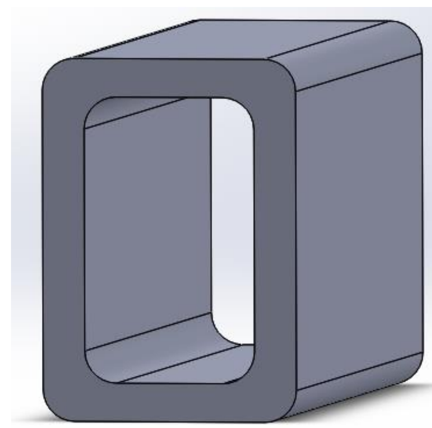


Figure 5. Cross section of basic tube frame design

Another type of frame often used in cars is the uni-body, or monocoque frame. It combines the frame and body of the vehicle, making it all one piece. It is able to withstand the forces and torques applied by the vehicle because of careful and precise engineering. Tubular shapes and cross braces make up the uni-body frame.

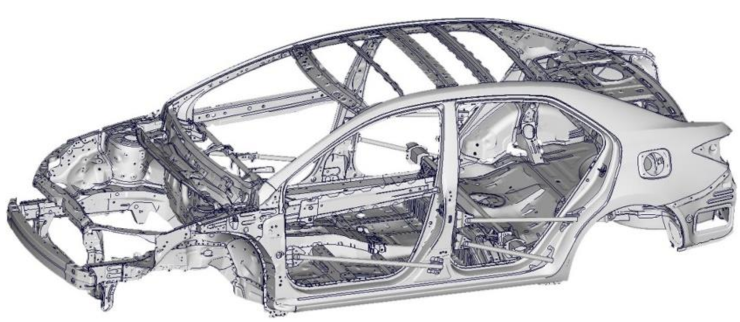


Figure 6. Example of uni-body frame

Many structural principles are combined to create a monocoque frame, shown in Figure 6. The uni-body is very specific to the car it is designed for, thus requiring a lot of engineering work to complete. Nevertheless, once the design is complete, it greatly reduces production costs and speeds up manufacturing time since there are fewer parts to assemble.

Many of the quarter-scale tractors made by our competitors have a sheet metal C-channel frame, as shown in Figure 7. This is a good design, because it provides the strength needed to support the tractor, but is also lightweight. It is very similar to a tube frame in its ability to handle bending stress. The single web is able to withstand the forces and moments applied to the frame. However, it requires some extra support members to handle torsion. These support members must be strategically placed in order to spread out the forces seen by the frame. Figures 8 and 9 show how support members can strengthen the design of a frame. In Figure 8, the frame is lacking support structures. High stress concentrations can be seen where the forces are applied. The frame in Figure 9 has properly placed support structures and there is significantly less stress concentrations present.

Figure 7. Cross Section of C-Channel Frame

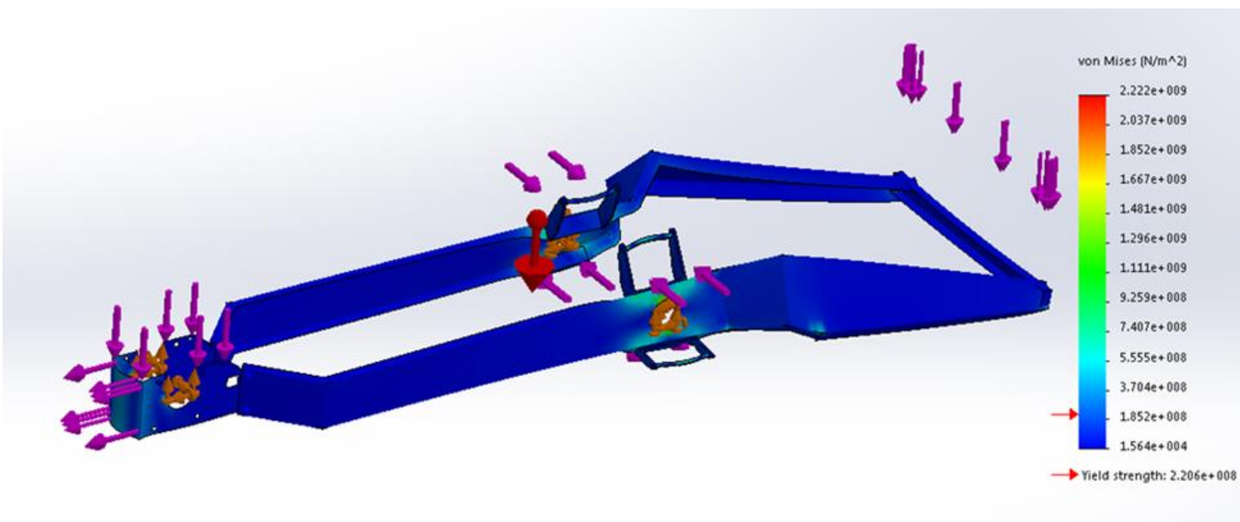


Figure 8. Frame without support structures

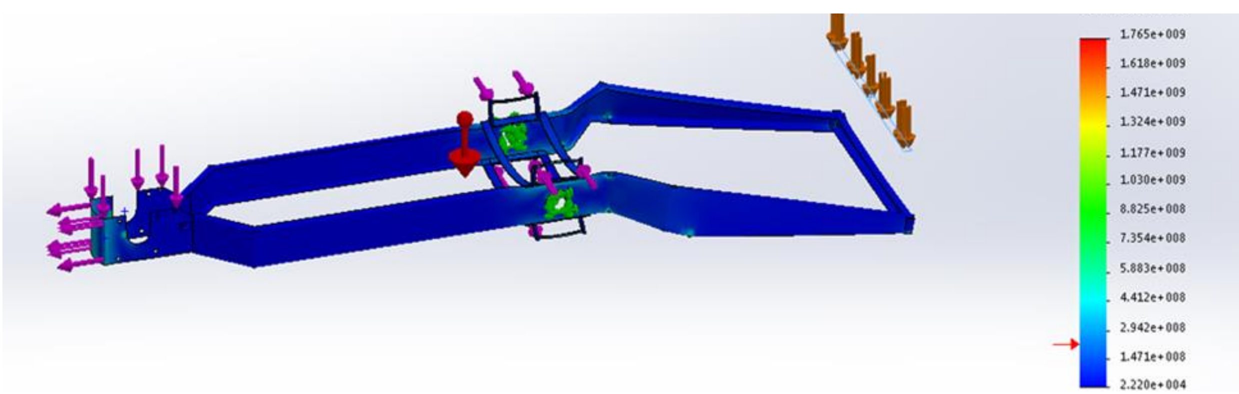


Figure 9. Frame with support structures

Types of Suspension

Air springs, air shocks, linear actuators, hydraulic cylinders, and coil over shocks are the most common forms of ride height adjustment. They can all work on different types of vehicles and applications, shown in Figure 10.

Figure 10. Example of air springs on a vehicle, from *Progressive Automotive*

Coil over shocks and hydraulic cylinders are good options for adjusting ride height of a vehicle. They are both used in the agricultural and automotive industries. Hydraulic cylinders can be configured in a circuit that allows them to be used as both a lifting mechanism and suspension. However, weight is the major downfall to this system with respect to the competition weight requirements. Hydraulic components and fluid are all heavy duty and therefore add a considerable amount of weight to any vehicle.

Coil over shock absorbers are proven in their ability to damp impact forces. They are also manually adjustable. However, the range of adjustment is limited. Air shocks, again, are proven products in industry. This suspension system has a shock absorber and lifting mechanism integrated together. Linear actuators are another strong candidate. They would work very well as a ride height adjustment device. They are easy to install and can handle the load of the front of the tractor. One major problem with this system is the electrical engineering necessary to make it function as a suspension system.

Air springs are commonly used on tractor trucks and their trailers. They are also used as aftermarket add on systems to trucks and SUV's that haul heavy loads. Air shocks can be configured in a system that functions as both a ride height adjustment and functional suspension. The air springs are lightweight, range from 2-4 pounds, and are reasonably priced at $200-$300 each.

Recommendation

Frame

Frame design selection began by looking at what is seen in industry and what could be the most applicable to our requirements. Previous designs were not going to be used just because it was done in the past. There would be research and causes behind our decisions. The three main frame types that apply to this project are the tube frame, uni-body frame, and C-channel frame. The description and uses can be seen in the Design Concepts section. Using a tube frame design would give our ¼ scale tractor all of the support it needs. However, the amount of material used in a tube frame makes it difficult to utilize the system and keep the total weight of the tractor under 800 pounds.

The C-channel frame is basically a tube frame without one of the side flanges. This reduction in weight is a tradeoff for strength. As mentioned in the Design Concepts section, the C-channel frame requires extra support members to be strong enough to handle the forces applied to the tractor. However, the combined weight of the C-channel frame and support structures could still be lighter than a tube frame. The C-channel frame has been used in previous models and has provided the opportunity to learn from the failures seen in those models.

A full uni-body design is not feasible for this project. The resources necessary to design a full uni-body frame are unavailable. However, some of the same ideas can be used for engineering mounting brackets and cross members. A slot and tab method, shown in Figure 11, will allow each component of the frame to be welded into place. This decreases assembly time and the chance of misalignment. Pieces can be assembled together without measuring, thus reducing the amount of possible human error. The components will also be designed to fit together in only one way, making it impossible to assemble incorrectly. Strength and serviceability are improved by designing the frame, mounting brackets, and braces as one piece. The strength is increased by transferring forces throughout the entire frame and reducing stress concentrations. Serviceability is increased because major components are directly bolted to the mounting brackets designed into the frame. The combination of C-channel frame with cross bracing similar to a uni-body is the recommended frame selection.

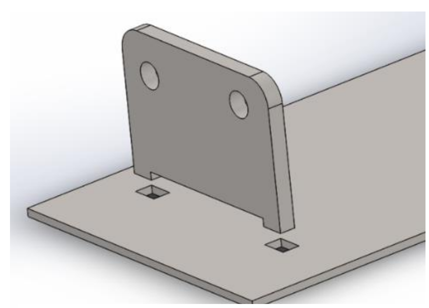


Figure 11. Example of the slot and tab method

By thoroughly examining the previous model's frame, shown in Figure 12, the new frame can be optimized. It was made of 14 gauge steel (.0747 in), 5" tall, had a 1" top and bottom flange, 91" in length, and 17" wide. It had 45° bends at the rear to fit around the rear differential mount. There were also no additional support structures designed into the prototype because of lack of analysis due to time constraints in the previous year.

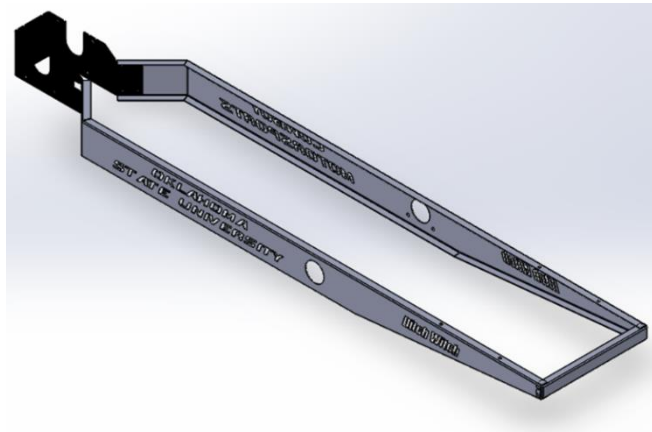


Figure 12. Previous model frame

Due to the absence of support structures, the previous design started to deform in multiple places. If left unattended, the structure would have eventually failed. The first place it deformed was in the 45° bends at the rear. When the sheet metal used for the frame was bent, a sharp corner, shown in Figure 13, was left at the 45° bends. When the rear wheels would go over a bump and apply torque to the frame, it caused the stresses to concentrate at those corners. This developed cracks along the bend, shown in Figure 14. If the sharp corner would have been welded together during assembly, it would have strengthened the frame at that area by eliminating the stress concentrations at those points. Additional support structures could have also been used to further strengthen the frame in the rear end.

Figure 13. Stress concentration in sharp corner

Figure 14. Cracking from sharp corners in frame

The other place the frame deformed was at the front differential. There were no supports around the differential causing the weight of the tractor to pull the bottom rails of the frame apart, while pushing the top rails together. This is represented by Figure 15. If left unresolved, the deflection would have caused the frame to be pushed past its ultimate strength. Figure 16 shows the stress distribution seen by the frame during the pulling events. Large stress concentrations can be seen around the front axle.

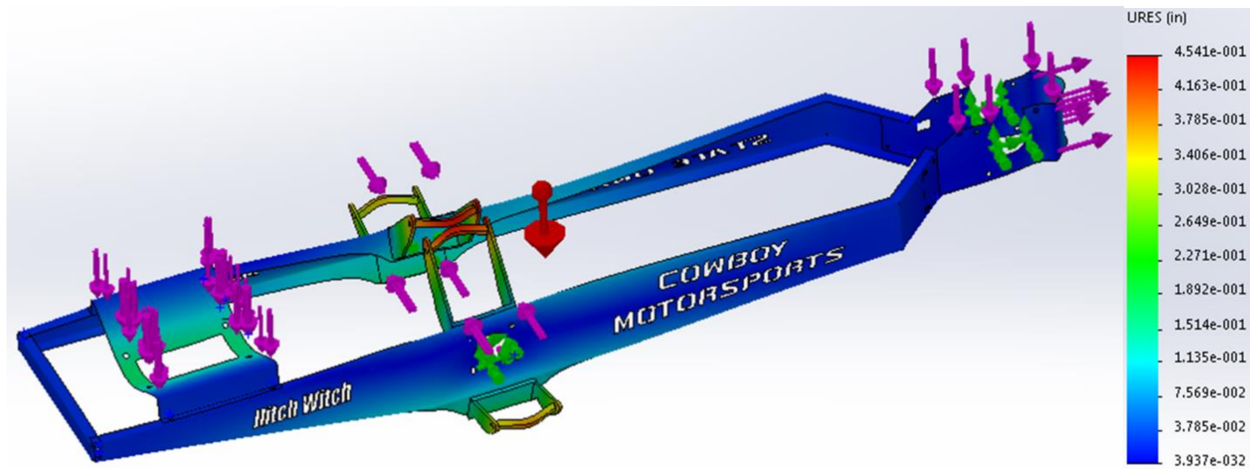


Figure 15. Exaggerated displacement of frame due to high stresses at front axle

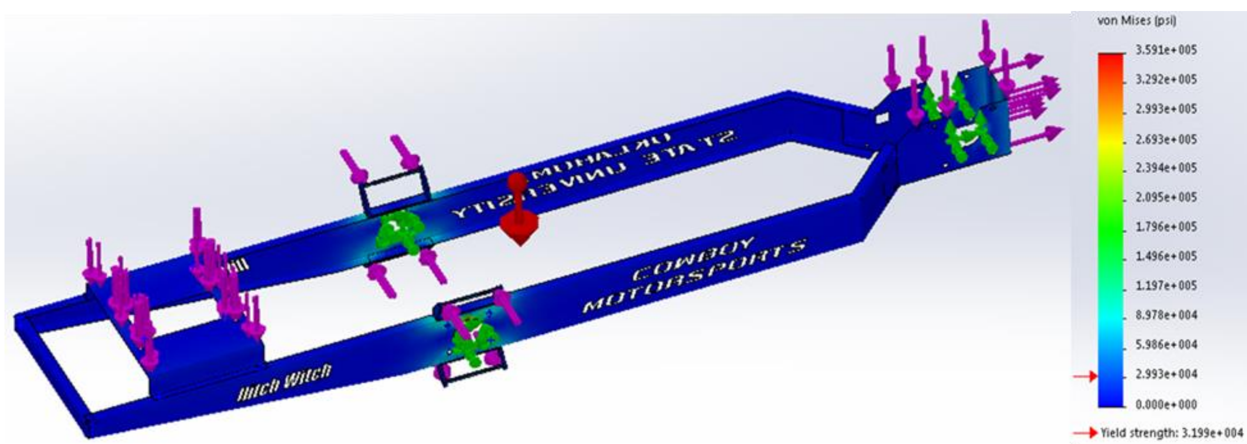


Figure 16. Stress distribution of forces applied to the frame during pulling events

To combat cracking in the rear end, changes were made to strengthen the frame in that area. The Solidworks simulation in Figure 17 shows where stresses act and how large they are. Comparing the simulation of the 45° bend to the 30° bend shows how the smaller angle reduces stress concentration. Also, reducing the angle allows more stress to be transferred down the length of the frame instead of acting perpendicular to it.

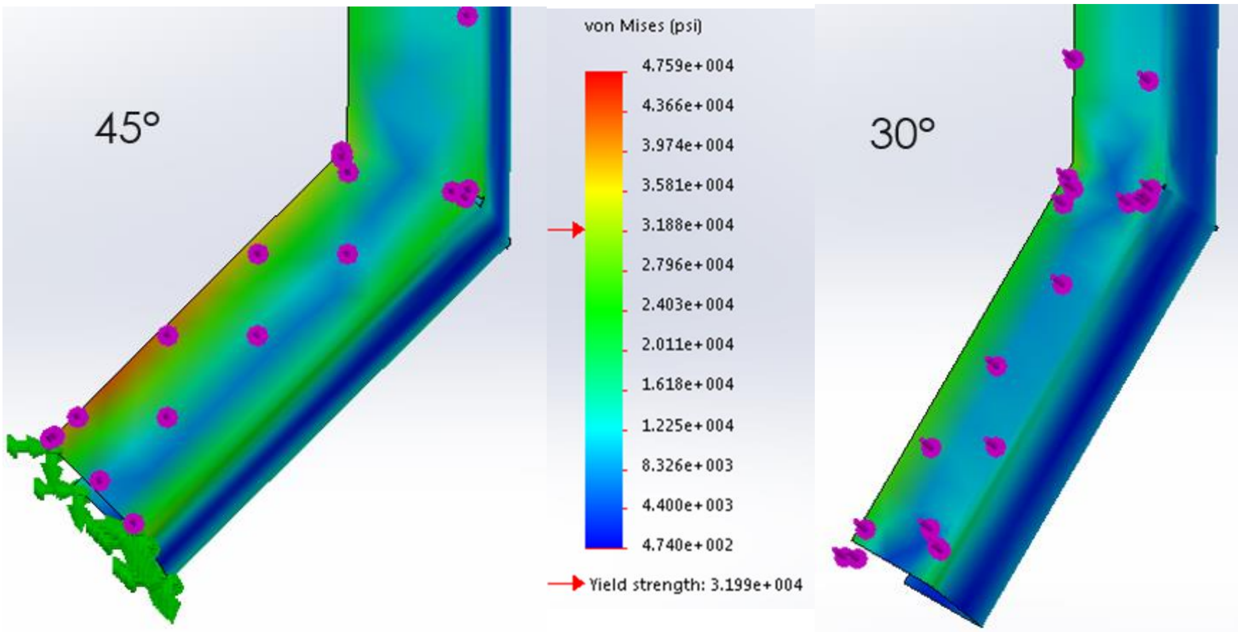


Figure 17. Comparison of stresses in 45° bends and 30° bends

The next change that was done to the rear end was to add a cross member at the 30° bends and at the end of the frame. These supports provide strength by boxing in the rear section of the frame. This stiffens that area and transfers stress to the rest of the frame rails. Boxing in the rear end meant the rear differential mount had to be redesigned. The new rear cross member no longer allows the differential to be pulled straight up out of its mount. To compensate for this the new design will bolt to the end of the frame using six 3/8” grade 8 UNC bolts. Calculations were done to determine bolt size and they showed a safety factor of over 200. Using bolts at this area will make servicing the rear end easier. Removing these six bolts will allow the entire rear axle and differential assembly to be rolled away from the frame.

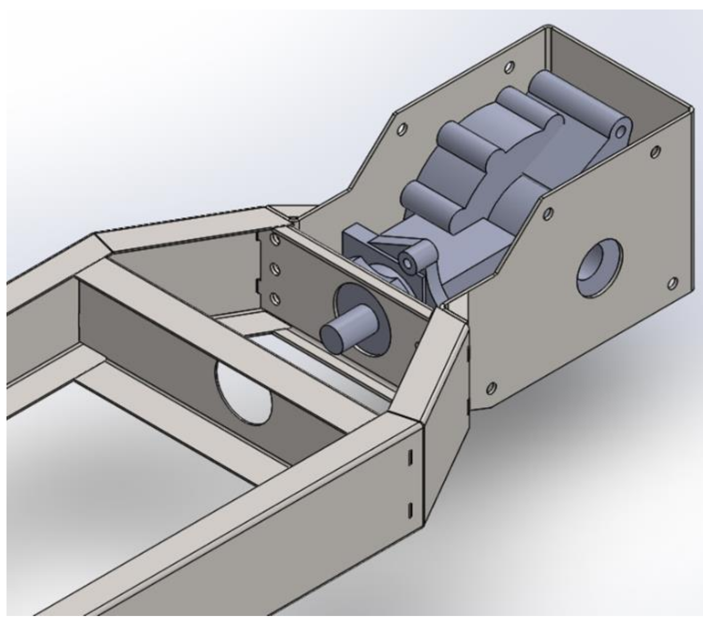


Figure 18: Supports at rear end to box in the frame

The area around the front differential has also been redesigned to improve stress distribution throughout the frame. To keep the bottom of the frame from pulling apart, cross bracing was incorporated with the A-arm mounting tabs. As shown in Figure 19, the A-arm mounts connect one frame rail to the other. The front differential mount also serves as a brace by running from one frame rail to another. The top A-arm mount will also be used as a brace to keep the top of the frame from moving closer together.

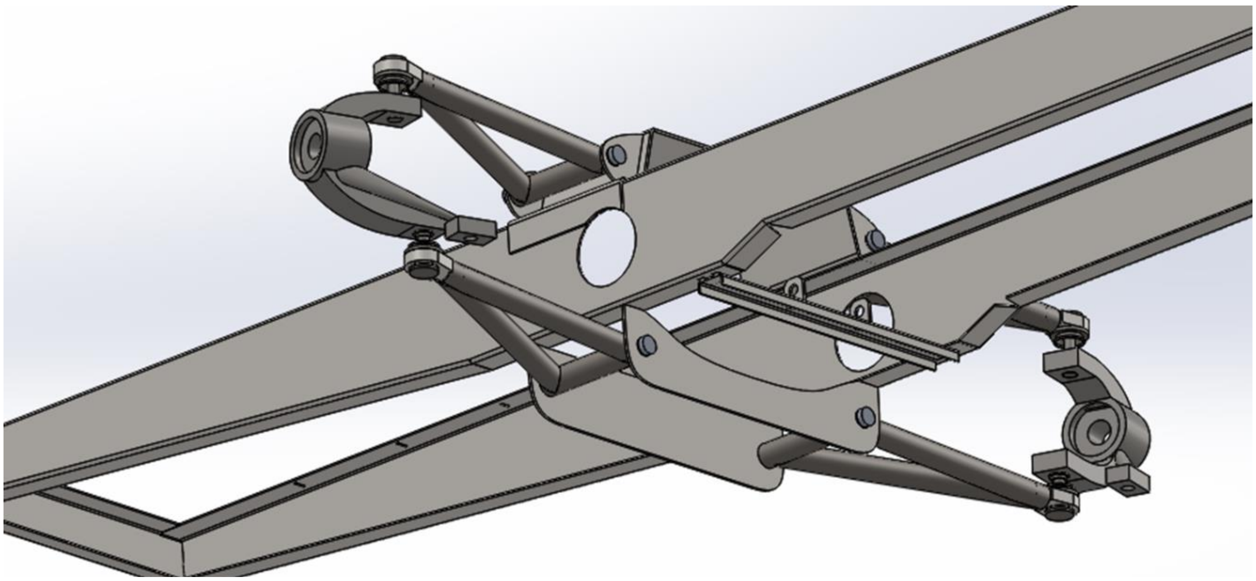


Figure 19: A-arm mounts used as cross braces

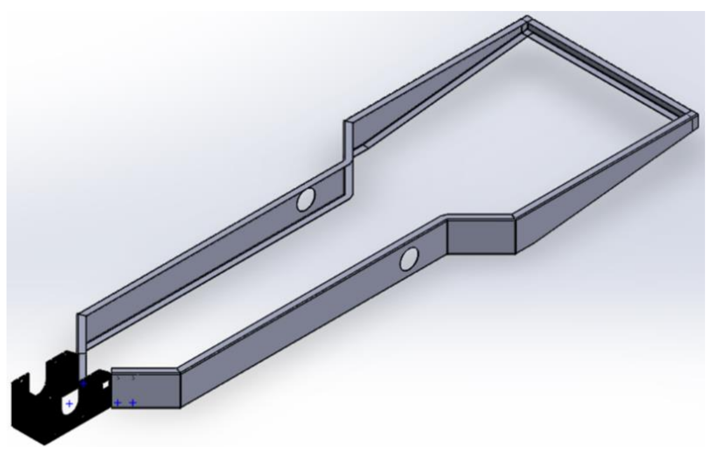


Figure 20. Wide engine frame

Initially, the shape of the frame rail went through a few iterations before a final design was decided. The first style considered was the wide engine frame, shown in Figure 20. It bent outward after the front axle to widen the frame for the engine and transmission to sit lower. The idea was to help lower the center of gravity and increase stability of the tractor. However, this design created complications for the powertrain design of the tractor. It required the differentials to move down to compensate for the engine and transmission moving, which in the end created more problems than it solved.

The next frame style considered was the Short Frame. This frame rail removed an inch of material before and after the front axle. It was designed this way to reduce weight. However, it also reduced the strength of the frame rail. Extra support structures would have been designed to compensate for the loss of strength. This design was not used because it was not compatible with the new front A-arm design.

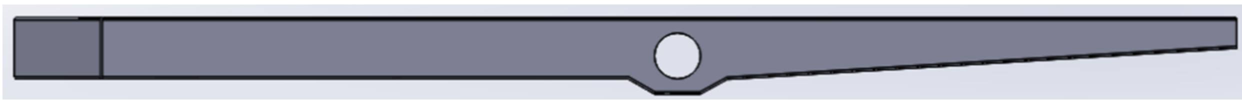


Figure 21: Short Frame

The third and final style was the spearhead frame, shown in Figure 22. In order to accommodate for the new front A-arm design, the length below the front axle was extended before and after the hole for the axle shaft. The height decreases after the front axle from 5” to 4”. It is 78.5” long, and is made of 14 gauge steel. Figure 23 shows Simulations in Solidworks that prove the spearhead frame design is strong enough to support a fully weighted tractor. Some high stress areas can be seen at the ends. This is because of the way it was fixed in Solidworks. It will not see those stress concentrations while in operation.

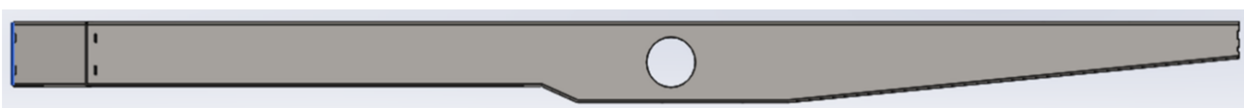


Figure 22. Spearhead frame

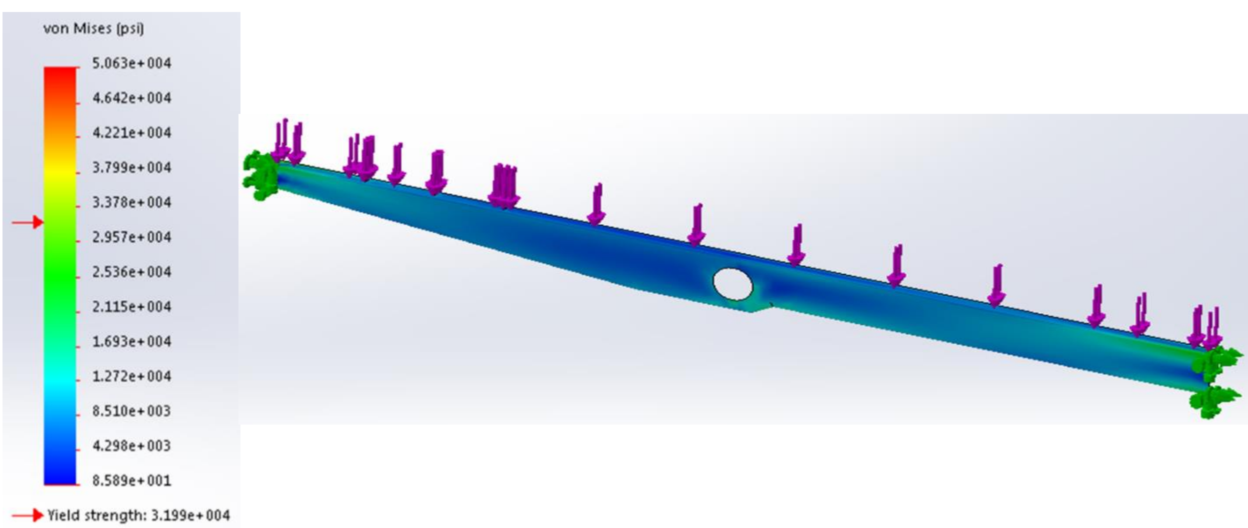


Figure 23: Spearhead frame under an 800 lb distributed load

Once the frame rail design was finalized a common assembly was made, as displayed in Figure 24, to see how all the parts fit together. The main difference between this design and the previous design is the width from one frame rail to the other was reduced from 17” to 14.5”. By reducing the frame width, the width of every cross member is reduced by 2.5”. This saves weight when the overall design is complete. Another advantage to reducing the width of the frame is that it reduced the length of the lever arm acting on the 30° bends. In this design, the bends do not have to come in as far to connect to the rear differential mount. It is 90” from the front cross member of the frame to the back of the rear differential mount.

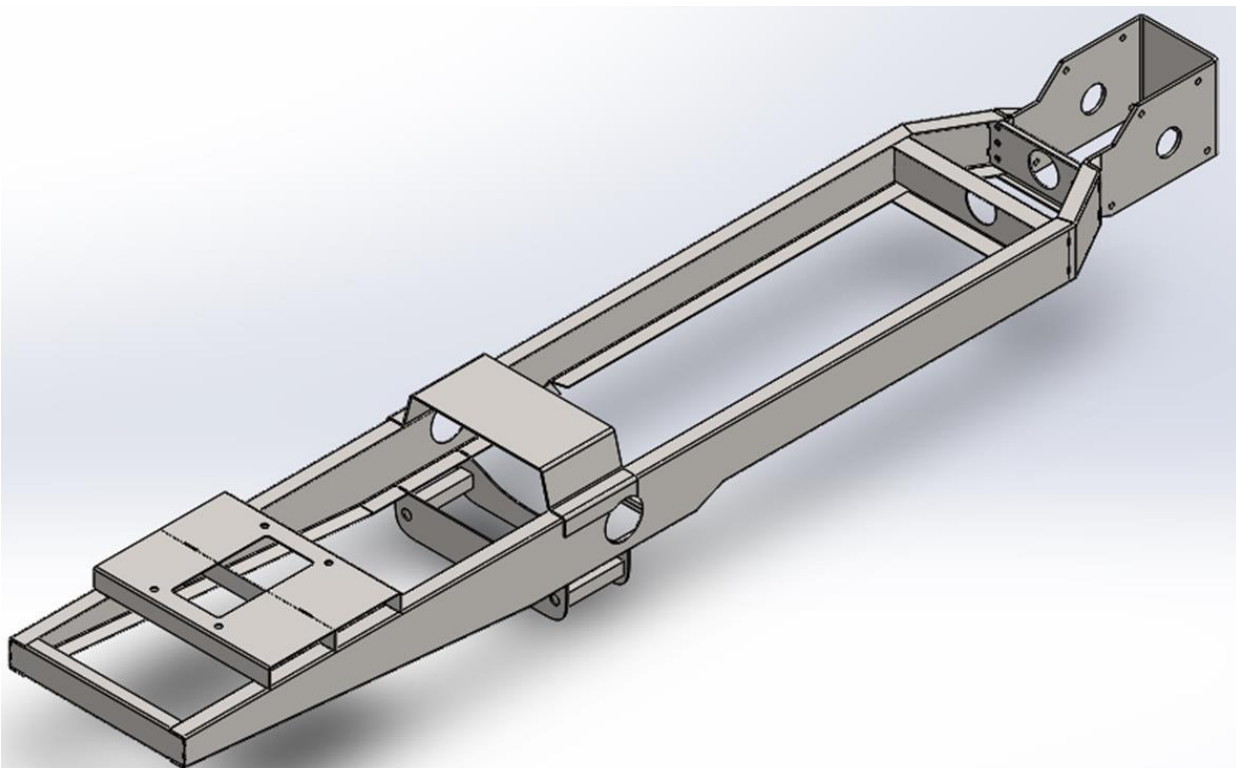


Figure 24: Overall assembly of frame and cross members

Once the main frame and cross members where in the assembly, simulations were done to see how the design would react to the forces applied to it. As shown in Figure 25, the new design easily handles the stresses presented to it.

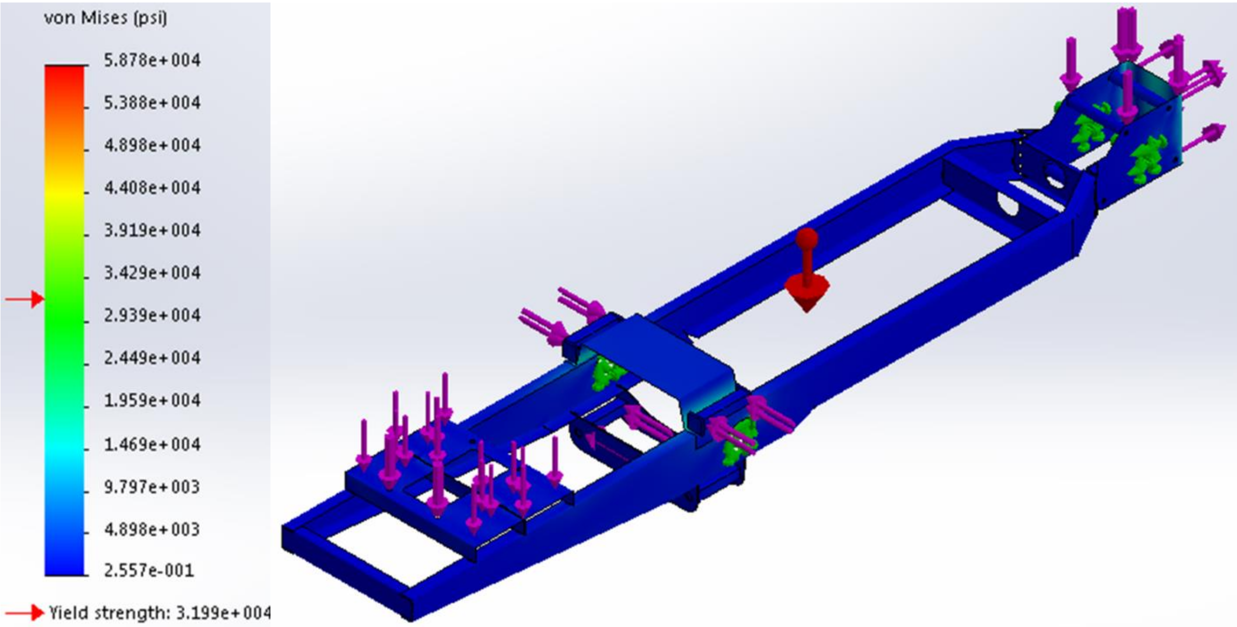


Figure 25: Simulations of forces applied during a pulling event on the overall frame assembly

Suspension

After careful consideration of the client's requirements and conditions the suspension will operate in, the air spring suspension was selected. The previous tractor was used as a means of testing the feasibility of the suspension concept in this application. The previous model was converted from a stiff suspension to using air springs. This comparison is shown in Figures 26 and 27. Placement of the air springs quickly became an area of concern. The first two iterations of air spring location brought clearance and leverage issues to the surface. Initially the spring was located near the outside of the frame rail. This allowed the weight of the tractor to use the leverage of the A-arm to gain a mechanical advantage over the air spring. When the air springs were pressurized, the force output was too small to overcome the weight of the tractor. To compensate for this problem, the air springs where relocated to the end of the A-arm. This reduced the leverage advantage and allowed the pressurized air springs to raise and lower the front of the tractor. However, the relocation created a clearance issue between the air spring and the front tire. The air spring was relocated once more and moved 1.5 inches toward the frame rail. This eliminated the clearance issue.

Figure 26. 2015-2016 solid suspension

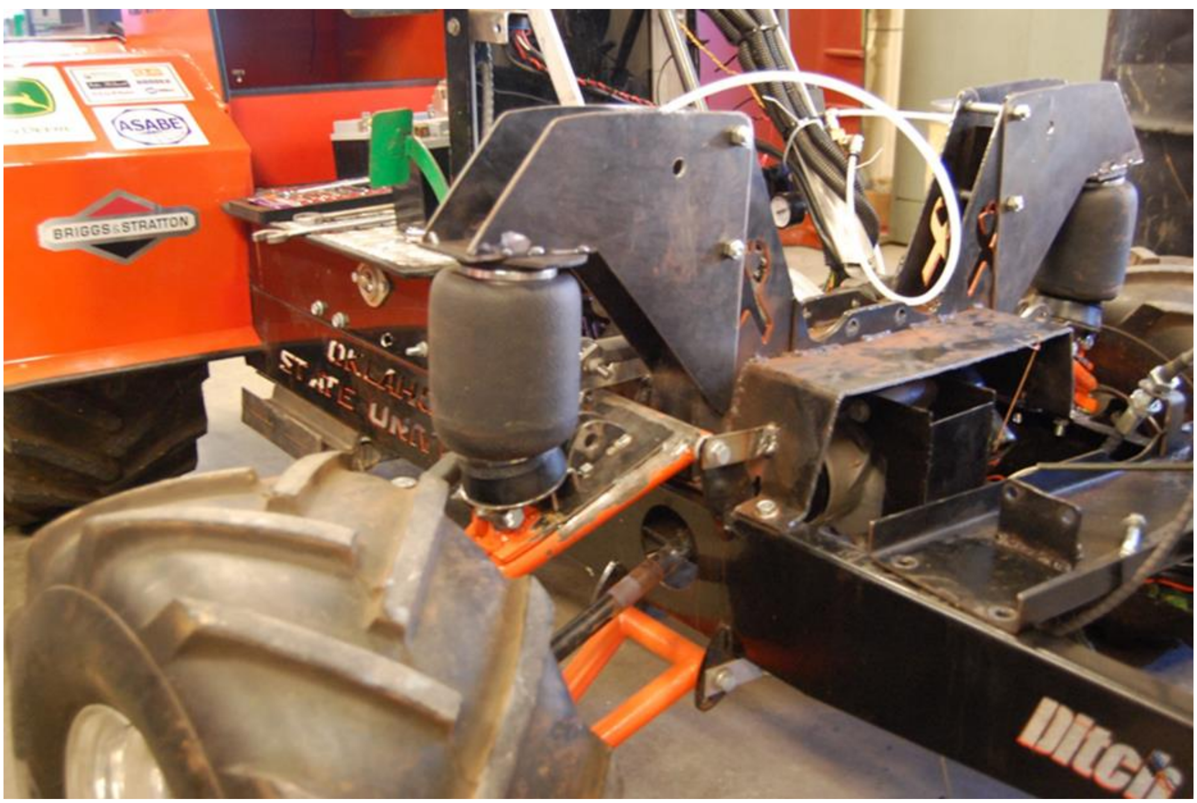


Figure 27. 2016-2017 prototype air spring suspension

Once the previous issues had been addressed, the tractor was tested on its ability to raise and lower the ride height while at 1500 pounds of total weight. The system failed to raise the ride height under this load. Using a four pad scale, the weight carried by the front tires could be monitored while removing weight until the suspension could raise the tractor. Equation 1 and 2 were developed by taking this weight, the length of the moment arm on the air spring, and the air springs location. Figure 28 models how each of the previously mentioned factors effect one another when changed.

Equation 1.

$$M_A = 0 = W \times (L + O) - (F \times M)$$

Equation 2.

$$F = \frac{W \times (L + O)}{M}$$

F = force required to lift the tractor

W = weight / front tire

R = max radius of air spring

C = clearance between air spring and ball joint

O = Length from center of tire to ball joint

L = Length of A-arm

A = pivot point on frame for A-arm

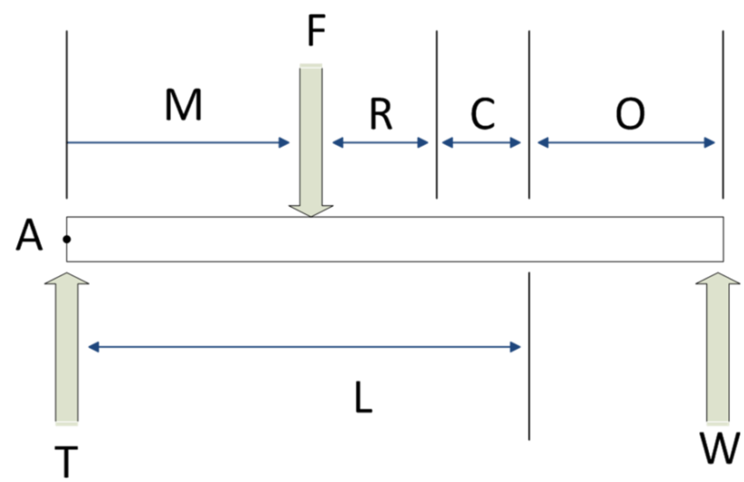


Figure 28. Diagram of the variables that effect the force needed to raise the ride height of the tractor

To test the performance of the damping action of air springs, the tractor was driven on multiple passes through a rough field. The individual front suspension had articulation and minimal bouncing. The operator reported a noticeable positive difference in ride quality with the prototype suspension. More testing on the final product will provide an accurate account of how well the suspension absorbs bumps. Further testing will cover ride quality, stability, lifting capabilities, as well as performance under pulling conditions.

Weight transfer to the rear tires was a known concern for this suspension system. To test the severity of the problem, the tractor was tested on the pulling track. Once the tractor was under load while pulling the weight of the sled, the suspension immediately rose to its maximum ride height. The test was stopped before damage to the front drive axles or air springs could occur. To compensate for this major problem, a suspension locking mechanism will be an added feature to the air spring suspension. This will allow the operator to pull with a solid suspension.

Steering

Through the process of researching the strengths and weaknesses of typical steering systems, the rack and pinion was decided as the best choice. It is a durable and reliable system. They are also readily available on the market and come in a range of sizes. Previous models have problems with heavy steering. The solution is to move the rack and pinion down in line with the steering knuckles. A 2:1 geared reduction also aids in the solution. Turning from lock to lock takes 2 turns, but with half the force of the previous design.

The tie rod ends that connect the rack and pinion to the steering knuckle are also redesigned. They are constructed out of lightweight chrome-moly steel tube with threaded ends to allow for fine tuning of the tire alignment. The previous design did not allow for a range of adjustments. Figure 29 details the preliminary design.

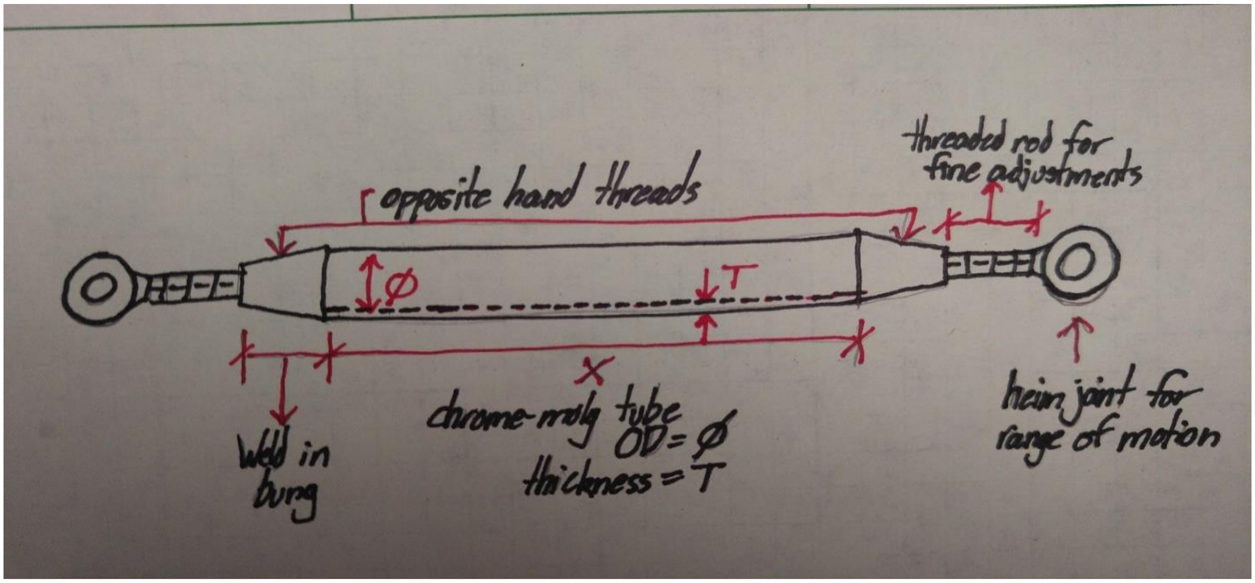


Figure 29. Adjustable tie rod end

Fabrication

Frame

Figure 30. Complete welded frame

Once the final design for the frame was completed, the drawings were sent to Ditch Witch in Perry, OK, so the parts could be cut and bent. Once the parts were done, they could easily be placed together at their correct locations without any measuring since the slot and tab method was used. This greatly sped up the fabrication process, and received praise from the fabricator because of its simplicity and ease. A total of 11 pieces were initially given to the shop workers in the BAE Laboratory to be welded together to form the first stage of the solid frame. It was finished a couple of days later without any problems. New parts were welded onto the frame when they were received from Ditch Witch. When the final stage of the solid frame was completed, it consisted of 27 individual pieces welded together into one, as shown in Figure 30.

Suspension

The suspension design was finalized and fabrication on all the suspension parts began. As stated in the recommendation section, the bottom structure was incorporated into the design of the frame for support. These pieces were welded together with the frame when it was fabricated. The remaining suspension structure was incorporated into a single removable structure (Figure 31). This structure is removable to allow access to the front differential for servicing or replacement. Slots and tabs were incorporated into the design allowing for quick and accurate fabrication of all sheet metal suspension parts.

Figure 31. Test fit of the removable suspension structure

The a-arms are made from 4130 chrome moly tubing. This is a strong and light weight material. The BAE lab used an end mill to cut the necessary angles for the joints.

Steering

After the design for the steering system was completed, the parts needed were ordered. These pieces were mocked up on the tractor before final assembly to ensure fitment. Once the parts were correctly assembled, the steering shafts were welded together and installed.

Testing

Frame

Initial torsion testing of the complete welded frame, which consisted of the rear end of the frame being clamped down while the front end was twisted back and forth by a team member, showed that it is much stiffer and deflected less than the previous year's frame when subjected to the same testing. No numerical values were recorded, the test was strictly visual. More testing and observations will be made once the entire tractor has been completed. The design will be a success if the frame is able to stay together without any failures or deformities while being under the forces and stresses seen at the various events of the competition.

Suspension

Testing the range of suspension movement required assembly of the a-arms, air springs, hubs, and the supporting structures (Figure 32). Once the range of movement was fine-tuned, plumbing the pneumatic control system began. During this process, a few minor adjustments were made, but the overall design concept discussed previously was successful.


Figure 32. Suspension travel and alignment testing

Next, the design of the pneumatic and electrical systems that control the air springs needed to be tested. Full diagrams of these systems can be found in Appendix A and B. The valves, pump, air springs, and all the fittings were assembled and installed on the tractor (Figure 33). Each switch, pressure relief, and safety interlock were tested to insure the system preformed the way it was designed.

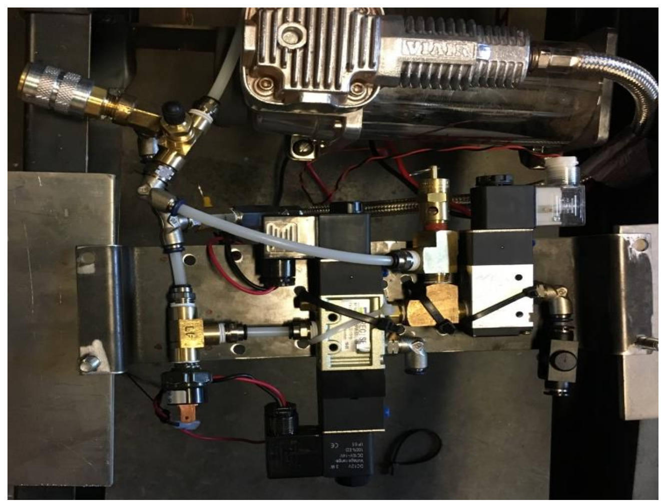


Figure 33. Pneumatic control system

The last step in testing the suspension was to fully ballast the tractor and cycle the suspension to ensure its functionality under a full load. The weight of the tractor was measured using vehicle scales (Figure 34). After they were zeroed, they were slid under each tire and weight was added until reaching 1500 lbs. The system was then raised and lowered five times to prove the design was successful.


Figure 34. Scales used to fully load the tractor

Steering

Before testing could be performed on the steering system, the tie rod ends where adjusted to match the length of the a-arms. This allows the a-arms and tie rods to be parallel (Figure 35) and reduces the change in toe alignment throughout the suspension cycle.

Figure 35. Parallel a-arm and steering tie rod

Once the toe in and out was set correctly, the Ackerman steering geometry needed to be set. As you can see in Figure 36, the inside (right) tire has a slightly larger turning angle then the outside (left) tire. This allows the tractor to turn using concentric radii, thus reducing tire scrub and provides easier turning.

The steering system has been tested with 1,000 pound and 1,500 pound tractor weight. A significant reduction in steering effort and tighter turning radius show that the improved geometry and relocation of the steering system was successful.

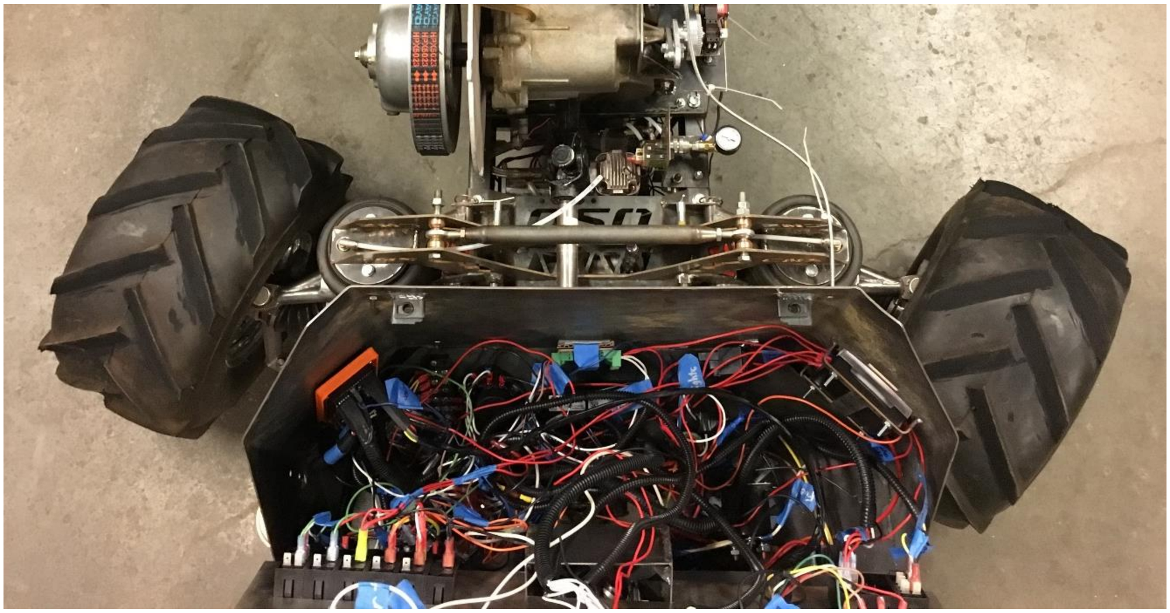


Figure 36. Ackerman steering geometry

Failure Mode and Effects Analysis (FMEA)

In order to further analyze the design created by the team, an FMEA was completed. For the design to be satisfactory, the RPN (risk priority number) must be below 99, a number regulated by the 2017 international quarter scale tractor competition rules. After analysis of the frame, steering, and suspension was completed, the designs were concluded to be satisfactory. The highest number seen was on the frame due to the potential for unforeseen impacts or conditions. To ensure operator safety, the team designed the components in a manner that injury and further system damage would be limited in the case of a failure. The failure would only result in loss of user comfort or primary function to reduce the risk of further damage. The FMEA can be seen in figure 37.

Item	Potential Failure Mode	Potential Effect of Failure	Severity	Potential Cause	Occurrence	Design Controls	Detectability	RPN
Suspension	air bag failure	rupture of air bags	7	over pressurized system	2	built in system relief	1	14
Suspension	air bag failure	puncture of air bags	7	foreign material in suspension	2	stock component	2	28
Suspension	electrical failure	air compressor failure	6	electrical system failure	3	appropriately sized wire and connections	1	18
Steering	steering column failure	bound steering	8	bound steering reducer/u-joint	2	appropriate clearance within system	2	32
Steering	tie rod/rack failure	tire rubbing	4	improperly tuned rack and tie rods	3	minmal/no adjustments required to stock components	1	12
Frame/Chassis	unpredictable forces/conditions	frame cracking	9	external force/trauma to frame	3	relief cuts and minimization of stress concentrations	2	54
Frame/Chassis	interal support failure	frame warping	9	external force/trauma to frame	2	multiple connection points and redundancies	2	36

Figure 37. FMEA of suspension, steering, and frame.

Freshman Involvement

Two teams of five freshman were assigned to help with this project. They are required to help with some small, but significant portion of the larger senior design project. One group, Micah Arthaud, Shyanna Hansen, Michael Leiterman, Nick Liegerot, and Heath Moorman, were tasked with developing a new rear differential mount. Figure 38 shows what the group designed. The second group, Jeremiah Foster, Brent Gwinn, Creston Moore, Austin Pickering, and Ross Ruark, were tasked with developing a new transmission mount. Figure 39 shows what the group designed.

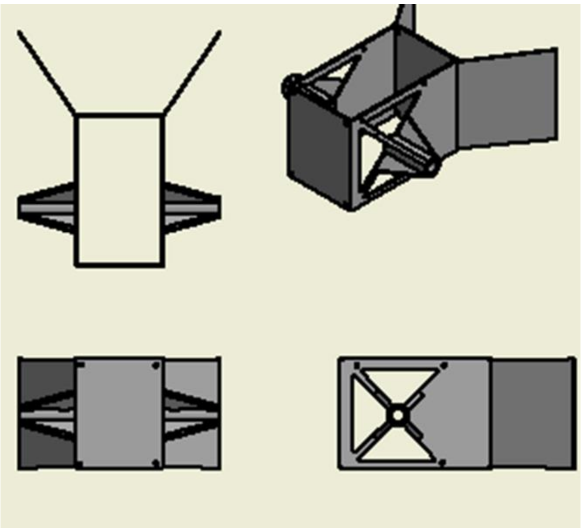


Figure 38. Freshman rear differential mount

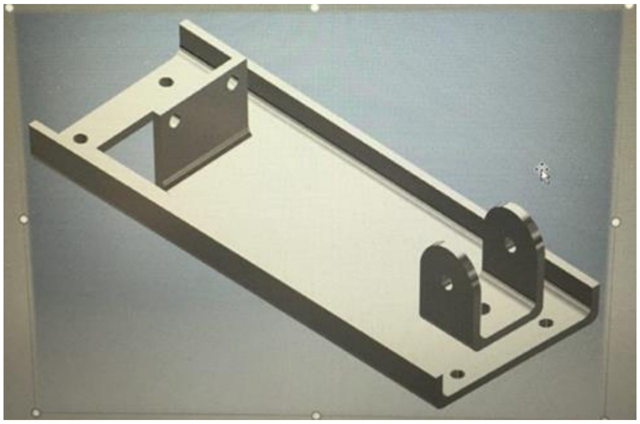


Figure 39. Freshman transmission mount

Budget

Cost Component	Price	Percent of Total Cost	Sub-Assembly	Price	Percent of Total Cost
Material cost	$ 87.45	3.55%	Suspension cost	$ 1,472.52	59.76%
Fabrication cost	$ 317.67	12.89%	Steering cost	$ 757.00	30.72%
Labor cost	$ 405.00	16.44%	Frame cost	$ 234.60	9.52%
Purchased parts	$ 1,654.00	67.12%	Total	$ 2,464.12	100.00%

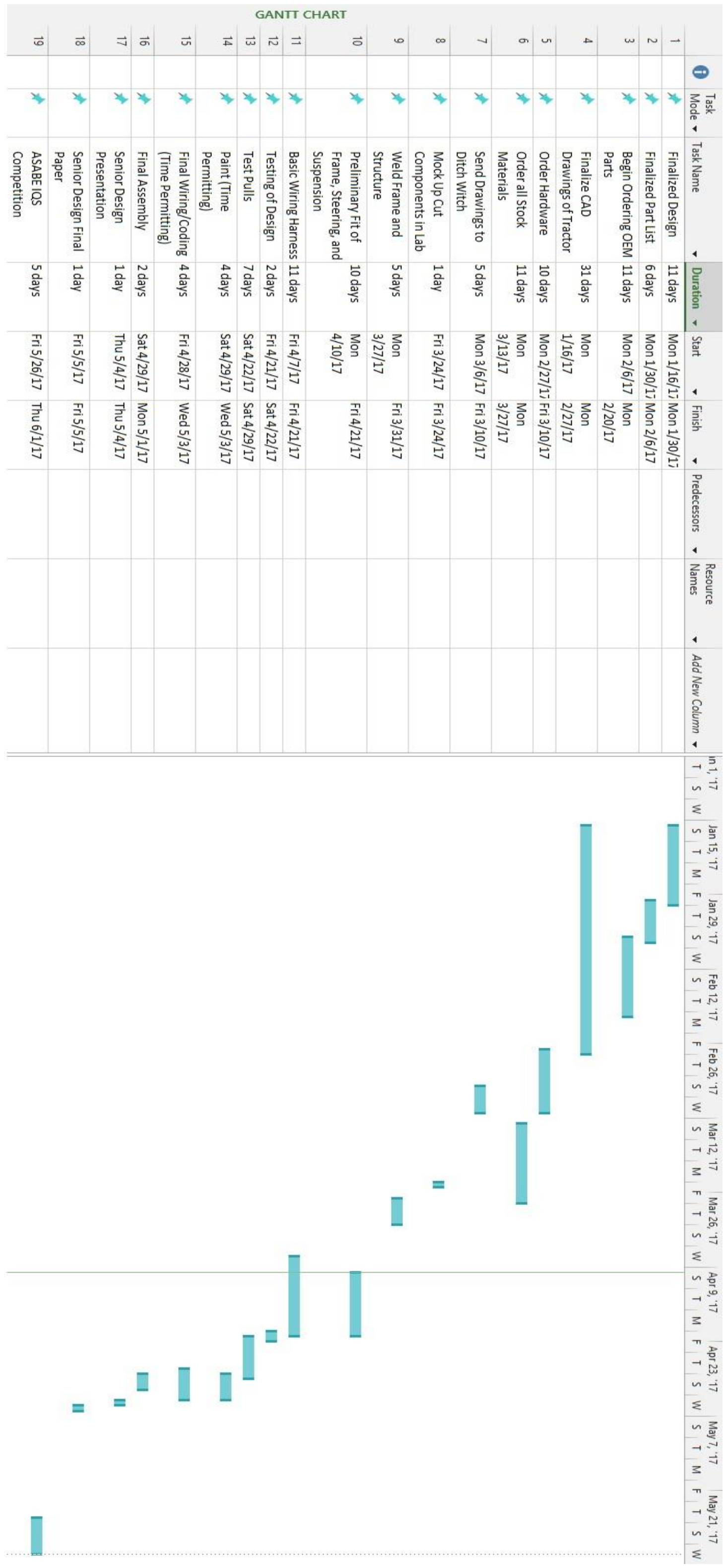
Aftermarket Parts	Cost Per Unit	#	Total Cost	Part number	Location
14" Rack and Pinion	$ 103.95	1	$ 103.95		dans performance parts
Howe Steering Adapters	$ 11.97	2	$ 23.94	5221	summitracing.com
Howe Steering Reducer	$ 86.99	1	$ 86.99	5224	jegs.com
Heim and Rod Kit (3/8 x 3/8-24 panhard w/ 0.058 bung) ea.	$ 19.20	2	$ 38.40		ebay-QS Components Inc.
Heim and rod kit (7/16 x 7/16-20 Panhard Bar Kit 7/16 Steel Cone Spacers .065 Bungs) each	$ 26.25	1	$ 26.25		ebay-QS Components Inc.
Hardware (cost estimate)	$ 50.00	1	$ 50.00		Fastenal
Air Springs	$ 163.42	2	$ 326.84	50252	Air lift
Air Compressor	$ 269.55	1	$ 269.55	16380	Air lift
Pneumatic Hoses and Fittings (estimate)	$ 150.00	1	$ 150.00		McMaster Carr
1/4 in Npt 3 way 2 position Pneumatic electric solenoid Valve DC	$ 19.95	2	$ 39.90		Amazon-U.S. Solid
Steering Wheel	$ 44.99	1	$ 44.99		speedwaymotors.com
Steering u-joints	$ 32.95	2	$ 65.90	425260	dans performance parts
5/8 in spline weld in shafts	$ 6.95	2	$ 13.90	425244-2	dans performance parts
90 degree gear box		1	$ 200.00		
retractable spring plunger with t-handle	$ 14.47	4	$ 57.88	31265A36	McMaster Carr
nylon fabric strip	$ 17.14	1	$ 17.14	87425k76	McMaster Carr
Pin with locking ring	$ 2.36	4	$ 9.44	90170A205	McMaster Carr
Borgeson shaft coupler	$ 19.49	1	$ 19.49	311800	summitracing.com
Knuckle	$ 54.72	2	$ 109.44		Country cat
Total			$1,654.00		

Assemblies	Total price of assembly	#	Inches of weld	Assembly of nut, bolt, and washer	Indiviual Fabricated parts
Price			0.15	0.1	
Top Plate Sutructure	$ 55.07	4	90	12	Top A-arm mount
		1			Top plate
		2			tower 2
		2			Tower part 2
		2			tower part 3
		2			bottom plate tower
Frame Structure	$ 177.63	2	200		Frame rail
		1			Front frame piece
		1			Engine Mount
		1			Engine mount brace
		1			bottom rear A-arm mount
		2			lower A-arm support 1
		1			Lower A-arm support 1M
		2			Lower A-arm support left
		2			Lower A-arm support right
		1			Rack and steering mount V2
		1			Mid shaft mount
		2			Air compressor mount
		1			Bottom front A-arm mount
		1			Front diff mount top
		2			Front diff mount top tab
		1			Mid section support
		4			side engine
		2			vert engine
		1			Rear diff connection plate
		1			90 degree gear box mount
		2			90 degree gear box mount piece 2
		1			right side gear box support
		1			left side gear box support
Transmission Mount	$ 24.18	1	20	60	Transmission mount
		2			Transmission mount tab1
		1			Transmission mount tab2
		2			vert trans
		4			side trans
Rear Differential Mount	$ 32.79	1	50	12	Rear diff mount
		1			Hitch plate
		2			Hitch bolt top and bottom
		1			Hitch back plate
		4			Rear Diff mount gusset
Lower A-arm Structures	$ 247.39	2	32		Ball joint tab
		2			A-arm mock up
		2			A-arm pin
		2			A-arm bushing
		4			A-arm pin bushing
Upper A-arm Structures	$ 261.54	2	60		Ball joint tab
		2			A-arm mock up
		2			A-arm pin
		2			A-arm bushing
		4			A-arm pin bushing
		2			Part 1 welded to a-arm
		4			side support lockout v2
Strut Brace	$ 6.71	1	6	2	Strut brace
Pnuematic Valve Cover	$ 4.82	1		4	Pnuematic valve cover
Totals	$ 810.12		458	90	

Sheet Metal Bends <3/8	Plate Bends >3/8	Plasma Cuts (in)	Labor (Hr)	Drilled Holes	14 ga steel sheet (sq in)	11 ga steel sheet (sq in)	1/4 in steel sheet (sq in)
0.05	0.1	0.1	45	0.35	0.0137081	0.0206597	0.0370903
		11				3.82	
6		80				110	
		75				58.53	
		13				5.24	
		15				8.04	
		7				2.85	
8		184			475		
2		38			54		
2		95			165		
		30			21		
		40.2				24	
		13.5				3.53	
		14.5				3.91	
2		9				4.6	
2		9				4.6	
1		32.5				33.2	
6		84.5			103		
1		18.04				5.6	
		52.5				35	
2		35.5			29		
		6			1		
2		51			85		
		12			3.88		
		8.8			4.2		
10		63				31	
2		63				58.34	
		14				10.62	
		14.5				3.75	
		14.5				4.5	
2		65				107	
		8					2
		13.5					7
		5.2			1.5		
		4.7			1		
4		101.5				178	
		26					
		15				11.63	
		14				8.06	
		3				0.602	
		10.2					
			1				
			0.5	1			
			0.25				
			0.25				
		10.2					
			1				
			0.5	1			
			0.25				
			0.25				
		21				12	
		12				5.5	
4		36			45		
69	0	2019.38	9	4	1484.92	890.928	11

1/2 in steel sheet (sq in)	1 in O.D. Chrome moly tubing .058 wall thickness (in)	1 in O.D. cold drawn steel rod (in)	Delrin 3/4 in rod (in)	Tube end prep (end)	Saw cut (in)	Total
0.0735973	0.203	0.791	0.1029	0.75	0.38	
						$ 4.72
						$ 10.57
						$ 17.42
						$ 2.82
						$ 3.33
						$ 1.52
						$ 50.62
						$ 4.64
						$ 11.86
						$ 3.29
						$ 4.52
						$ 2.85
						$ 1.53
						$ 2.19
						$ 2.19
						$ 3.99
						$ 10.16
						$ 3.94
						$ 5.97
						$ 4.05
						$ 1.23
						$ 6.37
						$ 5.01
						$ 1.88
						$ 7.44
						$ 7.61
						$ 3.24
						$ 1.53
						$ 1.54
						$ 8.81
						$ 1.75
						$ 1.61
						$ 1.08
						$ 1.93
						$ 14.03
15.78						$ 3.76
						$ 3.48
						$ 1.57
						$ 1.25
2.04						$ 2.34
	27			6	6	$ 114.52
		6			1	$ 55.95
			5		1.5	$ 24.67
			0.25			$ 45.10
2.04						$ 2.34
	27			6	6	$ 114.52
		6			1	$ 55.95
			5		1.5	$ 24.67
			0.25			$ 45.10
						$ 4.70
						$ 5.25
	16.5			2	2	$ 5.61
						$ 4.42
23.94	124.5	24	22	26	36	

Schedule



GANTT CHART

	Task Mode	Task Name	Duration	Start	Finish	Predecessors	Resource Names	Add New Column
1		Finalized Design	11 days	Mon 1/16/17	Mon 1/30/17			
2		Finalized Part List	6 days	Mon 1/30/17	Mon 2/6/17			
3		Begin Ordering OEM Parts	11 days	Mon 2/6/17	Mon 2/20/17			
4		Finalize CAD Drawings of Tractor	31 days	Mon 1/16/17	Mon 2/27/17			
5		Order Hardware	10 days	Mon 2/27/17	Fri 3/10/17			
6		Order all Stock Materials	11 days	Mon 3/13/17	Mon 3/27/17			
7		Send Drawings to Ditch Witch	5 days	Mon 3/6/17	Fri 3/10/17			
8		Mock Up Cut Components in Lab	1 day	Fri 3/24/17	Fri 3/24/17			
9		Weld Frame and Structure	5 days	Mon 3/27/17	Fri 3/31/17			
10		Preliminary Fit of Frame, Steering, and Suspension	10 days	Mon 4/10/17	Fri 4/21/17			
11		Basic Wiring Harness	11 days	Fri 4/7/17	Fri 4/21/17			
12		Testing of Design	2 days	Fri 4/21/17	Sat 4/22/17			
13		Test Pulls	7 days	Sat 4/22/17	Sat 4/29/17			
14		Paint (Time Permitting)	4 days	Sat 4/29/17	Wed 5/3/17			
15		Final Wiring/Coding (Time Permitting)	4 days	Fri 4/28/17	Wed 5/3/17			
16		Final Assembly	2 days	Sat 4/29/17	Mon 5/1/17			
17		Senior Design Presentation	1 day	Thu 5/4/17	Thu 5/4/17			
18		Senior Design Final Paper	1 day	Fri 5/5/17	Fri 5/5/17			
19		ASABE IQS Competition	5 days	Fri 5/26/17	Thu 6/1/17			

Figure 40. Past and future schedualed meetings and deadlines

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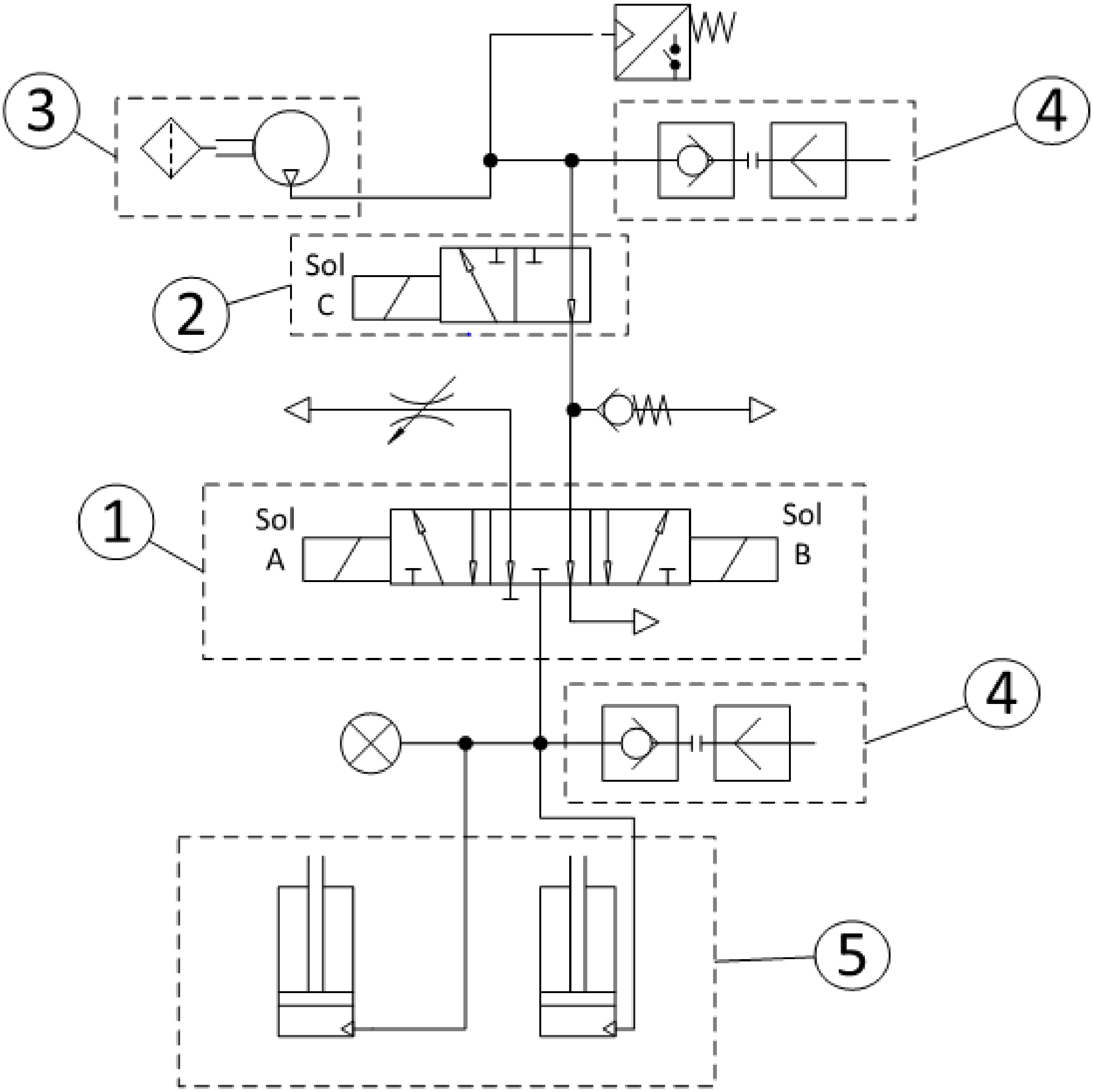
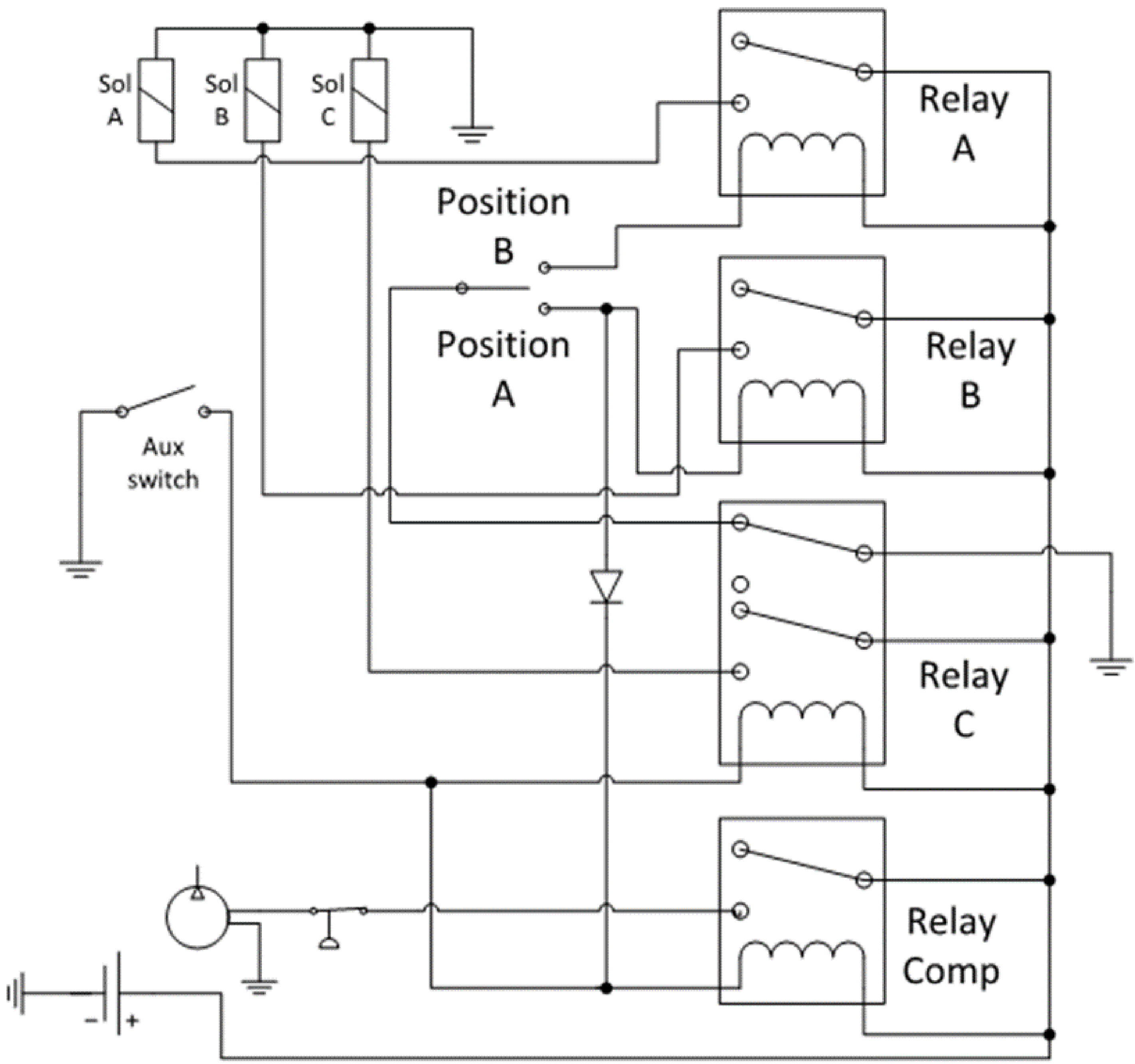
Appendix A

Suspension pneumatic diagram

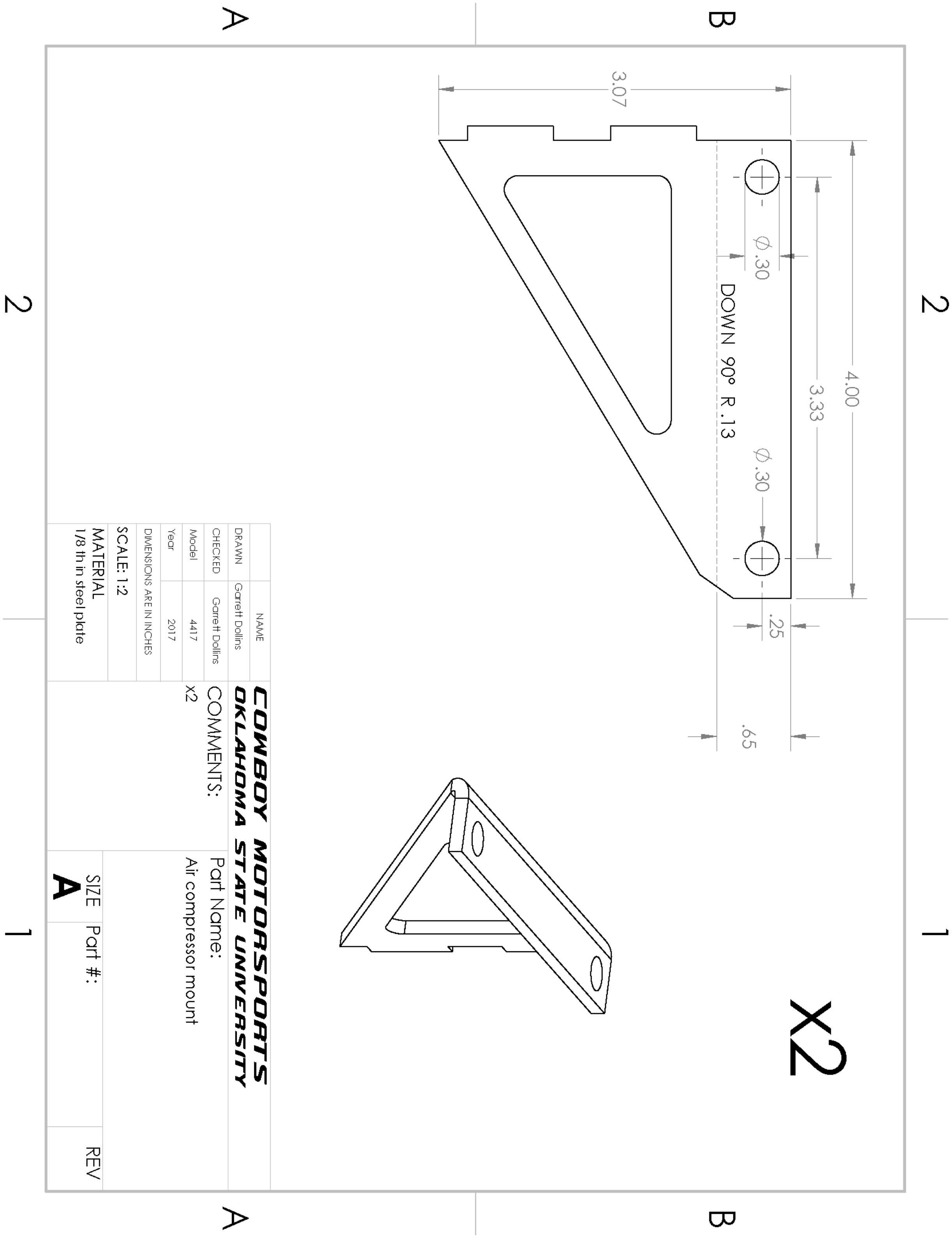


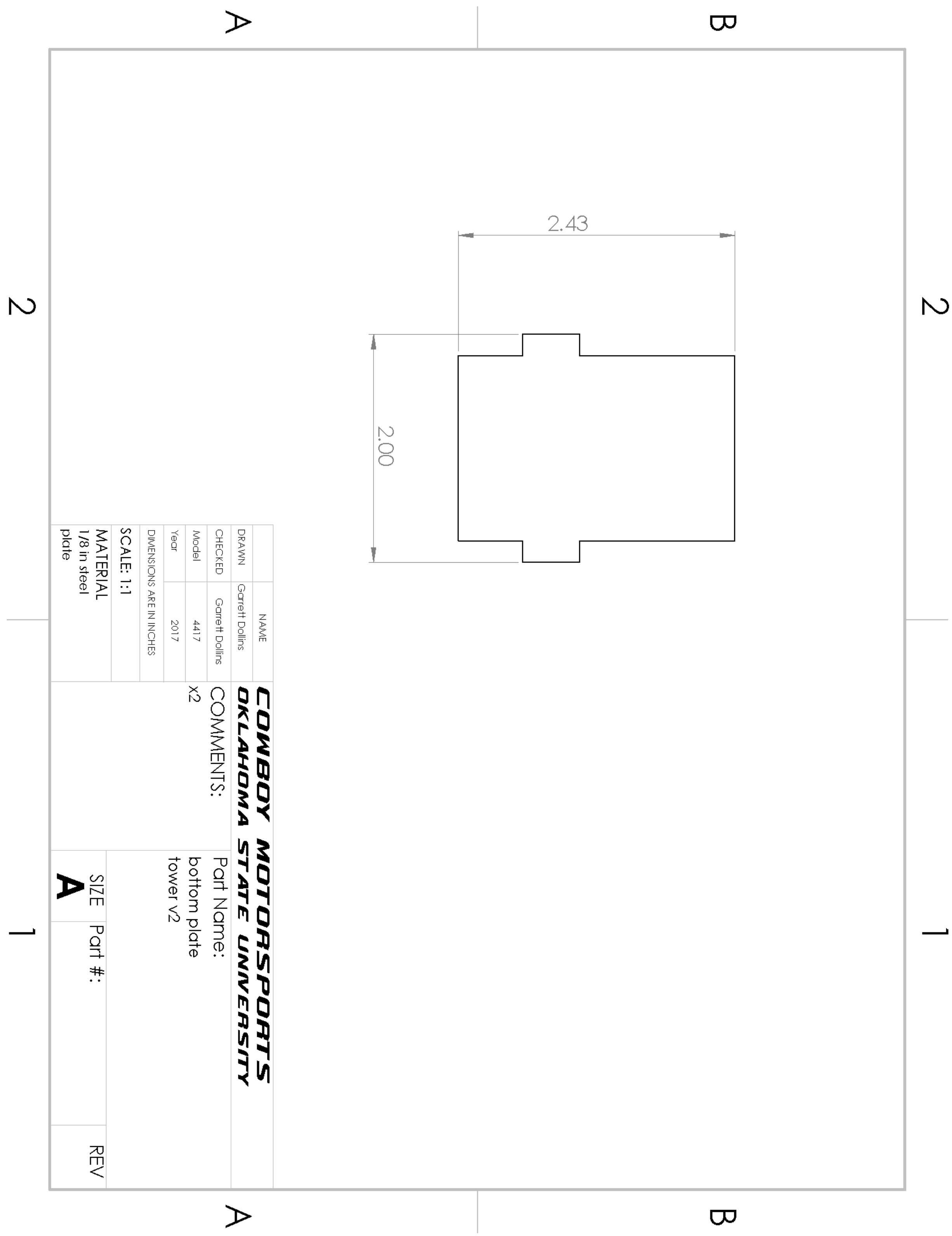
Appendix B

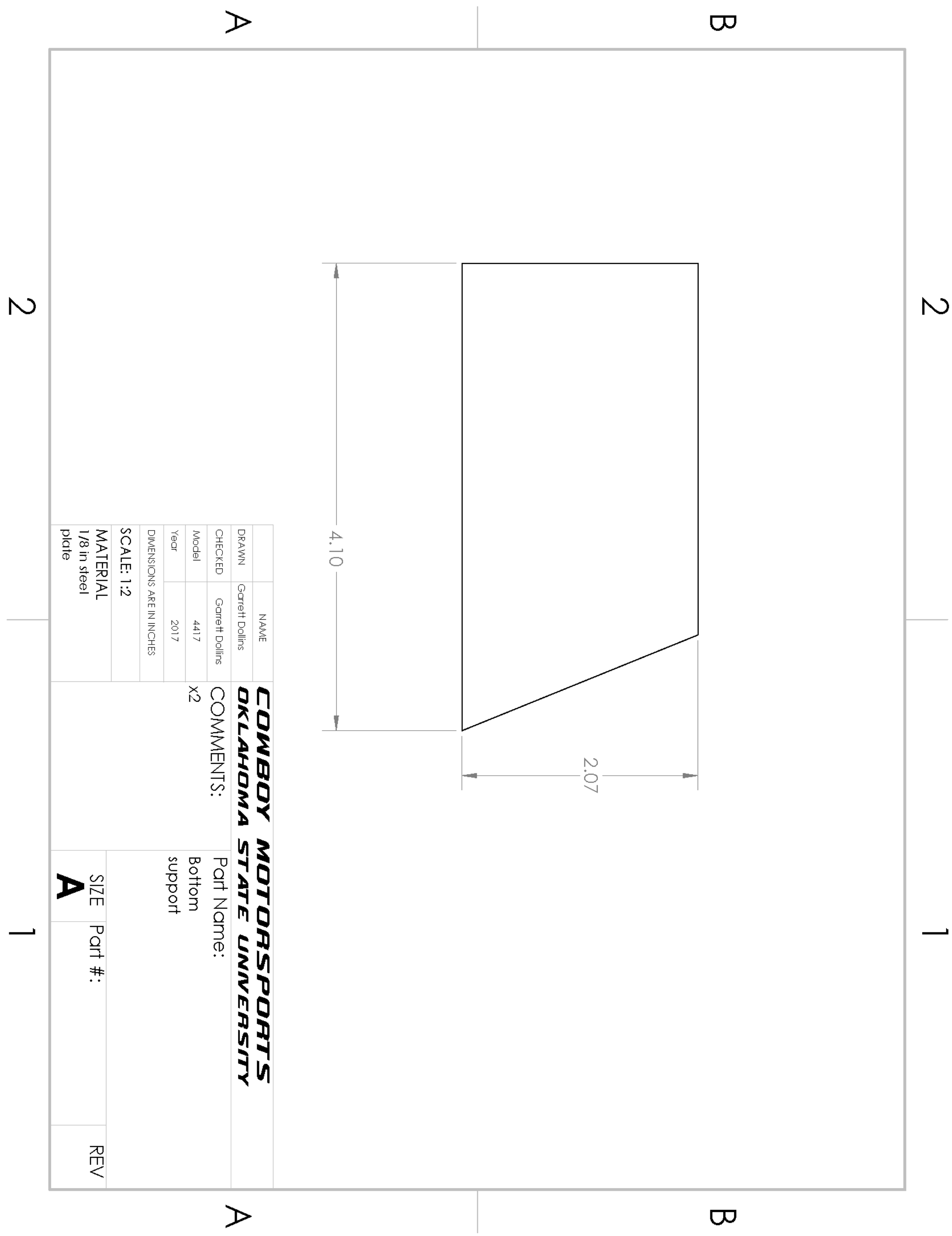
Suspension Circuit Diagram



Fabricated Parts




2.43
2.00
NAME
DRAWN Garrett Dollins
CHECKED Garrett Dollins
Model 4417
Year 2017
DIMENSIONS ARE IN INCHES
SCALE: 1:1
MATERIAL
1/8 in steel plate
COWBOY MOTORSPORTS
OKLAHOMA STATE UNIVERSITY
COMMENTS:
x2
Part Name:
bottom plate tower v2
SIZE
A
Part #:
REV
A
B
1
2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2.07
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OKLAHOMA STATE UNIVERSITY
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CHECKED Garrett Dollins
Model 417
Year 2017
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COMMENTS:
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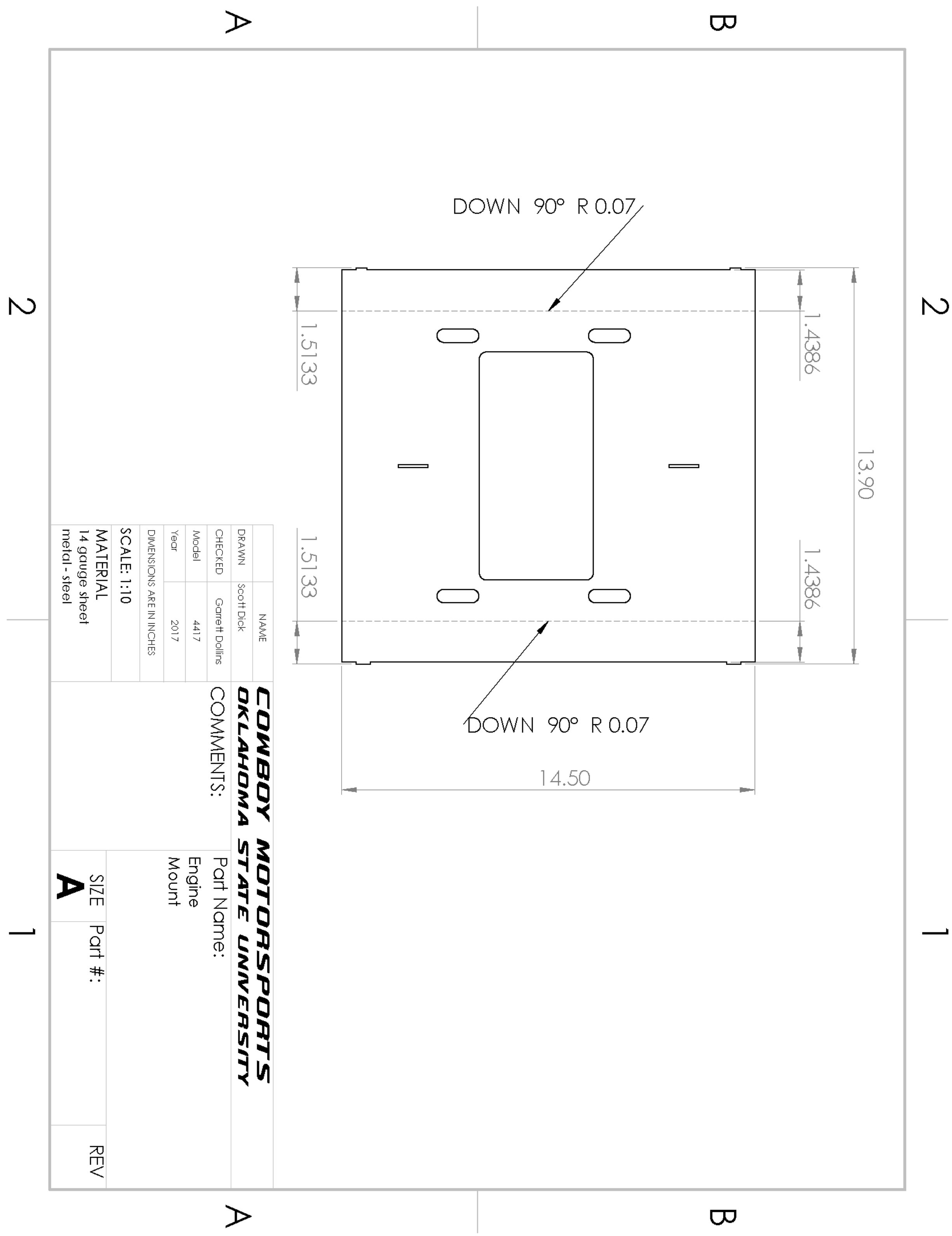
Part Name:
Bottom support
SIZE
A
Part #:
REV
A
B
1
2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DOWN 90° R 0.07
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1.5133
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13.90
14.50
COWBOY MOTORSPORTS
OKLAHOMA STATE UNIVERSITY
NAME
DRAWN Scott Dick
CHECKED Garrett Dollins
Model 417
Year 2017
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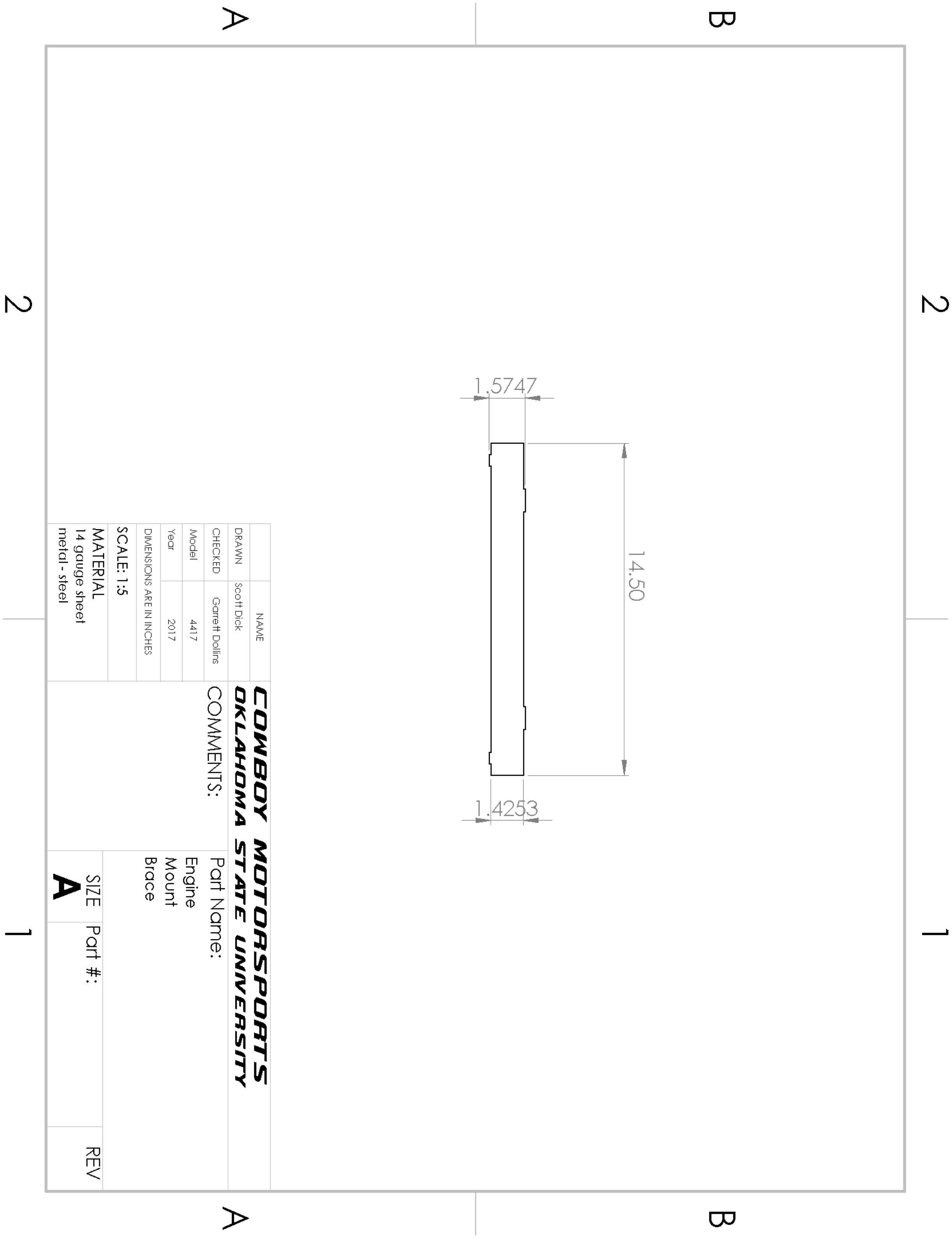
Engine Mount
SIZE
A
Part #:
REV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14.50
1.4253
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CHECKED Garrett Dollins
Model 417
Year 2017
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MATERIAL
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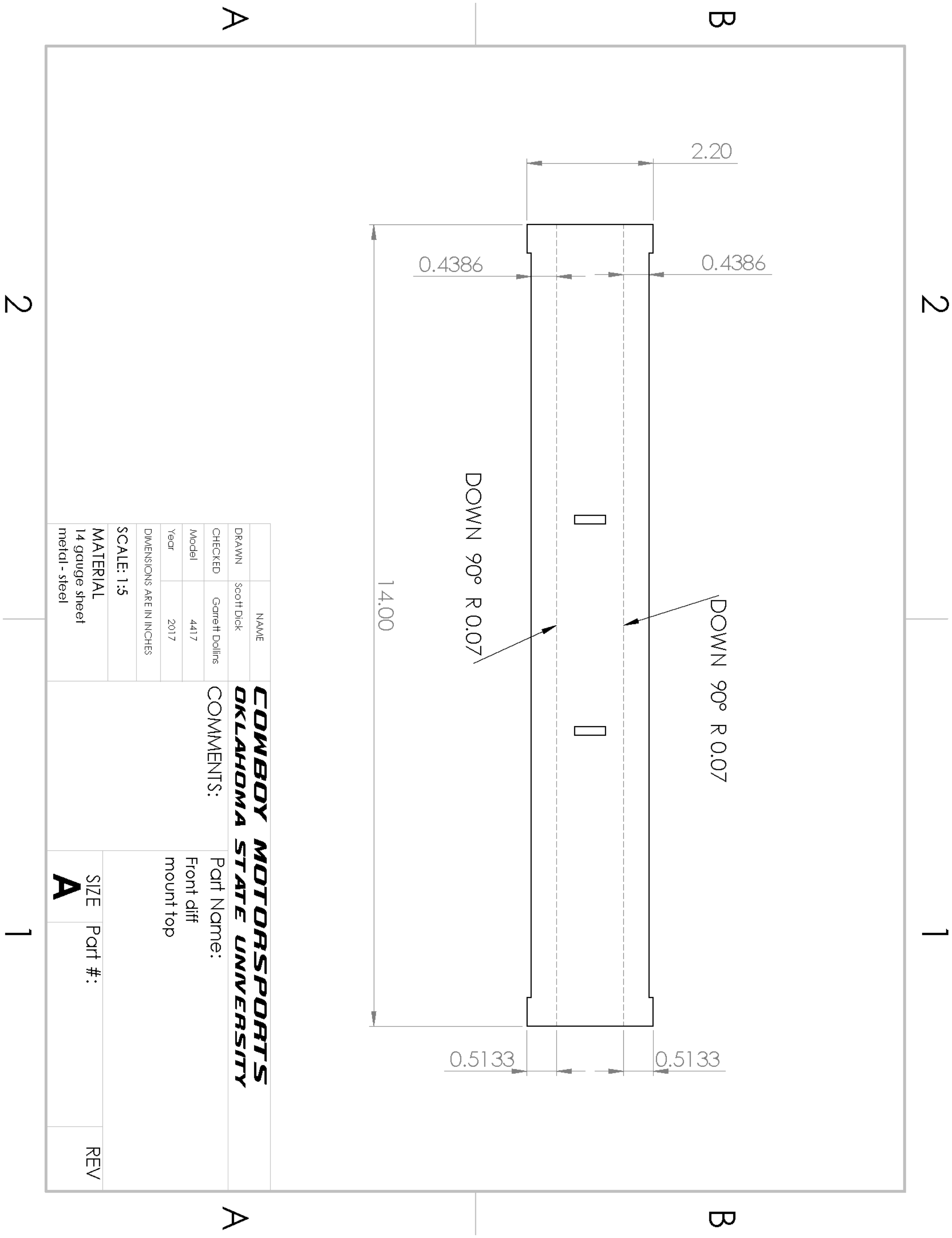
COWBOY MOTORSPORTS
OKLAHOMA STATE UNIVERSITY
COMMENTS:
Part Name:
Engine Mount Brace
SIZE
A
Part #:
REV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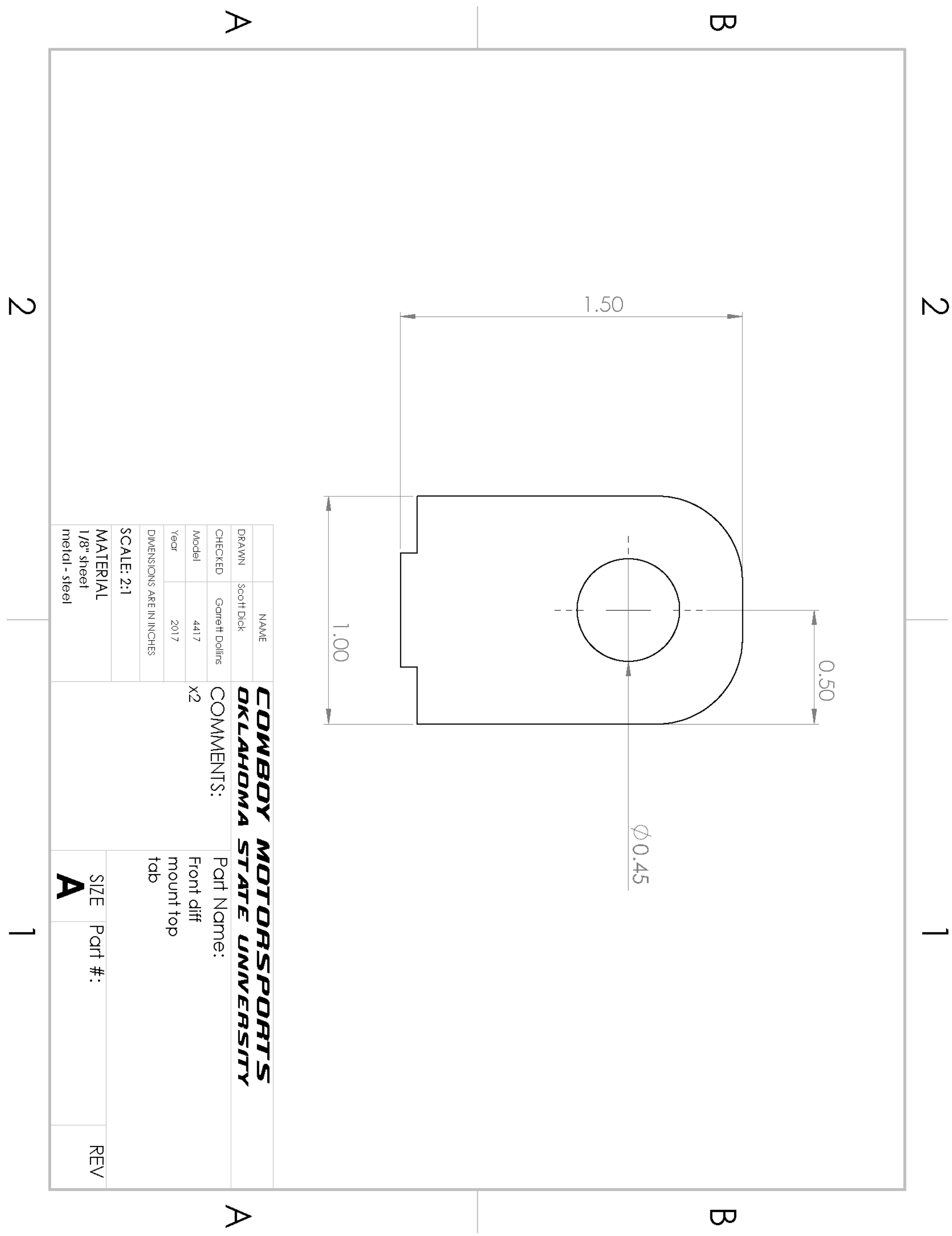
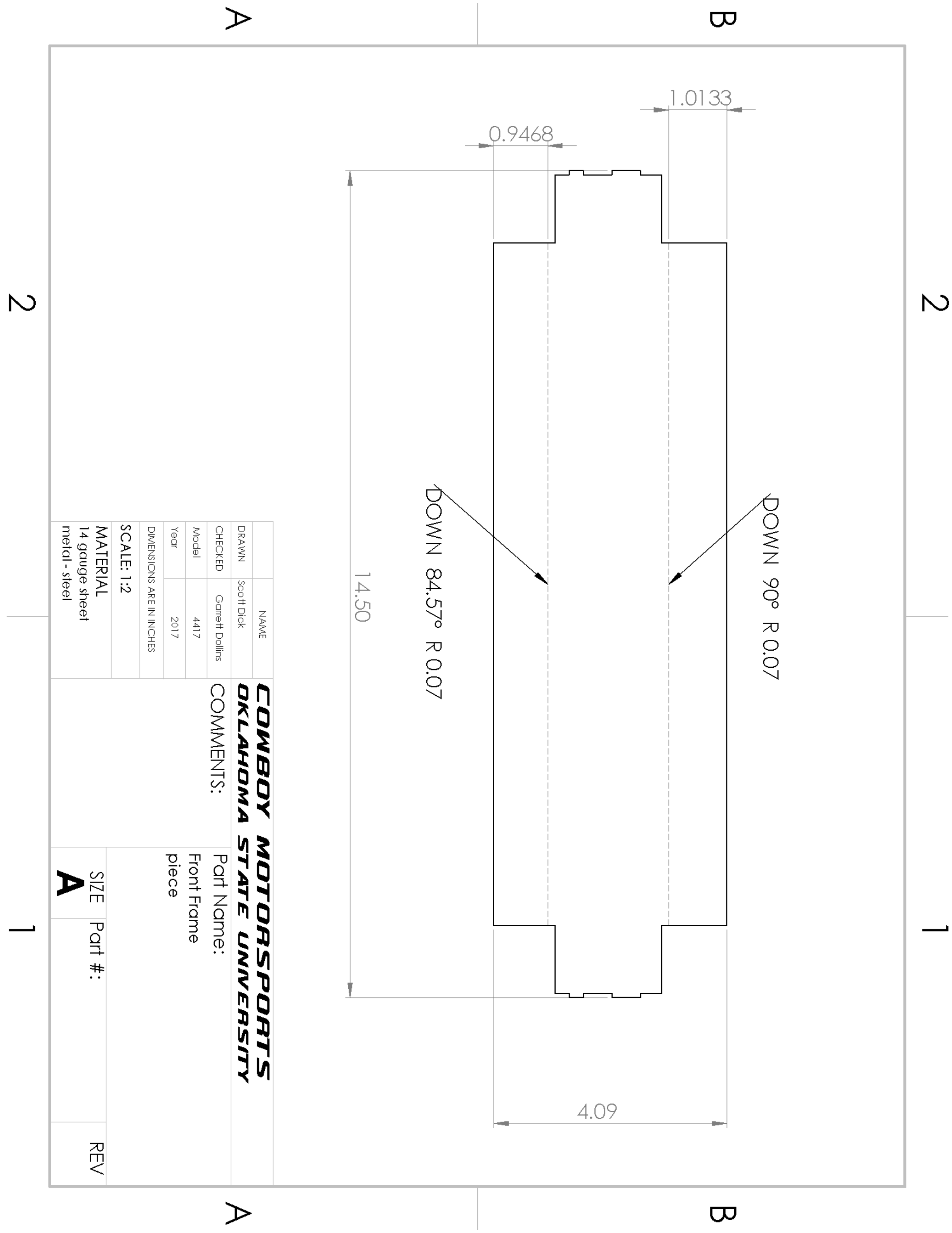
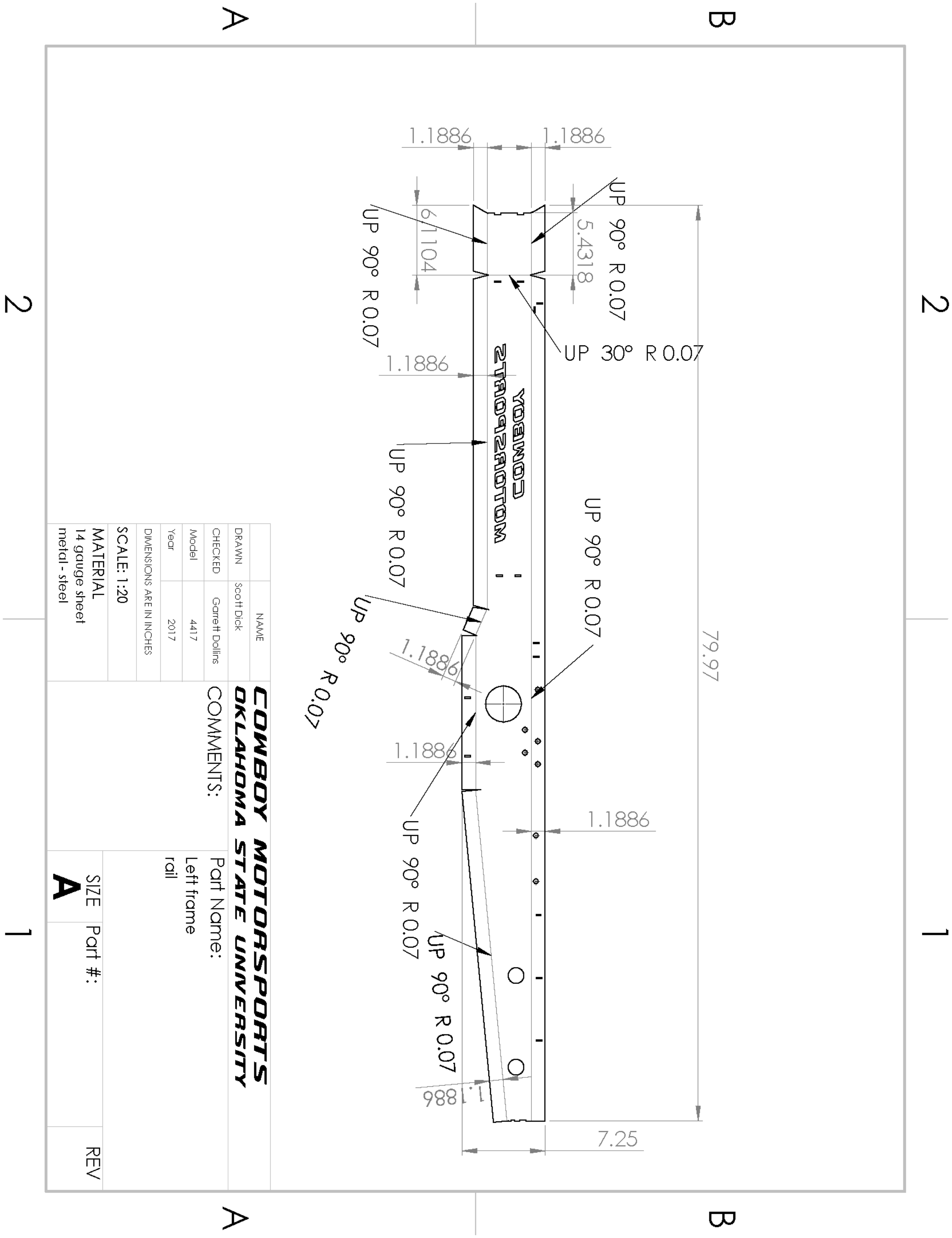
14.00
DOWN 90° R 0.07
DOWN 90° R 0.07
0.5133
0.5133
NAME
DRAWN Scott Dick
CHECKED Garrett Dollins
Model 4417
Year 2017
DIMENSIONS ARE IN INCHES
SCALE: 1:5
MATERIAL
14 gauge sheet metal - steel
COWBOY MOTORSPORTS
OKLAHOMA STATE UNIVERSITY
COMMENTS:
Part Name:
Front diff mount top
SIZE
A
Part #:
REV
A
B
2
1

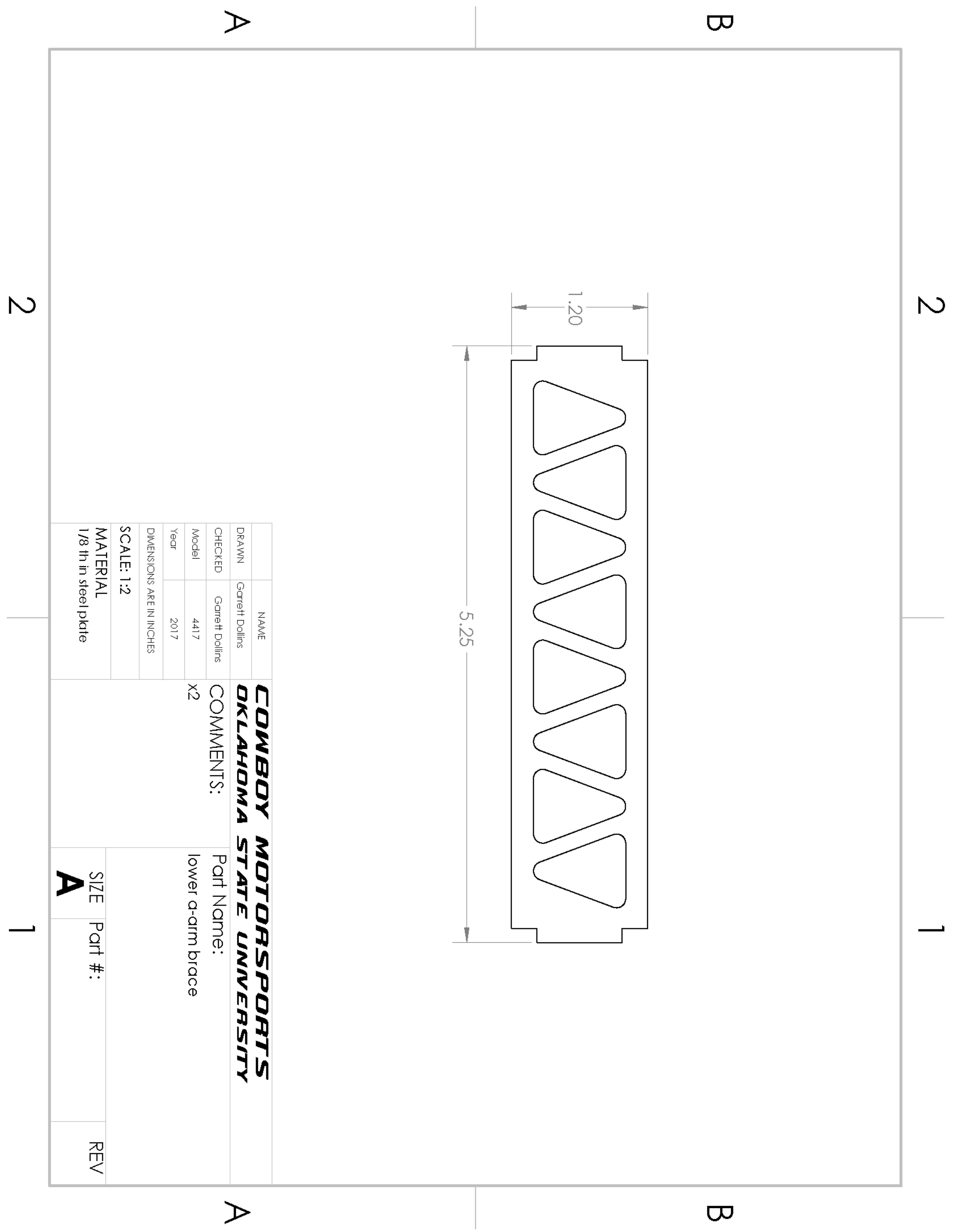


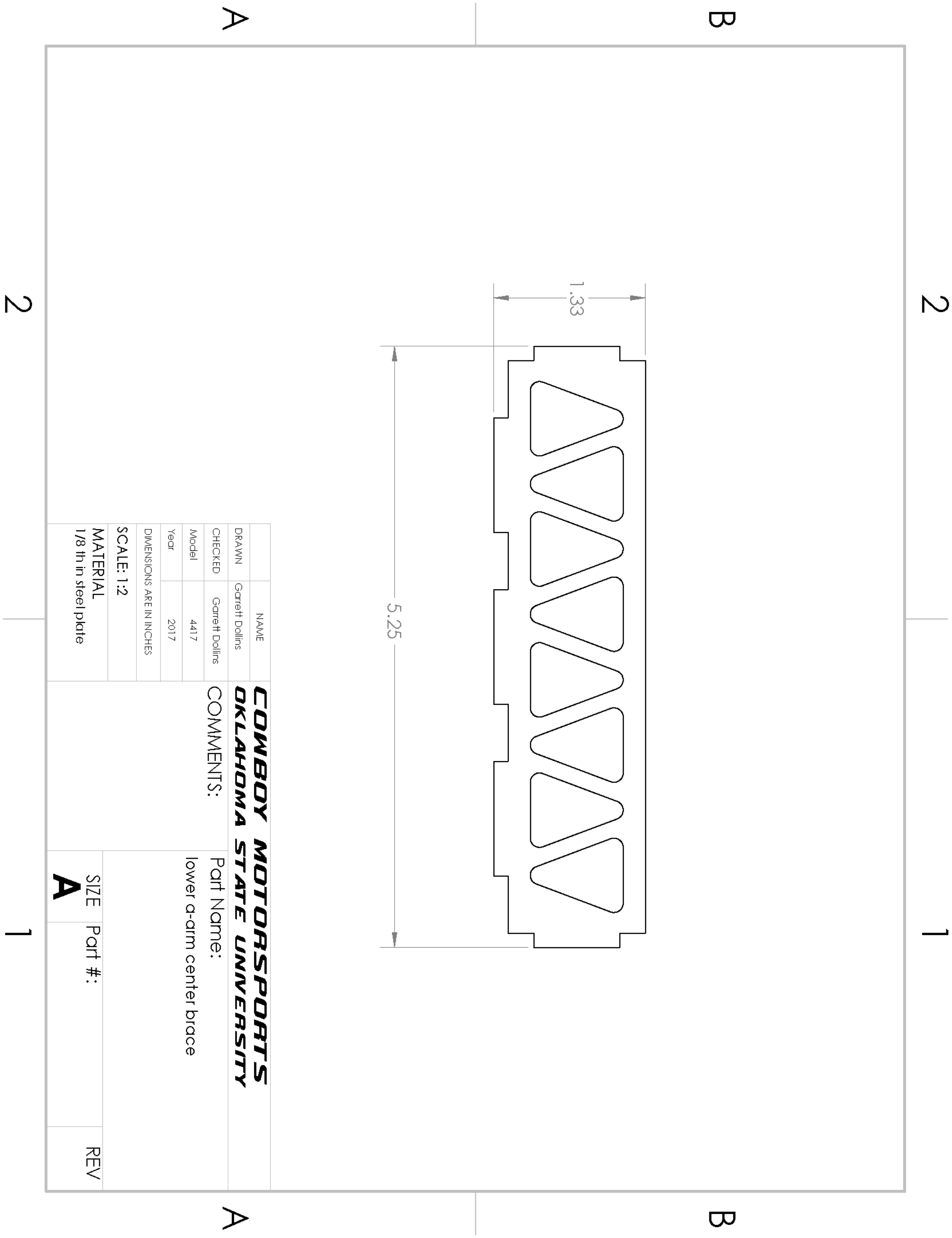
COWBOY MOTORSPORTS
OKLAHOMA STATE UNIVERSITY
NAME
DRAWN Scott Dick
CHECKED Garrett Dollins
Model 4417
Year 2017
DIMENSIONS ARE IN INCHES
SCALE: 2:1
MATERIAL
1/8" sheet metal - steel
COMMENTS:
x2
Part Name:
Front diff mount top tab
SIZE
A
Part #:
REV
1.50
1.00
0.50
∅0.45
A
B
2
1

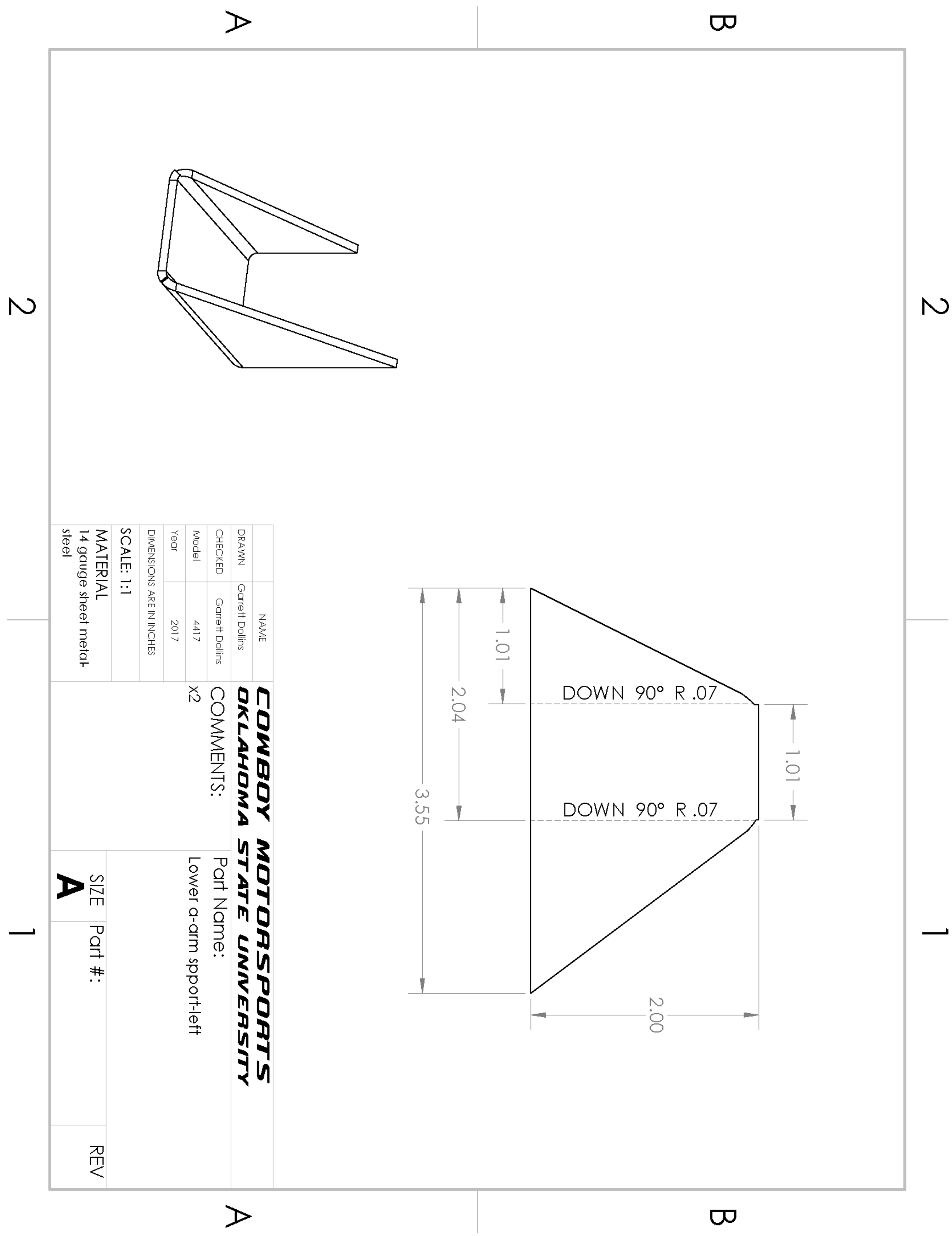

1.0133
0.9468
DOWN 90° R 0.07
DOWN 84.57° R 0.07
14.50
4.09
NAME
DRAWN Scott Dick
CHECKED Garrett Dollins
Model 4417
Year 2017
DIMENSIONS ARE IN INCHES
SCALE: 1:2
MATERIAL
14 gauge sheet metal - steel
COWBOY MOTORSPORTS
OKLAHOMA STATE UNIVERSITY
COMMENTS:
Part Name:
Front Frame piece
SIZE
A
Part #:
REV

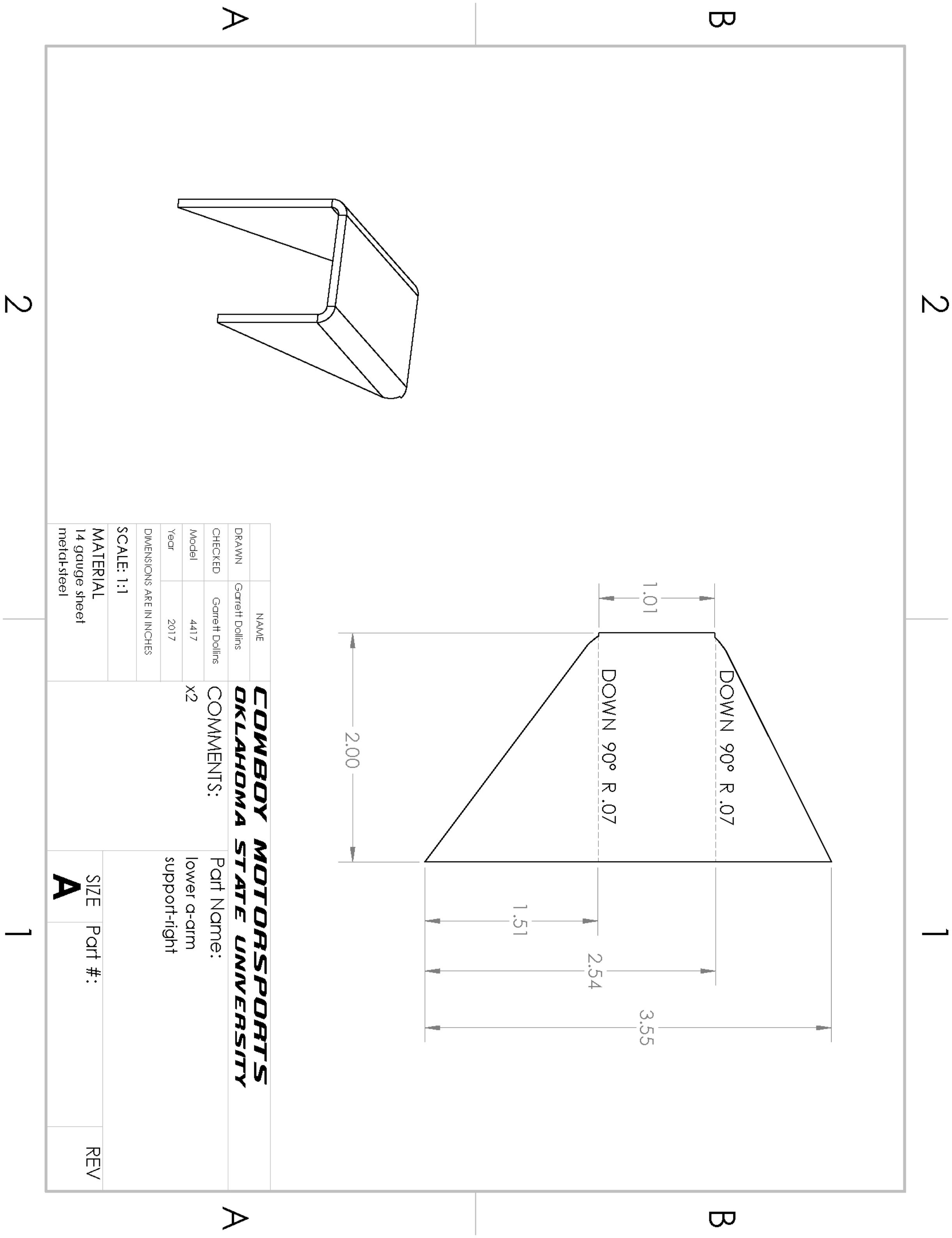


COWBOY MOTORSPORTS
OKLAHOMA STATE UNIVERSITY
COMMENTS:
Part Name:
Left frame rail
SIZE
A
Part #:
REV
NAME
DRAWN Scott Dick
CHECKED Garrett Dollins
Model 417
Year 2017
DIMENSIONS ARE IN INCHES
SCALE: 1:20
MATERIAL
14 gauge sheet metal - steel
UP 90° R 0.07
UP 30° R 0.07
79.97
7.25
1.1886
6.1104
5.4318

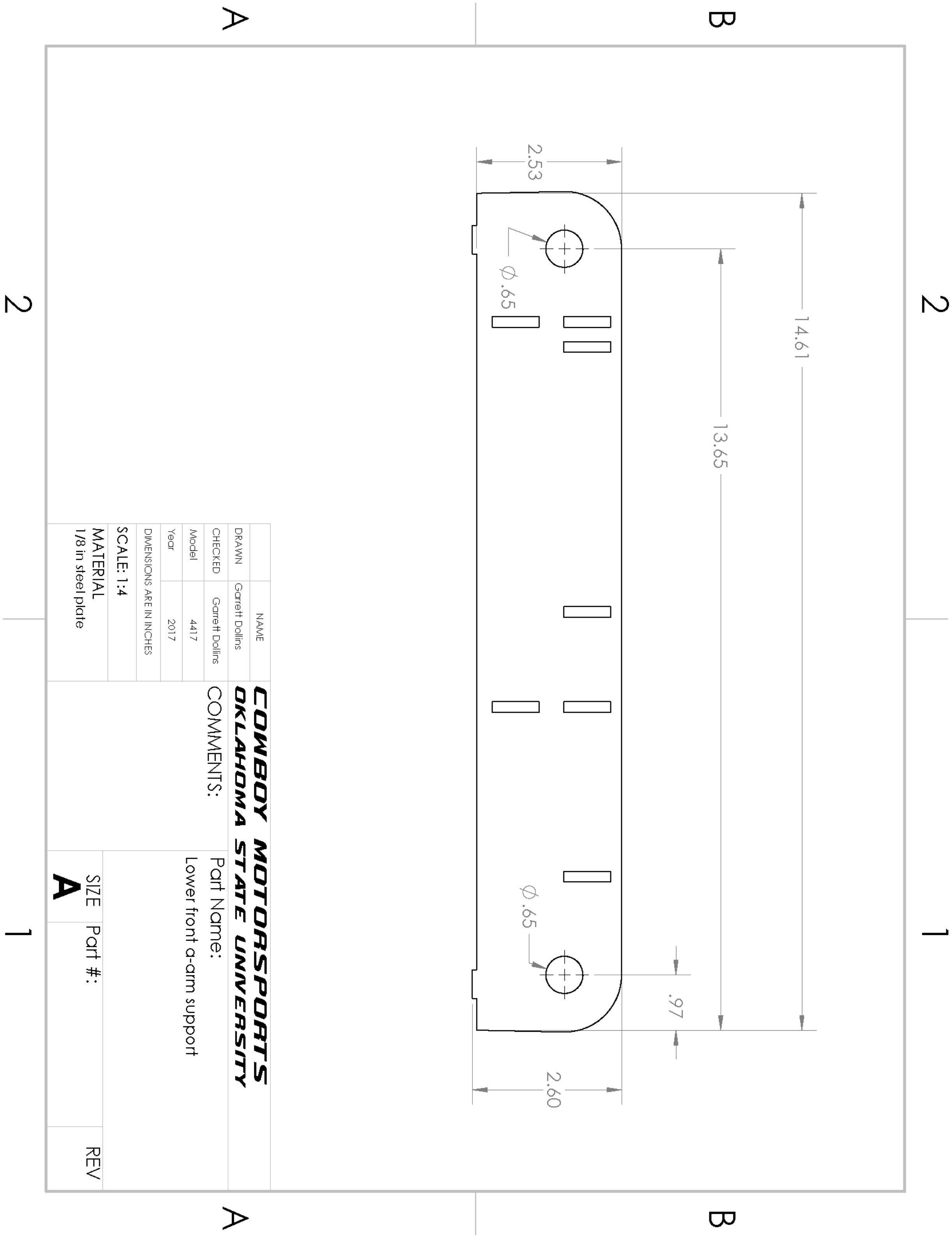

1.20
5.25
NAME
DRAWN Garrett Dollins
CHECKED Garrett Dollins
Model 4417
Year 2017
DIMENSIONS ARE IN INCHES
SCALE: 1:2
MATERIAL
1/8 th in steel plate
COWBOY MOTORSPORTS
OKLAHOMA STATE UNIVERSITY
COMMENTS:
x2
Part Name:
lower a-arm brace
SIZE
A
Part #:
REV

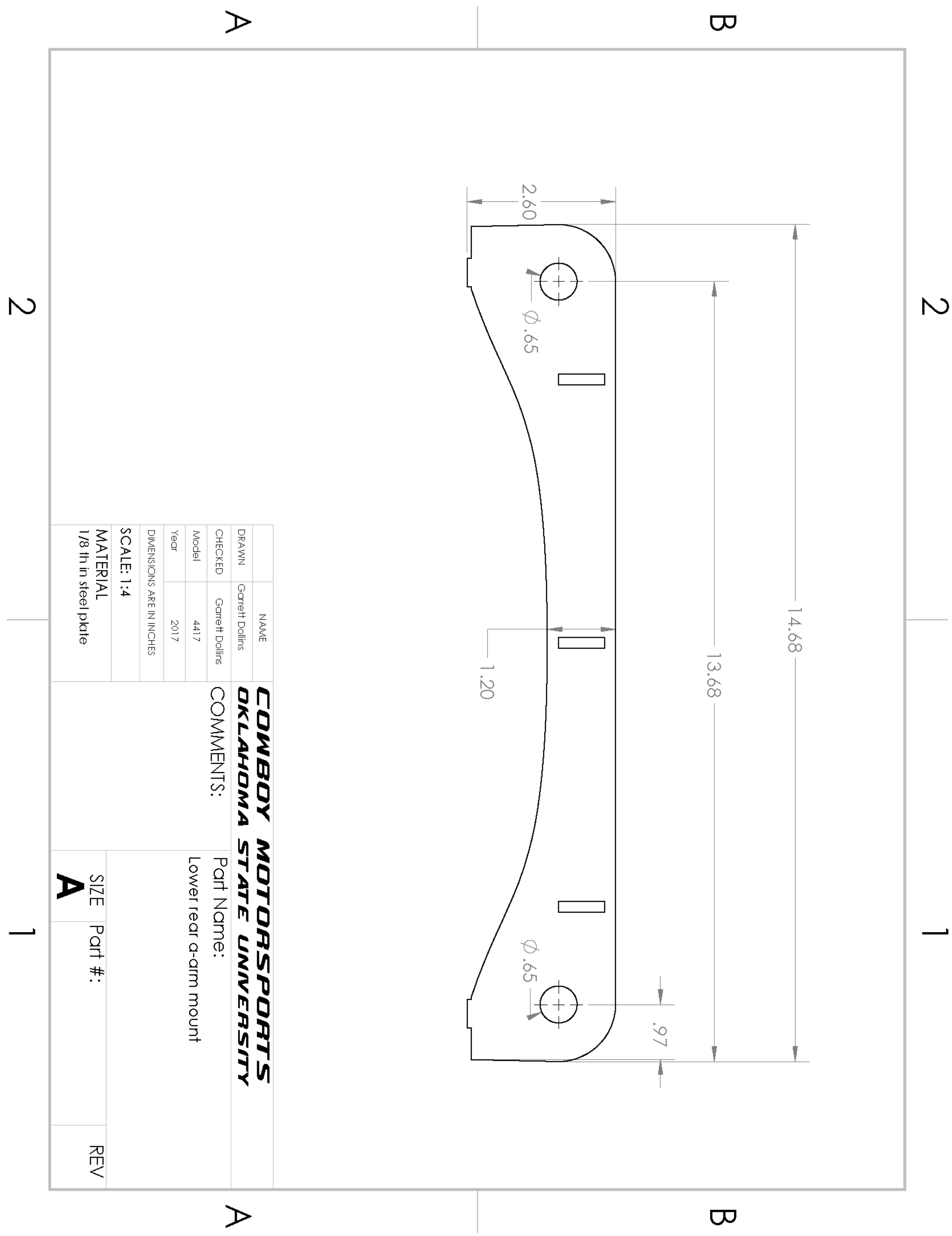

A
B
2
1
1.33
5.25
NAME
DRAWN Garrett Dollins
CHECKED Garrett Dollins
Model 4417
Year 2017
DIMENSIONS ARE IN INCHES
SCALE: 1:2
MATERIAL
1/8 th in steel plate
COWBOY MOTORSPORTS
OKLAHOMA STATE UNIVERSITY
COMMENTS:
Part Name:
lower a-arm center brace
SIZE
A
Part #:
REV

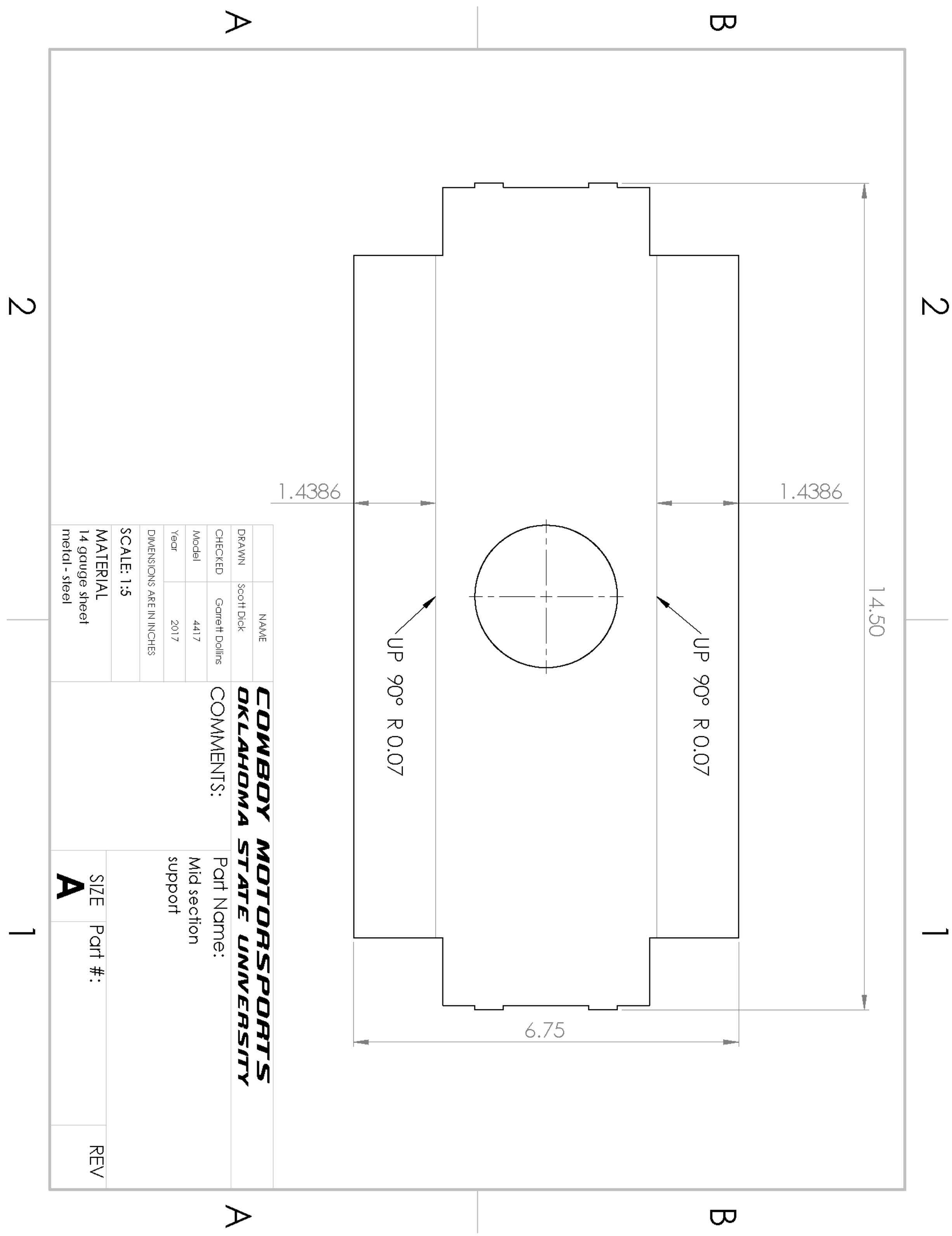

COWBOY MOTORSPORTS
OKLAHOMA STATE UNIVERSITY
NAME
DRAWN Garrett Dollins
CHECKED Garrett Dollins
Model 417
Year 2017
DIMENSIONS ARE IN INCHES
SCALE: 1:1
MATERIAL
14 gauge sheet metal-steel
COMMENTS:
x2
Part Name:
Lower a-arm spport-left
SIZE
A
Part #:
REV
DOWN 90° R .07
DOWN 90° R .07
1.01
2.04
3.55
1.01
2.00
A
B
1
2

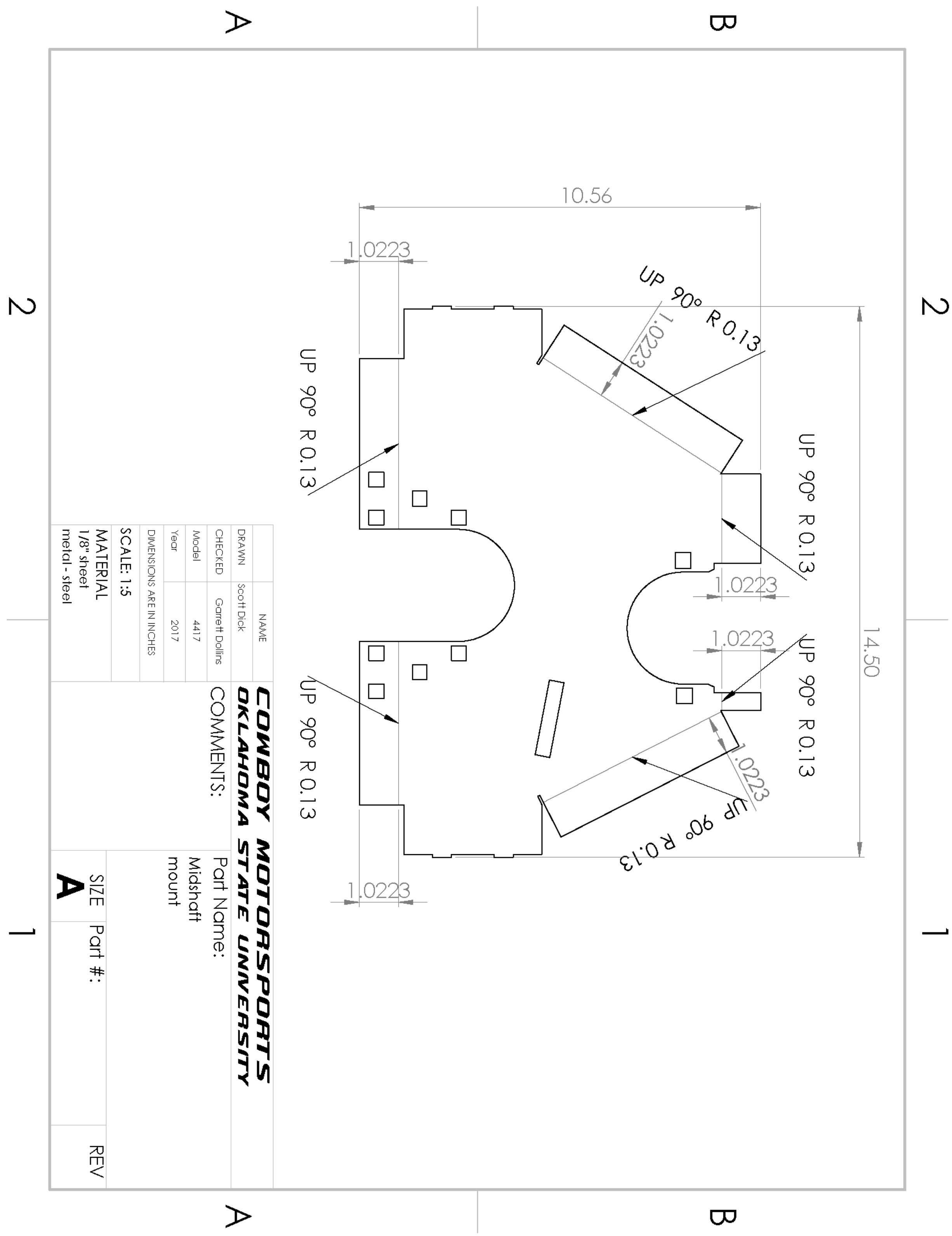


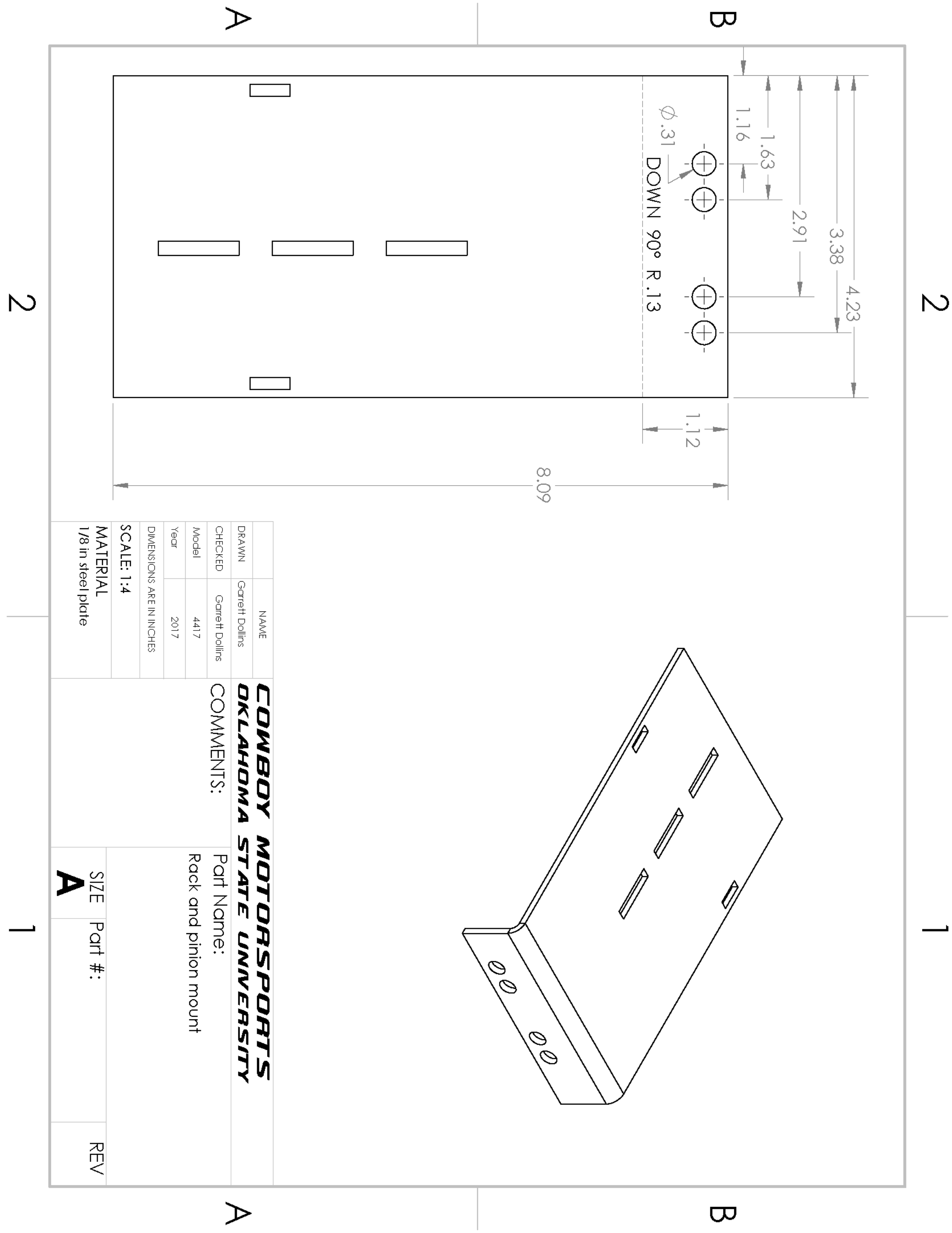
A
B
1
2
DOWN 90° R .07
DOWN 90° R .07
1.01
2.00
1.51
2.54
3.55
COWBOY MOTORSPORTS
OKLAHOMA STATE UNIVERSITY
NAME
DRAWN Garrett Dollins
CHECKED Garrett Dollins
Model 417
Year 2017
DIMENSIONS ARE IN INCHES
SCALE: 1:1
MATERIAL
14 gauge sheet metal-steel
COMMENTS:
x2
Part Name:
lower a-arm support-right
SIZE
A
Part #:
REV


2.53
14.61
13.65
∅.65
∅.65
.97
2.60
NAME
DRAWN Garrett Dollins
CHECKED Garrett Dollins
Model 4417
Year 2017
DIMENSIONS ARE IN INCHES
SCALE: 1:4
MATERIAL
1/8 in steel plate
COWBOY MOTORSPORTS
OKLAHOMA STATE UNIVERSITY
COMMENTS:
Part Name:
Lower front a-arm support
SIZE
A
Part #:
REV

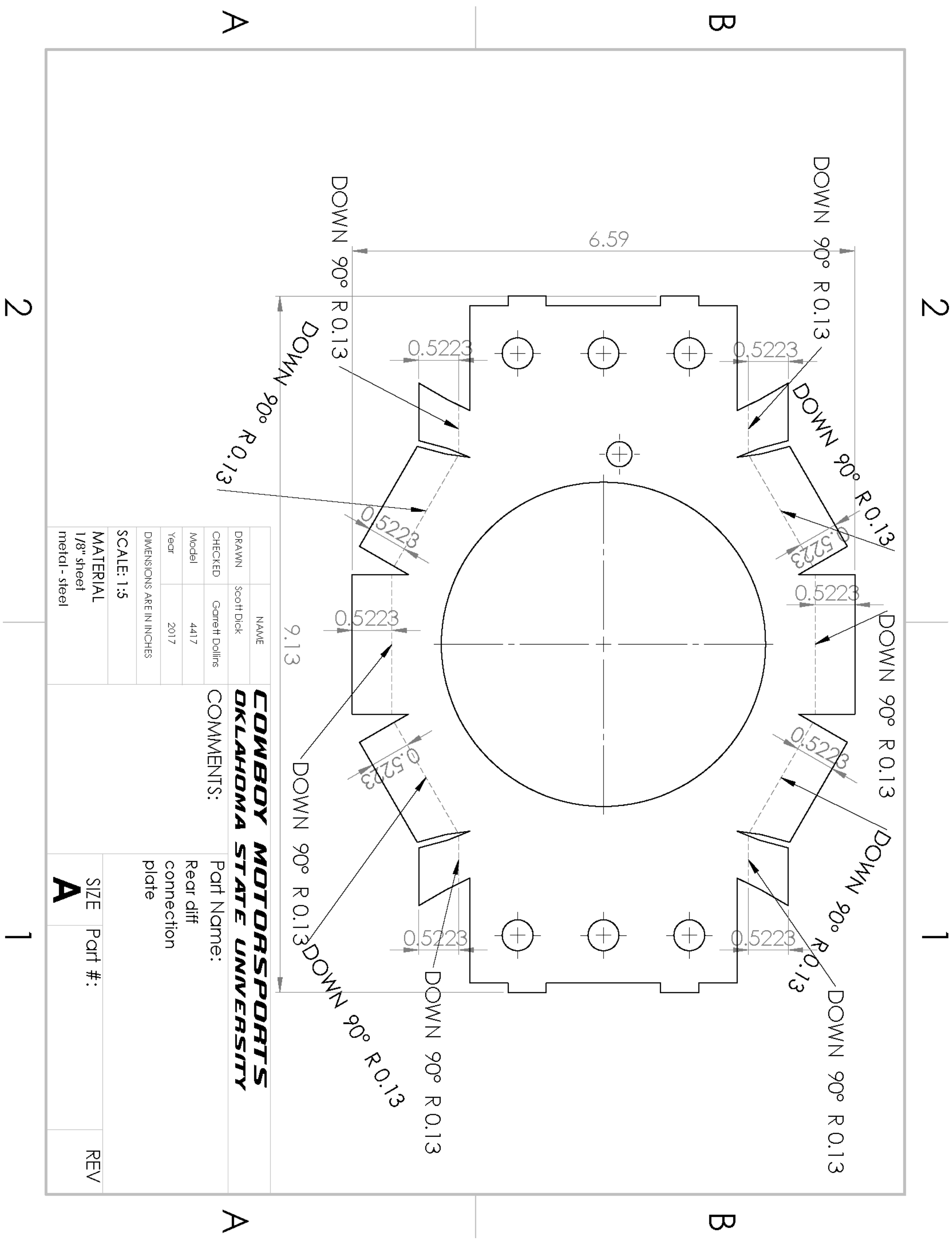

COWBOY MOTORSPORTS
OKLAHOMA STATE UNIVERSITY
NAME
DRAWN Garrett Dollins
CHECKED Garrett Dollins
Model 417
Year 2017
DIMENSIONS ARE IN INCHES
SCALE: 1:4
MATERIAL
1/8 th in steel plate
COMMENTS:
Part Name:
Lower rear a-arm mount
SIZE
A
Part #:
REV
2.60
14.68
13.68
1.20
.97
Ø.65
Ø.65
A
B
1
2

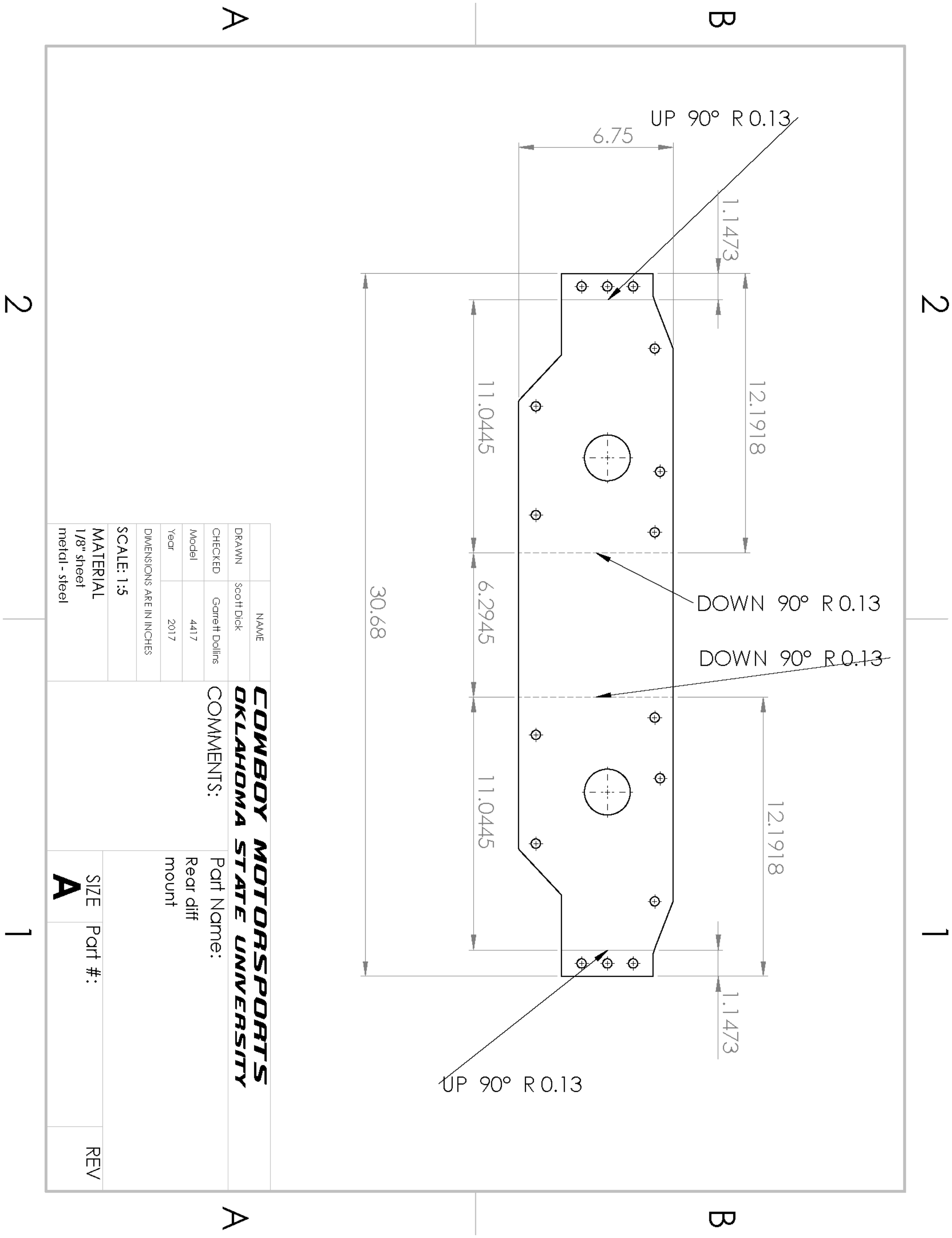

A
B
2
1
1.4386
1.4386
14.50
6.75
UP 90° R 0.07
UP 90° R 0.07
NAME
DRAWN Scott Dick
CHECKED Garrett Dollins
Model 417
Year 2017
DIMENSIONS ARE IN INCHES
SCALE: 1:5
MATERIAL
14 gauge sheet metal - steel
COWBOY MOTORSPORTS
OKLAHOMA STATE UNIVERSITY
COMMENTS:
Part Name:
Mid section support
SIZE
A
Part #:
REV

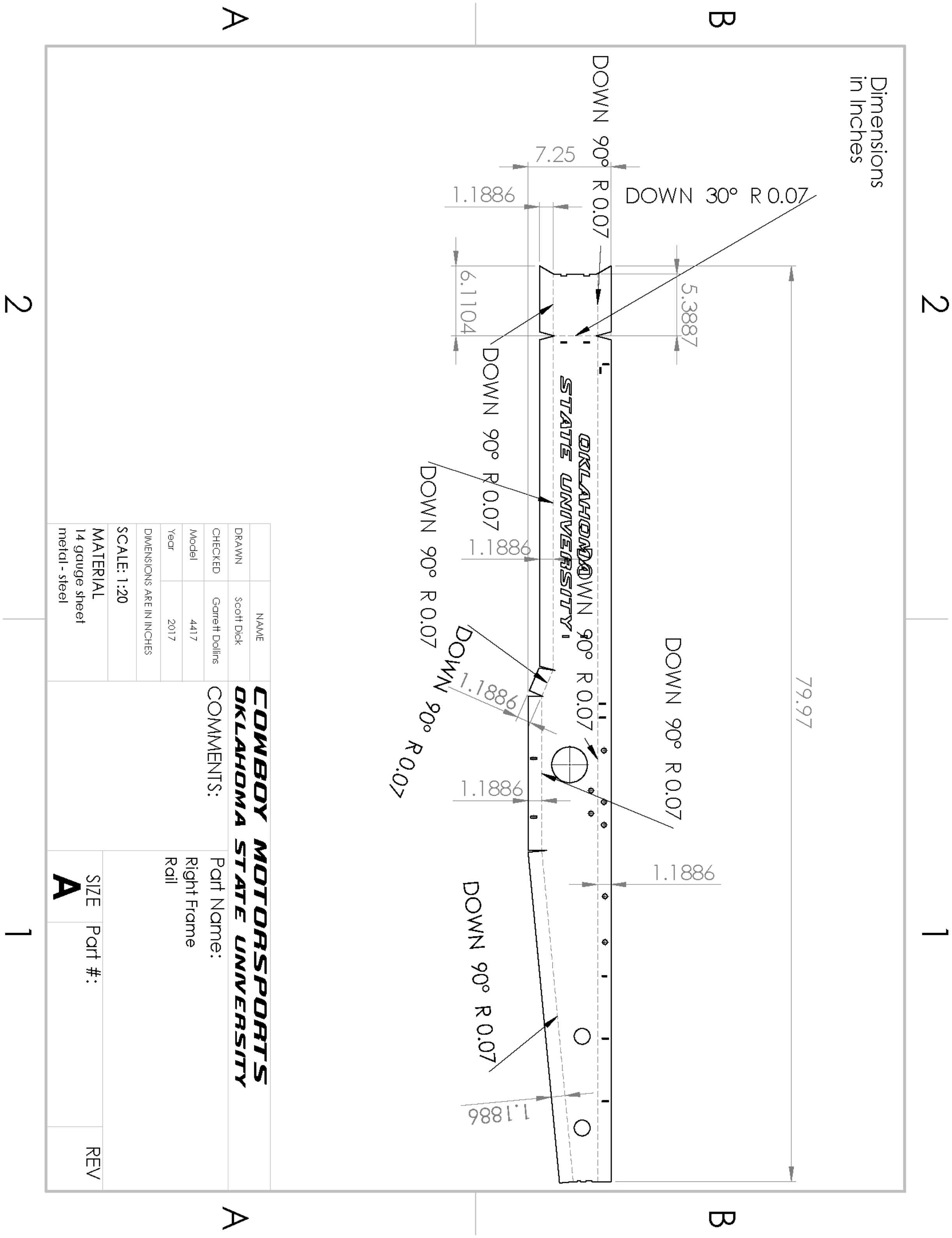

10.56
1.0223
14.50
UP 90° R 0.13
COWBOY MOTORSPORTS
OKLAHOMA STATE UNIVERSITY
COMMENTS:
NAME
DRAWN Scott Dick
CHECKED Garrett Dollins
Model 417
Year 2017
DIMENSIONS ARE IN INCHES
SCALE: 1:5
MATERIAL
1/8" sheet
metal - steel
Part Name:
Midshaft
mount
SIZE
A
Part #:
REV

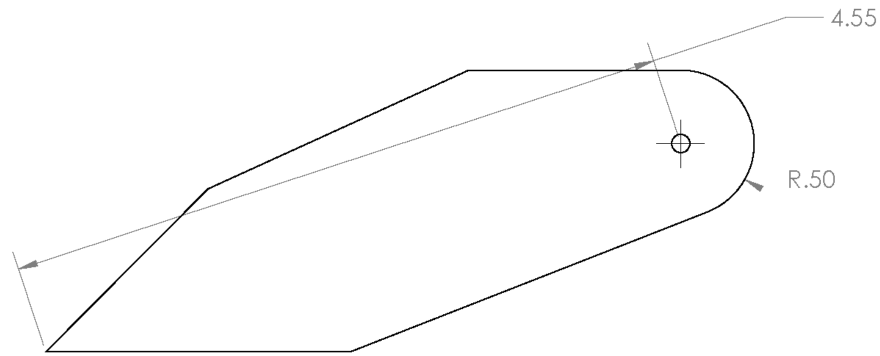


1.16
1.63
2.91
3.38
4.23
∅.31
DOWN 90° R.13
1.12
8.09
NAME
DRAWN Garrett Dollins
CHECKED Garrett Dollins
Model 417
Year 2017
DIMENSIONS ARE IN INCHES
SCALE: 1:4
MATERIAL
1/8 in steel plate
COWBOY MOTORSPORTS
OKLAHOMA STATE UNIVERSITY
COMMENTS:
Part Name:
Rack and pinion mount
SIZE
A
Part #:
REV


A
B
1
2
6.59
9.13
0.5223
DOWN 90° R 0.13
COWBOY MOTORSPORTS
OKLAHOMA STATE UNIVERSITY
NAME
DRAWN Scott Dick
CHECKED Garrett Dollins
Model 4417
Year 2017
DIMENSIONS ARE IN INCHES
SCALE: 1:5
MATERIAL
1/8" sheet metal - steel
COMMENTS:
Part Name:
Rear diff connection plate
SIZE
A
Part #:
REV


UP 90° R 0.13
6.75
1.1473
12.1918
11.0445
30.68
DOWN 90° R 0.13
6.2945
DOWN 90° R 0.13
11.0445
12.1918
1.1473
UP 90° R 0.13
COWBOY MOTORSPORTS
OKLAHOMA STATE UNIVERSITY
NAME
DRAWN Scott Dick
CHECKED Garrett Dollins
Model 417
Year 2017
DIMENSIONS ARE IN INCHES
SCALE: 1:5
MATERIAL
1/8" sheet metal - steel
COMMENTS:
Part Name:
Rear diff mount
SIZE
A
Part #:
REV
A
B
2
1


Dimensions in Inches
DOWN 90° R 0.07
DOWN 30° R 0.07
7.25
1.1886
6.1104
5.3887
79.97
OKLAHOMA STATE UNIVERSITY
DOWN 90° R 0.07
DOWN 90° R 0.07
DOWN 90° R 0.07
DOWN 90° R 0.07
DOWN 90° R 0.07
DOWN 90° R 0.07
DOWN 90° R 0.07
1.1886
1.1886
1.1886
1.1886
1.1886
COWBOY MOTORSPORTS
OKLAHOMA STATE UNIVERSITY
NAME
DRAWN Scott Dick
CHECKED Garrett Dollins
Model 4417
Year 2017
DIMENSIONS ARE IN INCHES
SCALE: 1:20
MATERIAL
14 gauge sheet metal - steel
COMMENTS:
Part Name:
Right Frame Rail
SIZE
A
Part #:
REV
A
B
2
1



	NAME
DRAWN	Garrett Dollins
CHECKED	Garrett Dollins
Model	4417
Year	2017

DIMENSIONS ARE IN INCHES

SCALE: 1:1

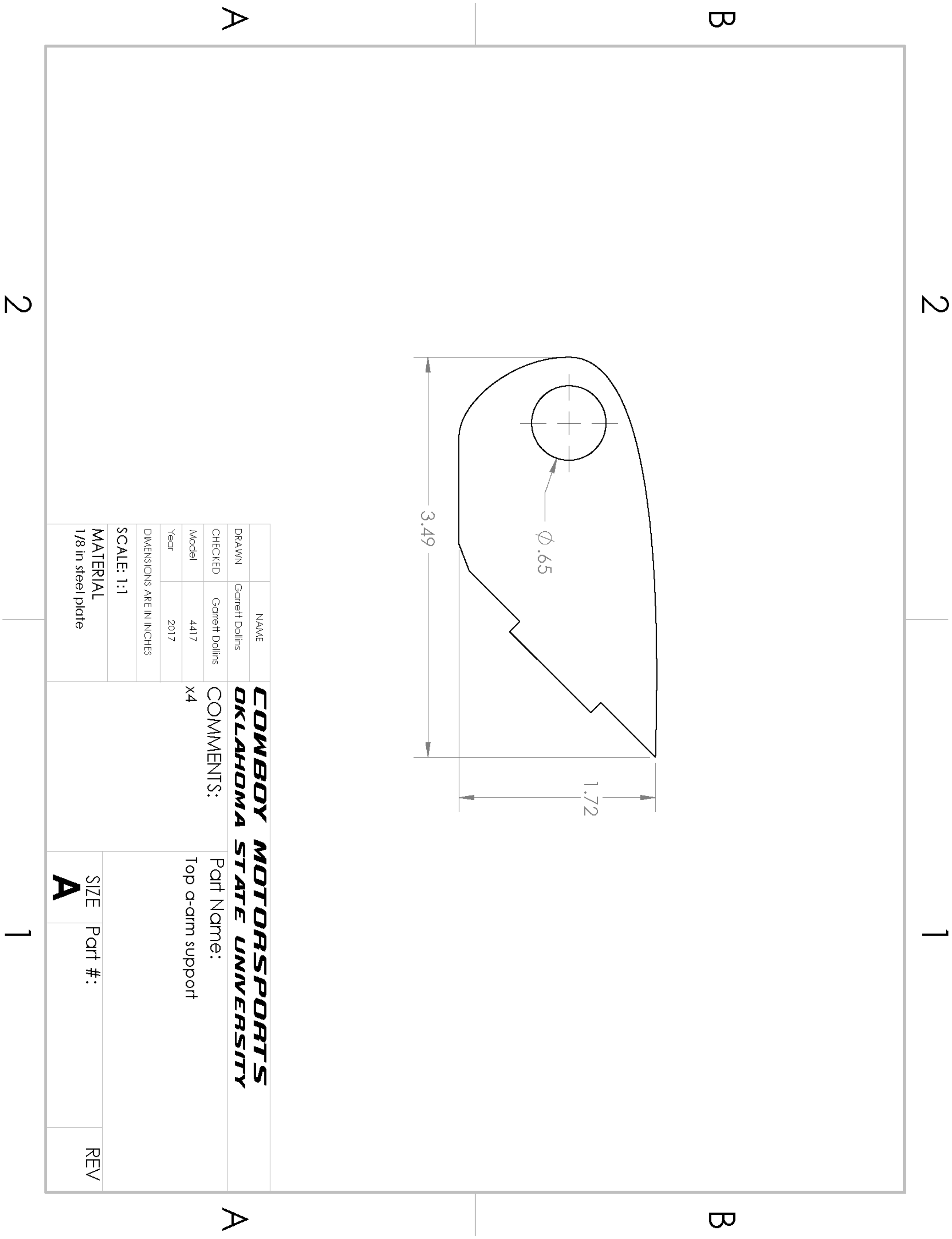
MATERIAL
1/8 in steel
plate

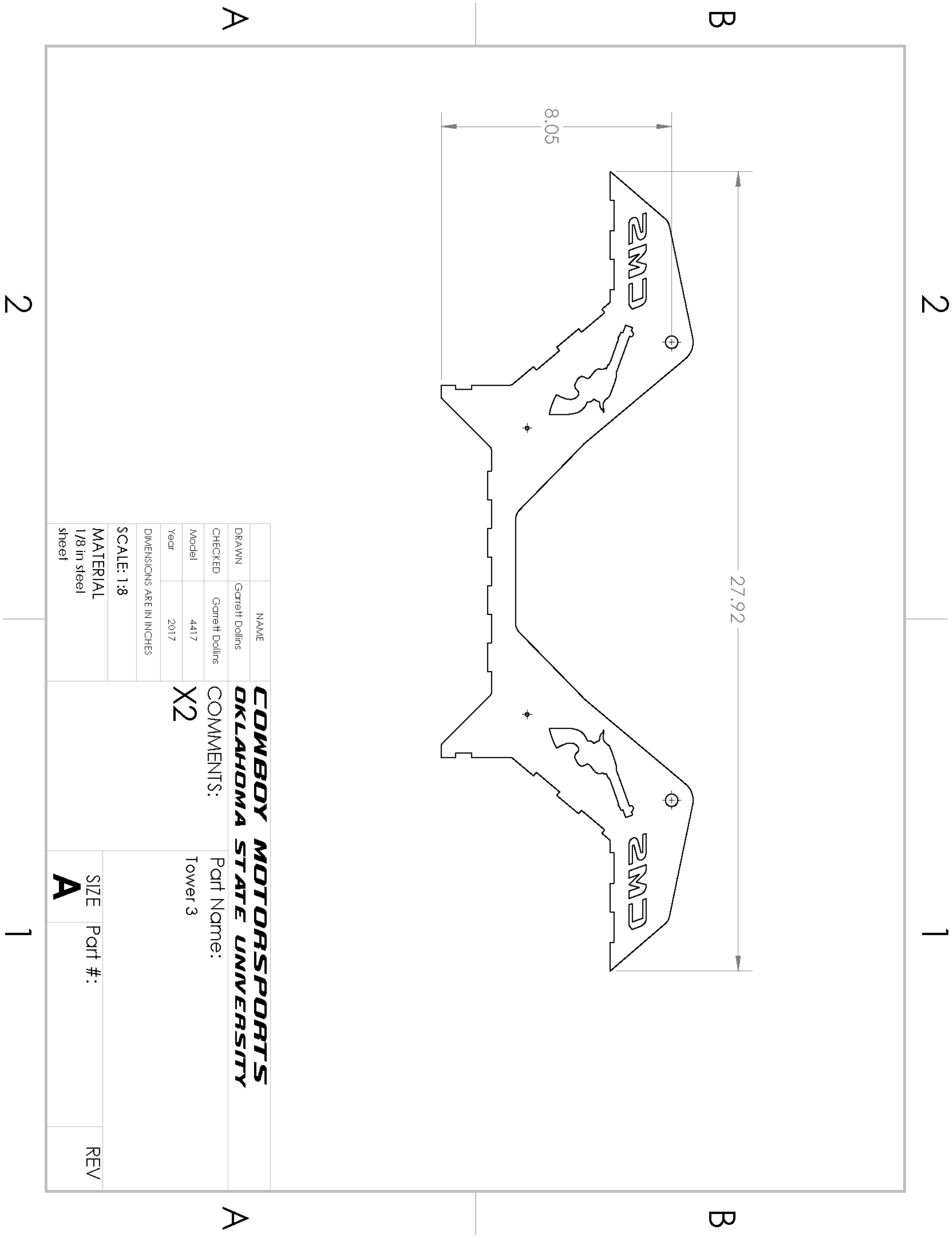
COWBOY MOTORSPORTS
OKLAHOMA STATE UNIVERSITY

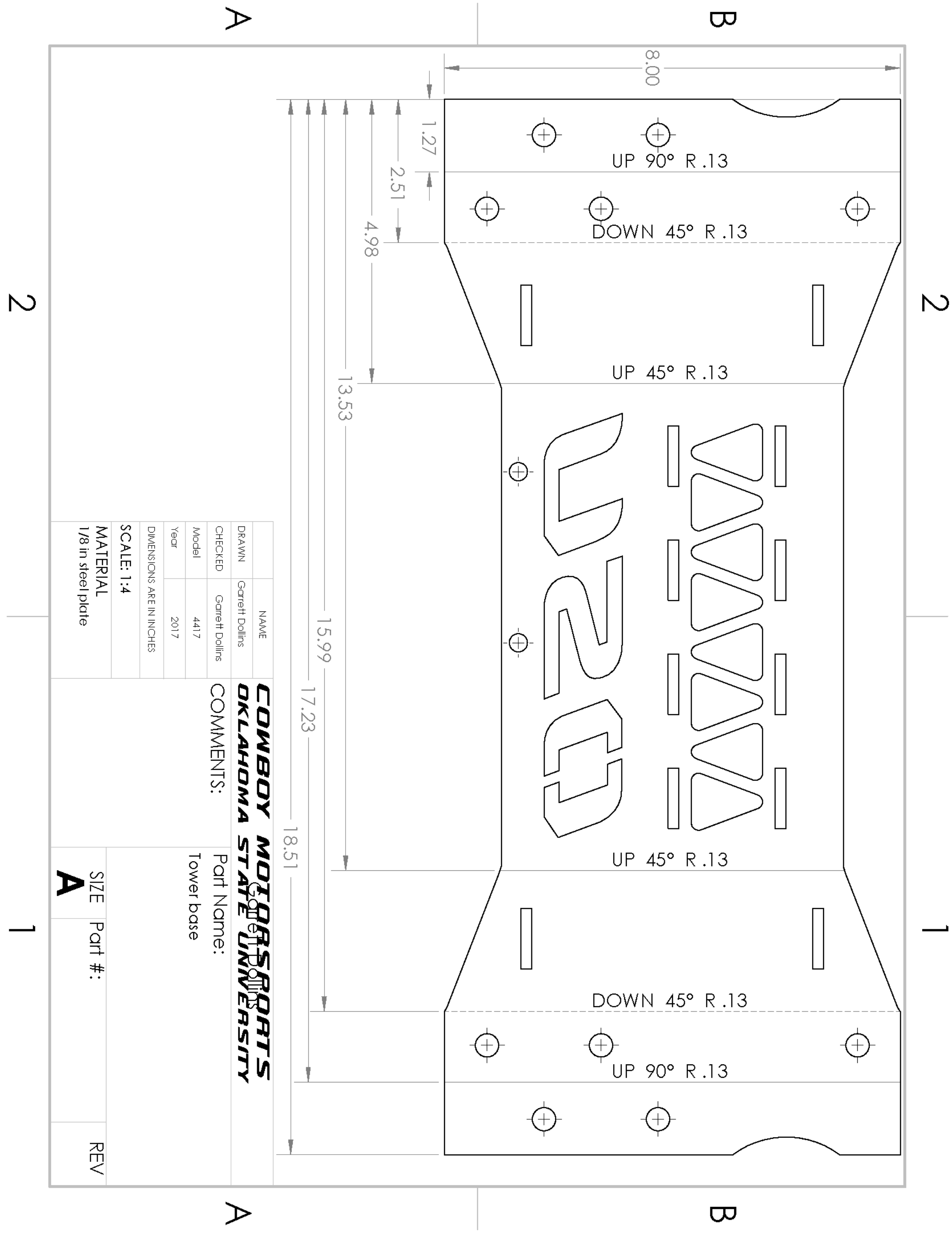
COMMENTS:
x4

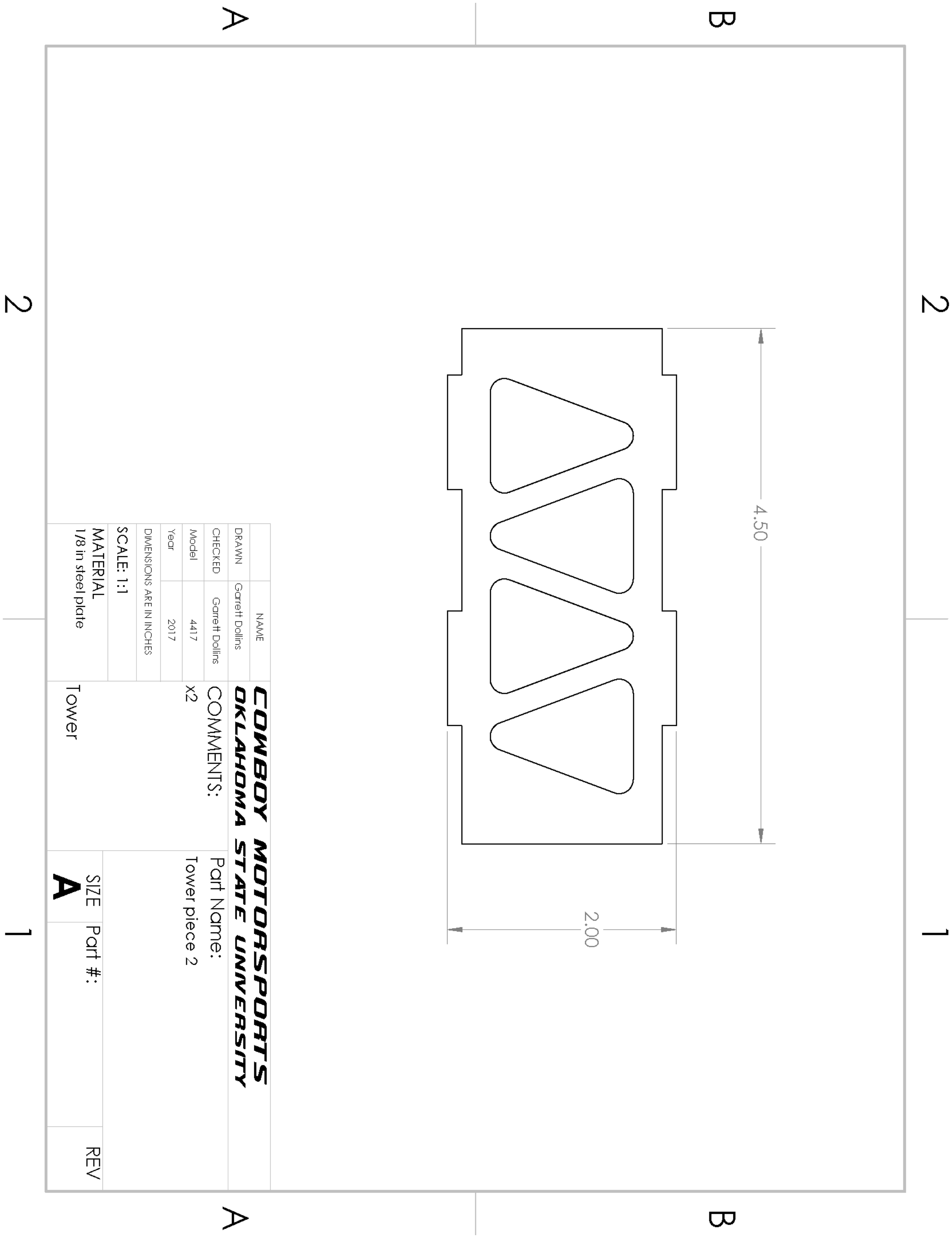
Part Name:
Side support
lock out 2

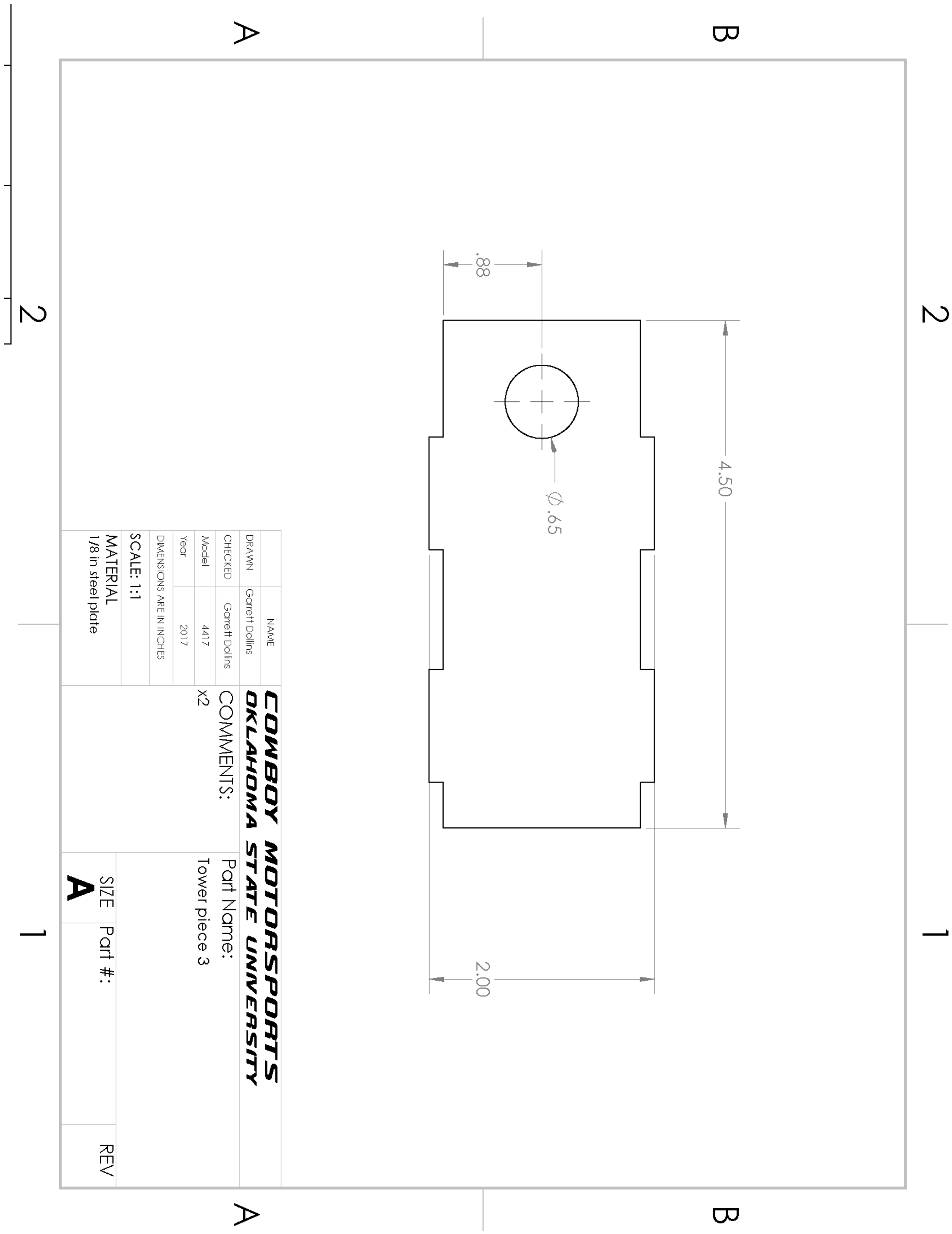
SIZE **A** Part #: REV

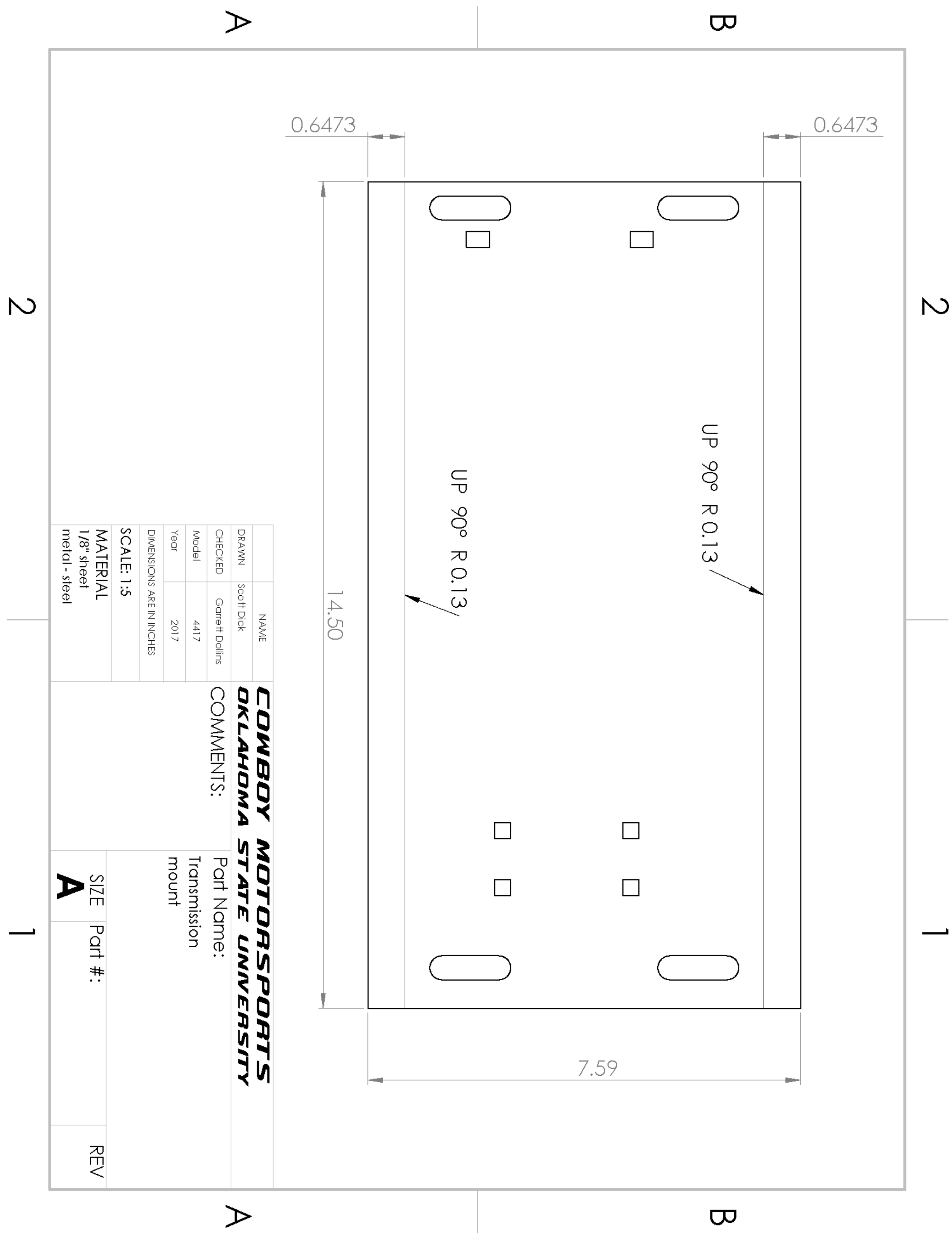

A
B
2
1
3.49
∅.65
1.72
COWBOY MOTORSPORTS
OKLAHOMA STATE UNIVERSITY
NAME
DRAWN Garrett Dollins
CHECKED Garrett Dollins
Model 4417
Year 2017
DIMENSIONS ARE IN INCHES
SCALE: 1:1
MATERIAL
1/8 in steel plate
COMMENTS:
x4
Part Name:
Top a-arm support
SIZE
A
Part #:
REV

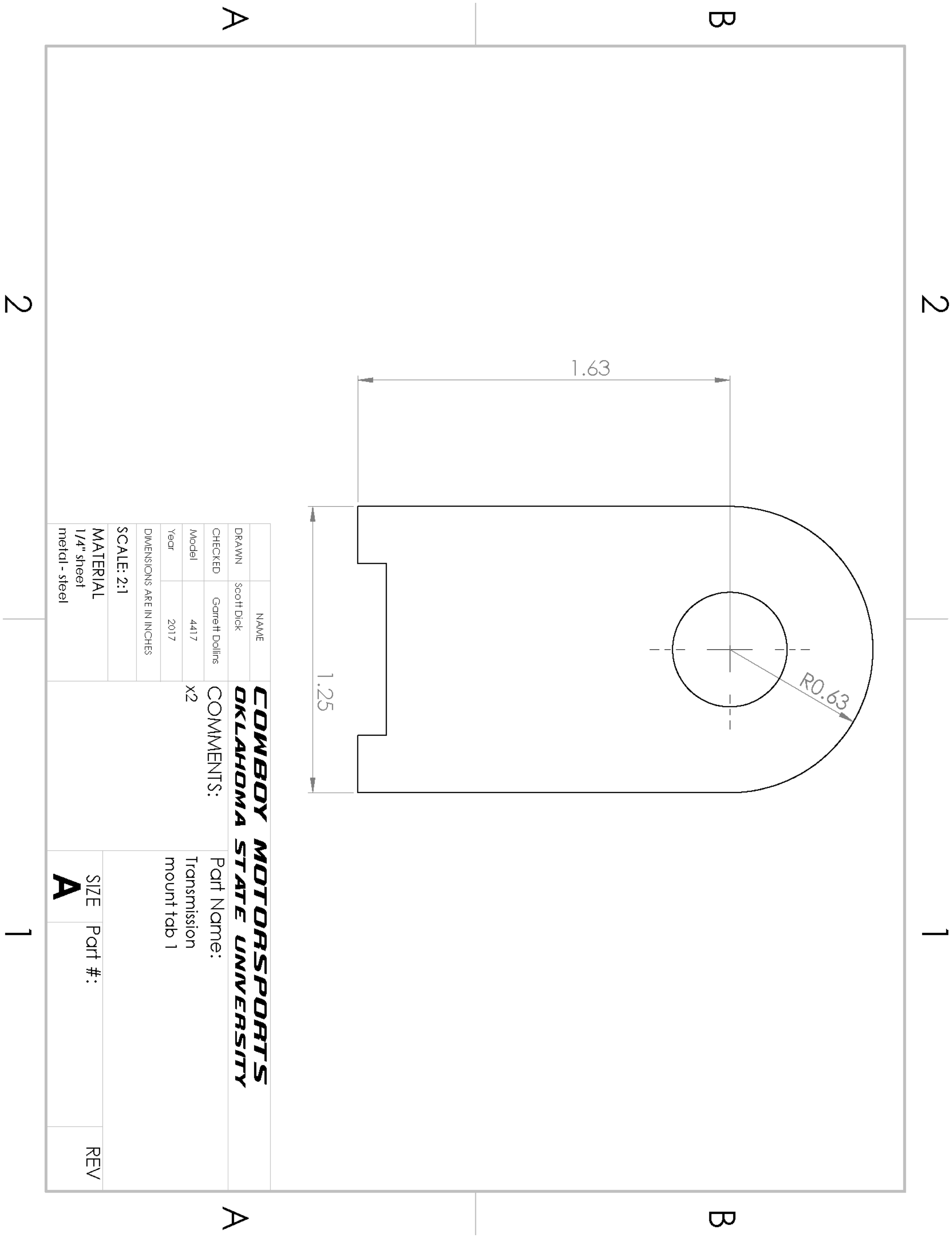

8.05
27.92
COWBOY MOTORSPORTS
OKLAHOMA STATE UNIVERSITY
NAME
DRAWN Garrett Dollins
CHECKED Garrett Dollins
Model 4417
Year 2017
DIMENSIONS ARE IN INCHES
SCALE: 1:8
MATERIAL
1/8 in steel sheet
COMMENTS:
X2
Part Name:
Tower 3
SIZE
A
Part #:
REV

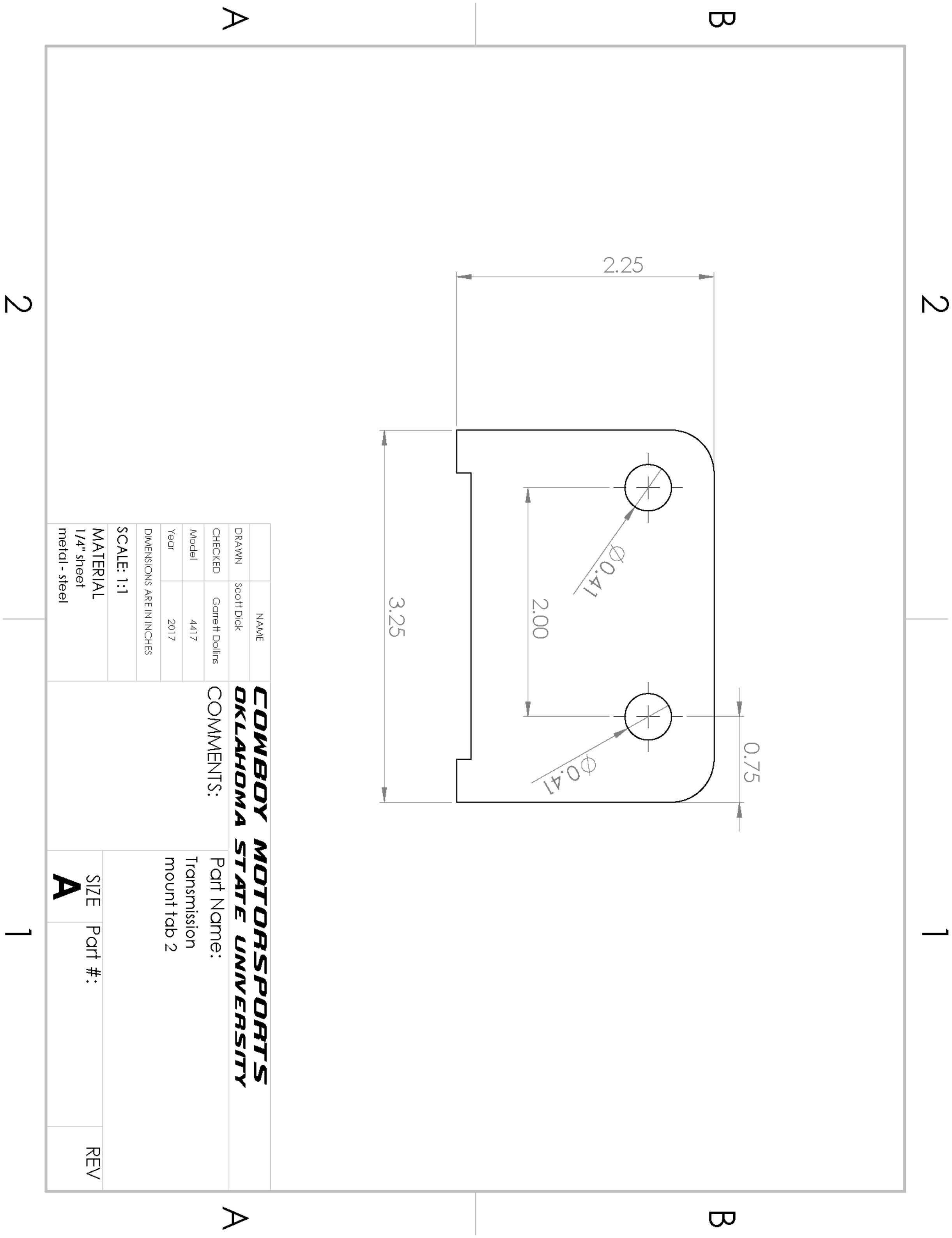

8.00
1.27
2.51
4.98
13.53
15.99
17.23
18.51
UP 90° R .13
DOWN 45° R .13
UP 45° R .13
UP 45° R .13
DOWN 45° R .13
UP 90° R .13
NAME
DRAWN Garrett Dollins
CHECKED Garrett Dollins
Model 417
Year 2017
DIMENSIONS ARE IN INCHES
SCALE: 1:4
MATERIAL
1/8 in steel plate
COWBOY MOTORSPORTS
OKLAHOMA STATE UNIVERSITY
COMMENTS:
Part Name:
Tower base
SIZE
A
Part #:
REV
A
B
2
1


4.50
2.00
A
B
2
1
NAME
DRAWN Garrett Dollins
CHECKED Garrett Dollins
Model 4417
Year 2017
DIMENSIONS ARE IN INCHES
SCALE: 1:1
MATERIAL
1/8 in steel plate
COWBOY MOTORSPORTS
OKLAHOMA STATE UNIVERSITY
COMMENTS:
x2
Tower
Part Name:
Tower piece 2
SIZE
A
Part #:
REV


COWBOY MOTORSPORTS
OKLAHOMA STATE UNIVERSITY
NAME
DRAWN Garrett Dollins
CHECKED Garrett Dollins
Model 417
Year 2017
DIMENSIONS ARE IN INCHES
SCALE: 1:1
MATERIAL
1/8 in steel plate
COMMENTS:
x2
Part Name:
Tower piece 3
SIZE
A
Part #:
REV
.88
4.50
Ø.65
2.00
A
B
1
2


0.6473
0.6473
14.50
7.59
UP 90° R 0.13
UP 90° R 0.13
COWBOY MOTORSPORTS
OKLAHOMA STATE UNIVERSITY
NAME
DRAWN Scott Dick
CHECKED Garrett Dollins
Model 417
Year 2017
DIMENSIONS ARE IN INCHES
SCALE: 1:5
MATERIAL
1/8" sheet
metal - steel
COMMENTS:
Part Name:
Transmission
mount
SIZE
A
Part #:
REV


1.63
1.25
R0.63
NAME
DRAWN Scott Dick
CHECKED Garrett Dollins
Model 4417
Year 2017
DIMENSIONS ARE IN INCHES
SCALE: 2:1
MATERIAL
1/4" sheet
metal - steel
COWBOY MOTORSPORTS
OKLAHOMA STATE UNIVERSITY
COMMENTS:
x2
Part Name:
Transmission
mount tab 1
SIZE
A
Part #:
REV


2.25
3.25
2.00
0.75
Ø0.41
Ø0.41
COWBOY MOTORSPORTS
OKLAHOMA STATE UNIVERSITY
NAME
DRAWN Scott Dick
CHECKED Garrett Dollins
Model 417
Year 2017
DIMENSIONS ARE IN INCHES
SCALE: 1:1
MATERIAL
1/4" sheet
metal - steel
COMMENTS:
Part Name:
Transmission
mount tab 2
SIZE
A
Part #:
REV
A
B
1
2

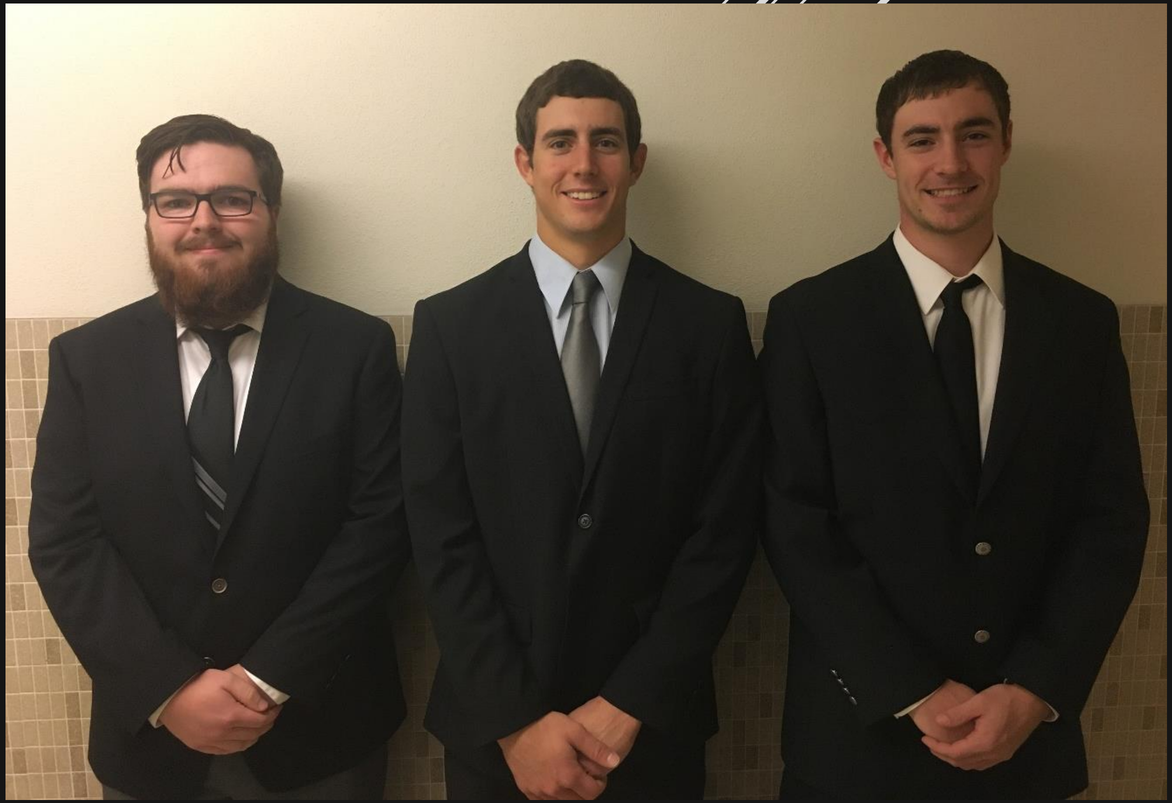
COWBOY MOTORSPORTS

SENIOR DESIGN 2016-2017

Scott Dick

Garrett Dollins

Logan Gary



2016-2017 ASABE INTERNATIONAL QUARTER SCALE TRACTOR STUDENT DESIGN COMPETITION

- Sponsored by the American Society of Agricultural and Biological Engineers (ASABE) and International Quarter Scale (IQS)
- 30 teams including some international participation

COMPETITION OVERVIEW

- Design report 500 pts
- Team presentation 500 pts
- Design judging 420 pts
- Technical inspection Pass/Fail
- Tractor pulls 600 pts
- Maneuverability 100 pts
- Durability event 200 pts
- Initial weigh in 100 pts

PROBLEM STATEMENT

To design and build a cost effective, reliable, and innovative frame, steering system, and suspension system for the Oklahoma State University Quarter Scale tractor team. The design will take into account the team's budget, timeline, and resources for the 2016-2017 competition.

FRAME REQUIREMENTS

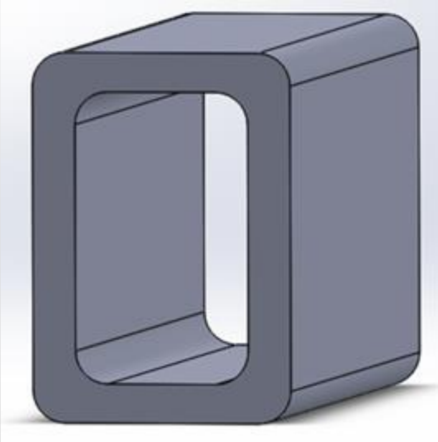
- Withstand weight of tractor and forces felt during competition
- Provide area to mount other components of tractor
- Less than 96 inches long
- Fully customized

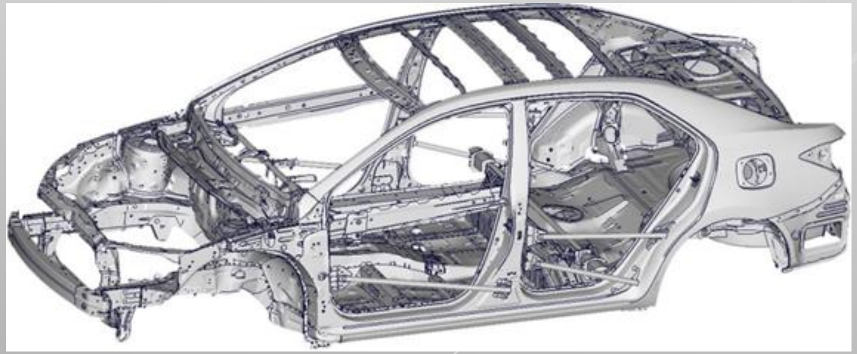
FRAME OBJECTIVES

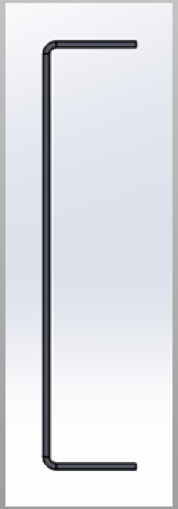
- Easily manufactured
- Fully welded together
- Lightweight
- Display school and club name

FRAME SELECTION

- Tube Frame
  - Strong, but heavy
- Unibody Frame
  - Very specific to each vehicle
  - Requires precise engineering
- C-Channel Frame
  - Lightweight
  - Not as strong as other options

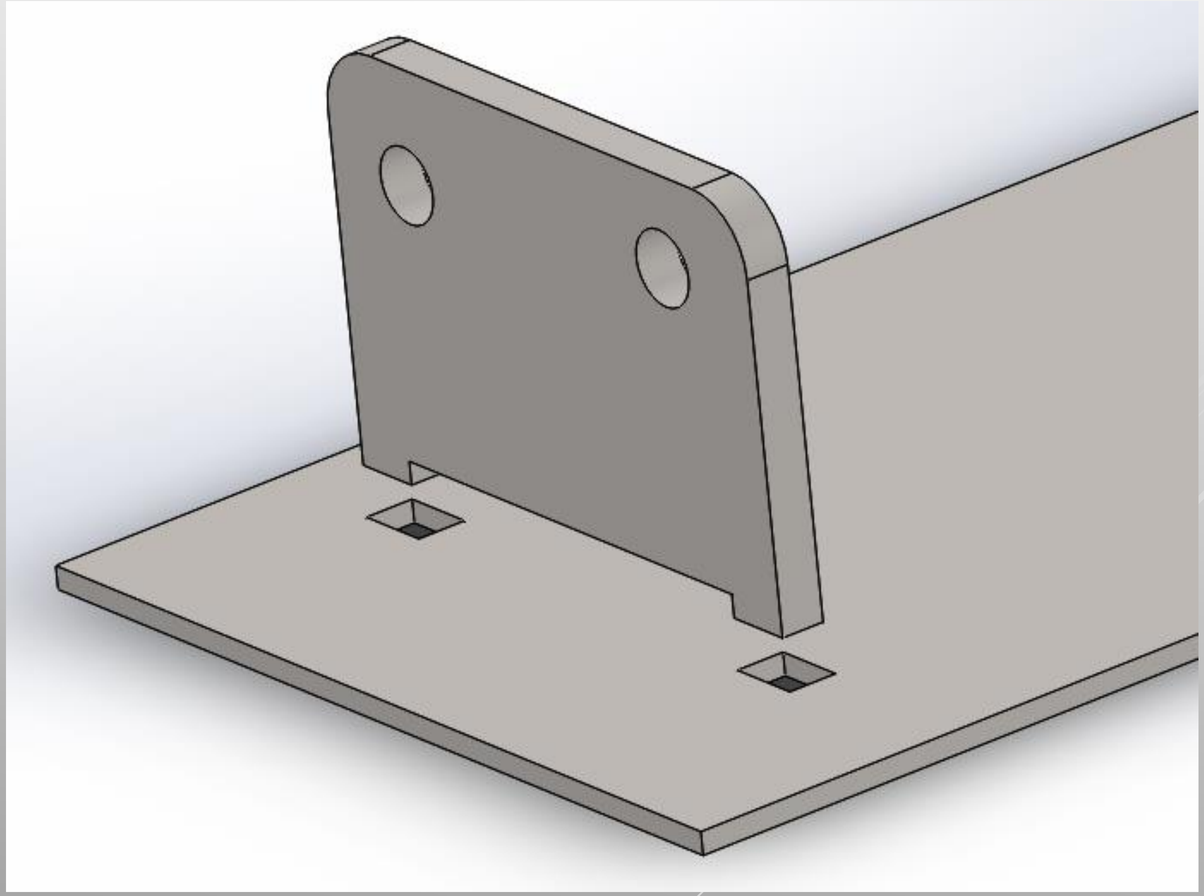






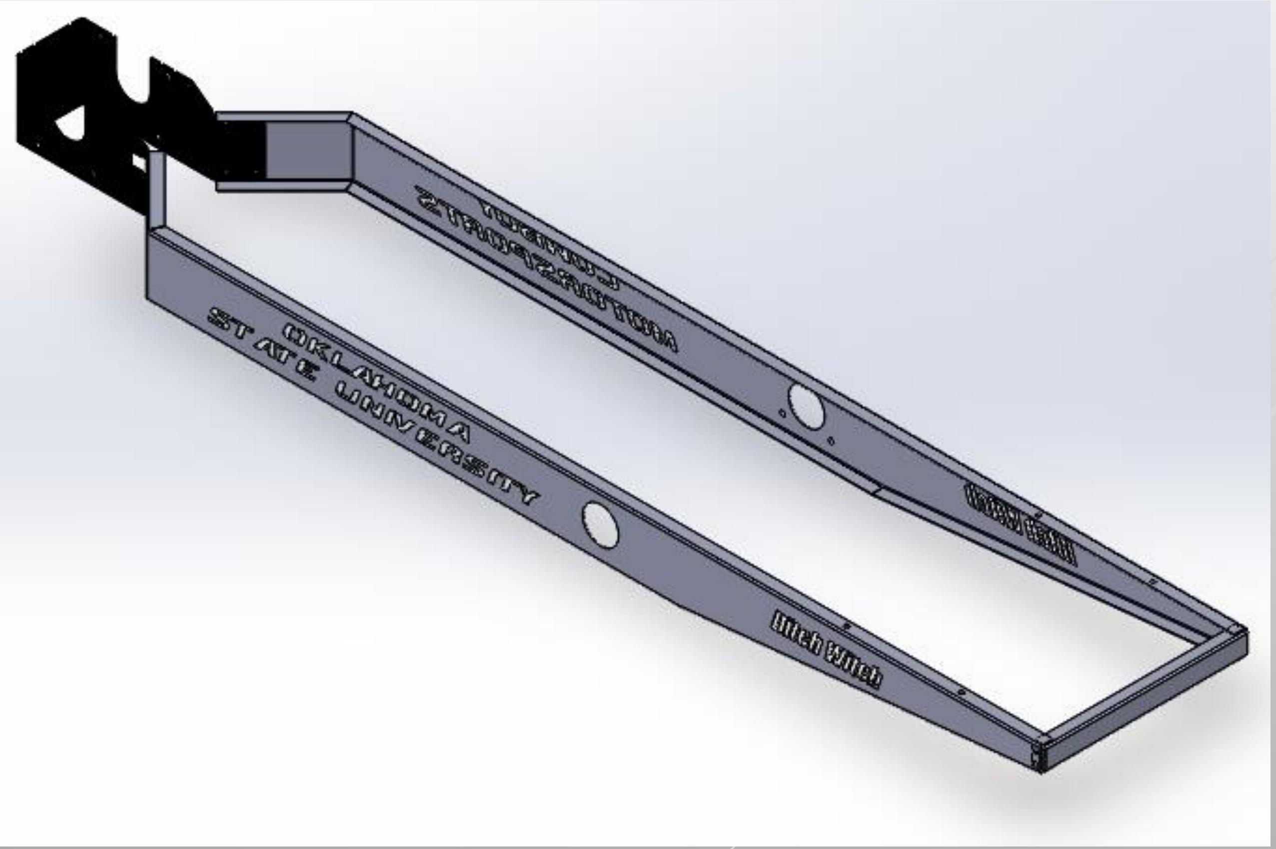
FRAME SELECTION

- C-channel System
  - Lightweight
  - Proven
- Easily Manufactured
  - Slot and Tab
  - Welded
  - Bolt on major components



PREVIOUS DESIGN

- 14 Gauge Steel
- 5" tall, 1" top and bottom flange
- 17" wide, 91" long
- 45° bends at rear
- Bolted together
- No additional support structures

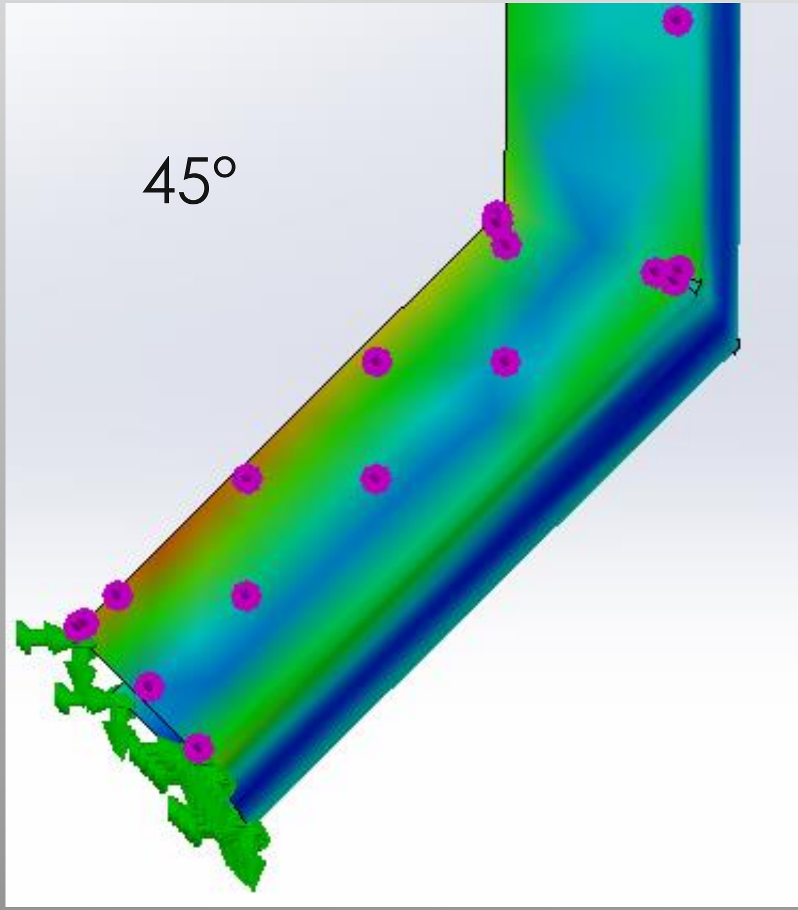


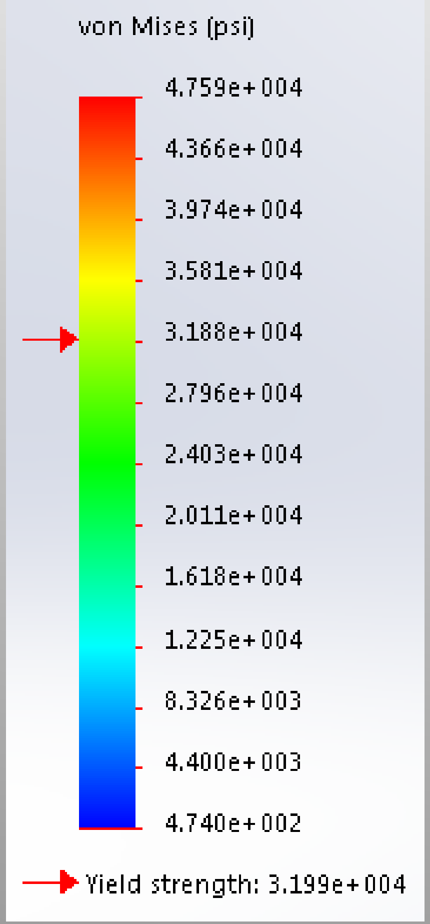
PREVIOUS DESIGN FAILURES

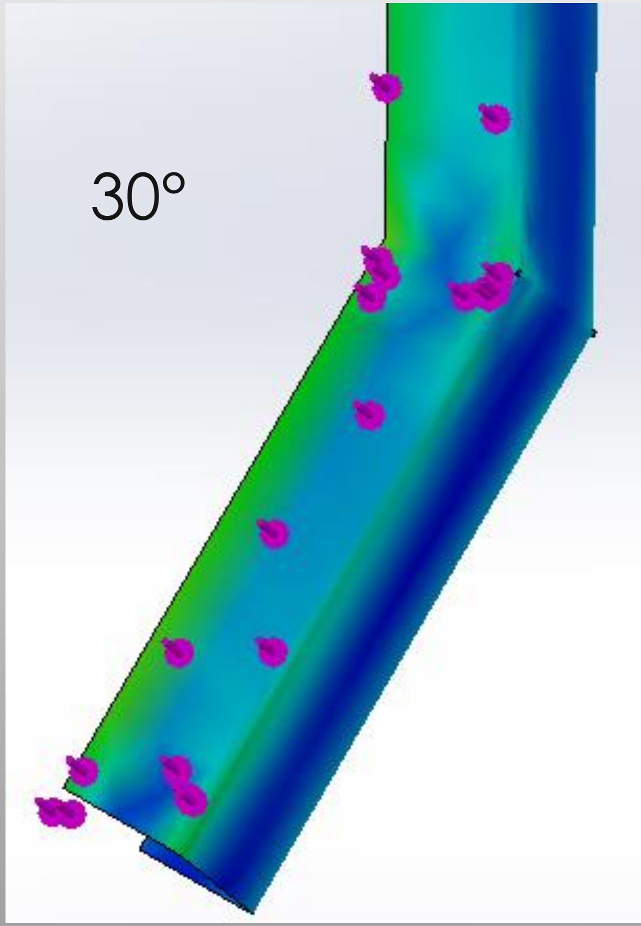
- Began cracking at 45 degree bends
- Stress concentrations due to sharp corner
- Could have been strengthened by welding the gaps

NEW DESIGN: REAR END

- Angle reduced from 45° to 30°

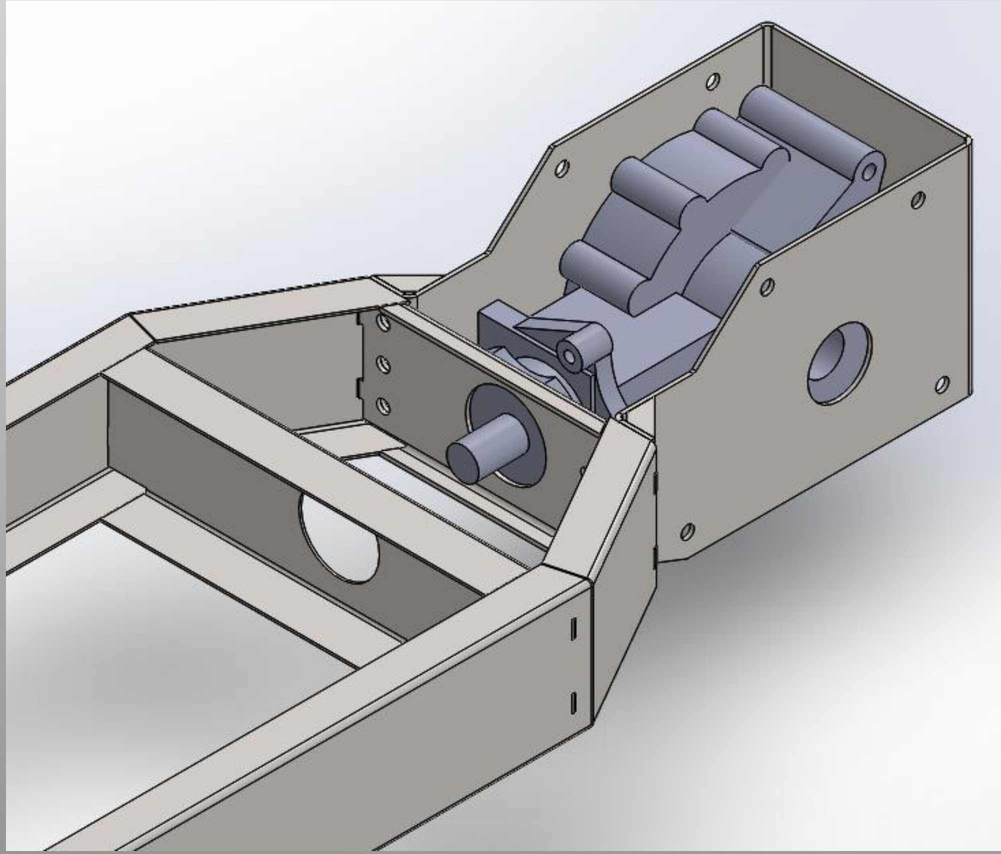


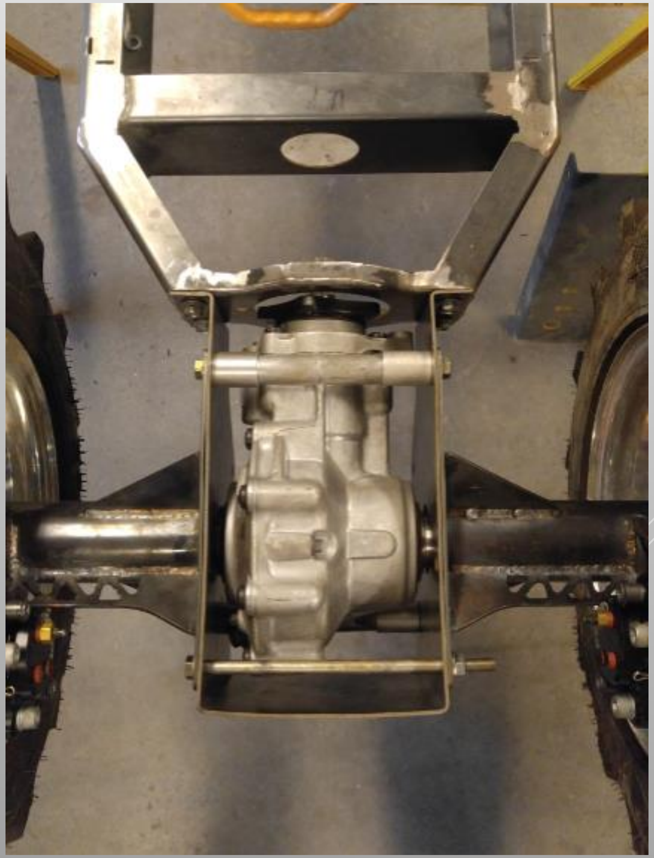




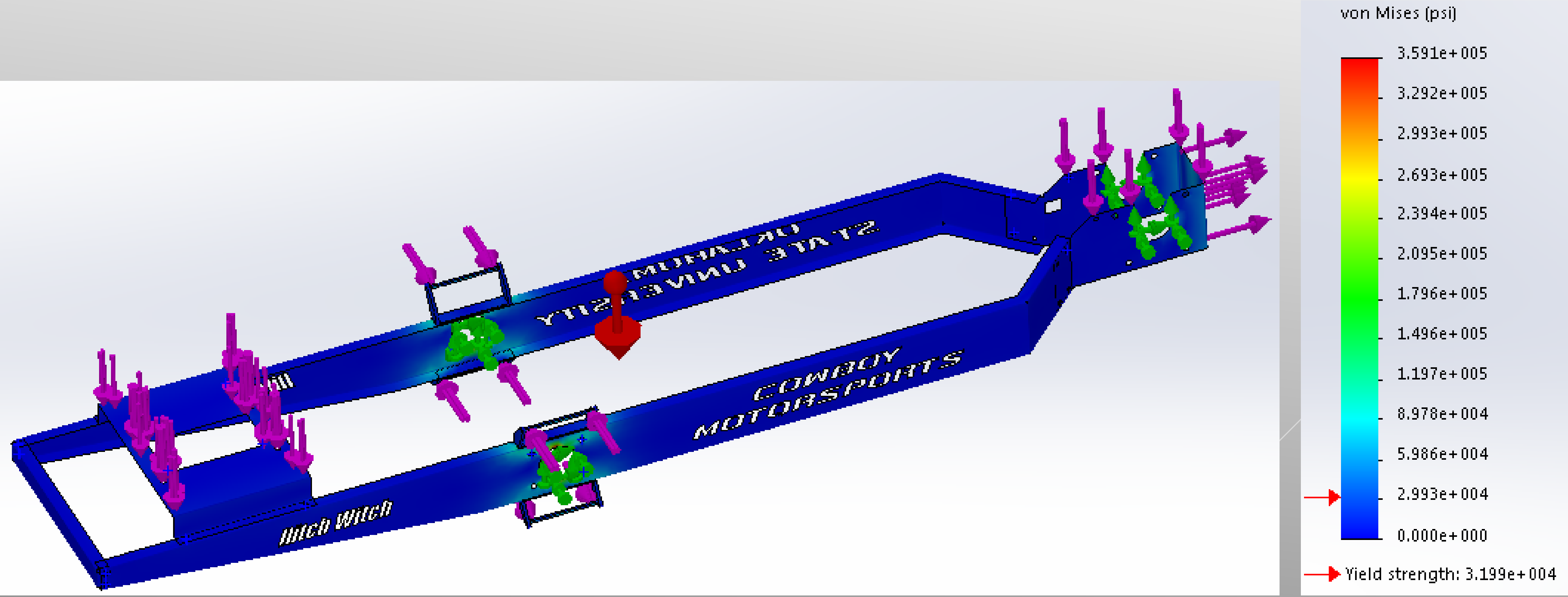
NEW DESIGN: REAR END

- Cross members to box in weak point
- Bolted Connection: Six 3/8" Grade 8 UNC Bolts

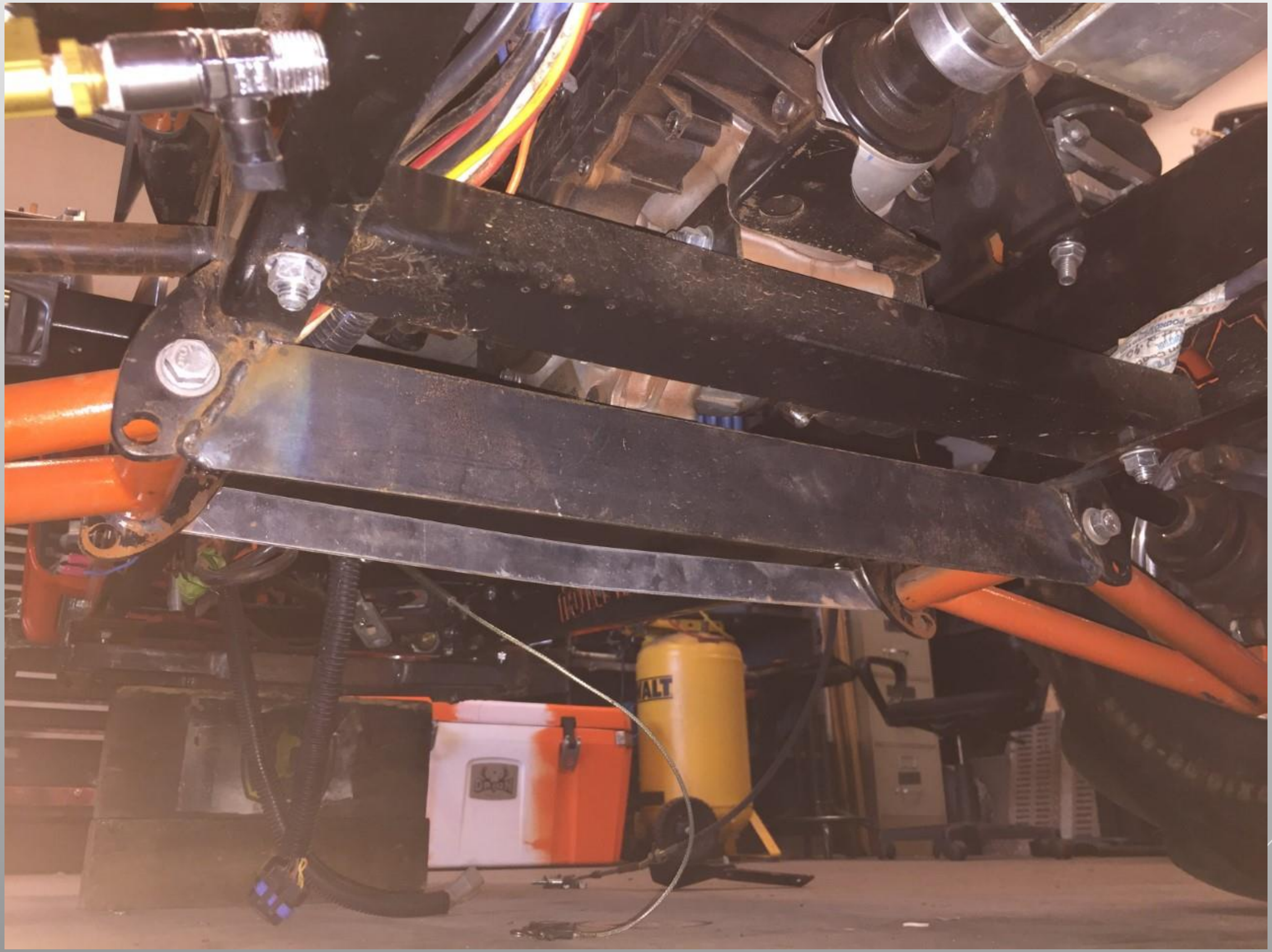




PREVIOUS DESIGN FAILURES

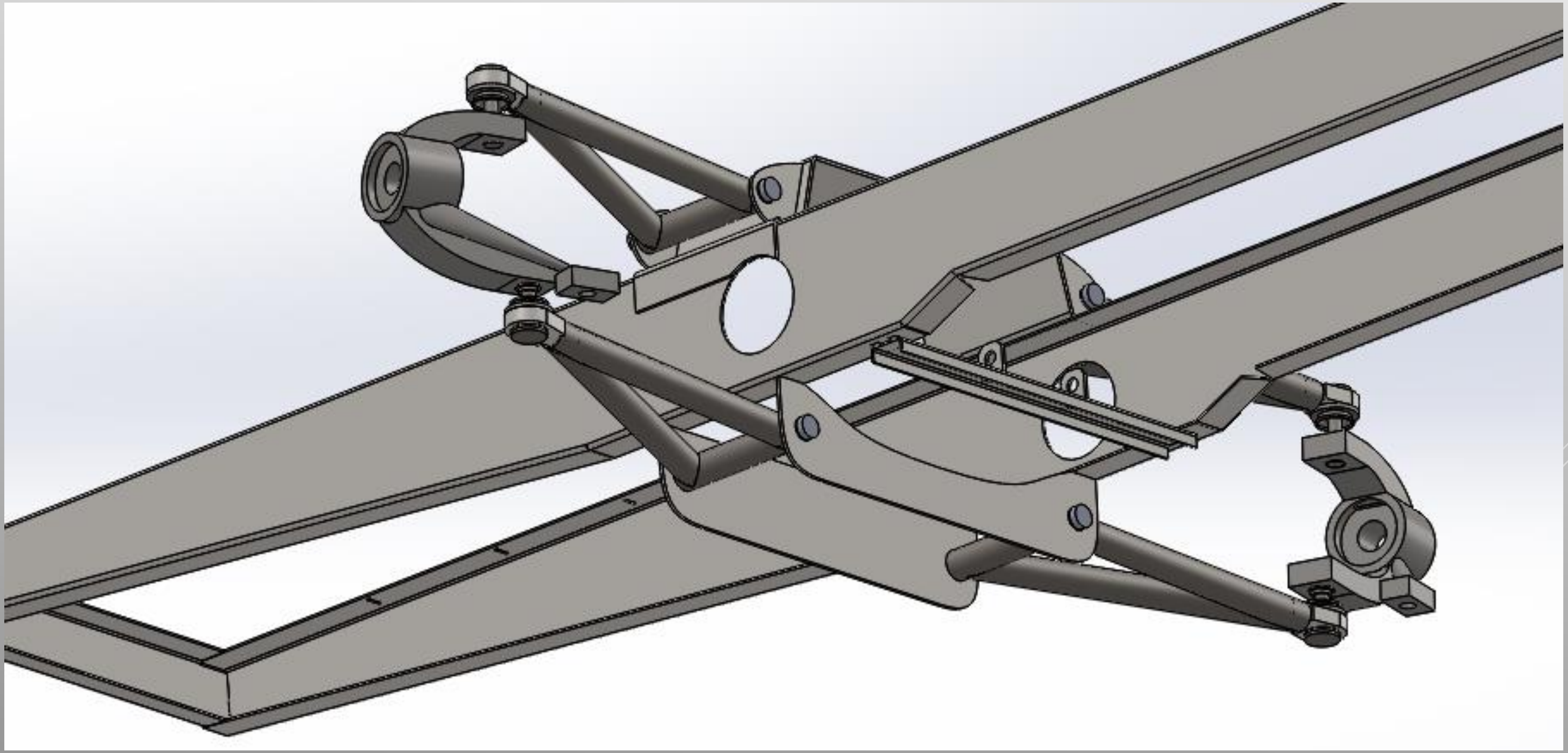


OLD DESIGN: FRONT AXLE



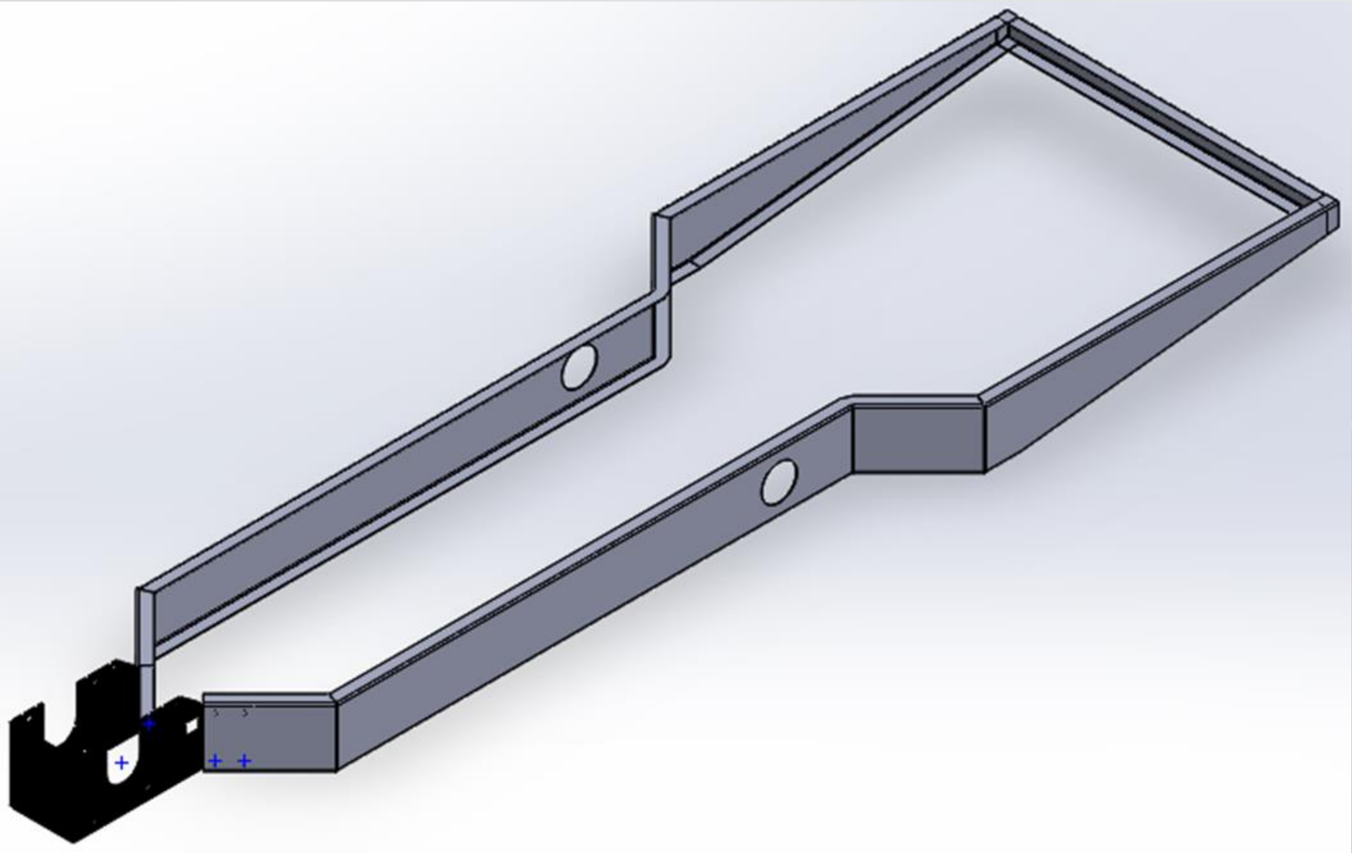
NEW DESIGN: FRONT AXLE

- Incorporated support structures



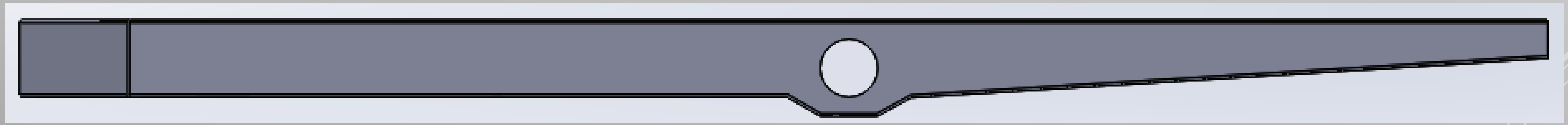
FRAME RAIL SELECTION

- Wide Engine Frame
  - Designed to lower the engine
  - Decided to not lower the engine



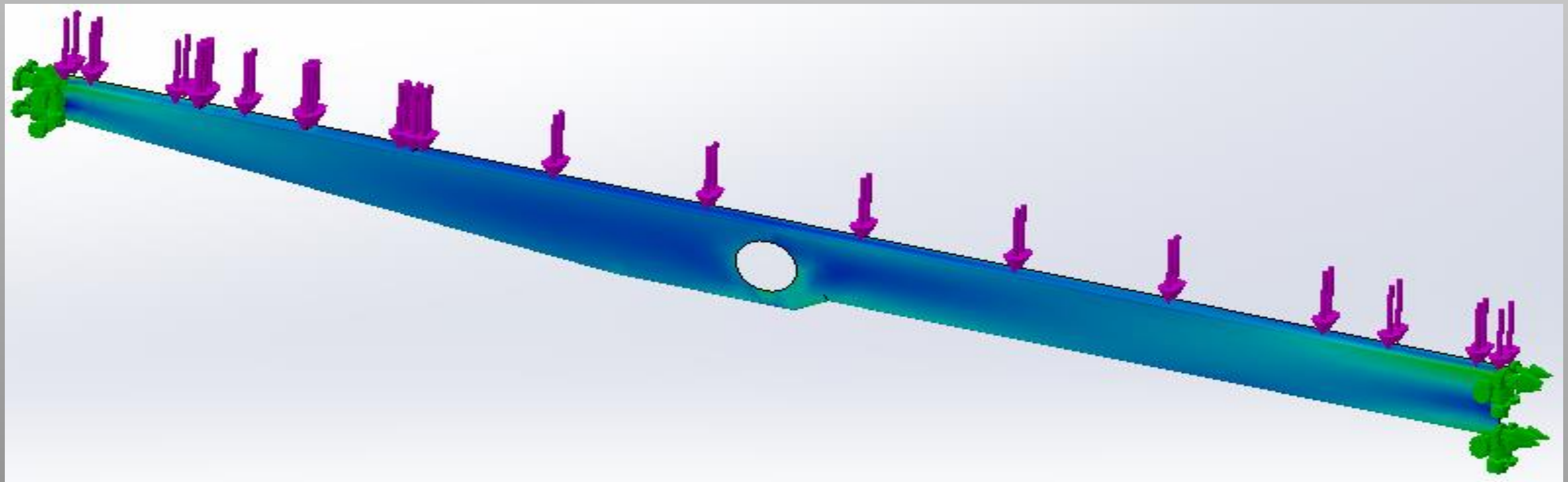
FRAME RAIL SELECTION

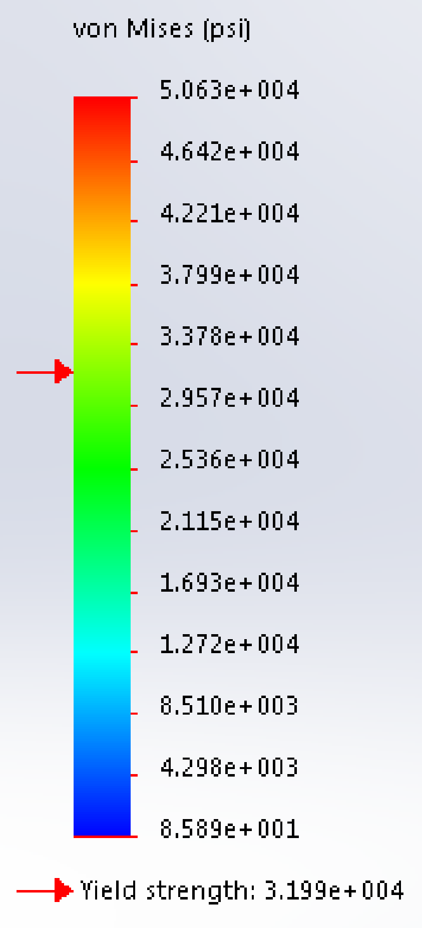
- Short Frame
  - Designed to reduce material
  - Did not fit with new front axle design



FRAME RAIL SELECTION

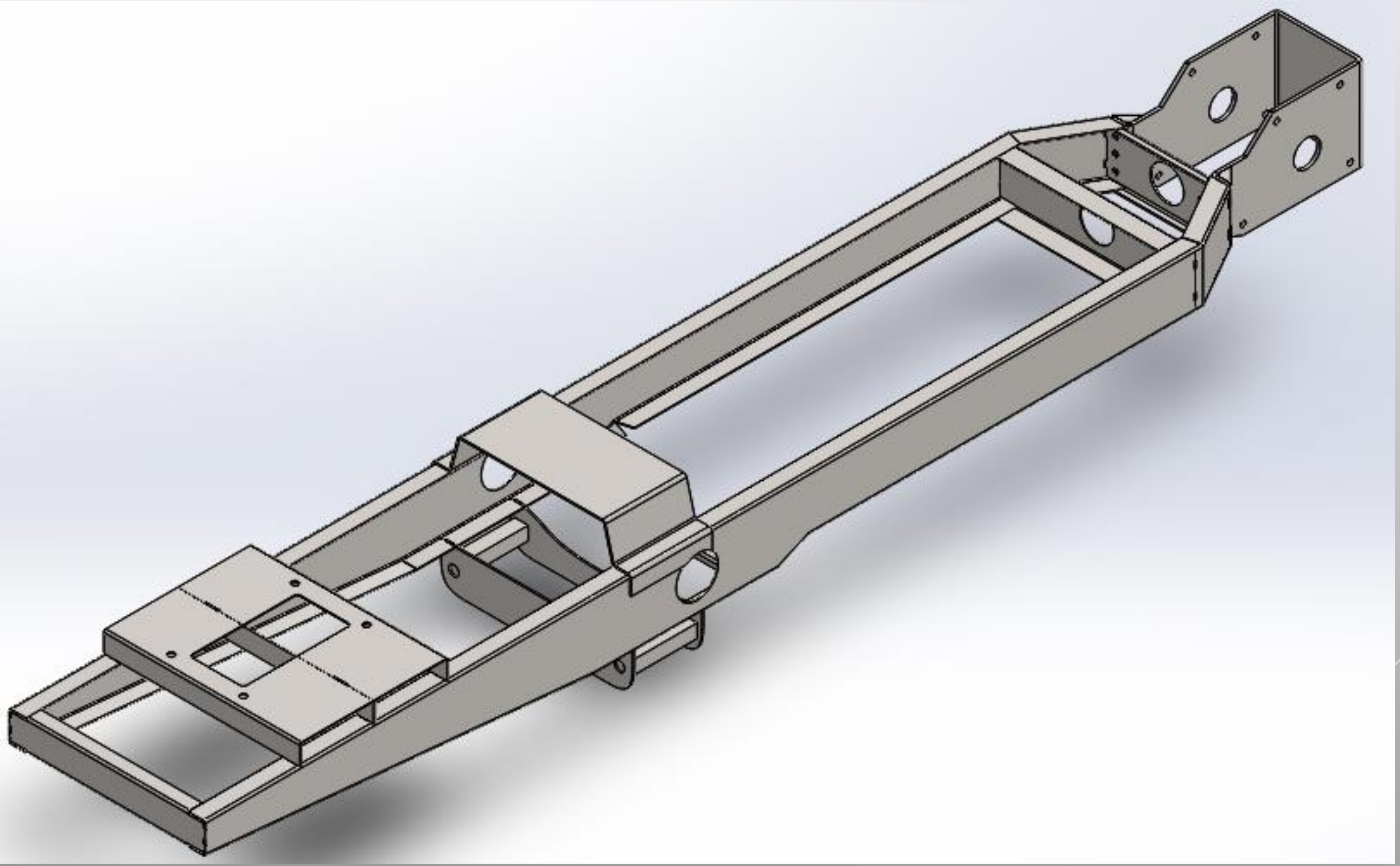
- Height decreases after front axle from 5" to 4"
- 78.5" long
- 14 gauge steel



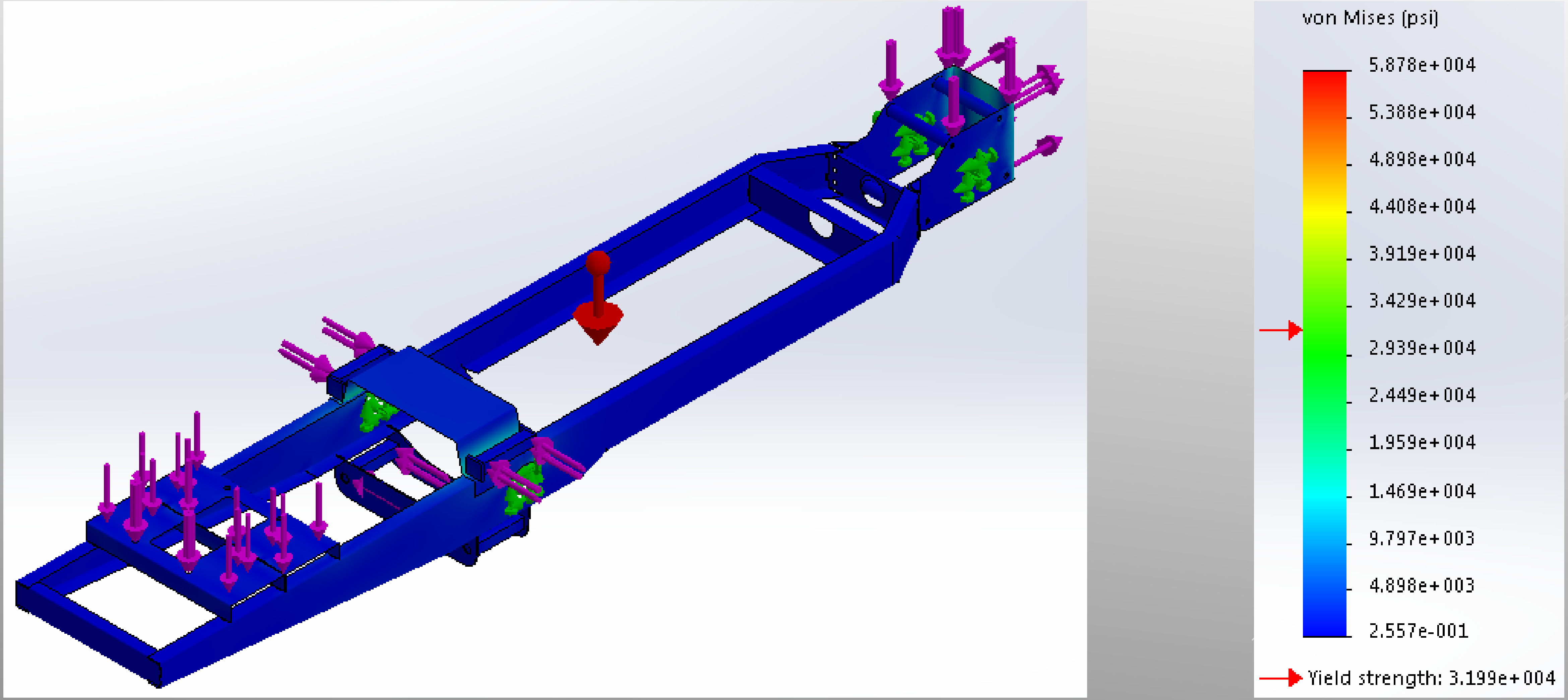


OVERALL ASSEMBLY

- Width reduced from 17” to 14.5” when compared to previous design
- 90” long



OVERALL ASSEMBLY SIMULATION

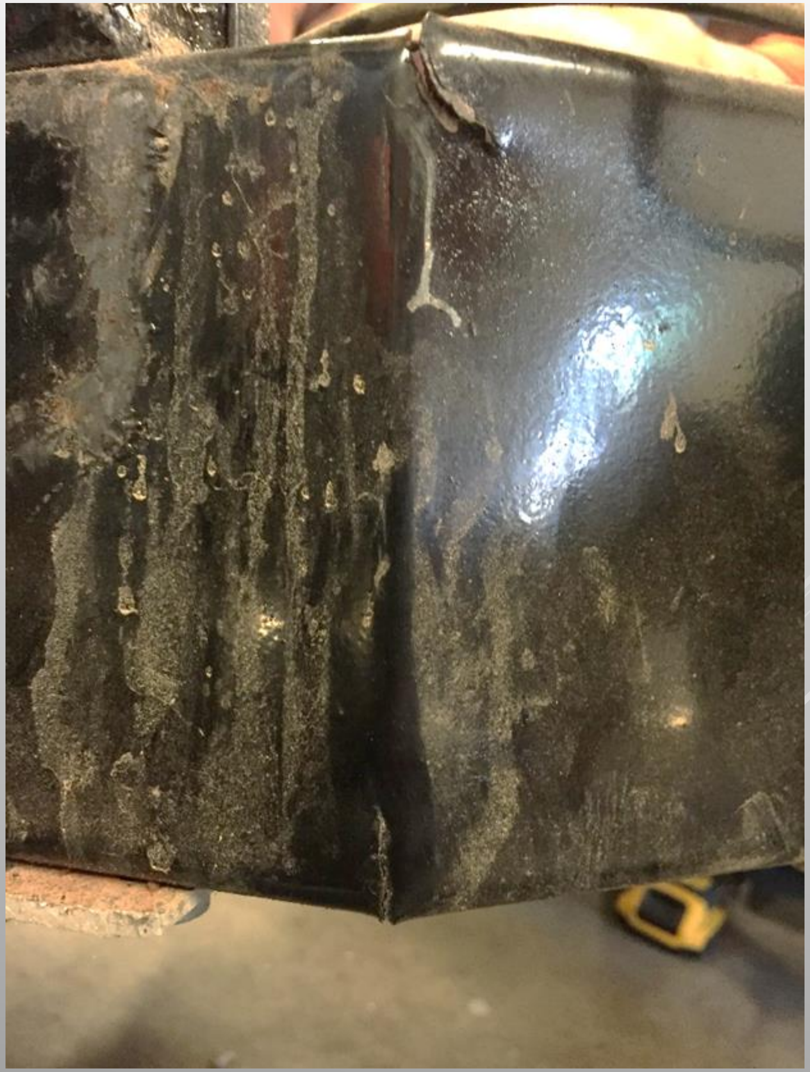


FRAME FABRICATION

- 27 total pieces welded together to make up entire frame assembly
- Took just over a day for BAE lab personnel to complete
- BAE lab personnel liked the slot and tab method, made it easier and faster to put together

FRAME TESTING

- Initial torsion testing showed frame is much stiffer than the previous year's frame
- More testing and observations will be made once tractor is completed
- Success will be no deformities or failures during testing or at competition

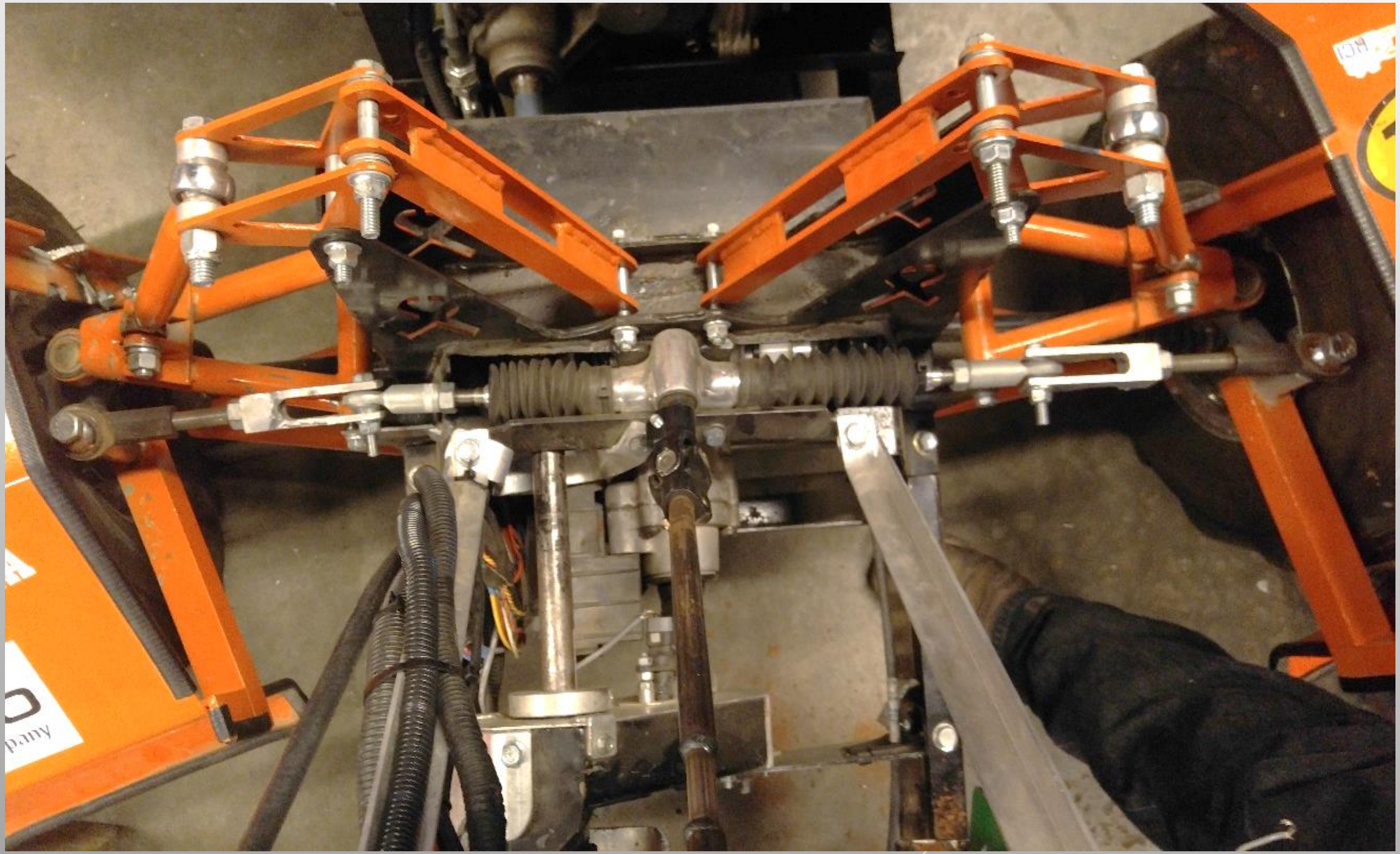


STEERING DESIGN GOALS

- Usability
- Adjustability
- Reliability
- Low maintenance

PREVIOUS DESIGN

- Strengths
  - Manufacturability
  - Simple
  - Lightweight
- Weaknesses
  - 1:1 ratio
  - Heavy steering
  - Poor turning radius



Steering assembly 2015-2016 competition year

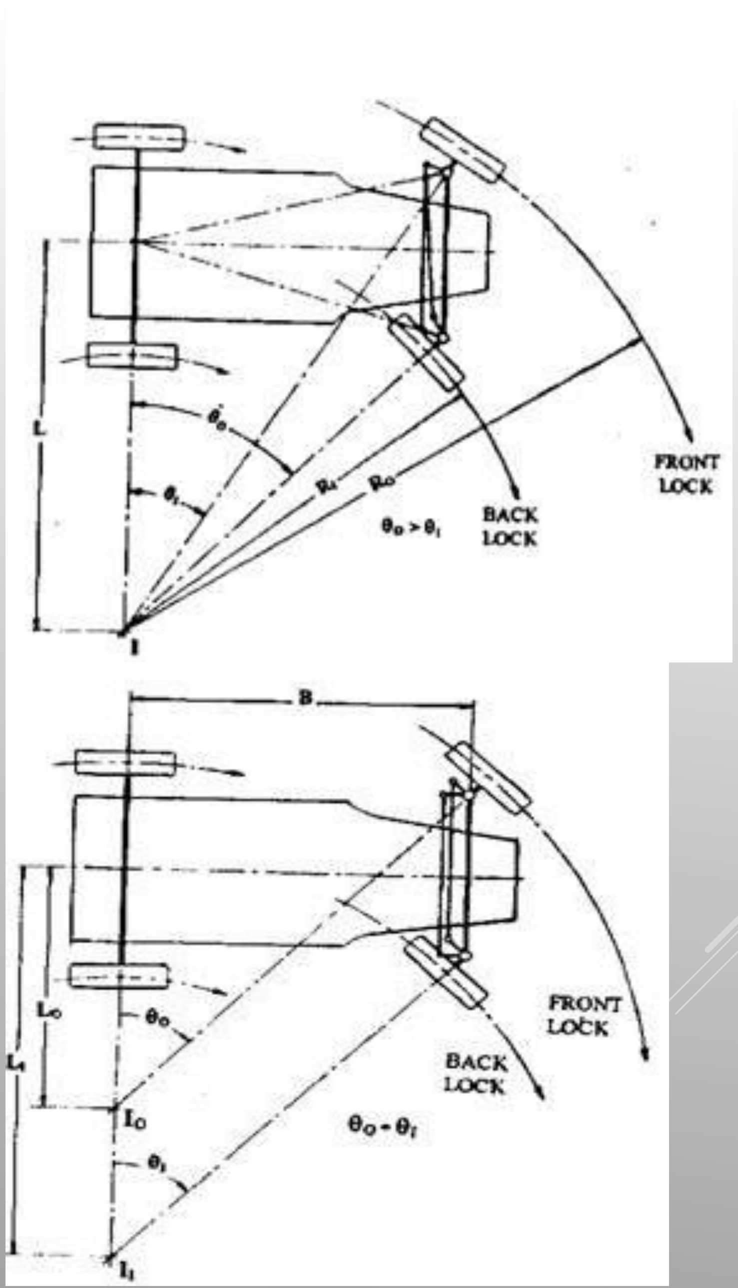
TOE ALIGNMENT PROBLEM

Air springs suspension fully inflated

Air springs suspension at pull height

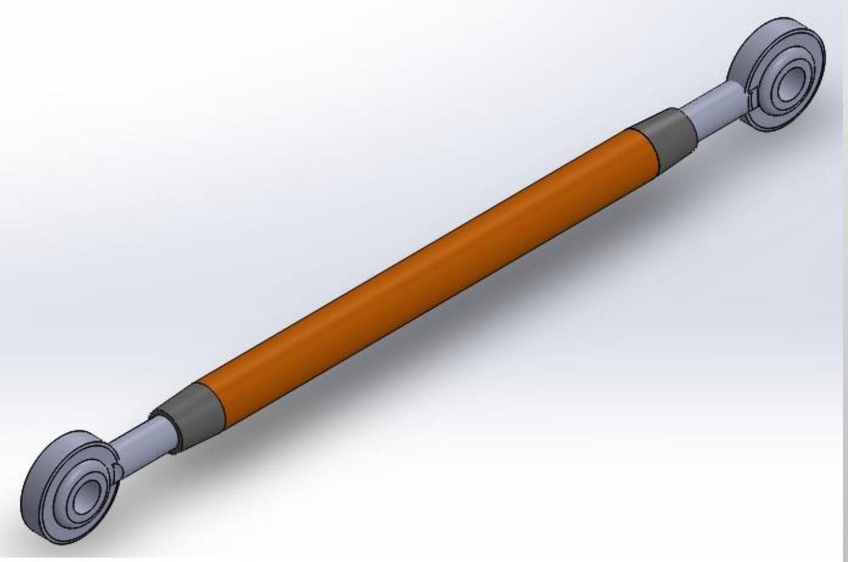
STEERING FACTORS

- Ackermann Geometry
- Parallel Set

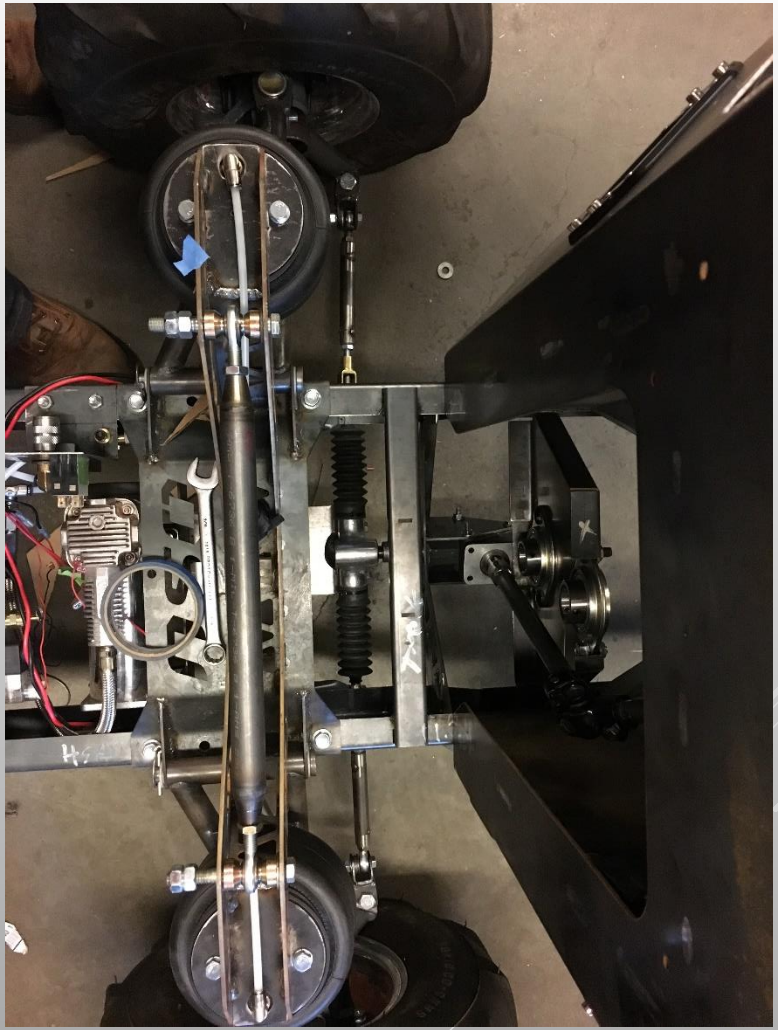


STEERING DESIGN

- Rack and pinion
- Chrome-moly turnbuckles
- Gear reduction
- Larger steering wheel
- Improved geometry

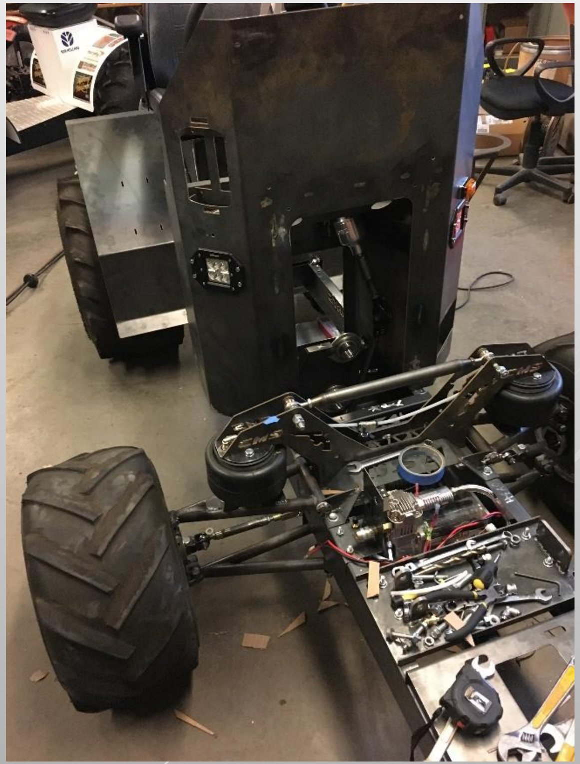


STEERING DESIGN CONT.



STEERING TESTING AND FABRICATION

- Tested gear reduction and noticed significantly decreased effort for turning
- More testing will be conducted as the tractor nears completion
- A reduction in overall effort to steer will signify a success



SUSPENSION OBJECTIVES

- Ride Height Adjustment
  - Scales, Brake test, Maneuverability, and Pulling
- Improve Ride Quality
  - Operator comfort and improve durability

PREVIOUS DESIGN

Rigid Suspension Lessons Learned

- Manually adjustable
- Light weight
- Limited potential travel
- No articulation
- No damping

INITIAL CONCEPTS

- Coil over shock absorber
- Linear actuators
- Hydraulic cylinders
- Air shocks
- Air springs

INITIAL CONCEPTS CONTINUED

Selection Criteria

- Cost
- Weight
- Strength
- Pulling performance
- Durability
- Adjustability
- Ride quality

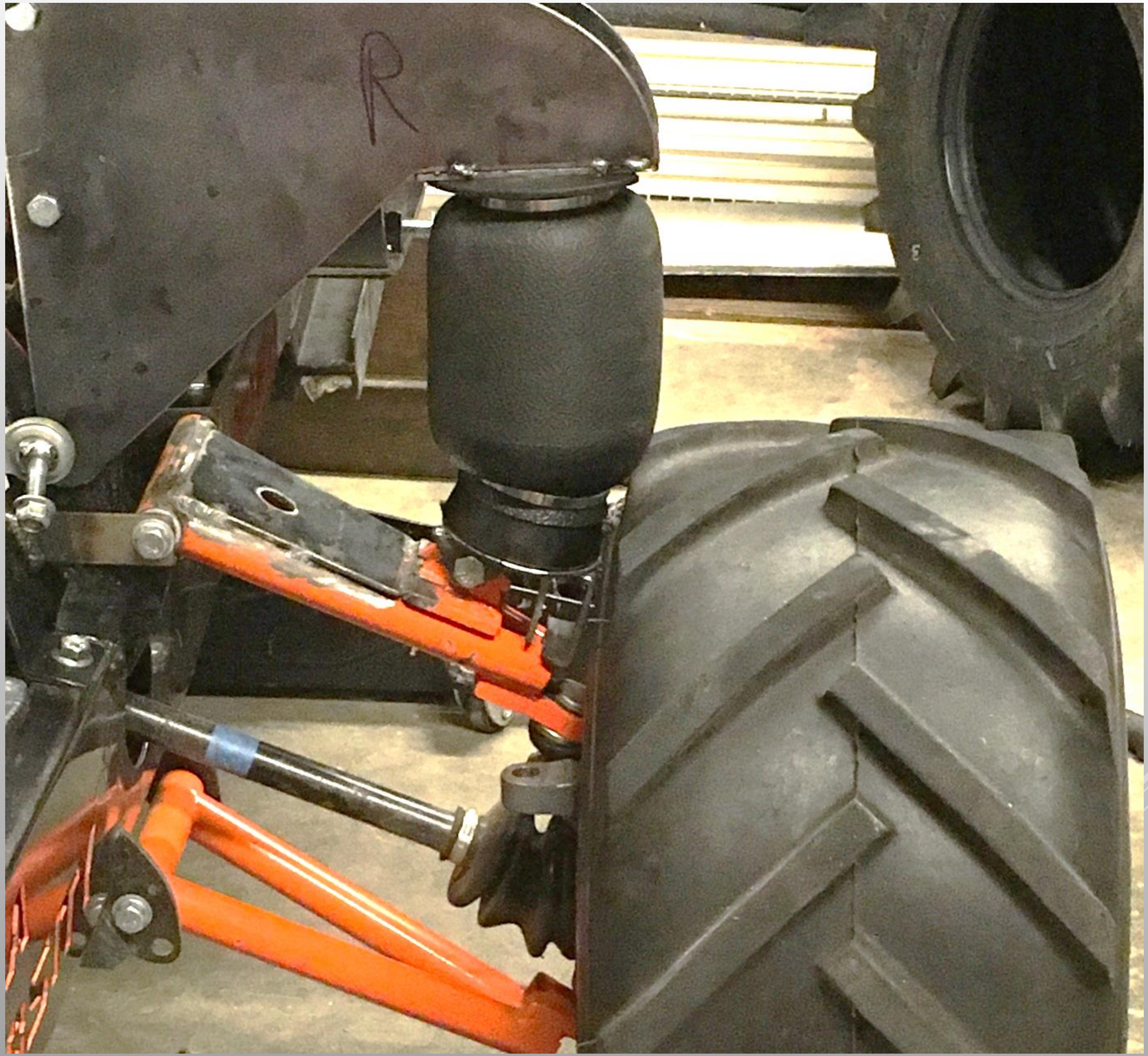
		Concept 1		Concept 2		Concept 3	
		Hydraulic Cylinders		Air Shocks		Air Springs	
Criteria	%	Rank	Score	Rank	Score	Rank	Score
Cost	15	1	15	3	45	3	45
Weight	20	1	20	1	20	2	40
Strength	10	3	30	1	10	2	20
Pulling Performance	15	3	45	2	30	1	15
Durability	15	3	45	2	30	3	45
Adjustability	10	3	30	3	30	3	30
Ride Quality	15	1	15	2	30	2	30
Total	100		200		195		225

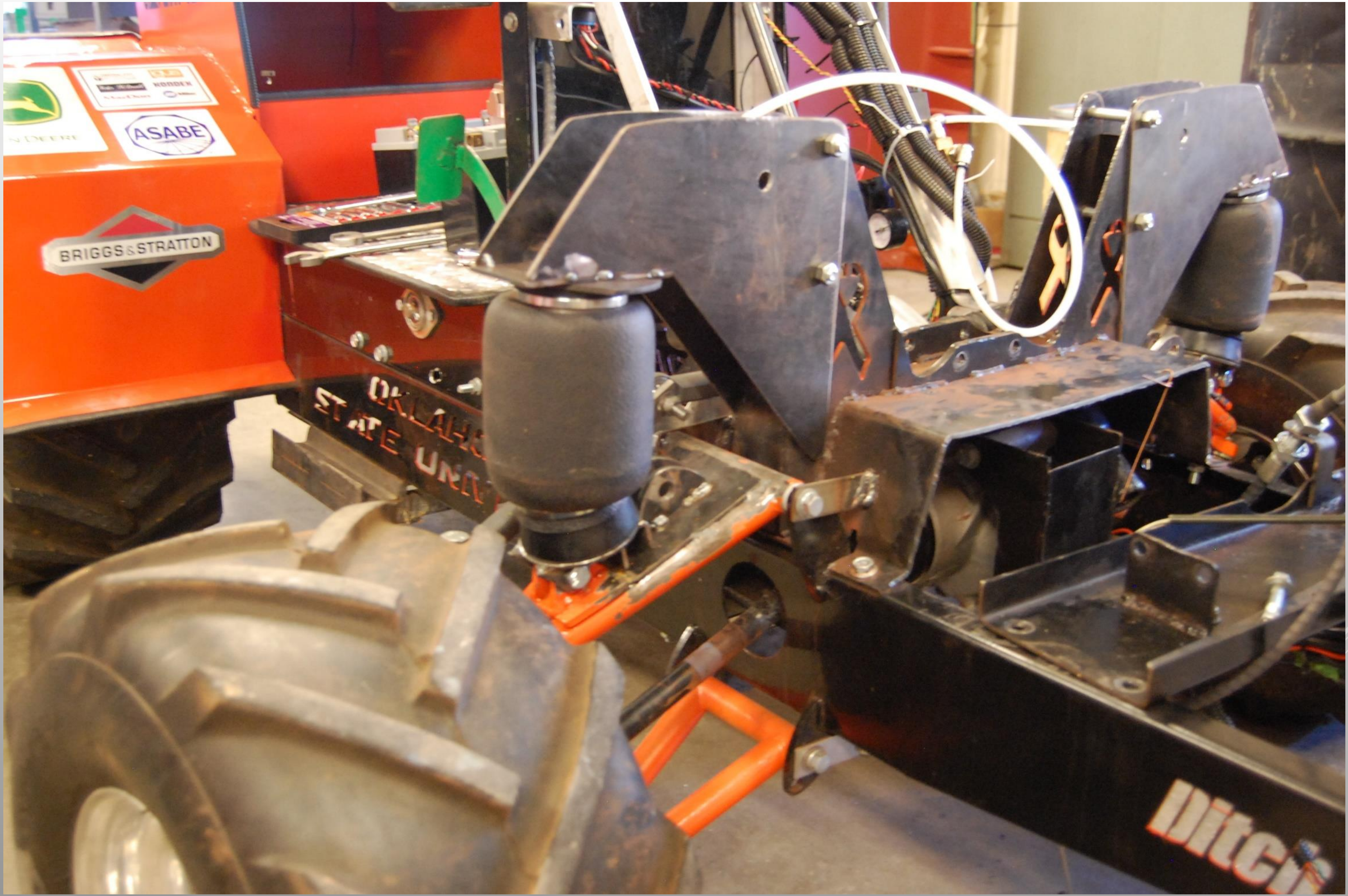
3 = Best in Category

1= Worst in Category

TESTING

- First Iteration
  - Overloaded
- Second Iteration
  - Clearance
- Third Iteration
  - Working prototype

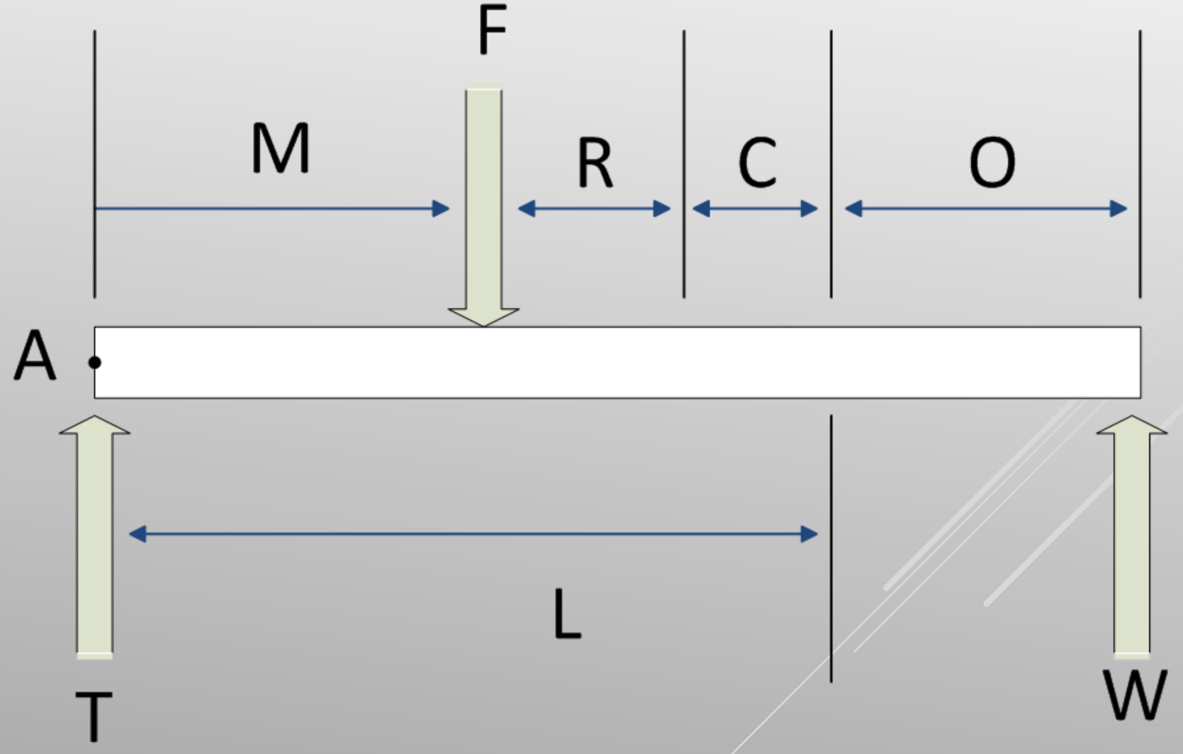



ASABE
BRIGGS&STRATTON


ASABE

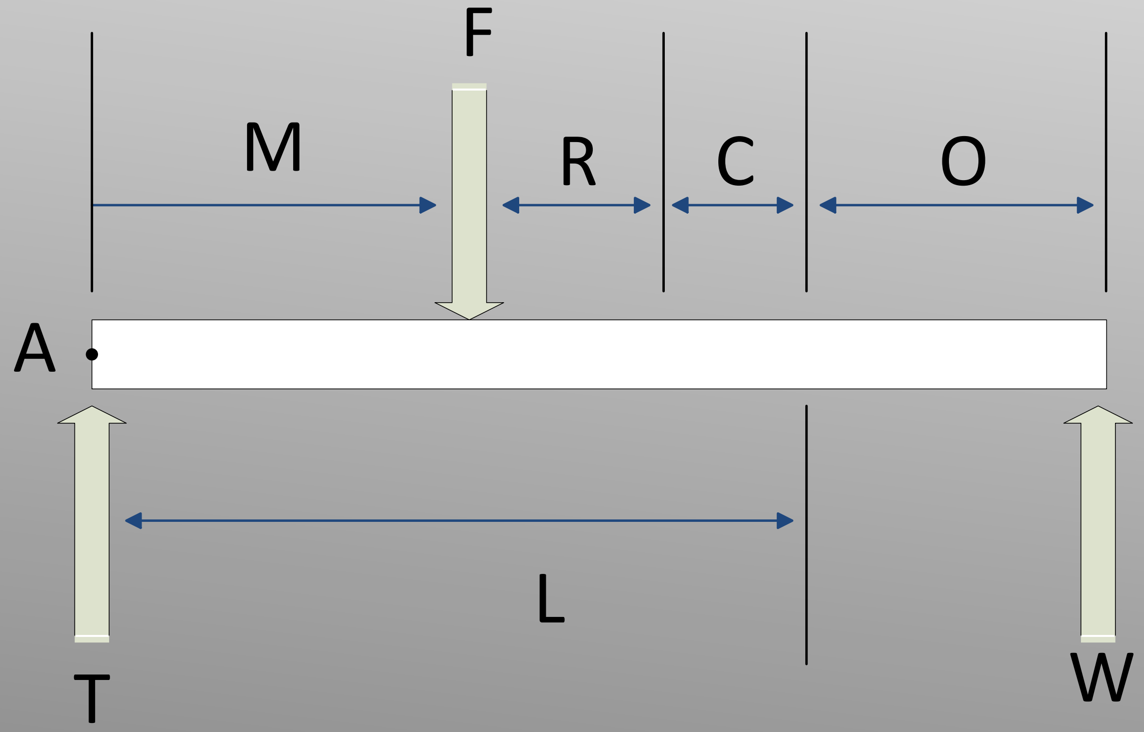
AIR SPRING SELECTION

- $M_A = 0 = (W)*(L+0) - (F)*(M)$
- $F = (W)*(L+0) / M$
- W= Reaction weight on each front tire
- T=Reaction weight on the tractor side
- L= Length of A-arm
- F= force required to lift the tractor
- M= distance from center of air spring to center of A-arm pivot point



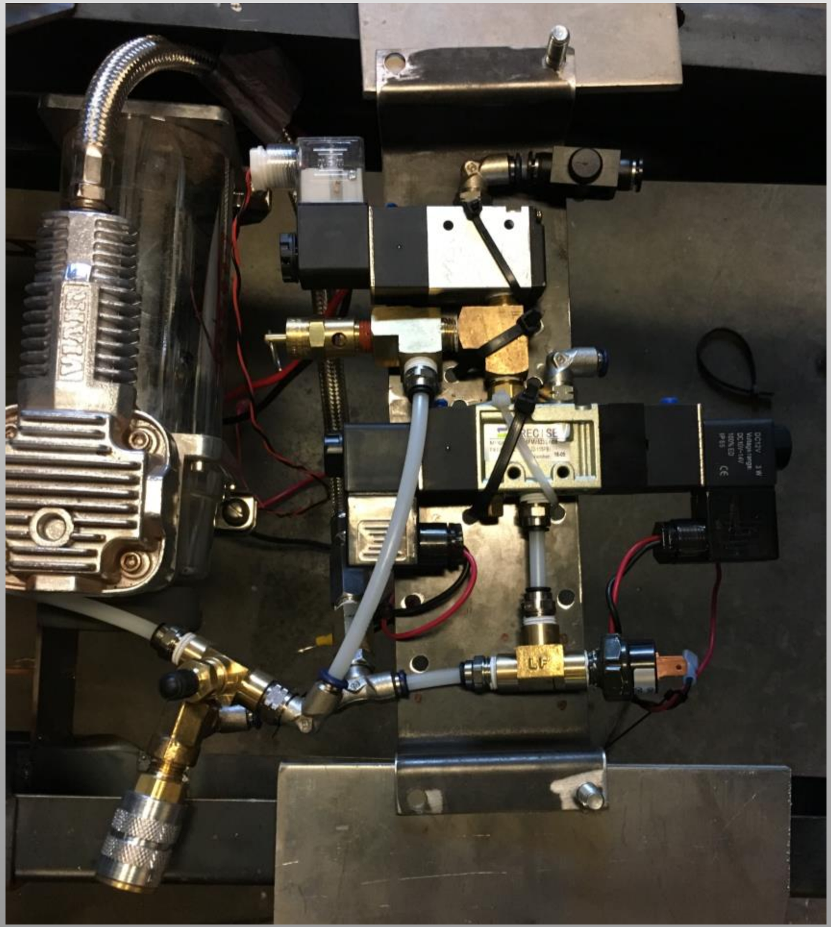
AIR SPRING SELECTION

Part number	Max load at 100 Psi	Max diameter (in)	R (in)	M (in)	Force needed (Lbf)	Safety factor
58407	2210	7	3.5	5.64	2144.7	1.03
58124	3340	9.4	4.7	4.44	2724.3	1.23
58616	3055	8	4	5.14	2353.3	1.30

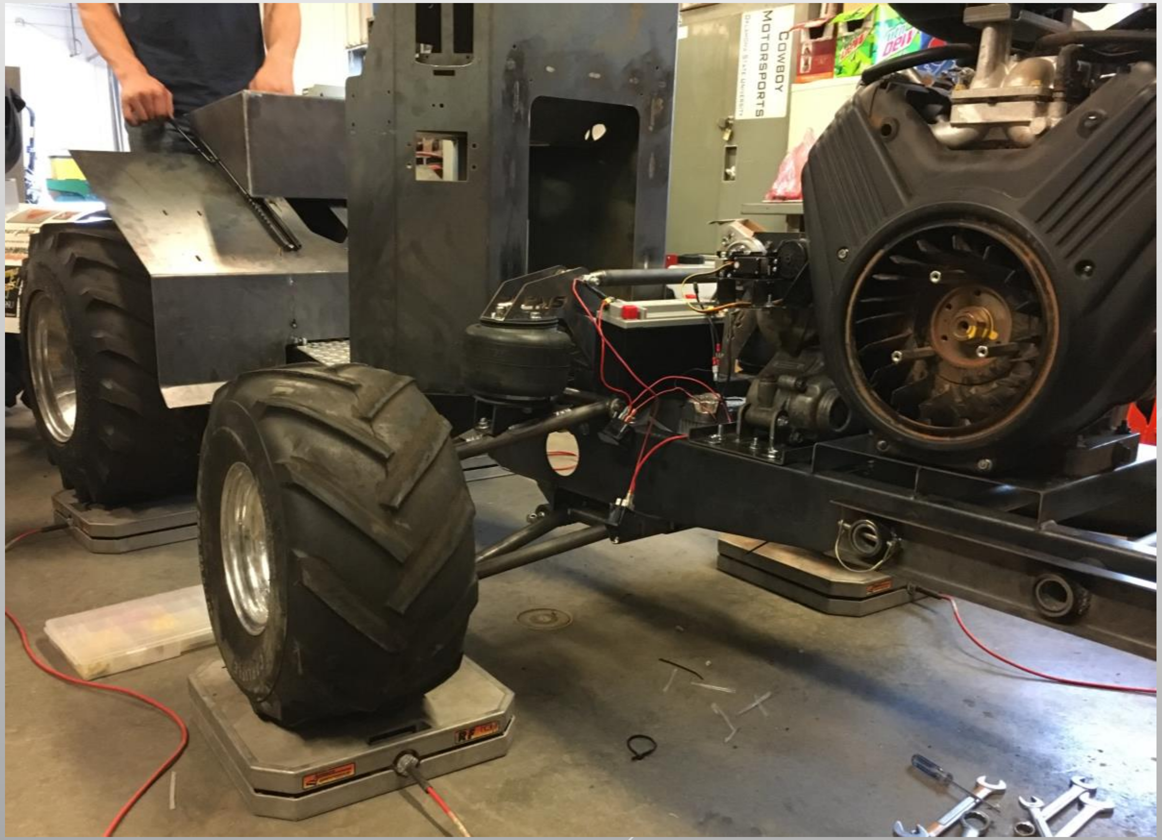


L (in)	O (in)	C (in)	W (Lbf)
11.64	5.64	2.5	700

SUSPENSION TESTING



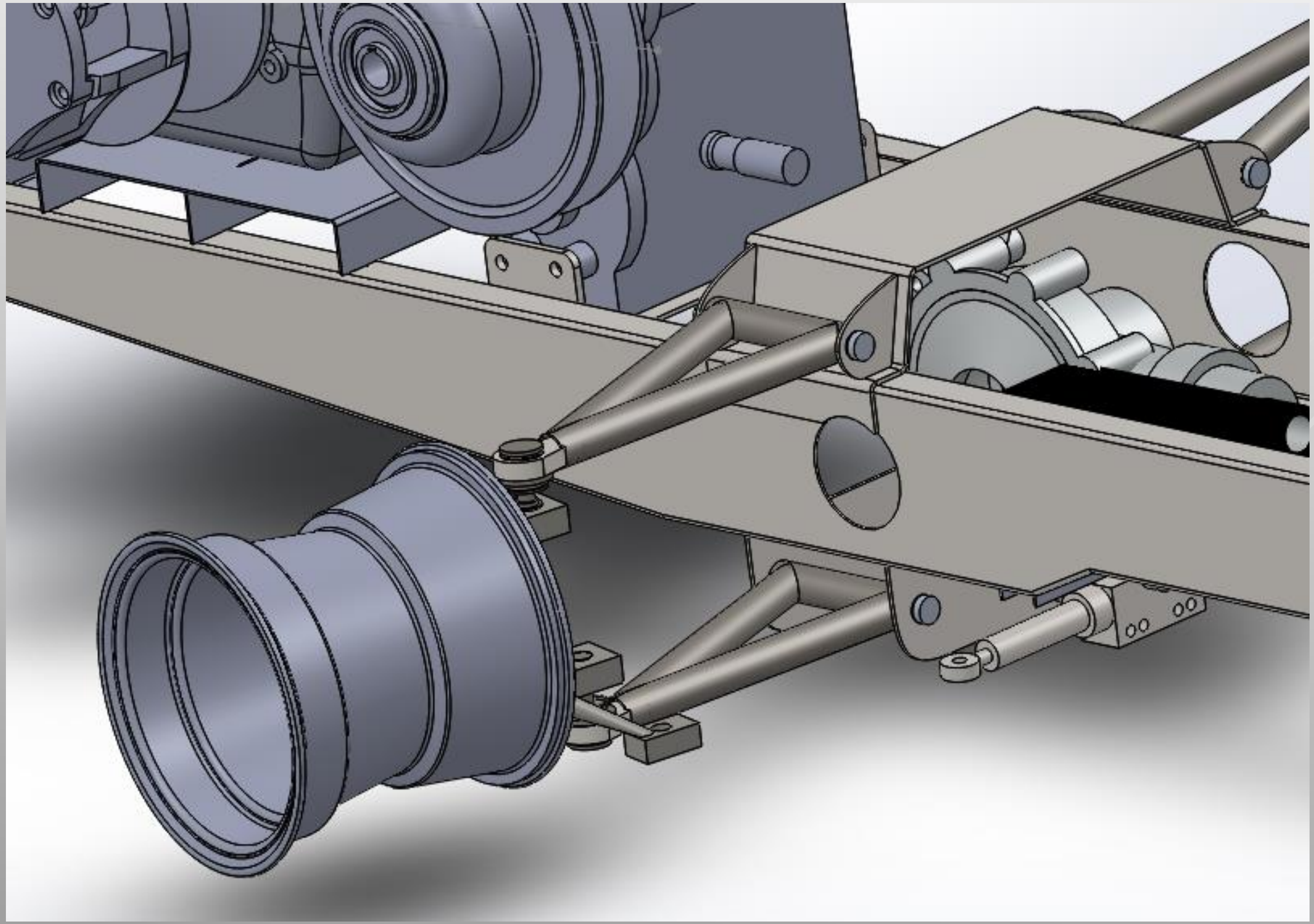
SUSPENSION TESTING



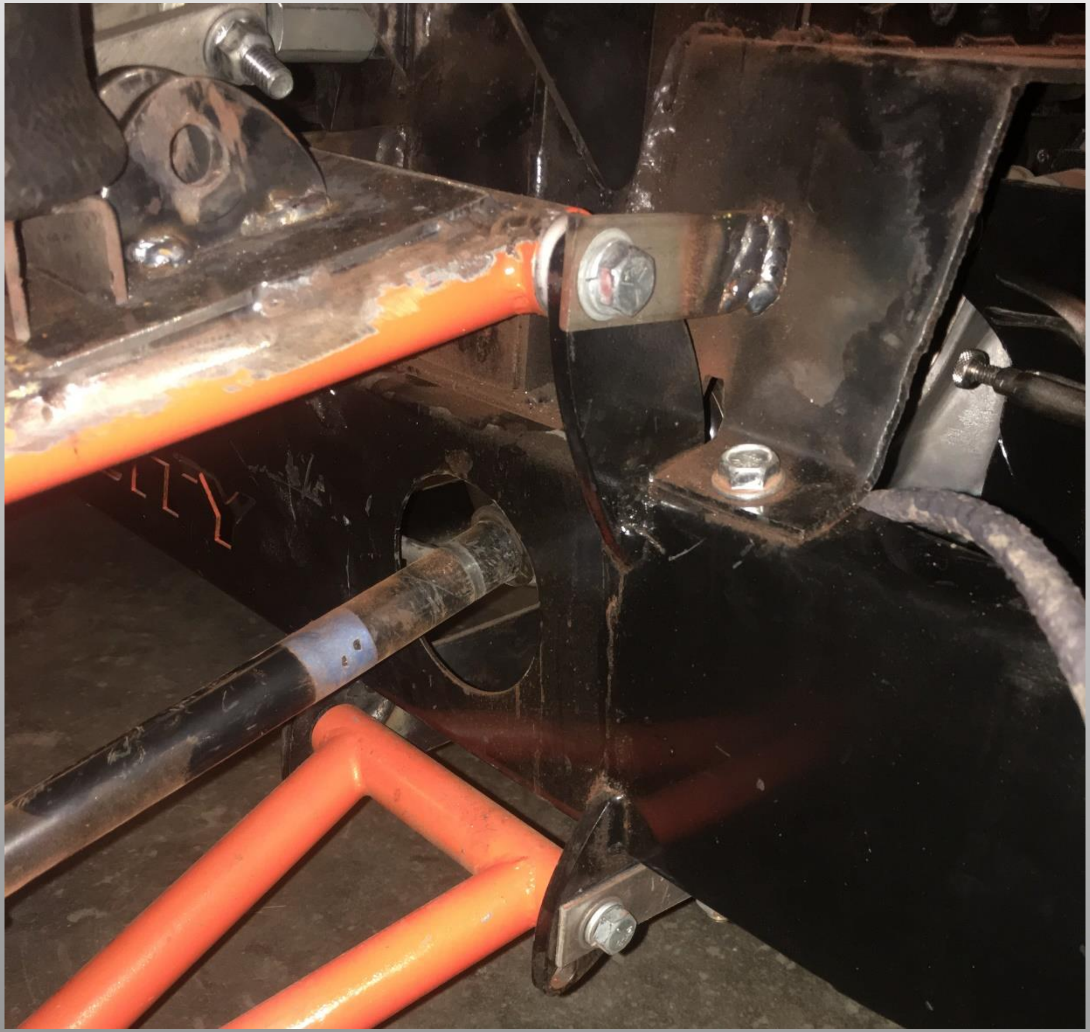

OSU
CMS
CAT

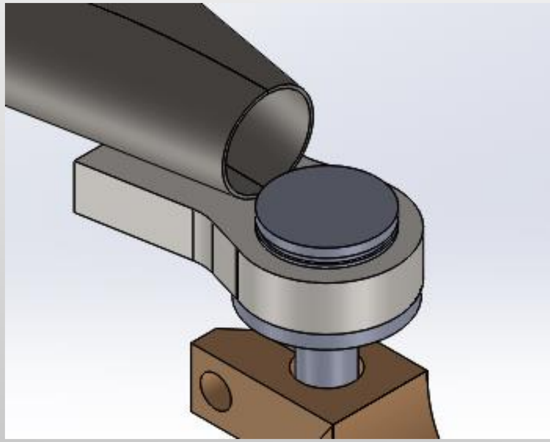
A-ARM DESIGN

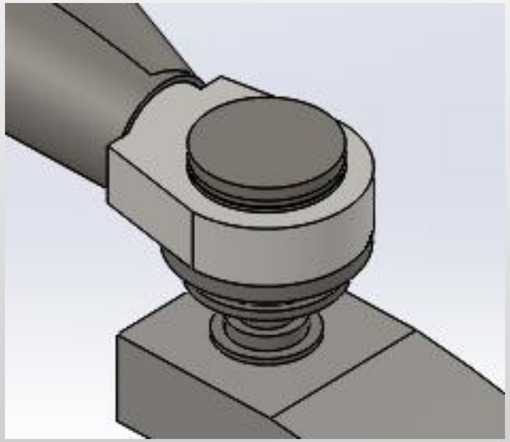
- 1in O.D. Chrome-moly tubing
- Right angle
- Double wishbone
- Improved serviceability
- Improved manufacturability

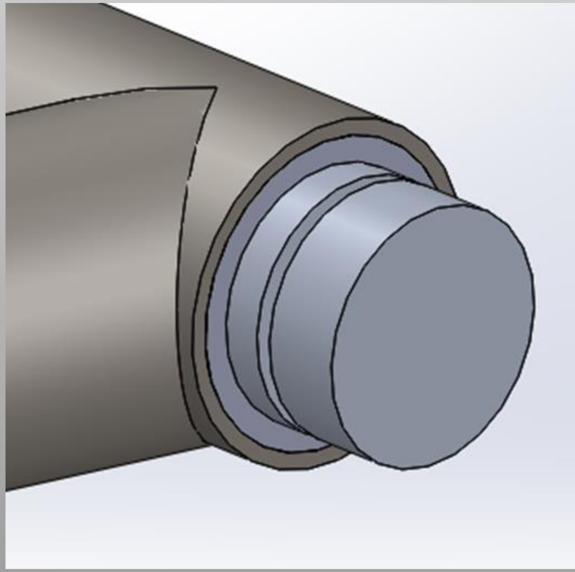


A-ARM DESIGN CONTINUED



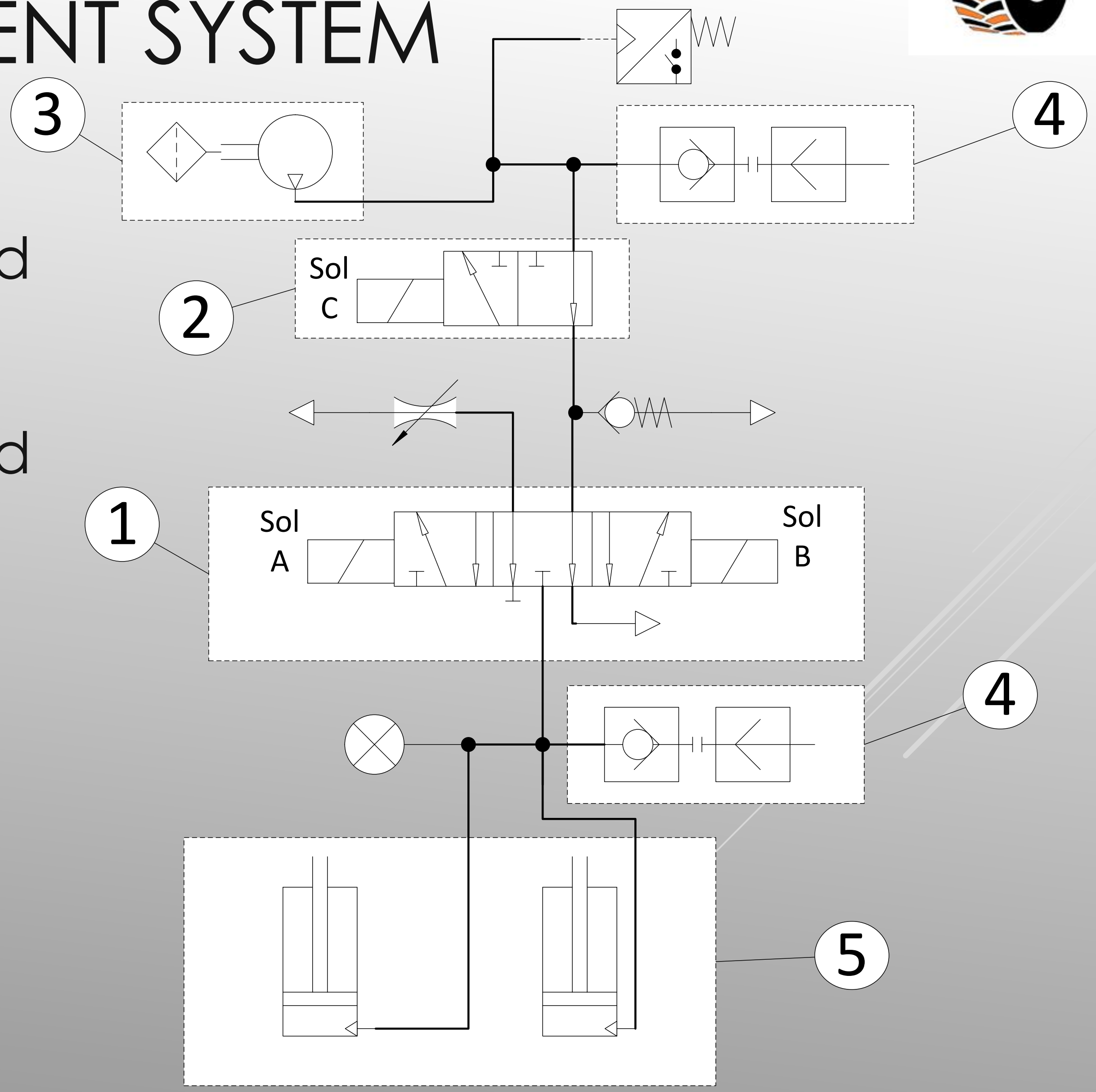






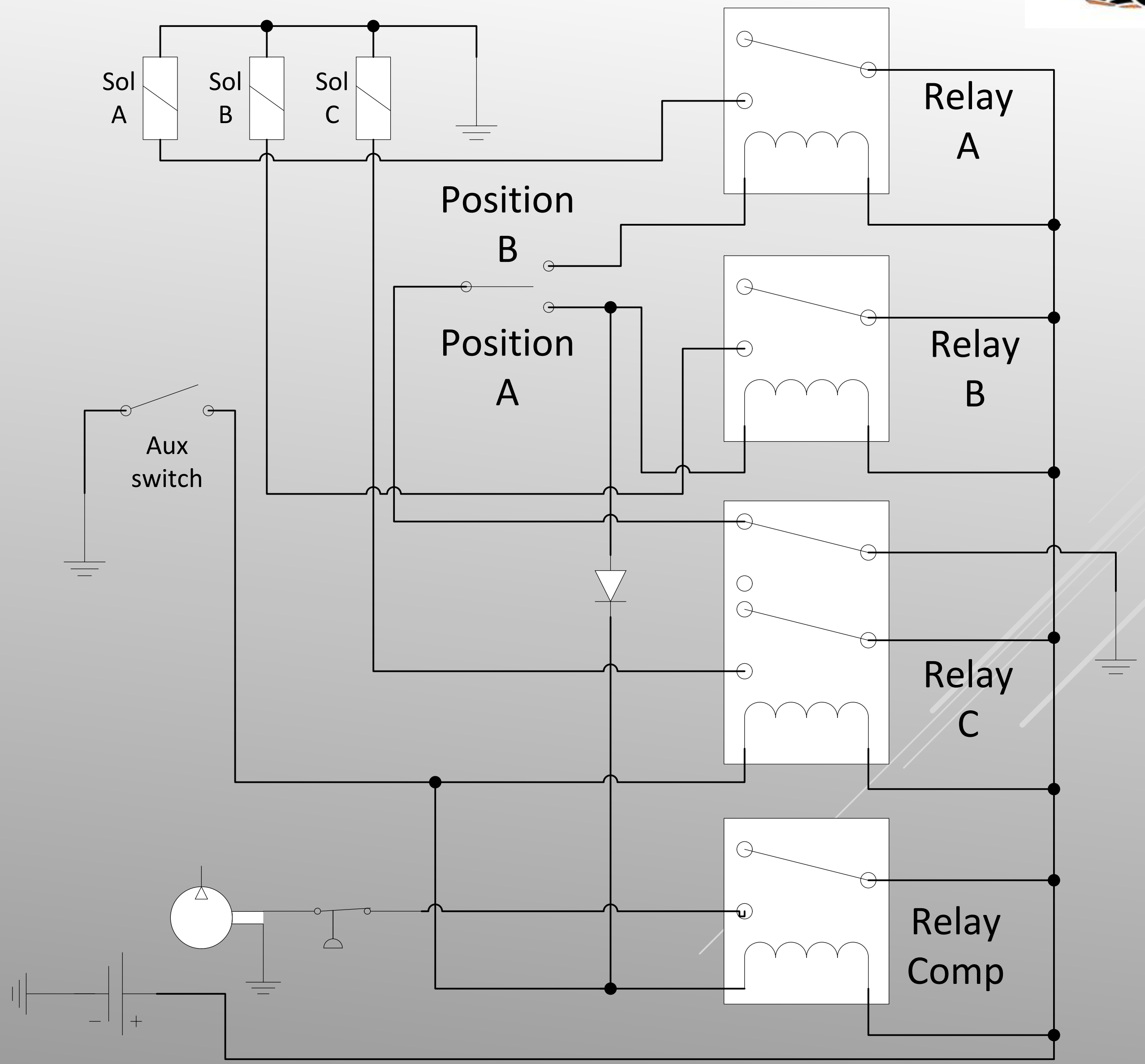
PNEUMATIC MANAGEMENT SYSTEM

- 1: 5 port, 3 way, solenoid controlled pneumatic valve
- 2: 3 port, 2 way, solenoid controlled pneumatic valve
- 3: 200 psi max air compressor
- 4: Auxiliary quick disconnect
- 5: Dual air springs



PNEUMATIC MANAGEMENT SYSTEM CONTINUED

- Inflate air springs
  - Switch position A
- Deflate air springs
  - Switch position B
- Fill aux reservoir
  - Activate Aux switch



COST BREAKDOWN

Cost Component	Price	Percent of Total Cost
Material cost	$ 87.45	3.55%
Fabrication cost	$ 317.67	12.89%
Labor cost	$ 405.00	16.44%
Purchased parts	$1,654.00	67.12%

Sub-Assembly	Price	Percent of Total Cost
Suspension cost	$1,472.52	59.76%
Steering cost	$ 757.00	30.72%
Frame cost	$ 234.60	9.52%
Total	$2,464.12	100.00%

Operations Cost Table

Operation	Cost (in U.S. Dollars)
CNC Machining	70.00/hr
Weld	0.06/cm (0.15/inch)
Saw cut	0.15/cm (0.38/inch)
Tube Bends	0.75/bend
Pre-welding preparation for tube ends	0.75/end
Drilled hole smaller than 1" dia., any depth	0.35/hole
Drilled hole larger than 1" dia.	0.35/inch greater than 1" per hole
Tapped Hole	0.35/hole
Sheet metal shearing	0.20/cut
Sheet metal bends (under 3/8 inch thickness)	0.05/bend
Plate bends (material over 3/8 inch thickness)	0.10/bend
Sheet metal punching	0.20/hole
Plasma cutting/ Lazer cutting	0.10 /inch
Fiberglass	$9.00/sq ft
*Powder coating	$55.00/hr
Chromeplating/anodizing/phosphate coating	$2.00/foot
**Painting	$50.00/hr
Assembly of nut,bolt, and washer	0.10/per
Component assembly (Time)	35.00/hr
Labor (machine set-up, load, unload)	45.00/hr
Miscellaneous operations	Obtain quote on 3000 pieces

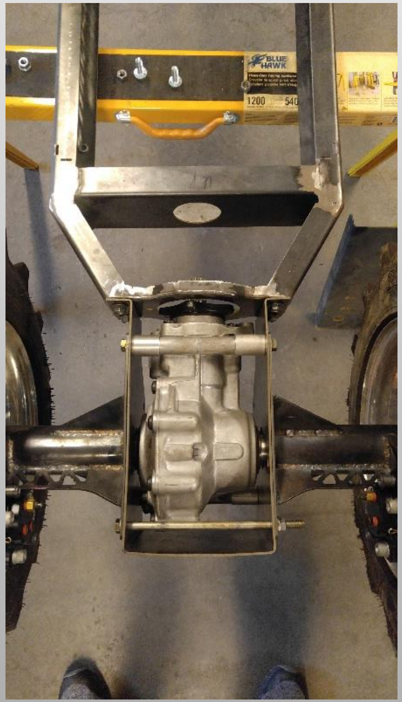
FAILURE MODE ANALYSIS

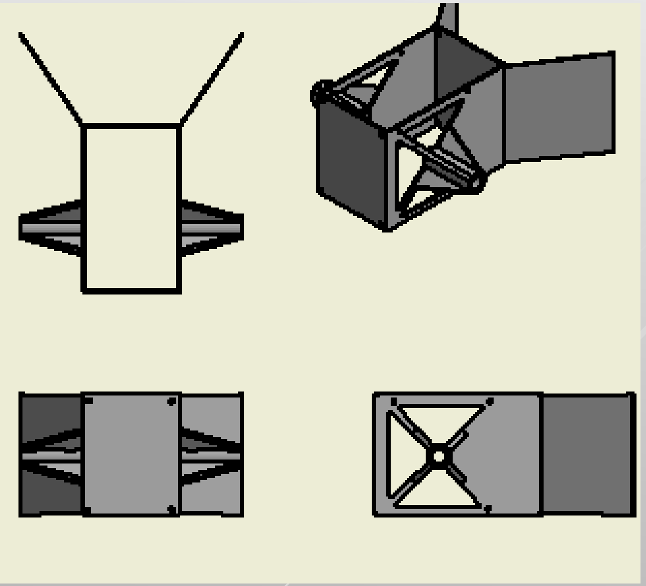
Item	Potential Failure Mode	Potential Effect of Failure	Severity	Potential Cause	Occurrence	Design Controls	Detectability	RPN
Suspension	air bag failure	rupture of air bags	7	over pressurized system	2	built in system relief	1	14
Suspension	air bag failure	puncture of air bags	7	foreign material in suspension	2	stock component	2	28
Suspension	electrical failure	air compressor failure	6	electrical system failure	3	appropriately sized wire and connections	1	18
Steering	steering column failure	bound steering	8	bound steering reducer/u-joint	2	appropriate clearance within system	2	32
Steering	tie rod/rack failure	tire rubbing	4	improperly tuned rack and tie rods	3	minmal/no adjustments required to stock components	1	12
Frame/Chassis	unpredictable forces/conditions	frame cracking	9	external force/trauma to frame	3	relief cuts and minimization of stress concentrations	2	54
Frame/Chassis	interal support failure	frame warping	9	external force/trauma to frame	2	multiple connection points and redundancies	2	36

In order to further analyze the design created by the team, an FMEA was completed. For the design to be satisfactory, the RPN (risk priority number) must be below 99, a number regulated by the 2017 international quarter scale tractor competition rules.

FRESHMAN INTERACTION

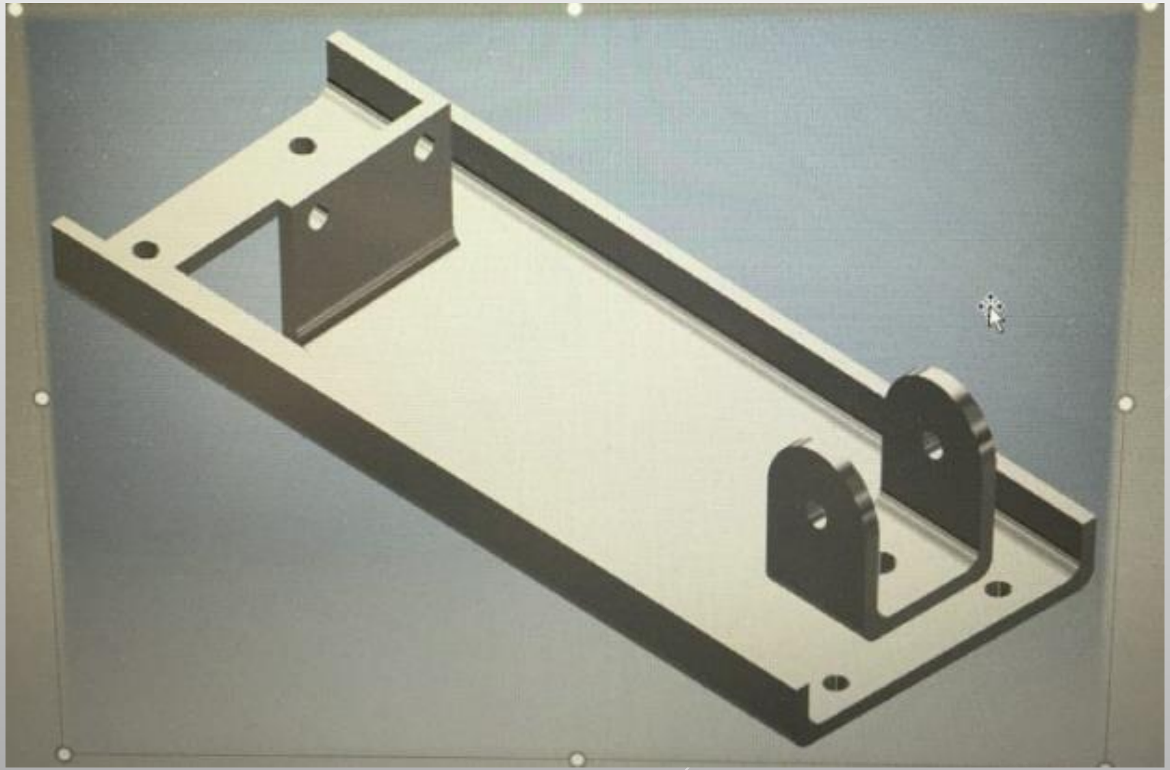
- Rear differential mount
  - Micah Arthaud, Shyanna Hansen, Michael Leiterman, Nick Liegerot, Heath Moorman





FRESHMAN INTERACTION CONTINUED

- Transmission mount
  - Jeremiah Foster, Brent Gwinn, Creston Moore, Austin Pickering, Ross Ruark



BEFORE COMPETITION

- Finalize fabrication
- Testing
- Paint

THANK YOU FOR YOUR TIME

QUESTIONS?

SOURCES

- Auto Dimensions Inc. (2016, Septermber 23). *Wheel Alignment Explained*. Retrieved from Anewtoronto.com: http://www.anewtoronto.com/wheel%20alignment.html
- *How the steering system works.* (2016, September 19). Retrived from How a Car Works: https://www.howacarworks.com/basics/how-the-steering-system-works
- *The Ackerman Principle as Applied to Steering*. (2016, September 19). Retrived from what-when-how: http://what-when-how.com/automobile/the-ackermann-principle-as-applied-to-steering-automobile/
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Cowboy Motorsports

Fall 2016 Report

Scott Dick Garrett Dollins Logan Gary

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Problem Statement

To design and build a cost effective, reliable, and innovative frame, steering system, and suspension system for the Oklahoma State University Quarter Scale tractor team. The design will take into account the team's budget, timeline, and resources for the 2016-2017 competition.

Figure 1. Team picture from the 2015-2016 competition

Introduction

Each year ASABE holds the international quarter scale tractor student design competition in Peoria, Illinois. This competition is designed to give students an opportunity to take a project from concept to finished product. The competition is made up of several parts. Each portion is assigned a maximum possible point value as seen in **Error! Reference source not found.** below.

Design Judging is an interactive portion of the competition where teams present their design's attributes in the particular category to the panel of judges. The judges may then ask questions for further details or provide comments for development of the team's next model. The design judging portion is made up of the following six categories; manufacturability, serviceability, ergonomics, safety, test and development, and sound judging. Each category is worth 70 points and are judged by professional engineers, technicians, or operators from industry.

Technical inspection is the pass or fail portion of the competition. All teams are required to pass a full technical inspection prior to participating in practice pulls or competing in any Performance Competition. This process is broken into two independent portions: Initial Weigh-in and a Detailed Technical Inspection. Technical inspection verifies compliance with the rules set forth by the competition committee. Operator safety and weight limit are the main focus of this inspection. The initial weigh-in will receive a 100 point bonus for starting and operating under its own power, having all shielding in place as best as possible, being on time to the scheduled tech time slot, and completing the inspection in under 24 hours from the end of the assigned time slot.

The pull performance event is comprised of a multi-stage tractor pull using a progressive sled. Points are gained by the number of feet the sled is pulled by the respective tractor. Each team will be allowed one scored pull in three separate heats.

The Maneuverability Course Event is held to encourage consideration for maneuverability in tractor design. The team(s) with the lowest number of overall 'course demarcations' will receive a maximum of 100 points (course demarcations indicate number of direction changes, distance traveled, and number of collisions with cones).

The Durability Event is conducted on an oval course setup on the pulling track that consists of bumps and loose sand. The bumps are no taller than 2.5 inches and set up in a random array to be determined at competition. The loose sand has a depth of approximately 6 inches. Teams will be required to tow a 4-wheel cart weighing up to 2000 lbs (with approximately 0% tongue weight) through the entire course. The cart attaches to the rear hitch of the tractor. Laps are 250 +/- 50 feet in length.

Points allotted to teams for sound level are based on the sound decibel level recorded during the team's first attempt in the sound level Tech Inspection station. The team with the lowest value below the required 91 decibel will receive the full 70 points. Other teams will receive points on a scale from 91 decibel to the lowest level, with allowed points weighted more heavily toward the lowest decibel value (i.e. this will not be a linear scale). No points will be awarded if the sound technical inspection is not passed during the first attempt.

Table 1. Competition points break down

Design Report	500 pts
Team Presentations	500 pts
Design Judging	420 pts
Technical Inspection	Pass/Fail
Tractor Pulls	600 pts
Maneuverability	100 pts
Durability Event	200 pts
Initial Weigh in	100 pts

The proposed project redesigns the main frame, support structures, suspension, and steering of the ¼ scale pulling tractor for the 2017 international competition. The basis of the project is to increase competitiveness of the tractor by increasing functionality of the frame, suspension, and steering. This is achieved by providing a product that makes use of CAD programs to model and test the product. By doing so, the design will have the added benefit of a seamless assembly while optimizing the use of materials required for the product.

Impact

This project is purely of the mechanical nature and part of a larger team design project for the ASABE International Quarter Scale Design Competition. It provides teams with insight into engineering in industry. The team must go through the engineering process and design solutions to address the challenges set forth by the competition. The competition rewards teams that design products with manufacturability, serviceability, ergonomics, and safety in mind. The project allows students to have a hands on experience with taking a concept all the way to production.

Competition Requirements

Our client requires us to follow the 2017 International ¼ Scale Tractor Student Design Competition Rules. These rules provide guidance on how the tractor can be designed and built. One of the two requirements it sets for the frame is that the tractor cannot be longer than 96 inches when measured from the center of the rear axle to the farthest part forward. The other requirement is that it has to be fully customized. This means a frame cannot be a modified frame from a similar vehicle. It must be designed by the team specifically for the ¼ scale tractor. Steering must be achieved with the front tires. Articulated tractors, tricycle front ends, and skid type steering are also against the rules. All steering components must use grade 5 or M8.8 fasteners and locking nuts with a minimum of two threads showing. The suspension falls under the same fastener rules as the steering components.

Client Requirements

The entire design needs to take into consideration what design features the judges look at. This includes manufacturability, serviceability, ergonomics, weight, cost, and strength. Each of these areas have points associated with them for the competition. All the events throughout the competition require the use of the frame, steering, and suspension. Therefore, the effectiveness of these components in each event need to be considered throughout the design process. The overall goal is to score as many points as possible throughout the entire competition.

Client requests

- Frame
    - 100% welded frame
    - Support structures that are incorporated with mounting brackets
    - Reduce weight from previous model
    - Display university and club name
- Steering
    - Reduce force needed to turn steering wheel
    - Improve alignment of steering components
    - Improve steering geometry
    - Improve adjustability
- Suspension
    - Incorporate an adjustable ride height
    - Improve damping of impact stresses applied to the tractor

Design Concepts

Factors of Steering and Handling

Steering is defined by the alignment of the tires and the geometry of the wheel base. Using the parameters of camber, caster, toe, steering axis inclination, included angle, scrub radius, and Ackermann steering geometry a vehicle's steering system can be tuned for the best performance based on the challenge at hand. The bulk of these parameters can be grouped into the category of wheel alignment, which by definition is the complex system of angles and adjustment of suspension components (Auto Dimensions Inc., 2016).

Camber is defined as the angle of the wheel, which is measured in degrees off of the true vertical plane. This angle can limit traction and act as a direct influence on toe angle (Auto Dimensions Inc., 2016). The angle of camber is largely determined by suspension travel and the type of control arm. Caster is the angle at which the steering knuckle pivots and can affect the straight line tracking of a vehicle. A positive angle results in difficult steering and steering wheel kick as the tire impacts obstacles. A negative angle causes difficulties maintaining a straight line (Auto Dimensions Inc., 2016). Toe is defined as the angle of the tires in respect to the centerline of a vehicle. For most rear wheel drive vehicles the toe is set positive to provide better straight line tracking. On the other hand front wheel drive vehicles are typically set negative to compensate for the forward movement of suspension (Auto Dimensions Inc., 2016). The steering axis inclination, included angle, and scrub radius are all affected by the camber, caster, and toe of a vehicle.

Ackermann steering geometry is simply defined as the two steering wheels pivoting at the ends of an axle beam at different angles so that the lines drawn through their stub-axles converge at a single point in-line with the rear axle (The Ackermann Principle as Applied to Steering, 2016). The idea behind Ackermann geometry is that the inner tire travels a shorter distance than the outer tire as is demonstrated in Figure 2.

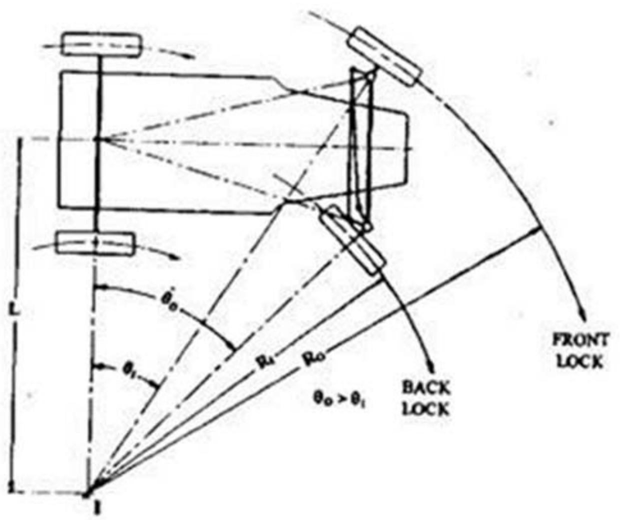

Figure 2. Ackermann steering geometry, from *The Ackermann Principle as Applied to Steering*

This particular steering setup is advantageous over a parallel steering system because Ackermann geometry keeps the two steering tires from fighting against each other during turns. In a parallel system, the two tire paths want to intersect as shown in Figure 3. This forces the tires to push against each other and causes unpredictable steering.

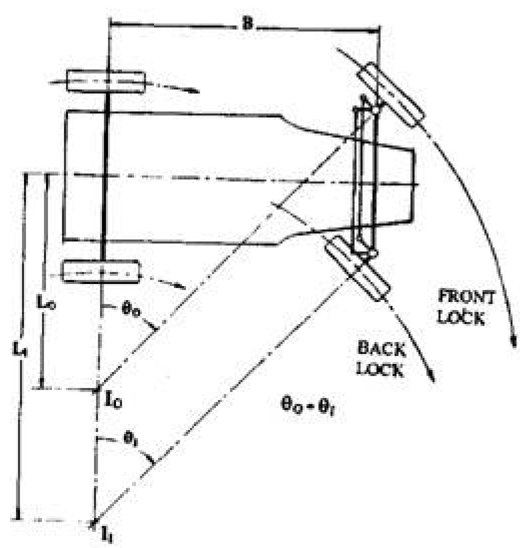

Figure 3. Parallel set steering geometry, from *The Ackermann Principle as Applied to Steering*

Steering Methods and Systems

There are a variety of steering systems used in industry. The most common are rack and pinion, steering box, power assisted steering (both hydraulic and electric), and electronically controlled steering.

As shown in Figure 4, the rack and pinon system makes use of a small pinion gear located at the bottom of the steering shaft. It is seated in a housing that contains a row of teeth. This system very simply changes the rotational movement of the steering wheel into lateral movement that is used to move the tires (How the steering system works, 2016). Smaller vehicles and equipment, like go karts and riding lawn mowers, often use a rack and pinion system. This type of system is best used on non-driven tires and can be more difficult to steer. This system is optimal for small machines due to its compact size and simplicity. The greatest drawback is that the fully manual steering can be cumbersome when the contact surface of the tire is increased.

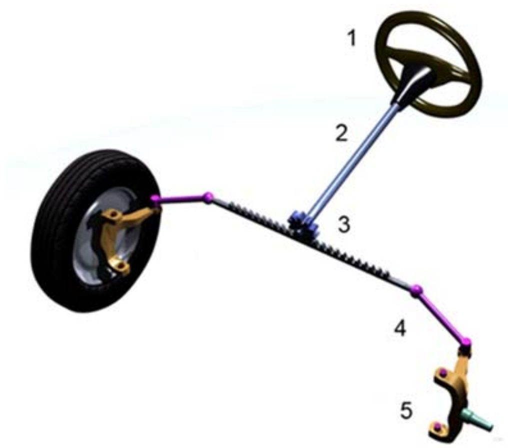

Figure 4. General rack and pinion system in modern vehicles

The steering box system is a bulkier version of the rack and pinion that makes use of a worm gear which controls a lever arm known as the pitman arm. The movement of the arm then controls a mechanical linkage that then steers the tires of the vehicle. This provides a less precise method of steering and more potential for wear (How the steering system works, 2016). The steering geometry is controlled by a drag link and tie rod that connects the hub assembly to the pitman arm. This sees most of its application in off-road vehicles and many rear wheel drive vehicles.

Power assisted steering is less its own system and more of an addition to the previous two. Using a hydraulic or an electrical system, the torque generated by the driver on the steering wheel is amplified in the steering box or rack to ease the steering. This method is widely used in the automotive and agricultural industries today. Having a mechanical system in place if damage occurs to the hydraulics or electronics is an important safety feature. The steering may become cumbersome, but the operator can still maintain control of the vehicle (How the steering system works, 2016). Electronically controlled steering is most commonly found in large ships, airplanes, and modern cars. This method strictly uses an electronic system to control actuators and motors to control the steering. This results in a very quick and light steering that can only be operated while electrical power is being supplied to the system.

Basic Frame Design

The frame is the main supporting structure of a motor vehicle. It is used to mount components and bare the weight of the machine. It needs to be strong enough to support the vehicle, but small enough to be economical. The manufacturers have to take into account where forces will be applied, how large they are, and how they can be spread throughout the frame to avoid overloading one area.

Frame Types

When looking at car and full size tractor frames, many of them are made up of large rectangular steel tubing, shown in Figure 5. Rectangular tubing is used because of its load bearing capacity. The webbing on both sides of the top and bottom flanges enable it to support forces and moments enacted on it, while keeping it at a reasonable size when compared to the overall machine. Its shape allows it to spread out the stress and torque applied to it. This type of shape is a good starting point because it provides the rigidity and durability needed for these machines.

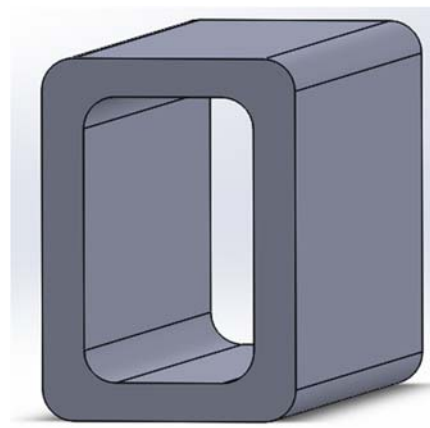

Figure 5. Cross section of basic tube frame design

Another type of frame often used in cars is the uni-body, or monocoque frame. It combines the frame and body of the vehicle, making it all one piece. It is able to withstand the forces and torques applied by the vehicle because of careful and precise engineering. Tubular shapes and cross braces make up the uni-body frame. Many structural principles are combined to create a monocoque frame, shown in Figure 6. The uni-body is very specific to the car it is designed for, thus requiring a lot of engineering work to complete. Nevertheless, once the design is complete, it greatly reduces production costs and speeds up manufacturing time since there are fewer parts to assemble.

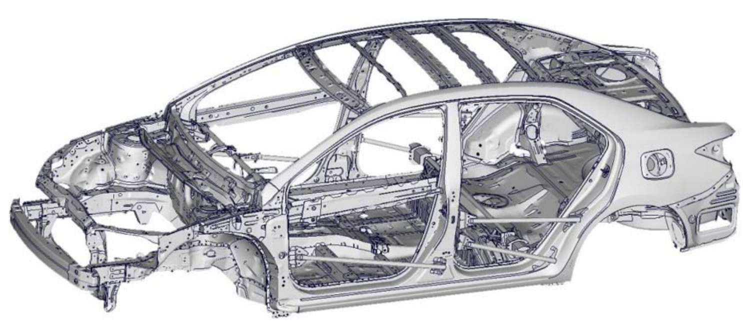

Figure 6. Example of uni-body frame

Many of the quarter-scale tractors made by our competitors have a sheet metal C-channel frame, as shown in Figure 7. This is a good design, because it provides the strength needed to support the tractor, but is also lightweight. It is very similar to a tube frame in its ability to handle bending stress. The single web is able to withstand the forces and moments applied to the frame. However, it requires some extra support members to handle torsion. These support members must be strategically placed in order to spread out the forces seen by the frame. Figures 8 and 9 show how support members can strengthen the design of a frame. In Figure 8, the frame is lacking support structures. High stress concentrations can be seen where the forces are applied. The frame in Figure 9 has properly placed support structures and there is significantly less stress concentrations present.

Figure 7. Cross Section of C-Channel Frame

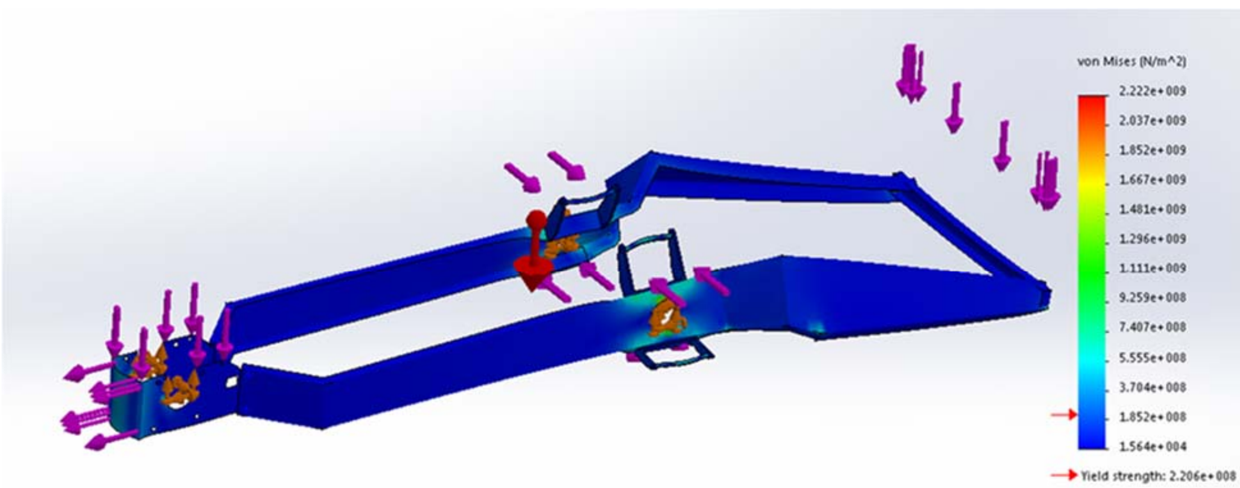

Figure 8. Frame without support structures

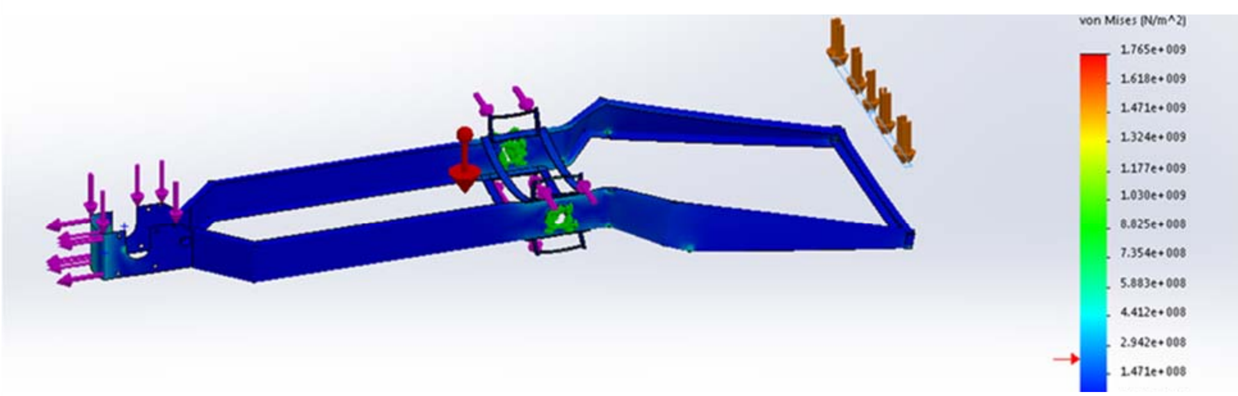

Figure 9. Frame with support structures

Types of Suspension

Figure 10. Example of air springs on a vehicle, from *Progressive Automotive*

Air springs, air shocks, linear actuators, hydraulic cylinders, and coil over shocks are the most common forms of ride height adjustment. They can all work on different types of vehicles and applications, shown in Figure 10.

Coil over shocks and hydraulic cylinders are good options for adjusting ride height of a vehicle. They are both used in the agricultural and automotive industries. Hydraulic cylinders can be configured in a circuit that allows them to be used as both a lifting mechanism and suspension. However, weight is the major downfall to this system. Hydraulic components and fluid are all heavy duty and therefore add a considerable amount of weight to any vehicle.

Coil over shock absorbers are proven in their ability to damp impact forces. They are also manually adjustable. However, the range of adjustment is limited. Air shocks, again, are proven products in industry. This suspension system has a shock absorber and lifting mechanism integrated together. Linear actuators are another strong candidate. They would work very well as a ride height adjustment device. They are easy to install and can handle the load of the front of the tractor. One major problem with this system is the electrical engineering necessary to make it function as a suspension system.

Air springs are commonly used on tractor trucks and their trailers. They are also used as aftermarket add on systems to trucks and SUV’s that haul heavy loads. Air shocks can be configured in a system that functions as both a ride height adjustment and functional suspension. The air springs are lightweight, range from 2-4 pounds, and are reasonably priced at $200-$300 each.

Recommendation

Frame

Frame design selection began by looking at what is seen in industry and what could be the most applicable to our requirements. Previous designs were not going to be used just because it was done in the past. There would be research and causes behind our decisions. The three main frame types that apply to this project are the tube frame, uni-body frame, and C-channel frame. The description and uses can be seen in the Design Concepts section. Using a tube frame design would give our ¼ scale tractor all of the support it needs. However, the amount of material used in a tube frame makes it difficult to utilize the system and keep the total weight of the tractor under 800 pounds.

The C-channel frame is basically a tube frame without one of the side flanges. This reduction in weight is a tradeoff for strength. As mentioned in the Design Concepts section, the C-channel frame requires extra support members to be strong enough to handle the forces applied to the tractor. However, the combined weight of the C-channel frame and support structures could still be lighter than a tube frame. The C-channel frame has been used in previous models and has provided the opportunity to learn from the failures seen in those models.

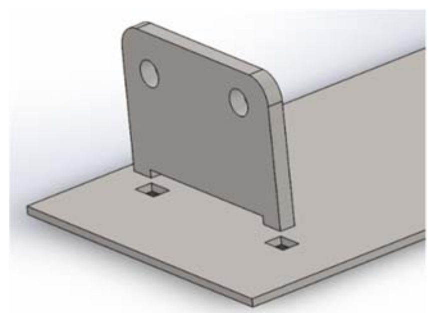

Figure 11. Example of the slot and tab method

A full uni-body design is not feasible for this project. The resources necessary to design a full uni-body frame are unavailable. However, some of the same ideas can be used for engineering mounting brackets and cross members. A slot and tab method, shown in Figure 11, will allow each component of the frame to be welded into place. This decreases assembly time and the chance of misalignment. Pieces can be assembled together without measuring, thus reducing the amount of possible human error. The components will also be designed to fit together in only one way, making it impossible to assemble incorrectly. Strength and serviceability are improved by designing the frame, mounting brackets, and braces as one piece. The strength is increased by transferring forces throughout the entire frame and reducing stress concentrations. Serviceability is increased because major components are directly bolted to the mounting brackets designed into the frame. The combination of C-channel frame with cross bracing similar to a uni-body is the recommended frame selection.

By thoroughly examining the previous model's frame, shown in Figure 12, the new frame can be optimized. It was made of 14 gauge steel (.0747 in), 5" tall, had a 1" top and bottom flange, 91" in length, and 17" wide. It had 45° bends at the rear to fit around the rear differential mount. There were also no additional support structures designed into the prototype because of lack of analysis due to time constraints.

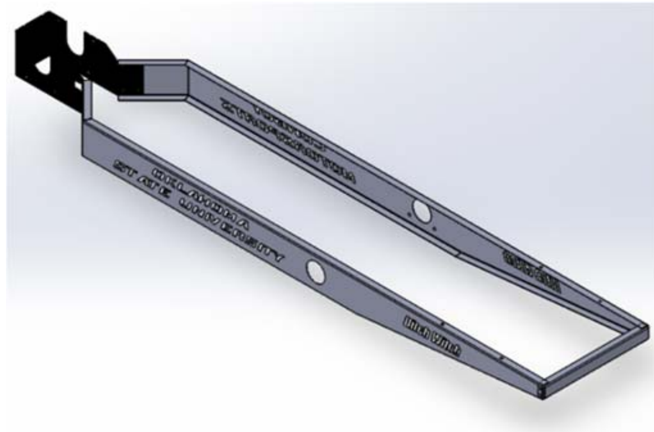

Figure 12. Previous model frame

Due to the absence of support structures, the previous design started to deform in multiple places. If left unattended, the structure would have eventually failed. The first place it deformed was in the 45° bends at the rear. When the sheet metal used for the frame was bent, a sharp corner, shown in Figure 13, was left at the 45° bends. When the rear wheels would go over a bump and apply torque to the frame, it caused the stresses to concentrate at those corners. This developed cracks along the bend, shown in Figure 14. If the sharp corner would have been welded together during assembly, it would have strengthened the frame at that area by eliminating the stress concentrations at those points. Additional support structures could have also been used to further strengthen the frame in the rear end.

Figure 13. Stress concentration in sharp corner

Figure 14. Cracking from sharp corners in frame

The other place the frame deformed was at the front differential. There were no supports around the differential causing the weight of the tractor to pull the bottom rails of the frame apart, while pushing the top rails together. This is represented by Figure 15. If left unresolved, the deflection would have caused the frame to be pushed past its ultimate strength. Figure 16 shows the stress distribution seen by the frame during the pulling events. Large stress concentrations can be seen around the front axle.

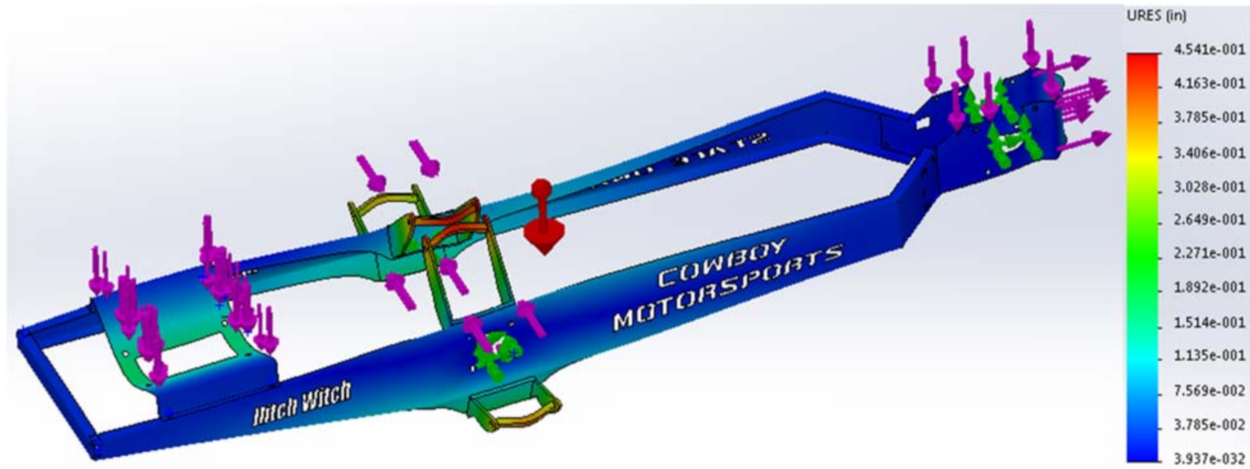

Figure 15. Exaggerated displacement of frame due to high stresses at front axle

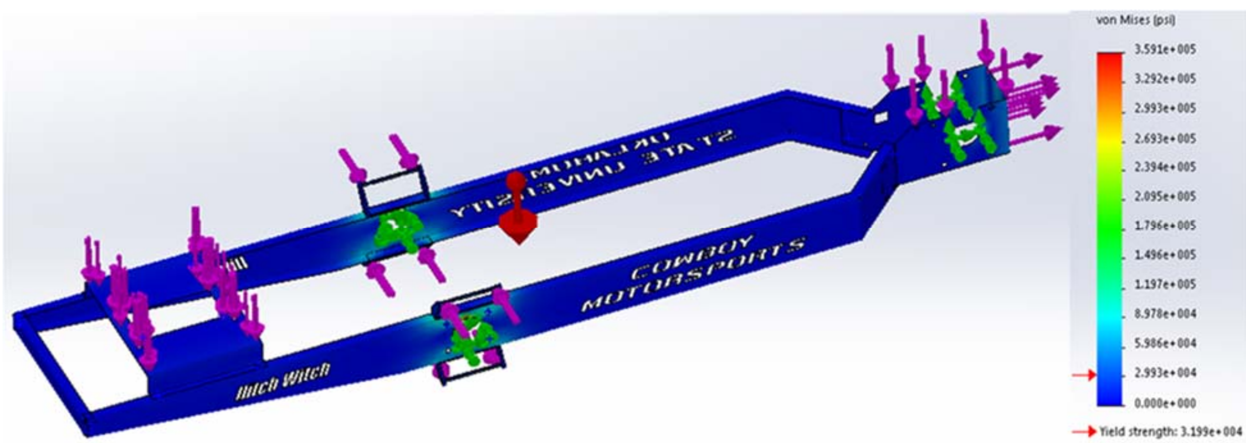

Figure 16. Stress distribution of forces applied to the frame during pulling events

To combat cracking in the rear end, changes were made to strengthen the frame in that area. The Solidworks simulation in Figure 17 shows where stresses act and how large they are. Comparing the simulation of the 45° bend to the 30° bend shows how the smaller angle reduces stress concentration. Also, reducing the angle allows more stress to be transferred down the length of the frame instead of acting perpendicular to it.

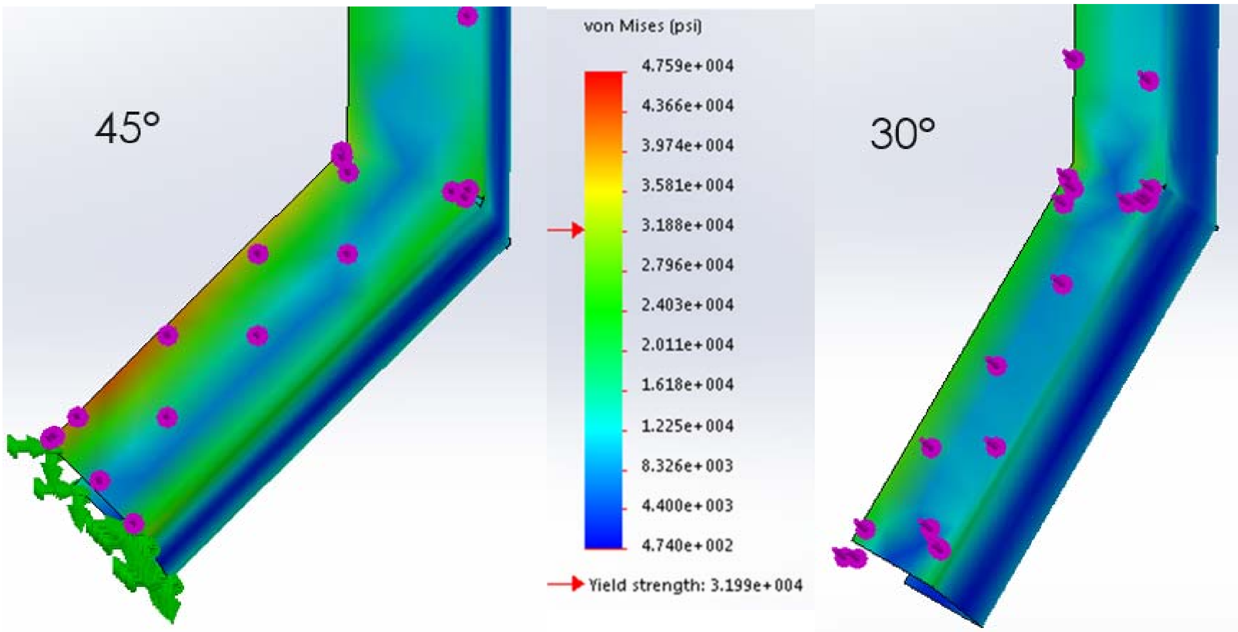

Figure 17. Comparison of stresses in 45° bends and 30° bends

The next change that was done to the rear end was to add a cross member at the 30° bends and at the end of the frame. These supports provide strength by boxing in the rear section of the frame. This stiffens that area and transfers stress to the rest of the frame rails. Boxing in the rear end meant the rear differential mount had to be redesigned. The new rear cross member no longer allows the differential to be pulled straight up out of its mount. To compensate for this the new design will bolt to the end of the frame using six 3/8" grade 8 UNC bolts. Calculations were done to determine bolt size and they showed a safety factor of over 200. Using bolts at this area will make servicing the rear end easier. Removing these six bolts will allow the entire rear axle and differential assembly to be rolled away from the frame.

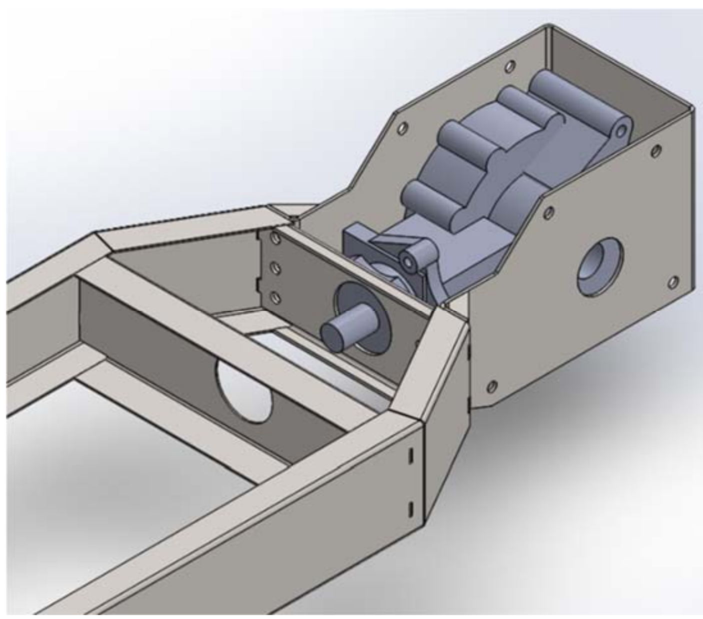

Figure 18: Supports at rear end to box in the frame

The area around the front differential has also been redesigned to improve stress distribution throughout the frame. To keep the bottom of the frame from pulling apart, cross bracing was incorporated with the A-arm mounting tabs. As shown in Figure 19, the A-arm mounts connect one frame rail to the other. The front differential mount also serves as a brace by running from one frame rail to another. The top A-arm mount will also be used as a brace to keep the top of the frame from moving closer together.

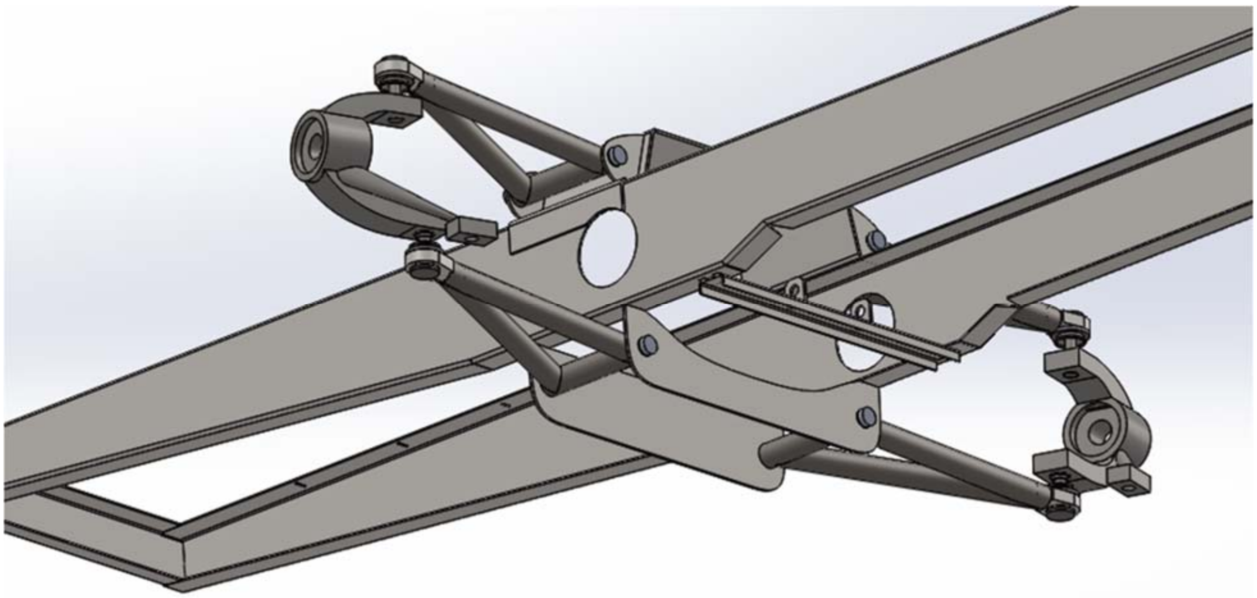

Figure 19: A-arm mounts used as cross braces

Initially, the shape of the frame rail went through a few iterations before a final design was decided. The first style considered was the wide engine frame, shown in Figure 20. It bent outward after the front axle to widen the frame for the engine and transmission to sit lower. The idea was to help lower the center of gravity and increase stability of the tractor. However, this design created complications for the powertrain design of the tractor. It required the differentials to move down to compensate for the engine and transmission moving, which in the end created more problems than it solved.

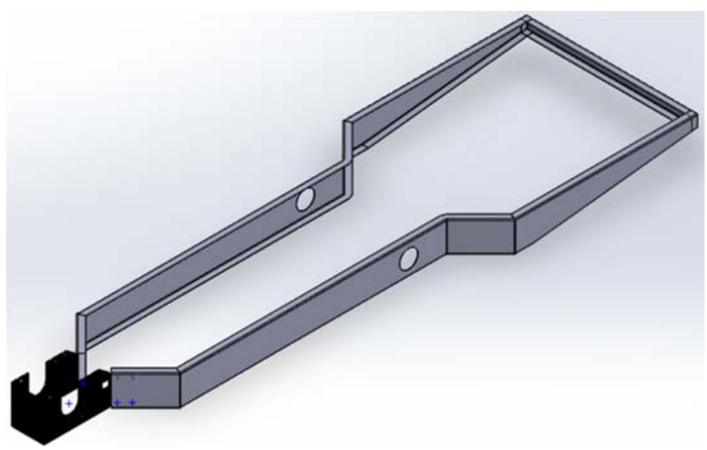

Figure 20. Wide engine frame

The next frame style considered was the Short Frame. This frame rail removed an inch of material before and after the front axle. It was designed this way to reduce weight. However, it also reduced the strength of the frame rail. Extra support structures would have been designed to compensate for the loss of strength. This design was not used because it was not compatible with the new front A-arm design.

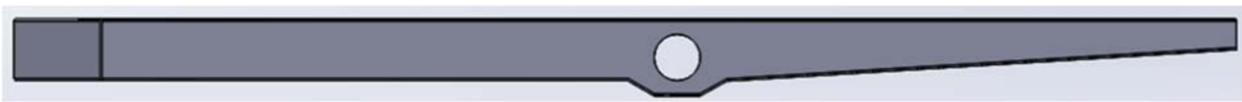

Figure 21: Short Frame

The third and final style was the spearhead frame, shown in Figure 22. In order to accommodate for the new front A-arm design, the length below the front axle was extended before and after the hole for the axle shaft. The height decreases after the front axle from 5” to 4”. It is 78.5” long, and is made of 14 gauge steel. Figure 23 shows Simulations in Solidworks that prove the spearhead frame design is strong enough to support a fully weighted tractor. Some high stress areas can be seen at the ends. This is because of the way it was fixed in Solidworks. It will not see those stress concentrations while in operation.

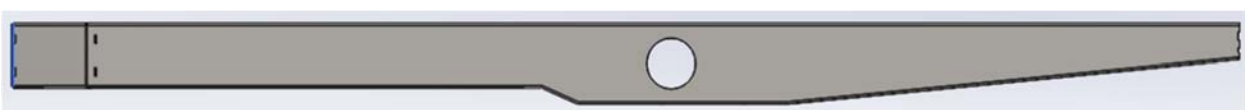

Figure 22. Spearhead frame

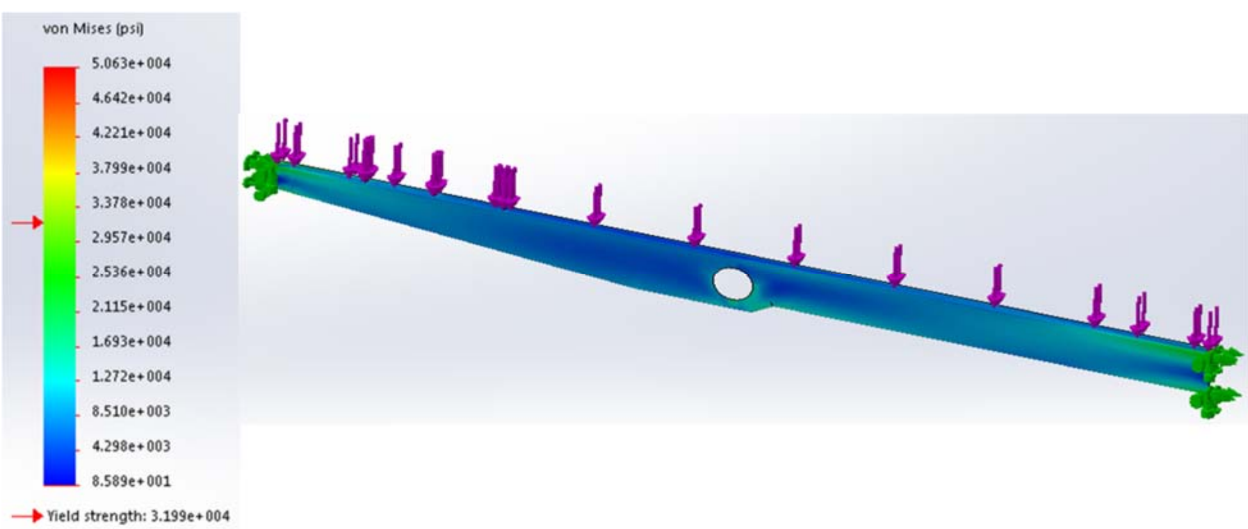

Figure 23: Spearhead frame under an 800 lb distributed load

Once the frame rail design was finalized a common assembly was made, as displayed in Figure 24, to see how all the parts fit together. The main difference between this design and the previous design is the width from one frame rail to the other was reduced from 17” to 14.5”. By reducing the frame width, the width of every cross member is reduced by 2.5”. This saves weight when the overall design is complete. Another advantage to reducing the width of the frame is that it reduced the length of the lever arm acting on the 30° bends. In this design, the bends do not have to come in as far to connect to the rear differential mount. It is 90” from the front cross member of the frame to the back of the rear differential mount.

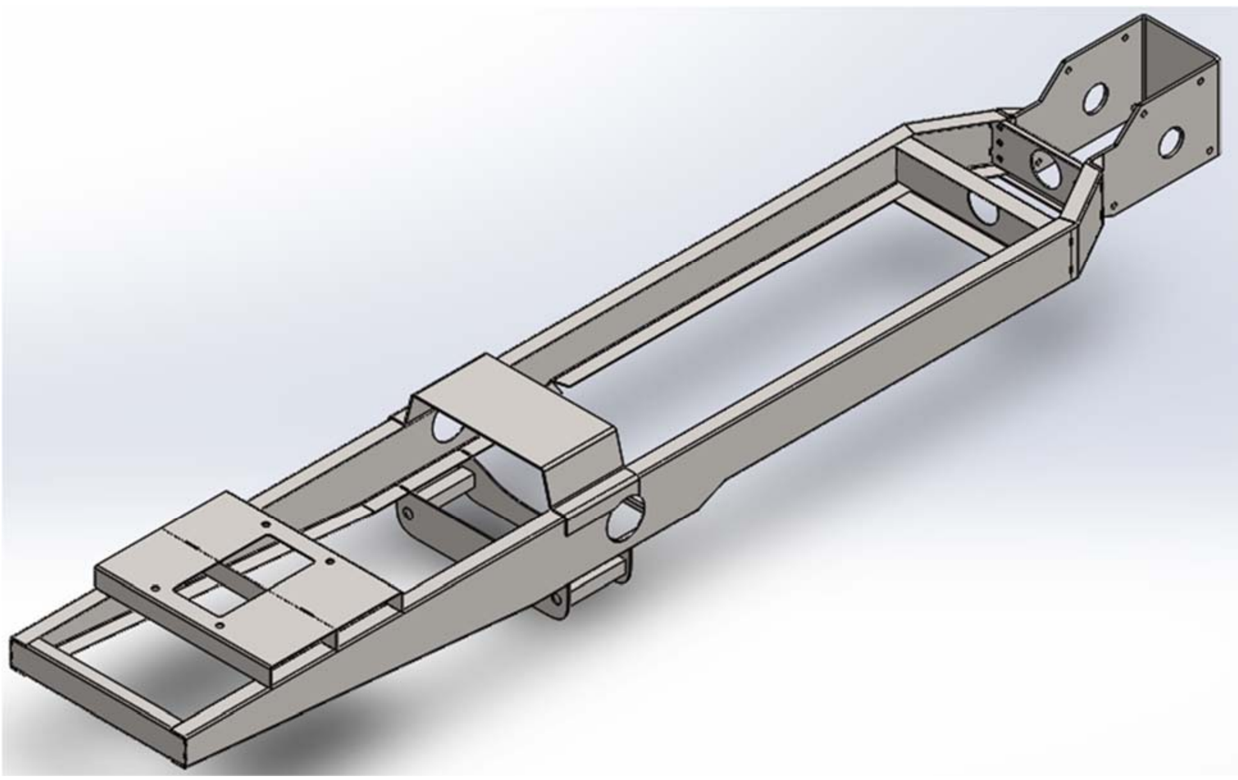

Figure 24: Overall assembly of frame and cross members

Once the main frame and cross members where in the assembly, simulations were done to see how the design would react to the forces applied to it. As shown in Figure 25, the new design easily handles the stresses presented to it.

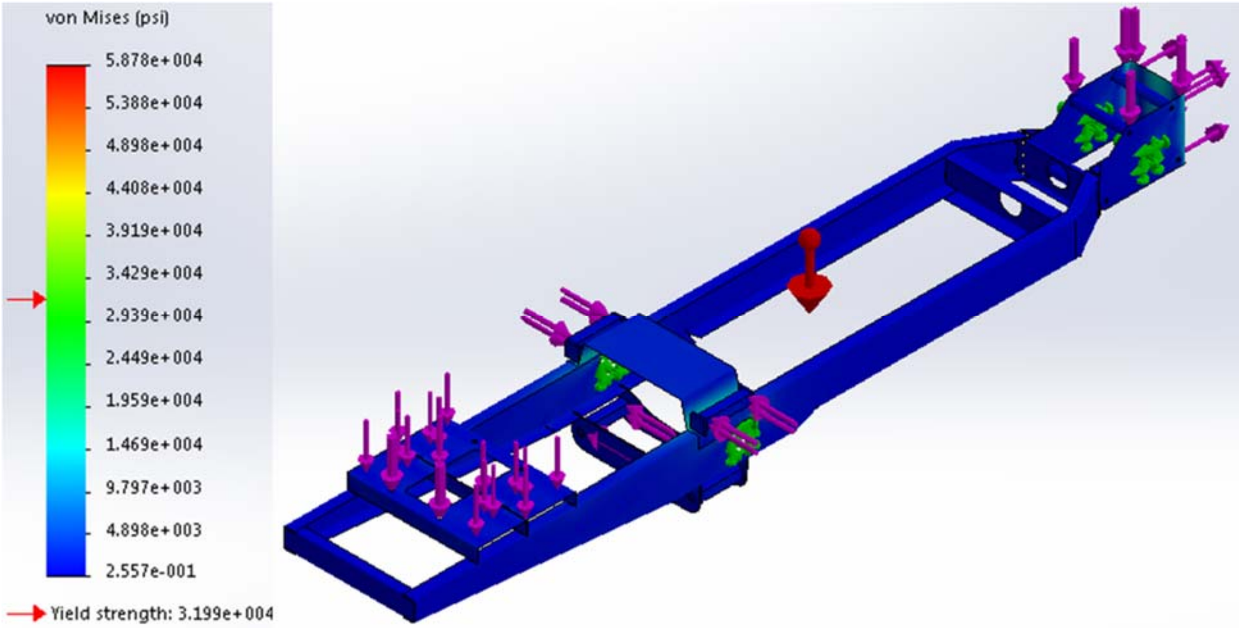

Figure 25: Simulations of forces applied during a pulling event on the overall frame assembly

Suspension

After careful consideration of the client's requirements and conditions the suspension will operate in, the air spring suspension was selected. The previous tractor was used as a means of testing the feasibility of the suspension concept in this application. The previous model was converted from a stiff suspension to using air springs. This comparison is shown in Figures 26 and 27. Placement of the air springs quickly became an area of concern. The first two iterations of air spring location brought clearance and leverage issues to the surface. Initially the spring was located near the outside of the frame rail. This allowed the weight of the tractor to use the leverage of the a-arm to gain a mechanical advantage over the air spring. When the air springs were pressurized, the force output was too small to overcome the weight of the tractor. To compensate for this problem, the air springs where relocated to the end of the a-arm. This reduced the leverage advantage and allowed the pressurized air springs to raise and lower the front of the tractor. However, the relocation created a clearance issue between the air spring and the front tire. The air spring was relocated once more and moved 1.5 inches toward the frame rail. This eliminated the clearance issue.

Figure 27. 2015-2016 solid suspension

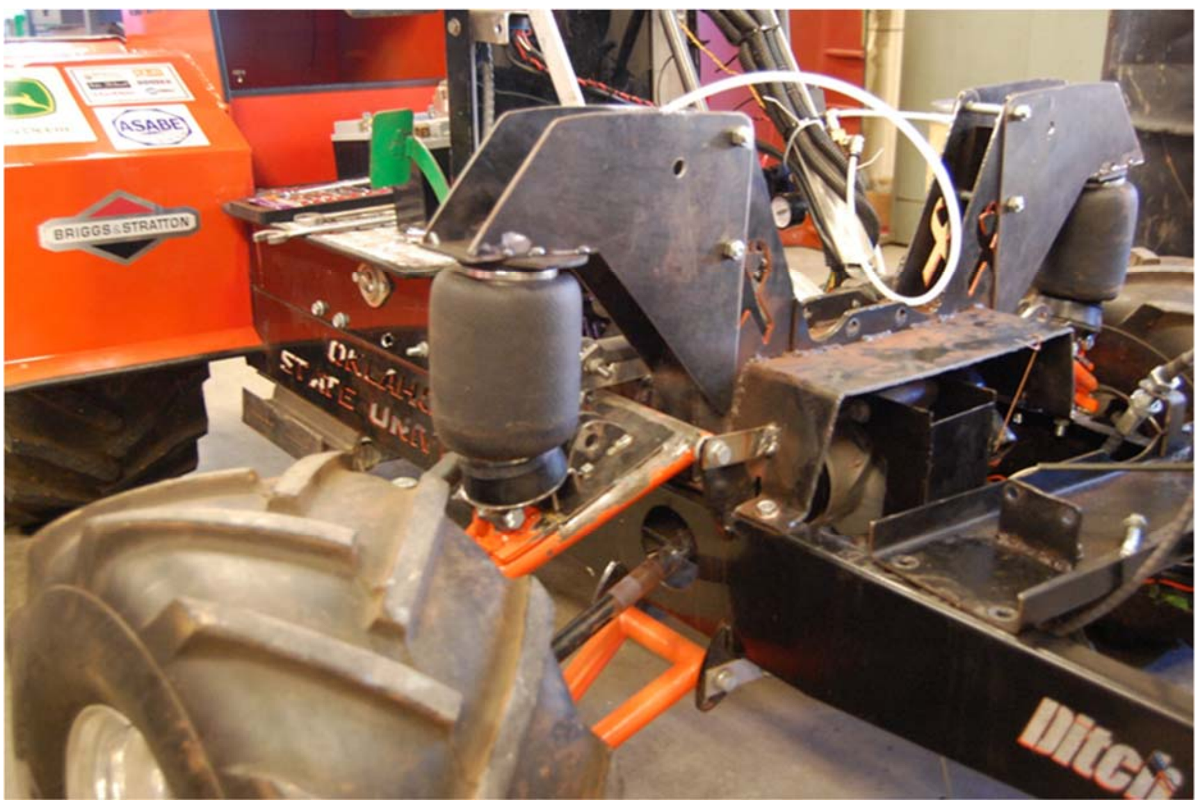

Figure 26. 2016-2017 prototype air spring suspension

Once the previous issues had been addressed, the tractor was tested on its ability to raise and lower the ride height while at 1500 pounds of total weight. The system failed to raise the ride height under this load. Using a four pad scale, the weight carried by the front tires could be monitored while removing weight until the suspension could raise the tractor. Taking this weight, the length of the moment arm on the air spring, and the air springs location, Equation 1 and 2 were developed. Figure 28 models how each of the previously mentioned factors effect one another when changed.

Equation 1.

$$M_A = 0 = W \times (L + O) - (F \times M)$$

Equation 2.

$$F = \frac{W \times (L + O)}{M}$$

F = force required to lift the tractor

W = weight / front tire

R = max radius of air spring

C = clearance between air spring and ball joint

O = Length from center of tire to ball joint

L = Length of A-arm

A = pivot point on frame for A-arm

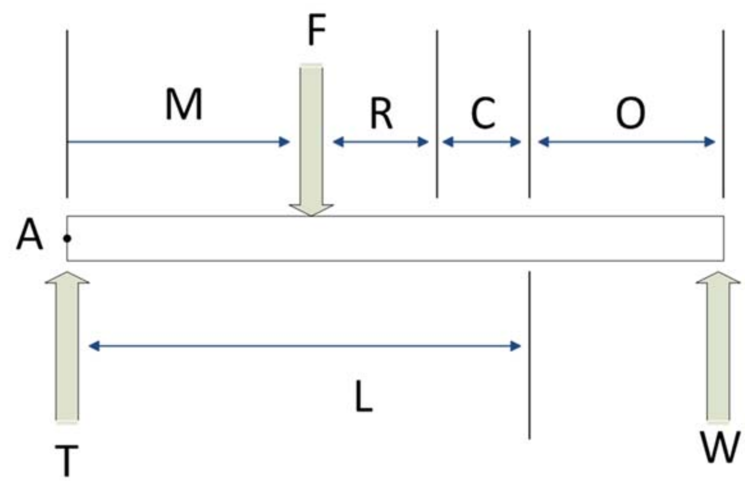

Figure 28. Diagram of the variables that effect the force needed to raise the ride height of the tractor

To test the performance of the damping action of air springs, the tractor was driven on multiple passes through a rough field. The individual front suspension had articulation and minimal bouncing. The operator reported a noticeable positive difference in ride quality with the prototype suspension. More testing on the final product will provide an accurate account of how well the suspension absorbs bumps.

Weight transfer to the rear tires was a known concern for this suspension system. To test the severity of the problem, the tractor was tested on the pulling track. Once the tractor was under load while pulling the weight of the sled, the suspension immediately rose to its maximum ride height. The test was stopped before damage to the front drive axles or air springs could occur. To compensate for this major problem, a suspension locking mechanism will be an added feature to the air spring suspension. This will allow the operator to pull with a solid suspension.

Steering

Through the process of researching the strengths and weaknesses of typical steering systems, the rack and pinion was decided as the best choice. It is a durable and reliable system. They are also readily available on the market and come in a range of sizes. Previous models have problems with heavy steering. The solution is to move the rack and pinion down in line with the steering knuckles. A 2:1 geared reduction also aids in the solution. Turning from lock to lock takes 2 turns, but with half the force of the previous design.

The tie rod ends that connect the rack and pinion to the steering knuckle are also redesigned. They are constructed out of lightweight chrome-moly steel tube with threaded ends to allow for fine tuning of the tire alignment. The previous design did not allow for a range of adjustments. Figure 29 details the preliminary design.

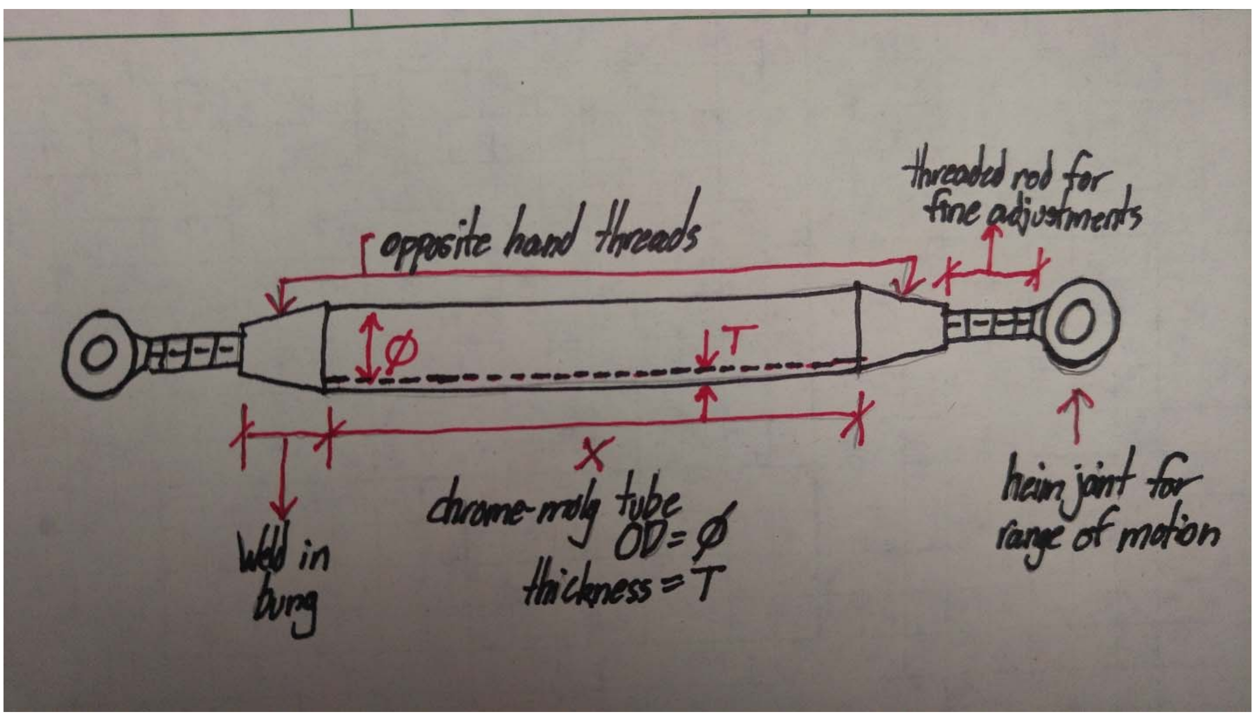

Figure 29. Adjustable tie rod end

Freshman Involvement

Two teams of five freshman were assigned to help with this project. They are required to help with some small, but significant portion of the larger senior design project. One group, Micah Arthaud, Shyanna Hansen, Michael Leiterman, Nick Liegerot, and Heath Moorman, were tasked with developing a new rear differential mount. Figure 30 shows what the group designed. The second group, Jeremiah Foster, Brent Gwinn, Creston Moore, Austin Pickering, and Ross Ruark, were tasked with developing a new transmission mount. Figure 31 shows what the group designed.

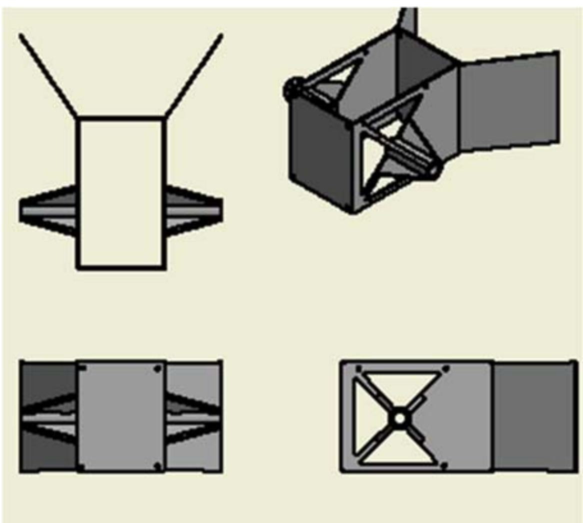

Figure 30. Freshman rear differential mount

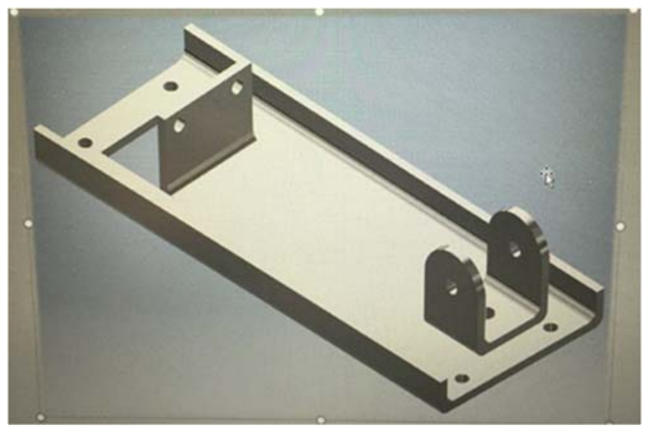

Figure 31. Freshman transmission mount

Spring Semester Goals

- Finish Solidworks model
- Send parts to be manufactured
- Assemble prototype
- Test prototype

Budget

Table 2. Cost breakdown for fabricated parts

Fabricated parts	Weld	sheet metal shearing	sheet metal bends	Plasma cutting	Labor	drilled holes	Total
Frail rail	$ 5.40	$ 0.40	$ 0.70	$ 42.00	$45.00	$ -	$ 93.50
Engine mount	$ 0.60	$ 0.60	$ 0.10	$ 8.20	$11.25	$ 1.40	$ 22.15
A-arm mounts	$ 2.40	$ 2.40	$ 0.30	$ 7.00			$ 12.10
Rear cross member	$ 4.80	$ 0.80	$ 0.20	$ 1.20			$ 7.00
Rear differential mo	$ 2.40	$ 0.80	$ 0.20	$ 5.20		$ 2.80	$ 11.40
							$ -
A-arm	$ 2.00				$90.00		$ 92.00
A-arm pivot					$15.00		$ 15.00
Ball joint tab				$ 1.00			$ 1.00

Table 3. Cost breakdown for aftermarket parts

Aftermarket Parts	Cost Per Unit	Total Cost	Location
14" Rack and Pinion	$ 100.00	$ 100.00	desertkarts.com
Howe Steering Reducer	$ 92.00	$ 92.00	jegs.com
4130 Chome-Moly Round 1" OD, 0.049" wall (cost per foot)	$ 3.40	$ 34.00	stockcarsteerl.com
4130 Chome-Moly Round 3/4" OD, 0.058" wall (cost per foot)	$ 3.64	$ 21.84	stockcarsteerl.com
14 Gauge Steel Sheet (cost per square foot) -estimate	$ 3.00	$ 250.00	Ditch Witch
Heim and Rod Kit (3/8 x 3/8-24 panhard w/ 0.058 bung) ea.	$ 19.20	$ 38.40	QS Components Inc.
Hardware (cost estimate)			Fastenal
Air Springs	$ 200.00	$ 400.00	
Air Compressor	$ 270.00		
Pneumatic Hoses and Fittings (estimate)	$ 150.00		
Pneumatic Valves	$ 40.00	$ 80.00	
Steering Wheel (cost ea.)	$ 50.00	$ 50.00	jegs.com
Fully welded frame	$ 200.00		

Schedule

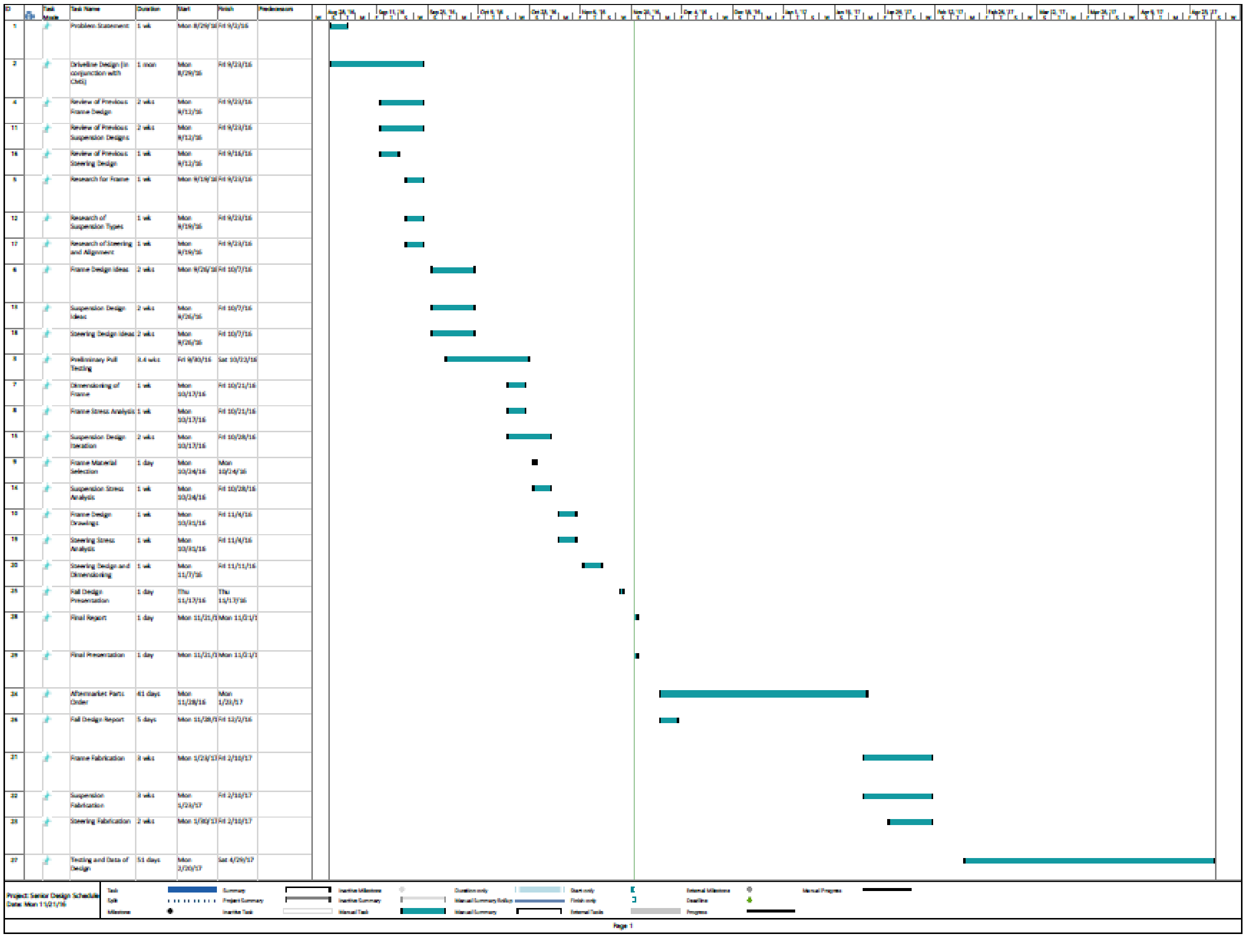

Figure 32. Past and future schedualed meetings and deadlines

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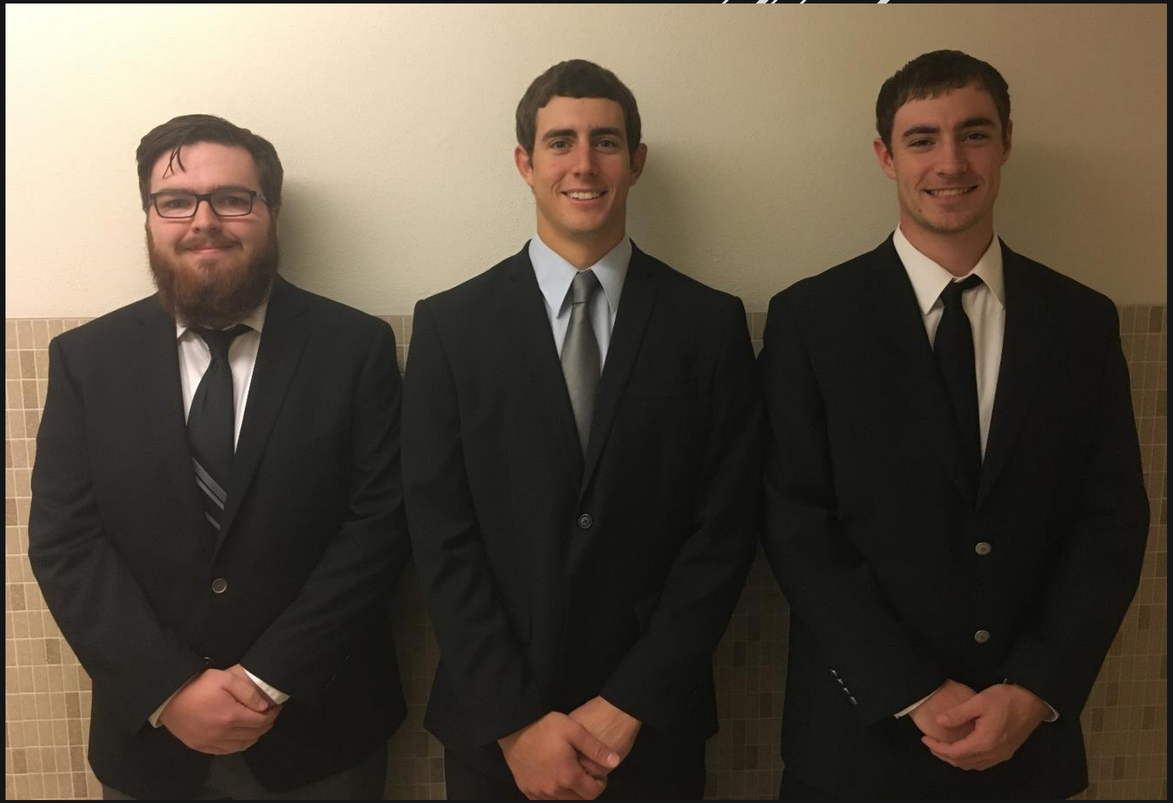
COWBOY MOTORSPORTS
SENIOR DESIGN 2016-2017

Scott Dick

Garrett Dollins

Logan Gary



2016-2017 ASABE INTERNATIONAL QUARTER SCALE TRACTOR STUDENT DESIGN COMPETITION

COMPETITION OVERVIEW

- Design report 500 pts
- Team presentation 500 pts
- Design judging 420 pts
- Technical inspection Pass/Fail
- Tractor pulls 600 pts
- Maneuverability 100 pts
- Durability event 200 pts
- Initial weigh in 100 pts

PROBLEM STATEMENT

To design and build a cost effective, reliable, and innovative frame, steering system, and suspension system for the Oklahoma State University Quarter Scale tractor team. The design will take into account the team’s budget, timeline, and resources for the 2016-2017 competition.

FRAME REQUIREMENTS

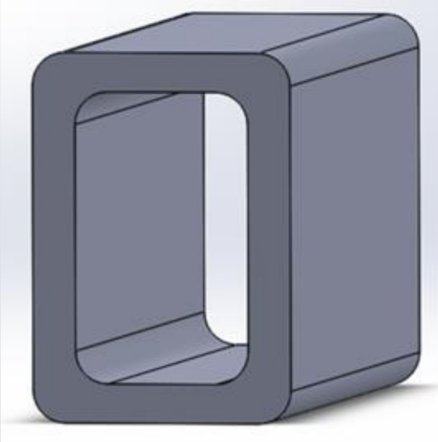
- Withstand weight of tractor and forces felt during competition
- Provide area to mount other components of tractor
- Less than 96 inches long
- Fully customized

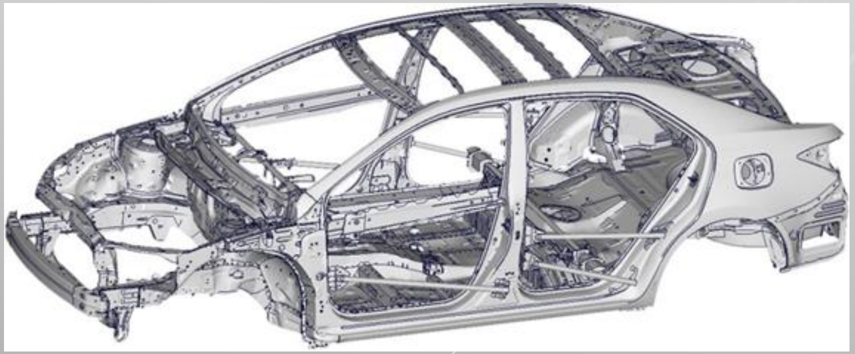
FRAME OBJECTIVES

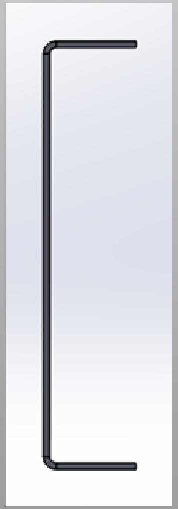
- Easily manufactured
- Fully welded together
- Lightweight
- Display School and club name

FRAME SELECTION

- Tube Frame
  - Strong, but heavy
- Unibody Frame
  - Very specific to each vehicle
  - Requires precise engineering
- C-Channel Frame
  - Lightweight
  - Not as strong as other options

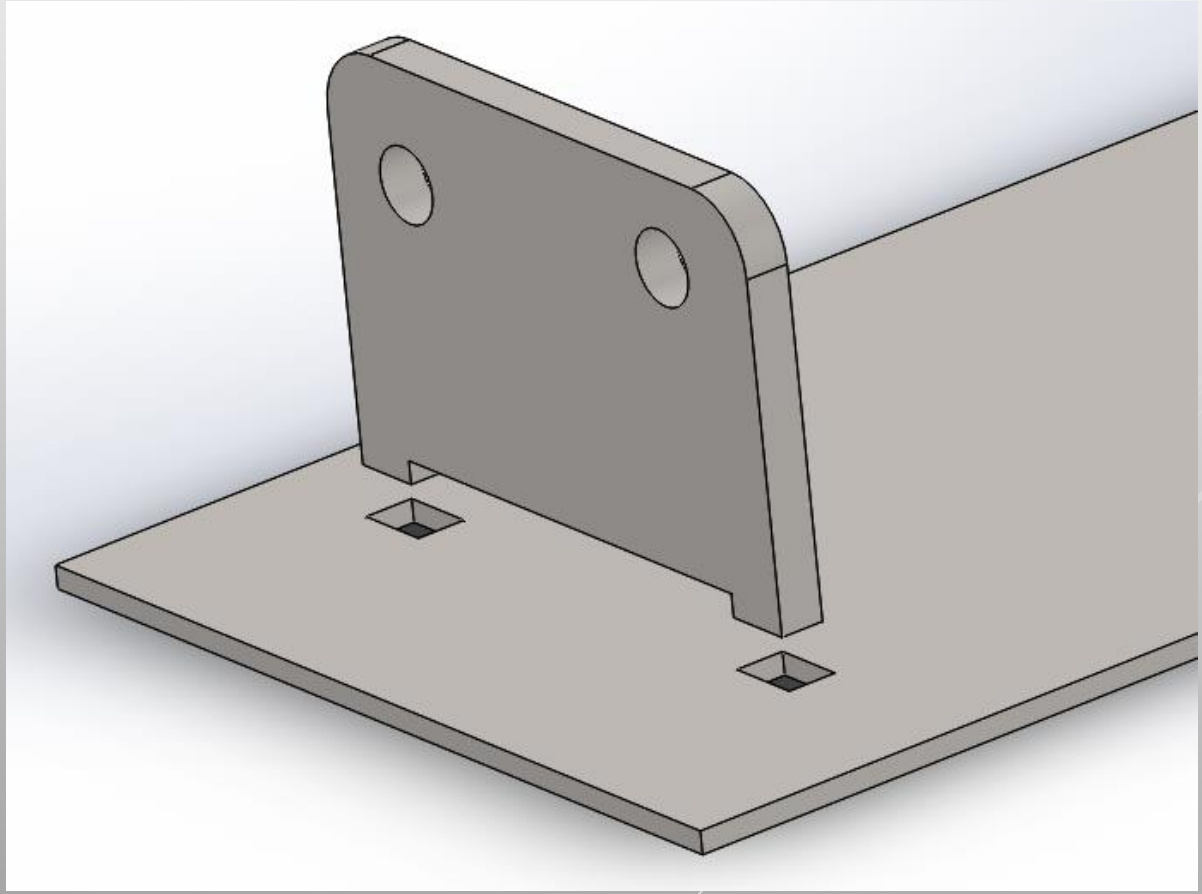






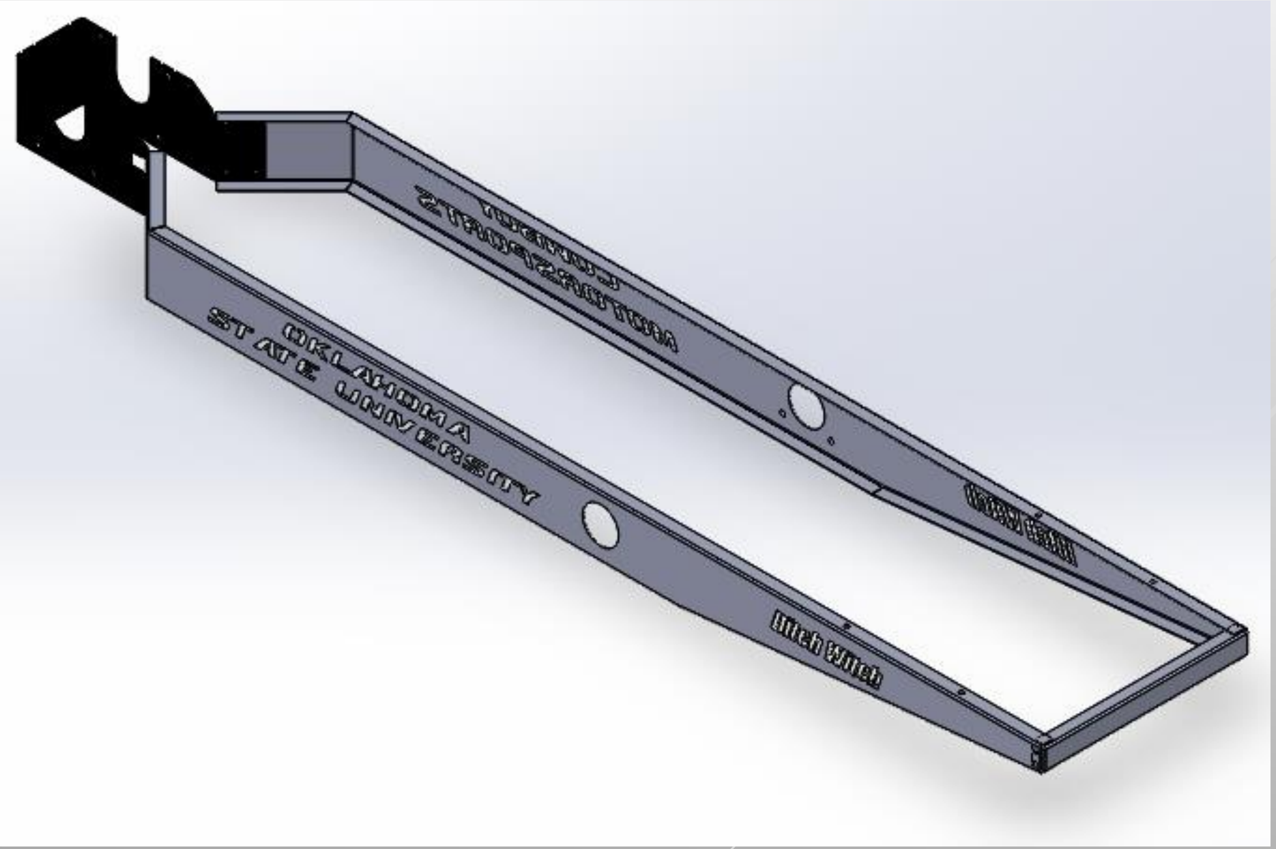
FRAME SELECTION

- C-channel System
    - Lightweight
    - Proven
- Unibody Concepts
    - Slot and Tab
    - Welded
    - Bolt on major components



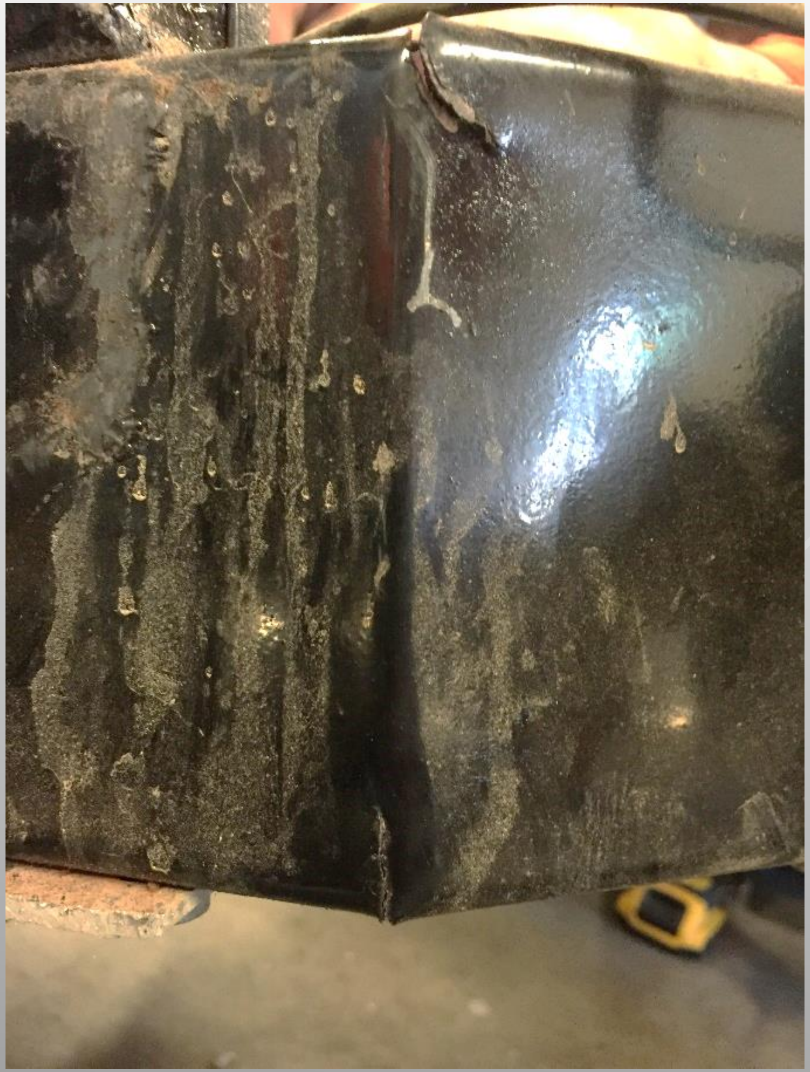
PREVIOUS DESIGN

- 14 Gauge Steel
- 5" tall, 1" top and bottom flange
- 17" wide, 91" long
- 45° bends at rear
- Bolted together
- No additional support structures

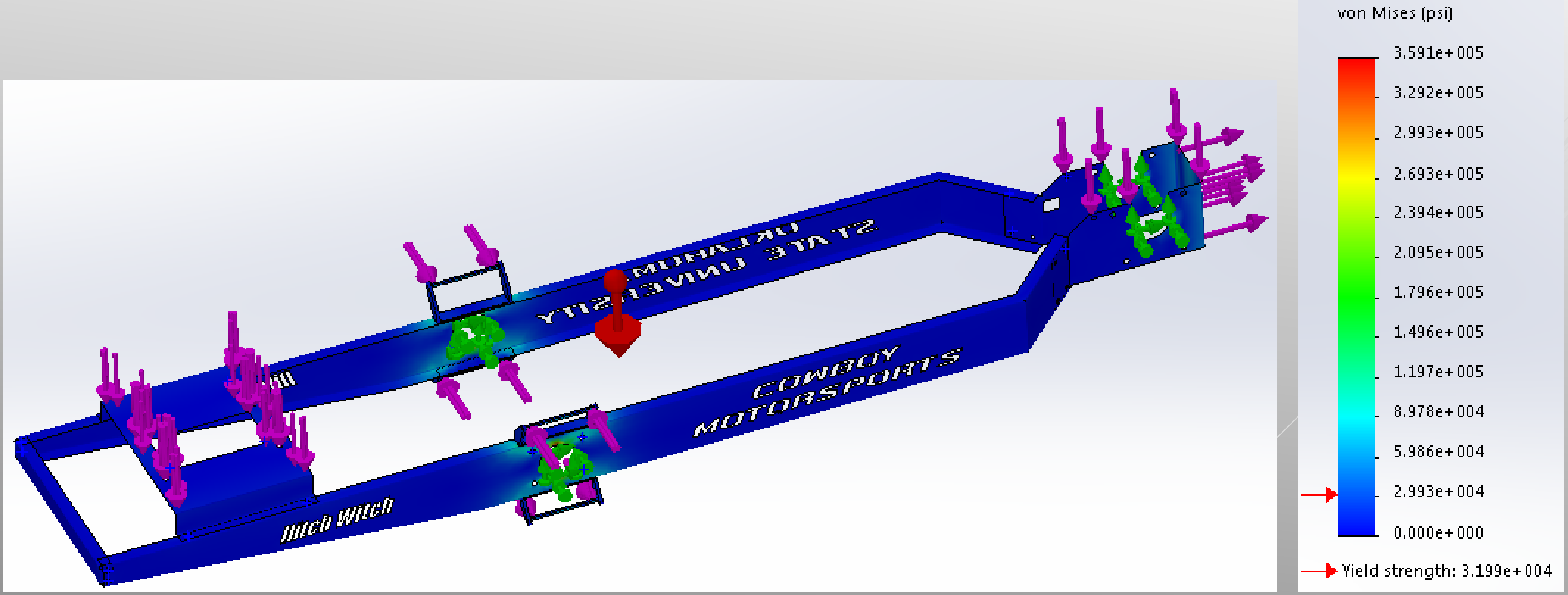


PREVIOUS DESIGN FAILURES

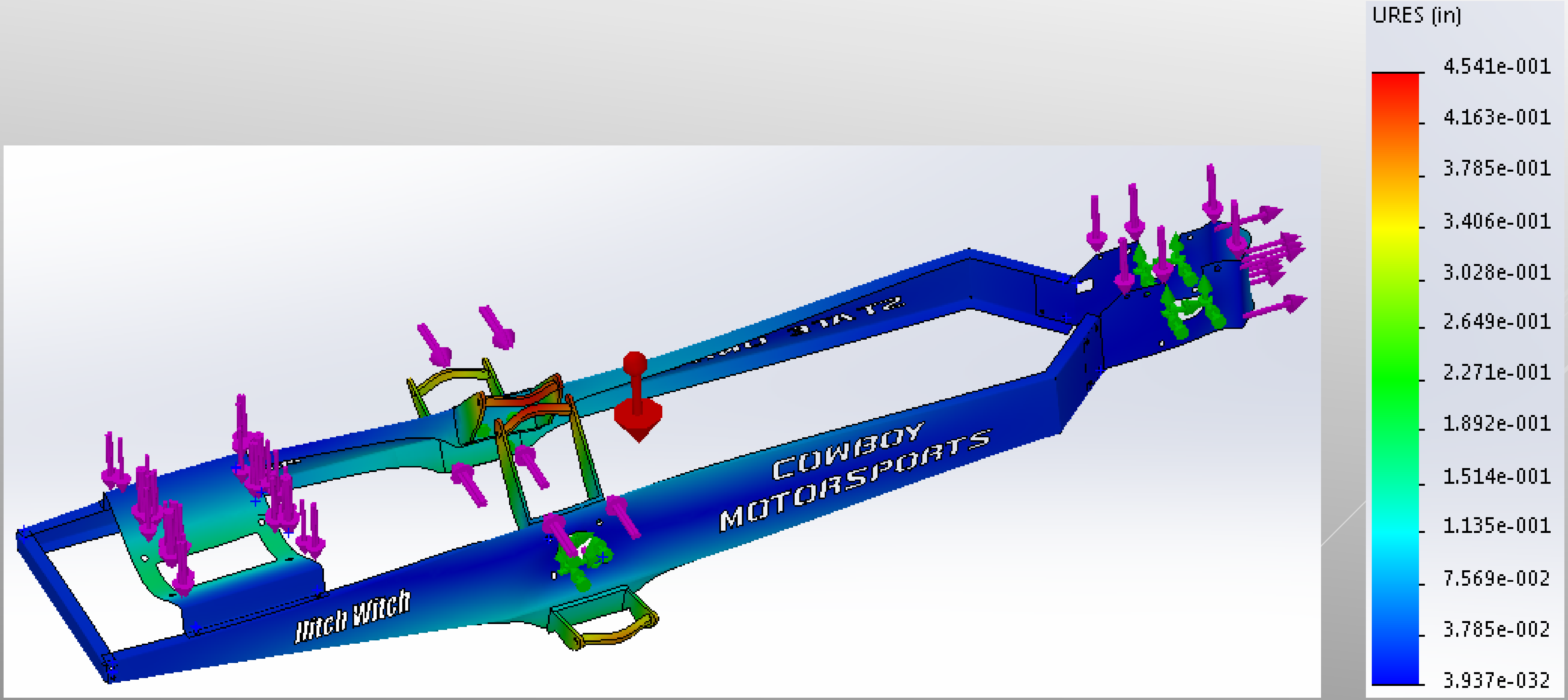
- Began cracking at 45 degree bends
- Stress concentrations due to sharp corner
- Could have been strengthened by welding the gaps



PREVIOUS DESIGN FAILURES

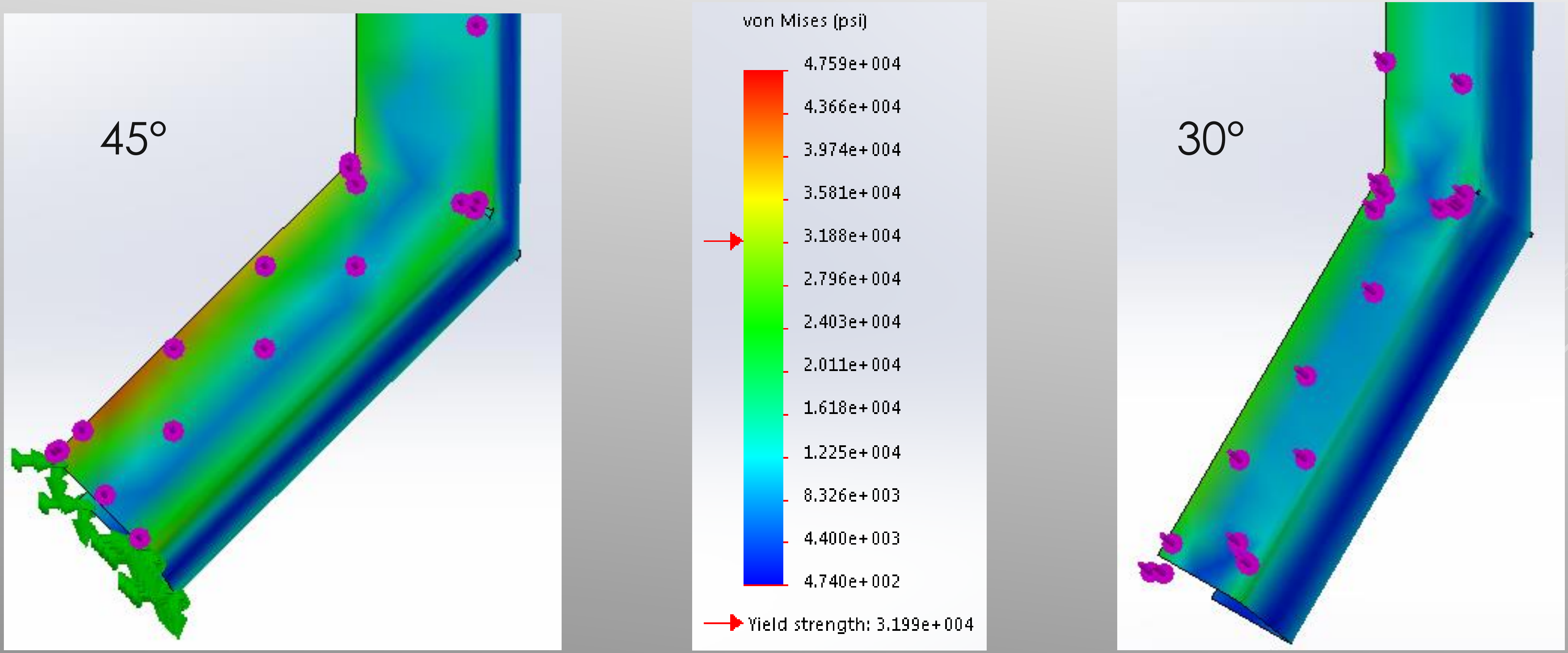


PREVIOUS DESIGN FAILURES



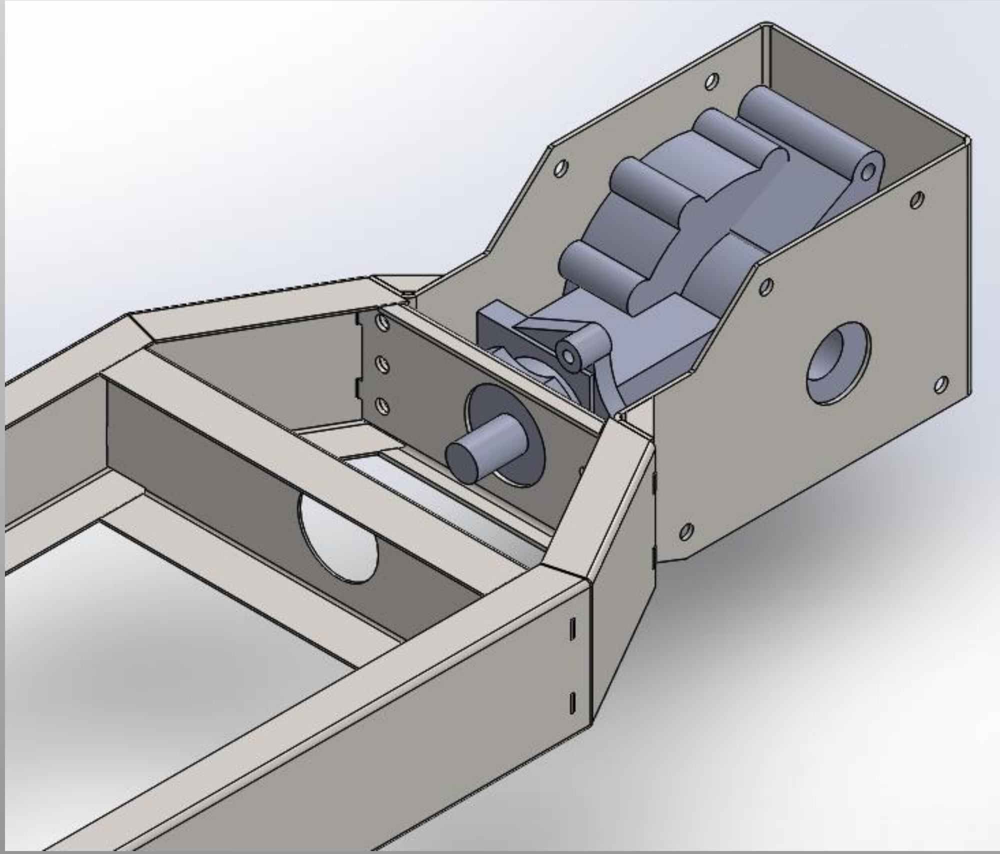
NEW DESIGN: REAR END

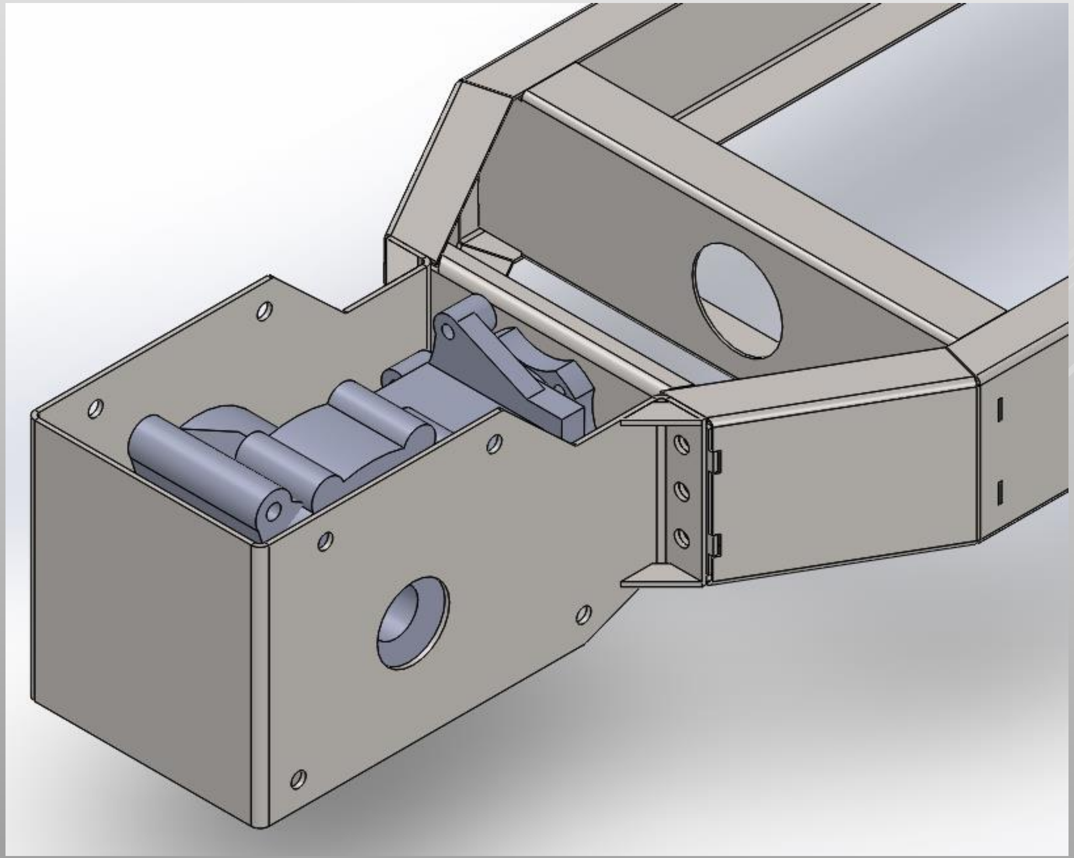
- Angle reduced from 45° to 30°



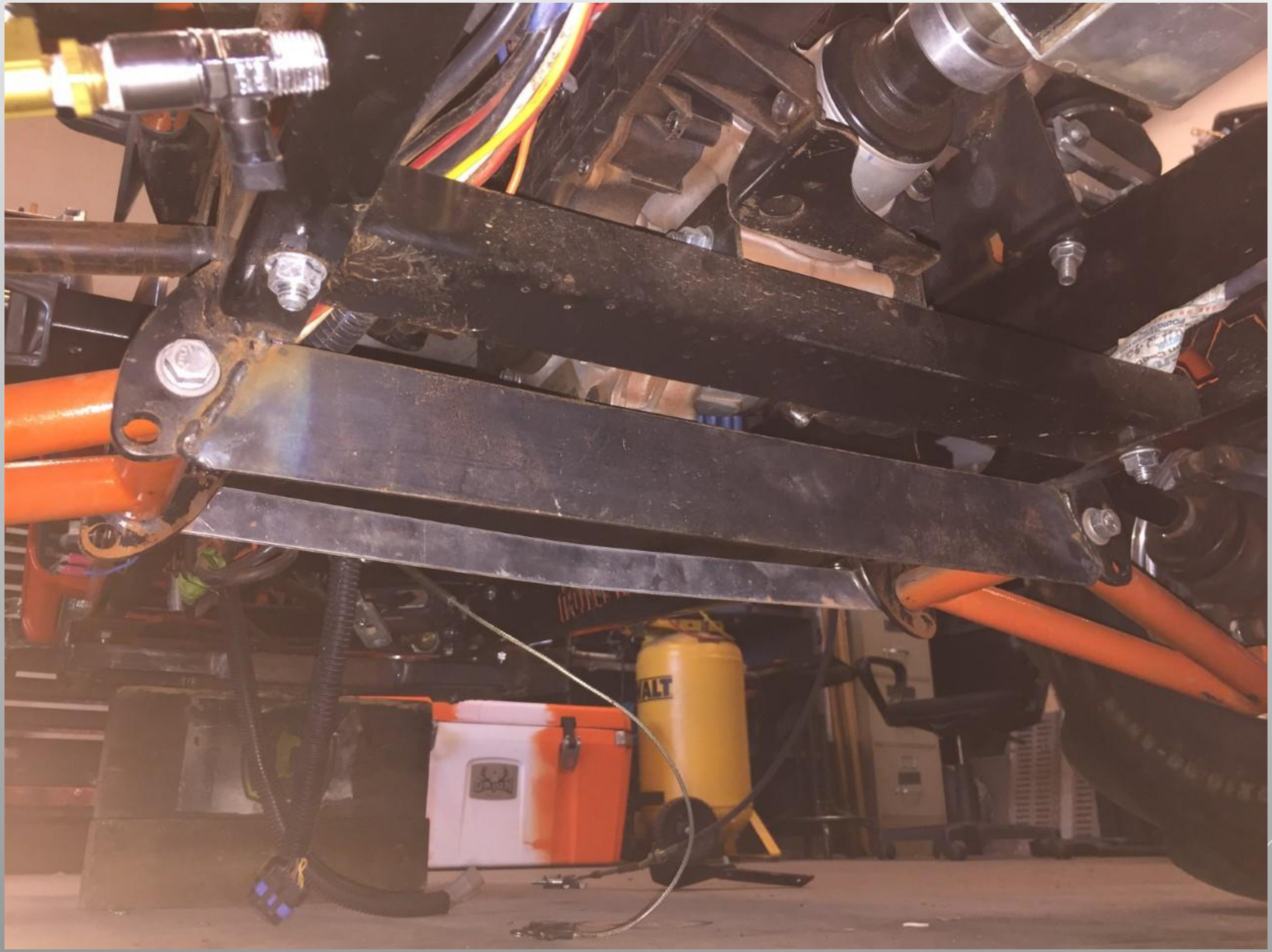
NEW DESIGN: REAR END

- Bolted Connection: Six 3/8" Grade 8 UNC Bolts



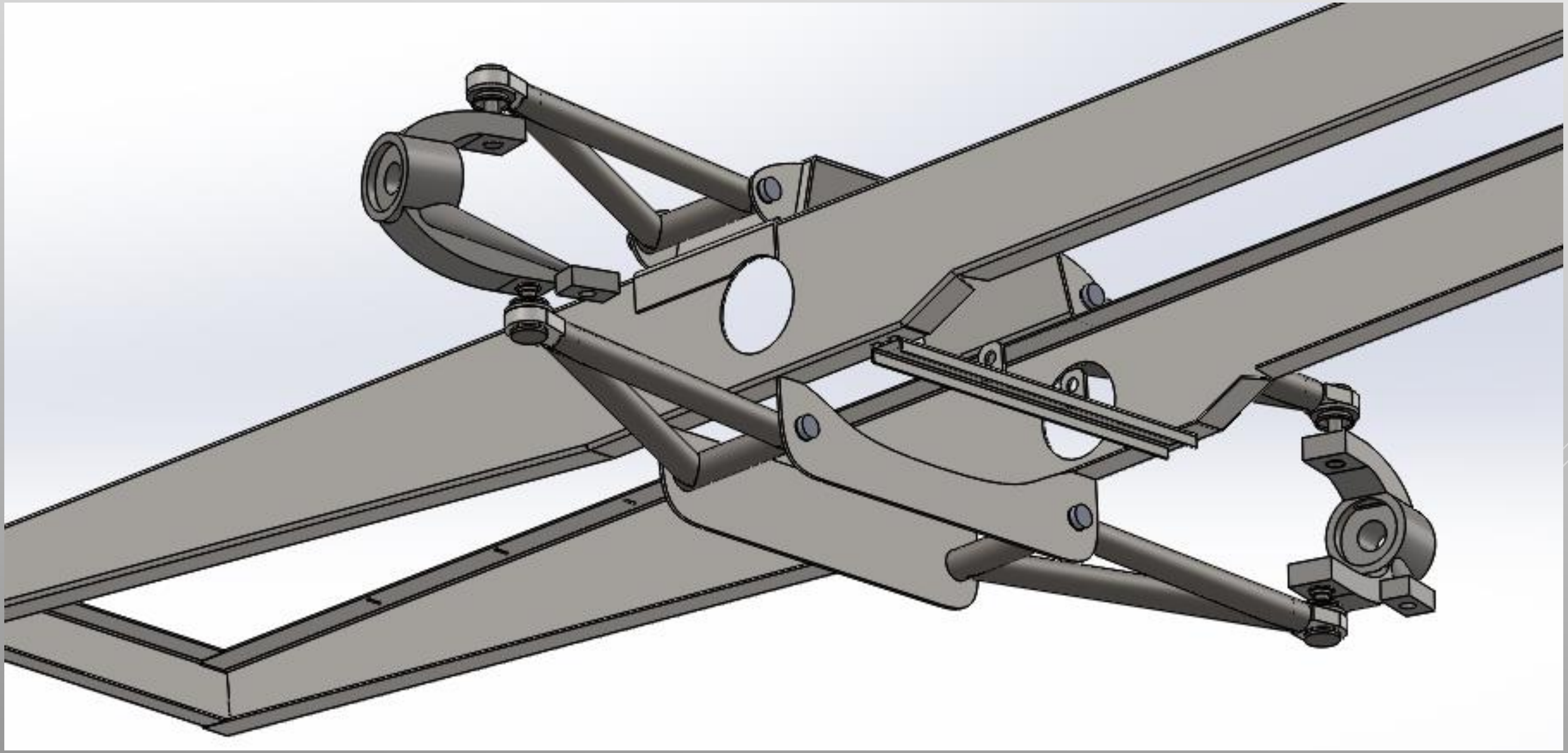


OLD DESIGN: FRONT AXLE



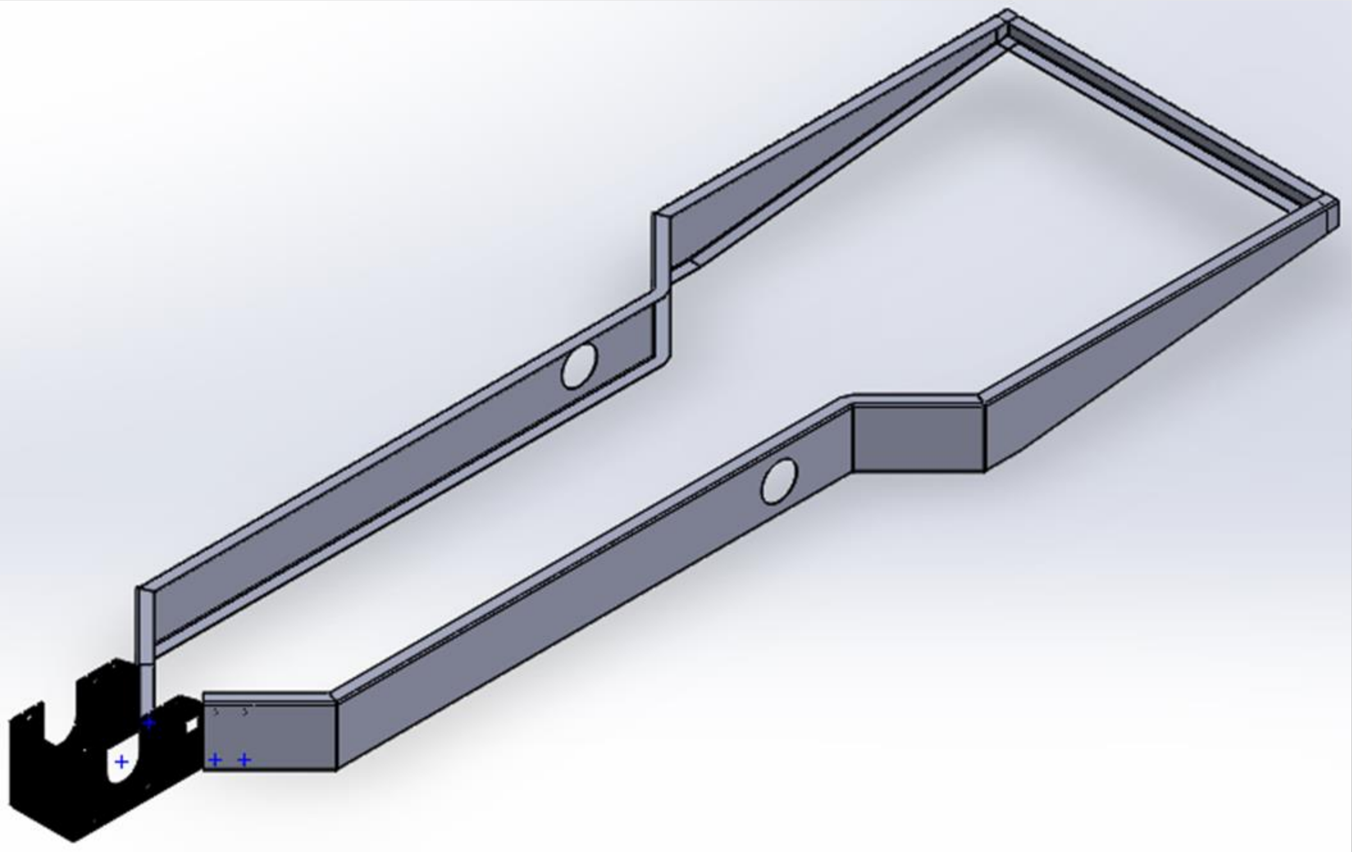
NEW DESIGN: FRONT AXLE

- Incorporated support structures



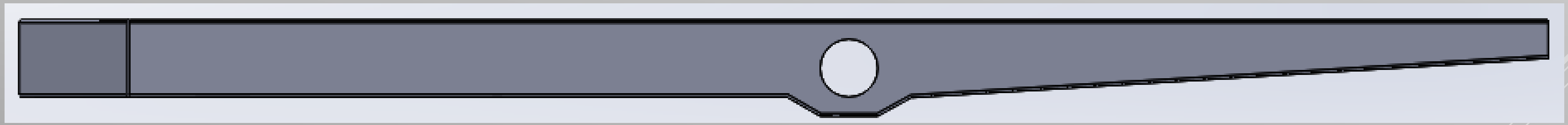
FRAME RAIL SELECTION

- Wide Engine Frame
  - Designed to lower the engine
  - Decided to not lower the engine



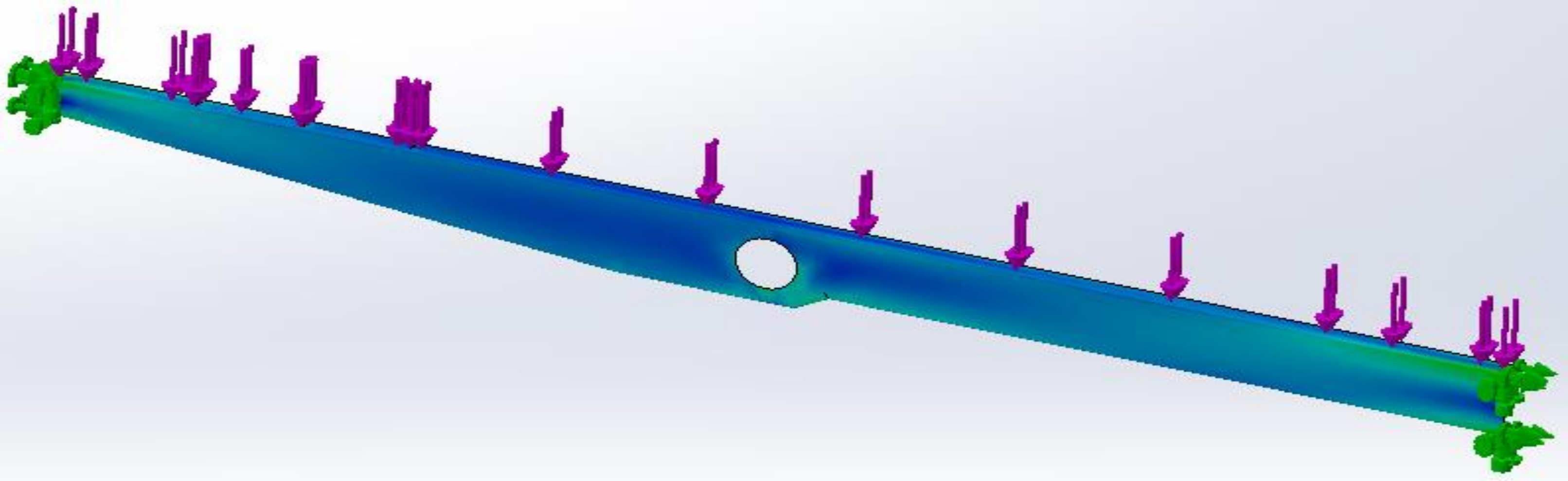
FRAME RAIL SELECTION

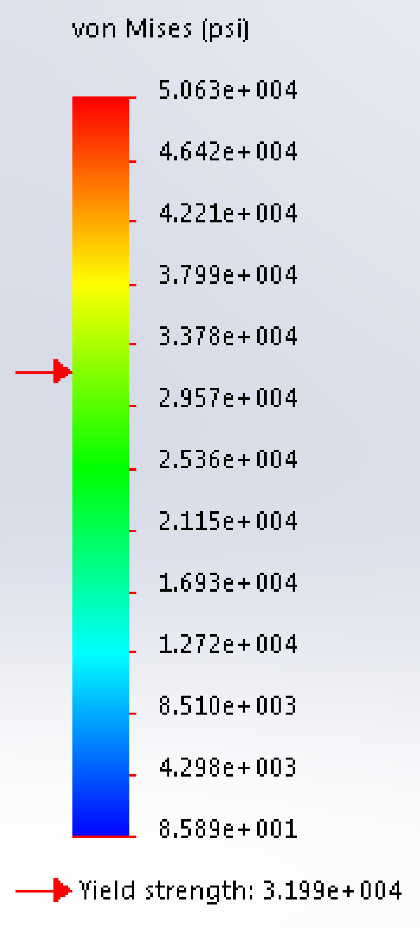
- Short Frame
  - Designed to reduce material
  - Did not fit with new front axle design



FRAME RAIL SELECTION

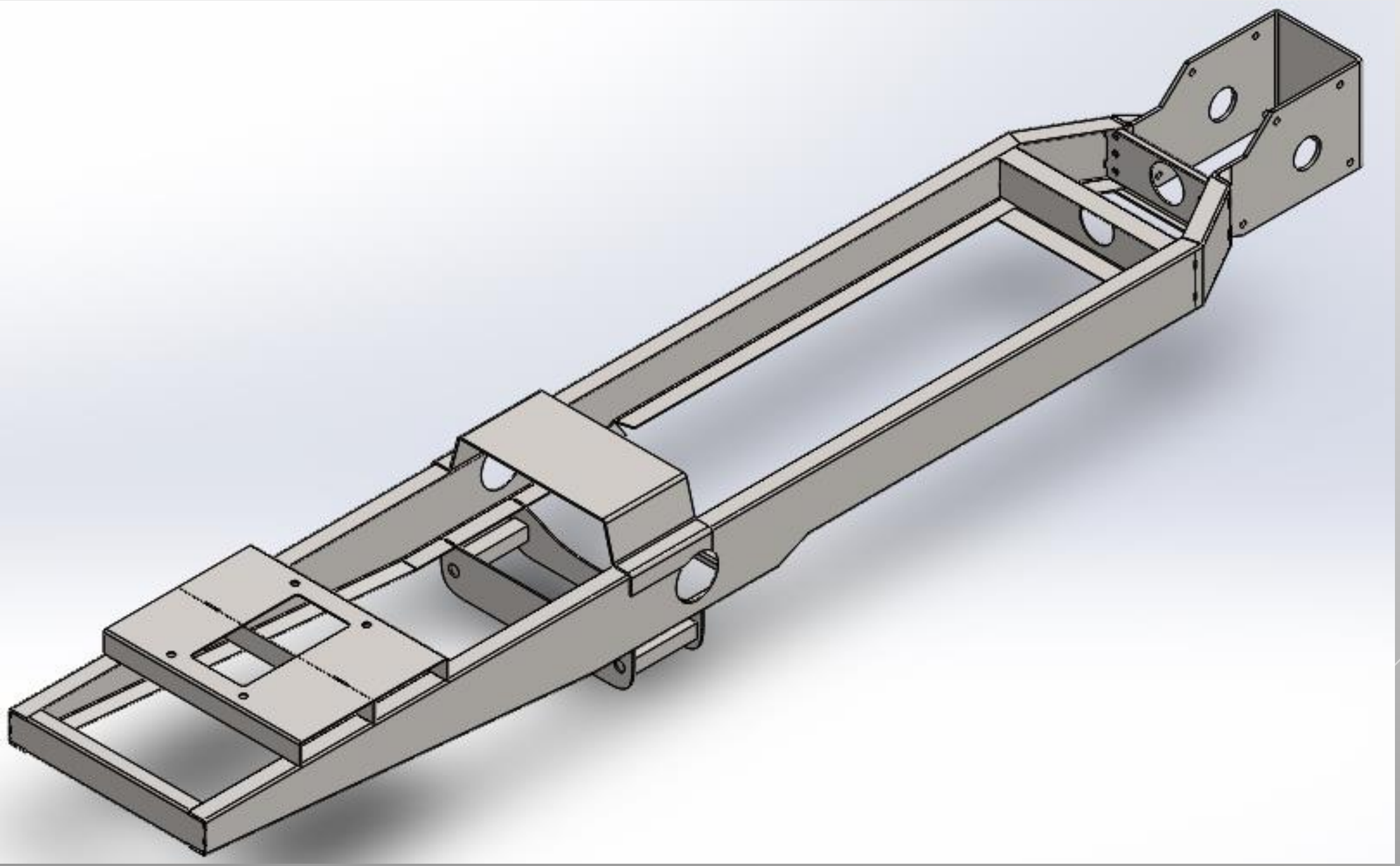
- Height decreases after front axle from 5" to 4"
- 78.5" long
- 14 gauge steel



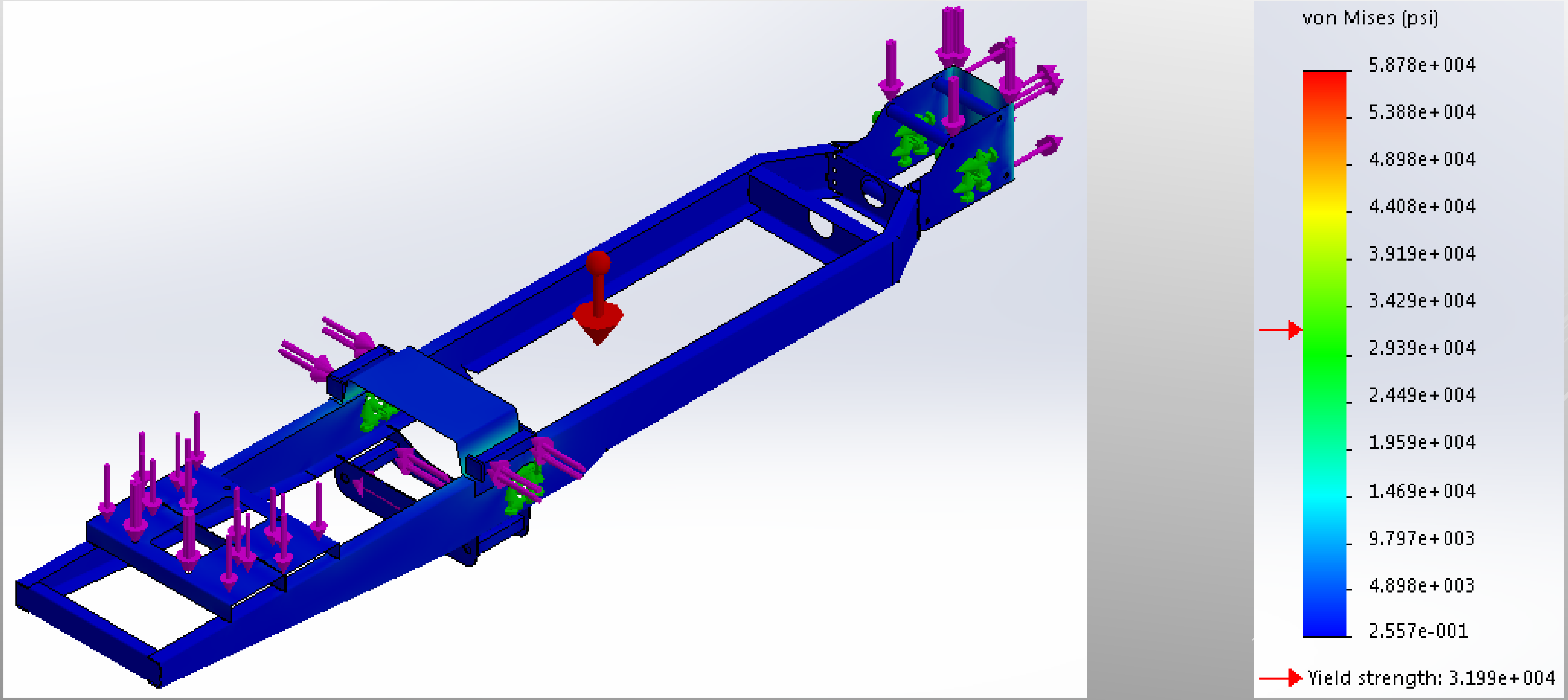


OVERALL ASSEMBLY

- Width reduced from 17" to 14.5" when compared to previous design
- 90" long



OVERALL ASSEMBLY SIMULATION

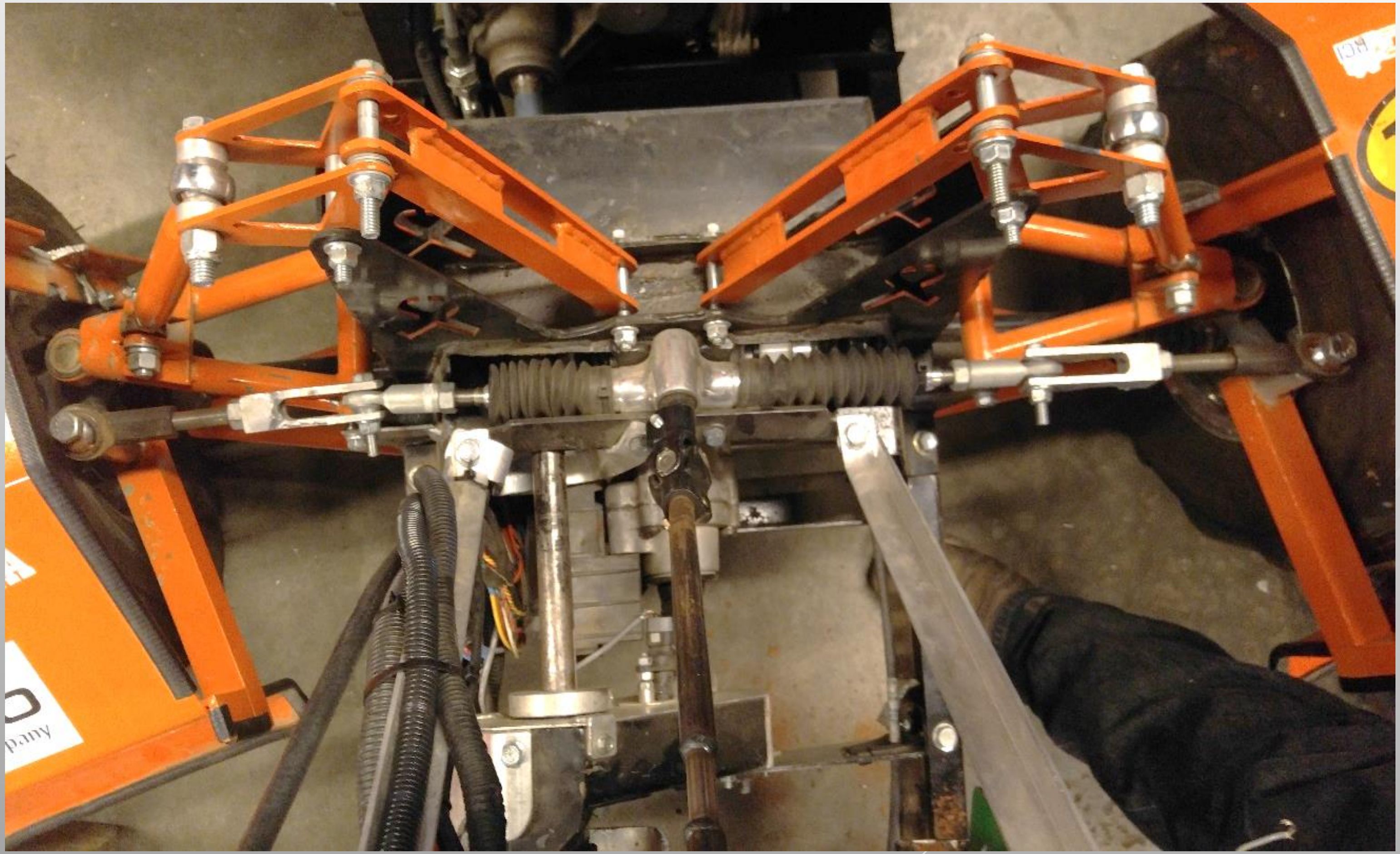


STEERING DESIGN GOALS

- Ease of steering
- Adjustability
- Reliability
- Low maintenance

PREVIOUS DESIGN

- Strengths
  - Manufacturability
  - Simple
  - Lightweight
- Weaknesses
  - 1:1 ratio
  - Heavy steering
  - Poor turning radius



Steering assembly 2015-2016 competition year

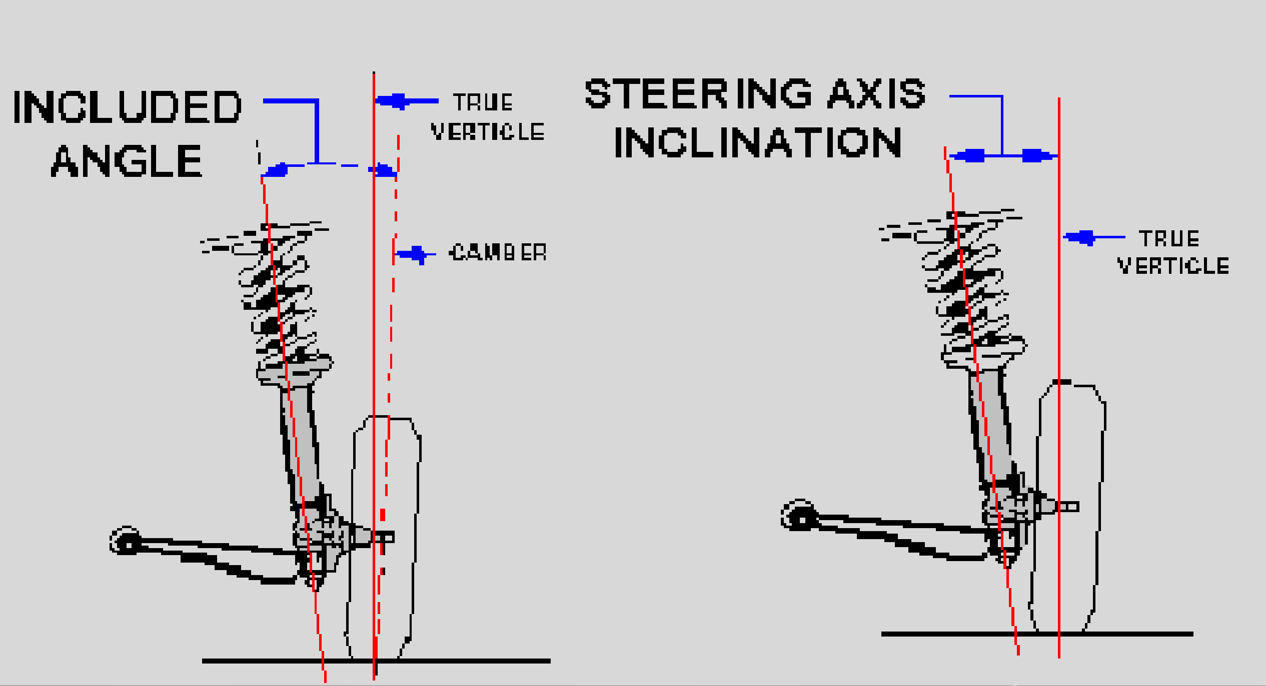
TOE ALIGNMENT PROBLEM

Air springs suspension fully inflated

Air springs suspension at pull height

STEERING FACTORS AND ALIGNMENT

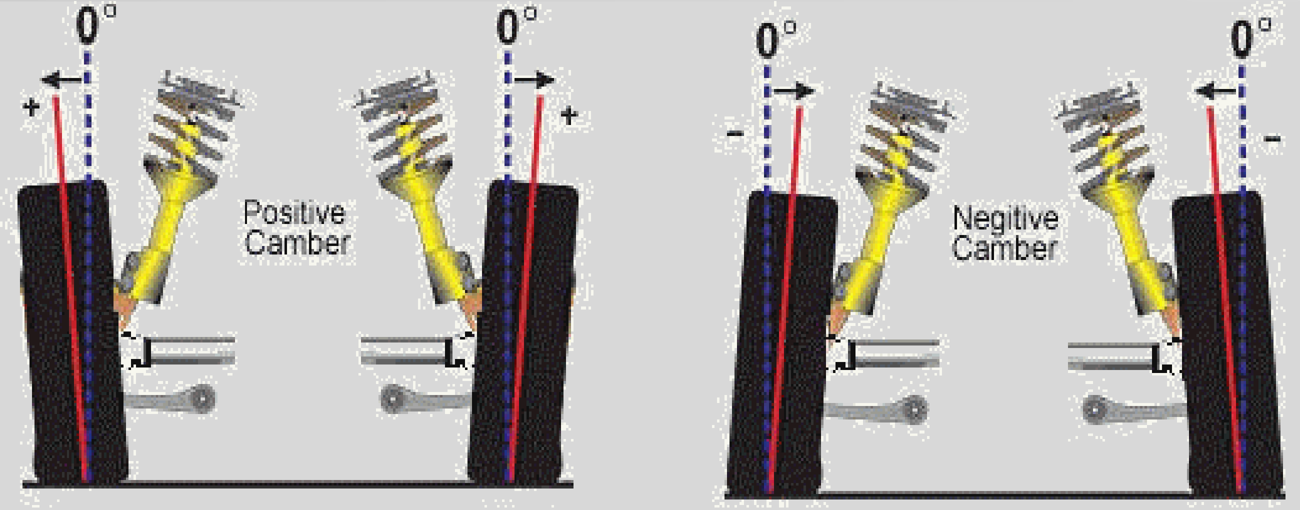
- Camber
- Caster
- Toe
- Geometry
- Systems



From: Auto Dimensions Inc.

CAMBER

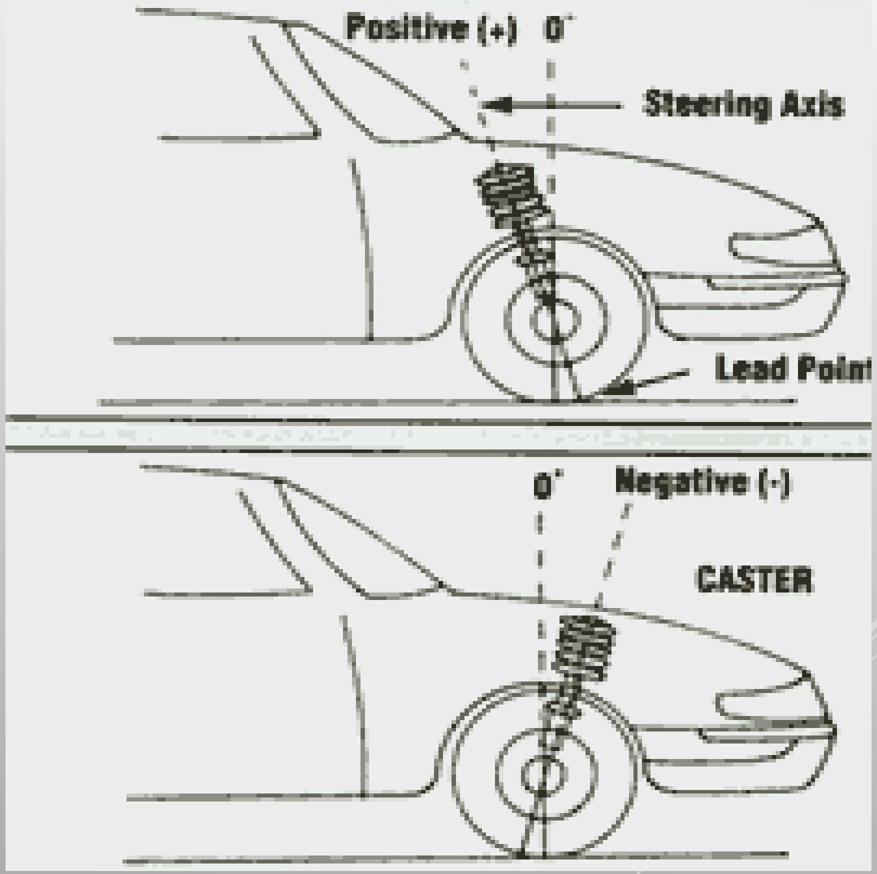
- Angle between true vertical and centerline of tire
- Direct effect on toe
- Can change with ride height



From: Auto Dimensions Inc.

CASTER

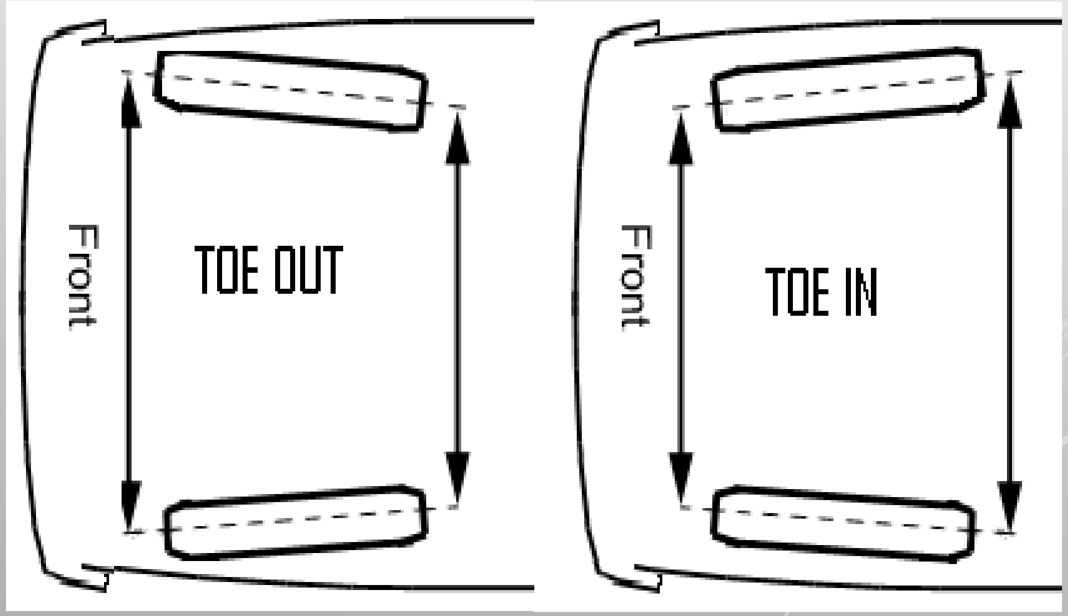
- Angle of the steering pivot
- Effects straight line tracking
- Steering Effort
  - Lower angle for less effort
  - Positive steering is heavy
  - Negative steering is light



From: Auto Dimensions Inc.

TOE

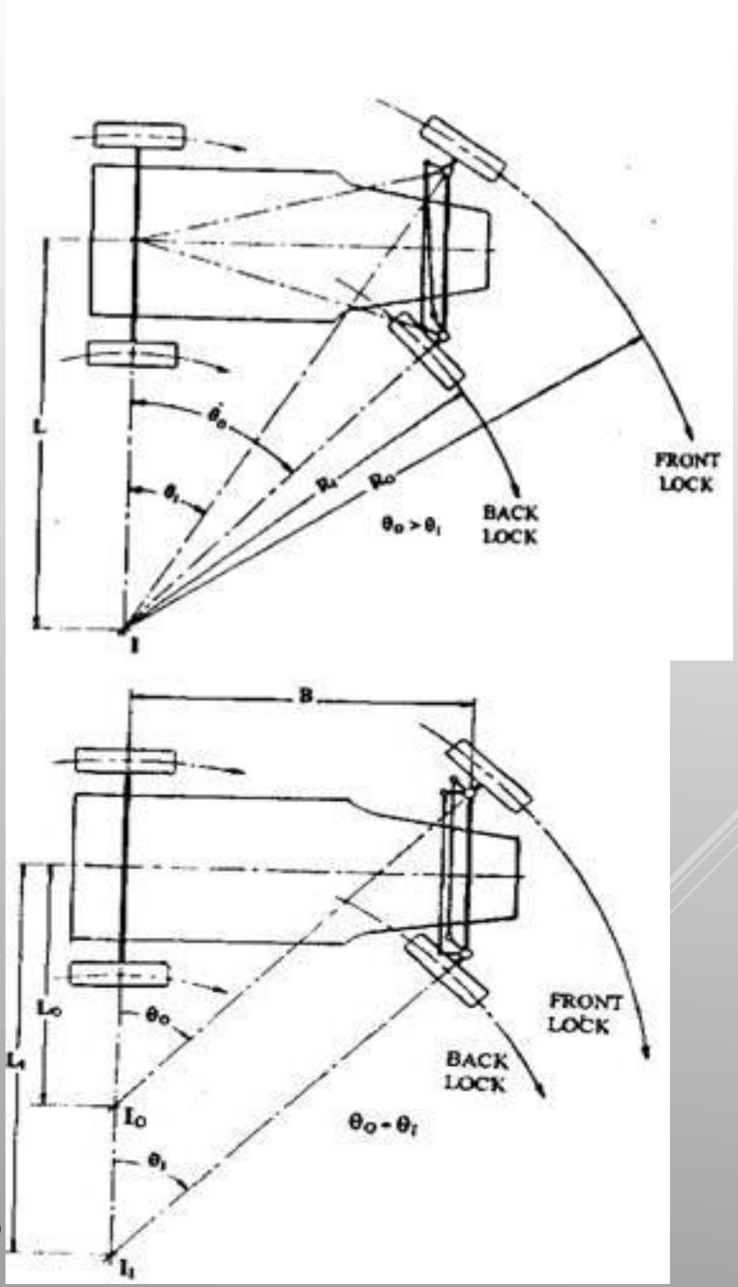
- Changes with ride height
- Steering characteristics
  - Toe-in increased understeer
  - Toe-out increased oversteer
- Vehicle stability



From: Auto Dimensions Inc.

STEERING GEOMETRY

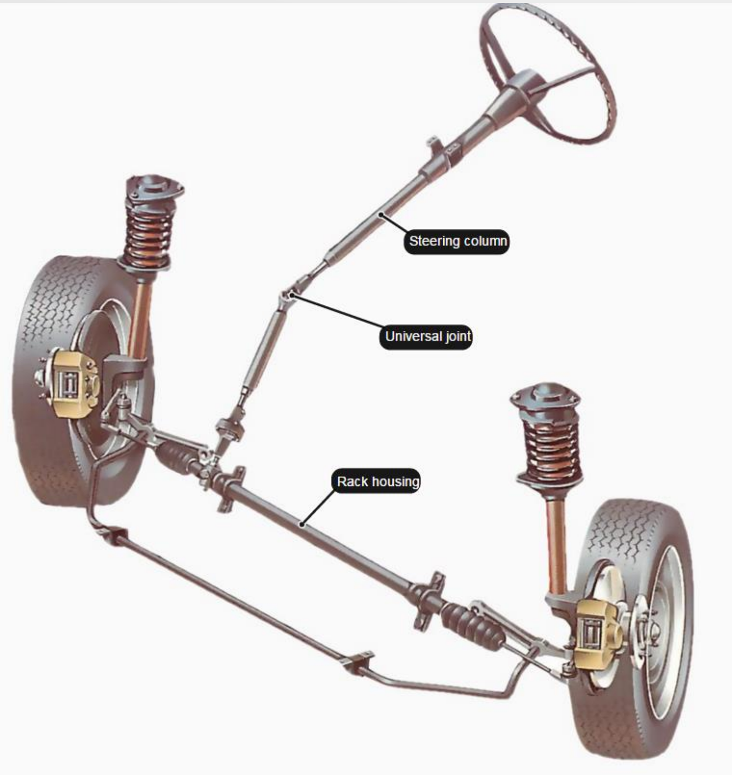
- Ackerman
  - Minimizes tire slip
  - Pure geometry is never used
- Parallel Set
  - Wheels turn same angle
  - Easiest to produce



From: The Ackermann Principle as Applied to Steering

STEERING SYSTEMS

- Rack and pinion
- Steering box
- Electric power assist
- Electronic steering
- Hydraulic



From: *How the Steering System Works*

STEERING SYSTEMS COMPARISON

Mechanism	Mech. Linkage	Steering Box	e-Power Assist	Electronic steering	Hydraulics
Cost	5	3	2	3	1
Parts Availability	4	3	2	5	5
Weight	2	2	4	5	1
Steering Ease	3	3	4	5	5
Reliability	5	5	4	1	3
Feasibility	5	4	4	0	0
Safety	4	4	4	1	3
Total score	28	24	24	20	18

Numbers based on scale from 1-5
Cost (High to Low)
Parts (Low to High)
Weight (High to Low)
Ease of Steering (Hard to Easy)
Reliability (Low to High)
Feasibility (Low to High)
Safety (Low to High)

STEERING DESIGN

- Rack and pinion
  - Improve previous design
    - Line of force
    - Geometry
  - Lessons learned
- Chrome-moly turnbuckles
  - Weight to strength ratio
  - Team experience
- Gear reduction

SIZING THE TURNBUCKLES

4130 CHROME-MOLY

- Cost per foot under $4
- Lightest per foot
- Hardware

Chrome-Moly Tube Steering Analysis (4130)						
OD (in)	ID (in)	T (in)	Cost Per Foot ($)	Weight Per Foot (lb)	Max Shear (psi)	Safety Factor
0.500	0.430	0.035	3.590	0.181	86345	0.731
0.500	0.402	0.049	3.450	0.236	67189	0.939
0.500	0.384	0.058	3.480	0.267	59980	1.052
0.500	0.370	0.065	3.500	0.289	55866	1.129
0.500	0.310	0.095	8.630	0.353	45895	1.375
0.500	0.260	0.120	5.680	0.374	42199	1.495
0.625	0.555	0.035	2.890	0.233	52951	1.192
0.625	0.527	0.049	3.330	0.310	40498	1.558
0.625	0.509	0.058	4.050	0.354	35754	1.765
0.625	0.495	0.065	5.420	0.386	33017	1.911
0.625	0.385	0.120	7.960	0.554	23394	2.697
0.750	0.680	0.035	3.280	0.286	35742	1.765
0.750	0.652	0.049	3.180	0.383	27023	2.335
0.750	0.634	0.058	3.640	0.441	23682	2.664
0.750	0.620	0.065	4.030	0.484	21743	2.902
0.750	0.584	0.083	4.200	0.582	18326	3.443

SUSPENSION OBJECTIVES

- Ride Height Adjustment
  - Scales, Brake test, Maneuverability, and Pulling
- Improve Ride Quality
  - Operator comfort and improve durability

PREVIOUS DESIGN

Rigid Suspension Lessons Learned

- Manually adjustable
- Light weight
- Limited potential travel
- No articulation
- No damping

INITIAL CONCEPTS

- Coil over shock absorber
- Linear actuators
- Hydraulic cylinders
- Air shocks
- Air springs

INITIAL CONCEPTS CONTINUED

Selection Criteria

- Objectives
- Feasibility
- Weight
- Weight transfer
- Price

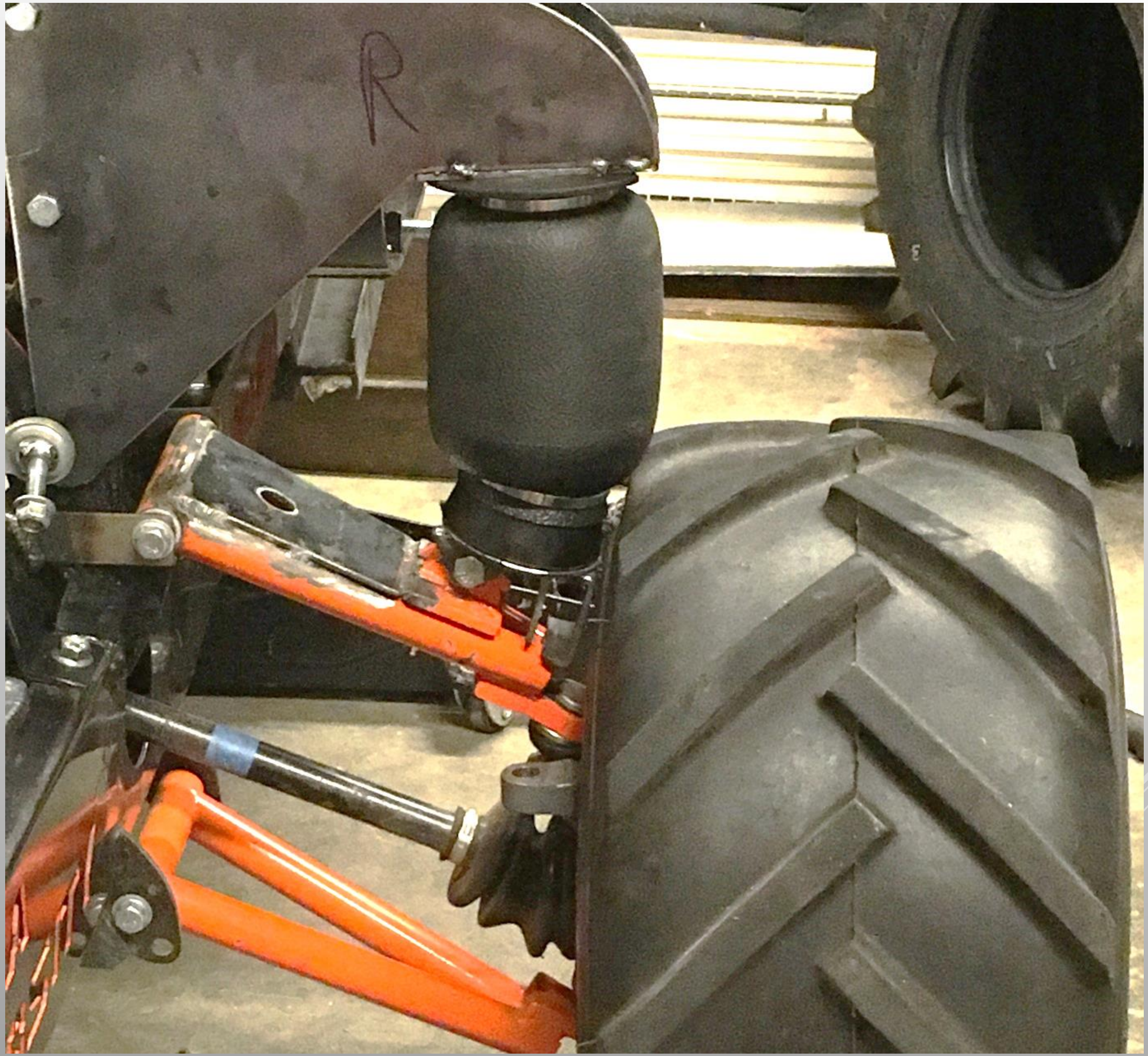
Design Concept	Lift Mechanism	Ride Quality	Feasibility	Weight	Weight Transfer	Price	Total
Coilover shock abs.	1	5	4	3	3	3	19
Linear Actuator	4	1	5	5	4	2	21
Hydraulic cylinders	5	2	1	1	5	1	15
Air shocks	2	3	2	2	2	4	15
Air springs	3	4	3	4	1	5	20

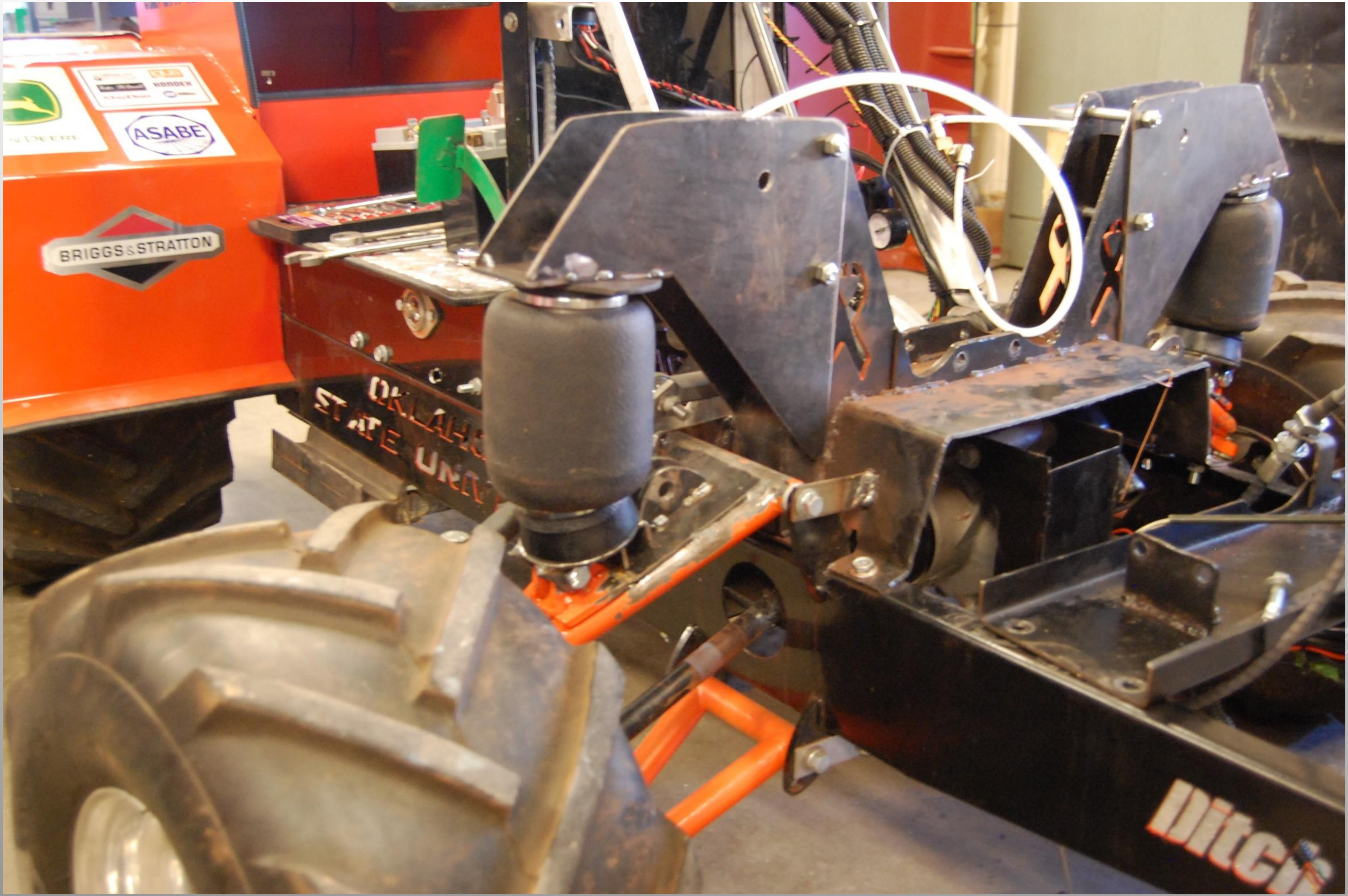
5 = Best in Category

1= Worst in Category

TESTING

- First Iteration
  - Overloaded
- Second Iteration
  - Clearance
- Third Iteration
  - Working prototype

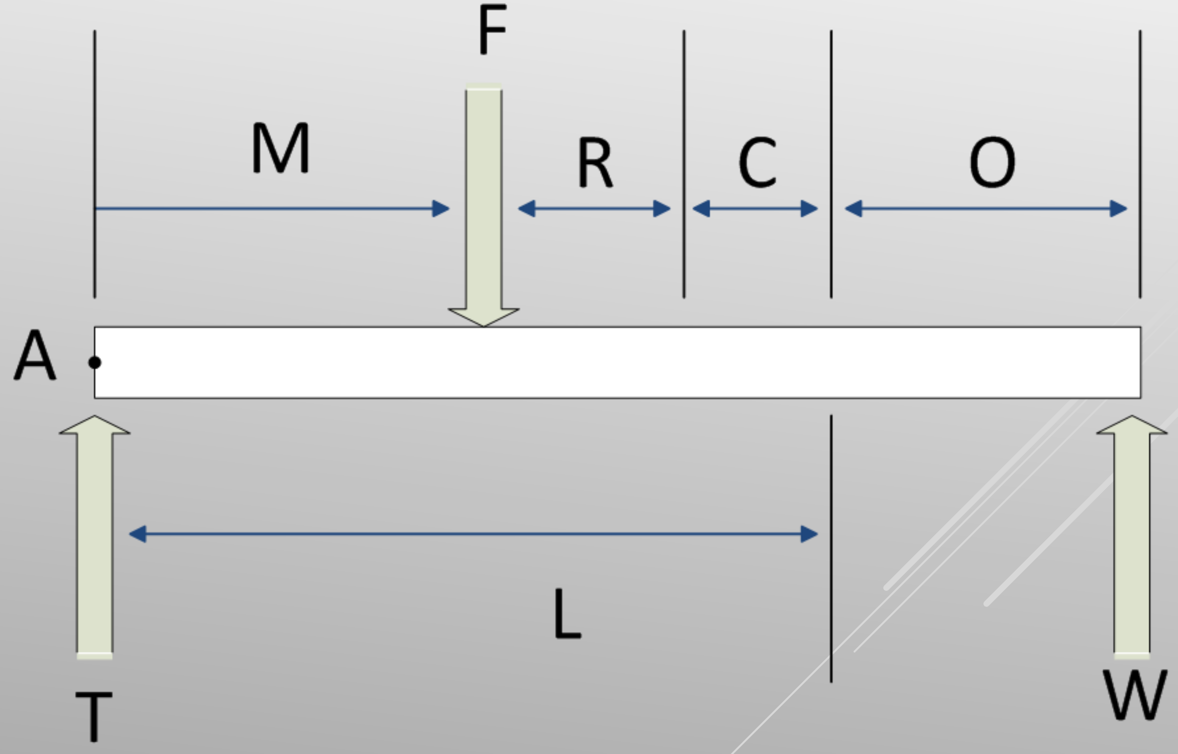



BRIGGS&STRATTON
ASABE


ASABE
OSU
EMS

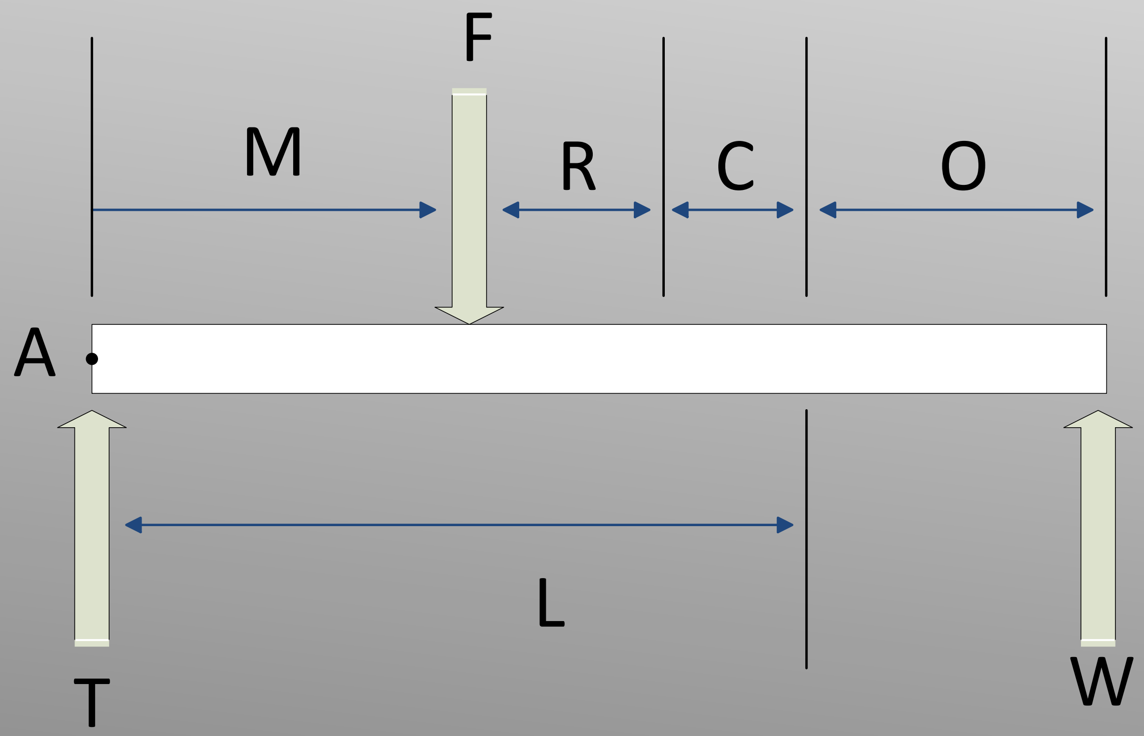
AIR SPRING SELECTION

- $M_A = 0 = (W)*(L+0) - (F)*(M)$
- $F = (W)*(L+0)/M$
- W= Weight on each front tire
- L= Length of A-arm
- F= force required to lift the tractor
- M= distance from center of air spring to center of A-arm pivot point



AIR SPRING SELECTION

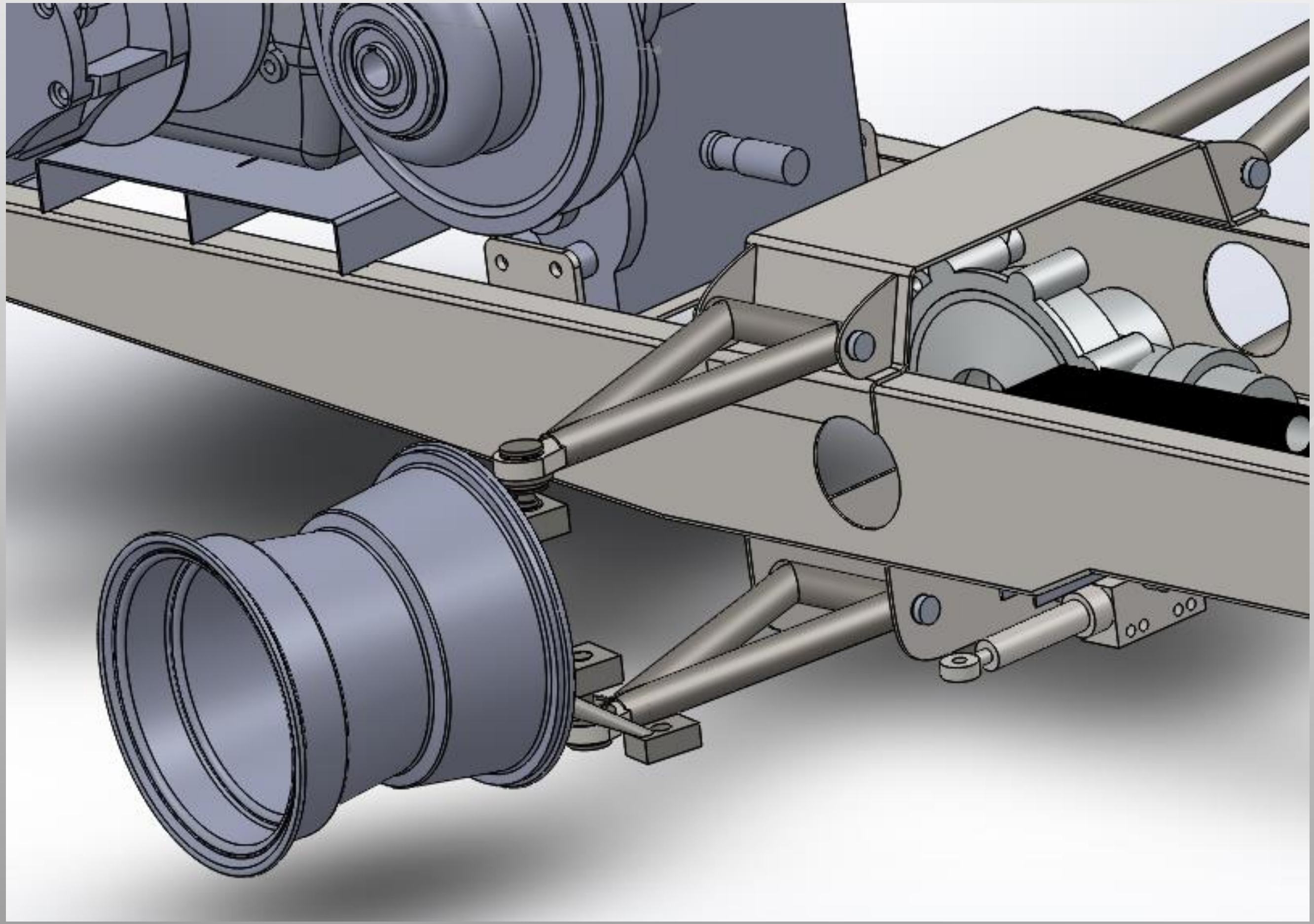
Part number	Max load at 100 Psi	Max diameter (in)	R (in)	M (in)	Force needed (Lbf)	Safety factor
58407	2210	7	3.5	5.64	2144.7	1.03
58124	3340	9.4	4.7	4.44	2724.3	1.23
58616	3055	8	4	5.14	2353.3	1.30



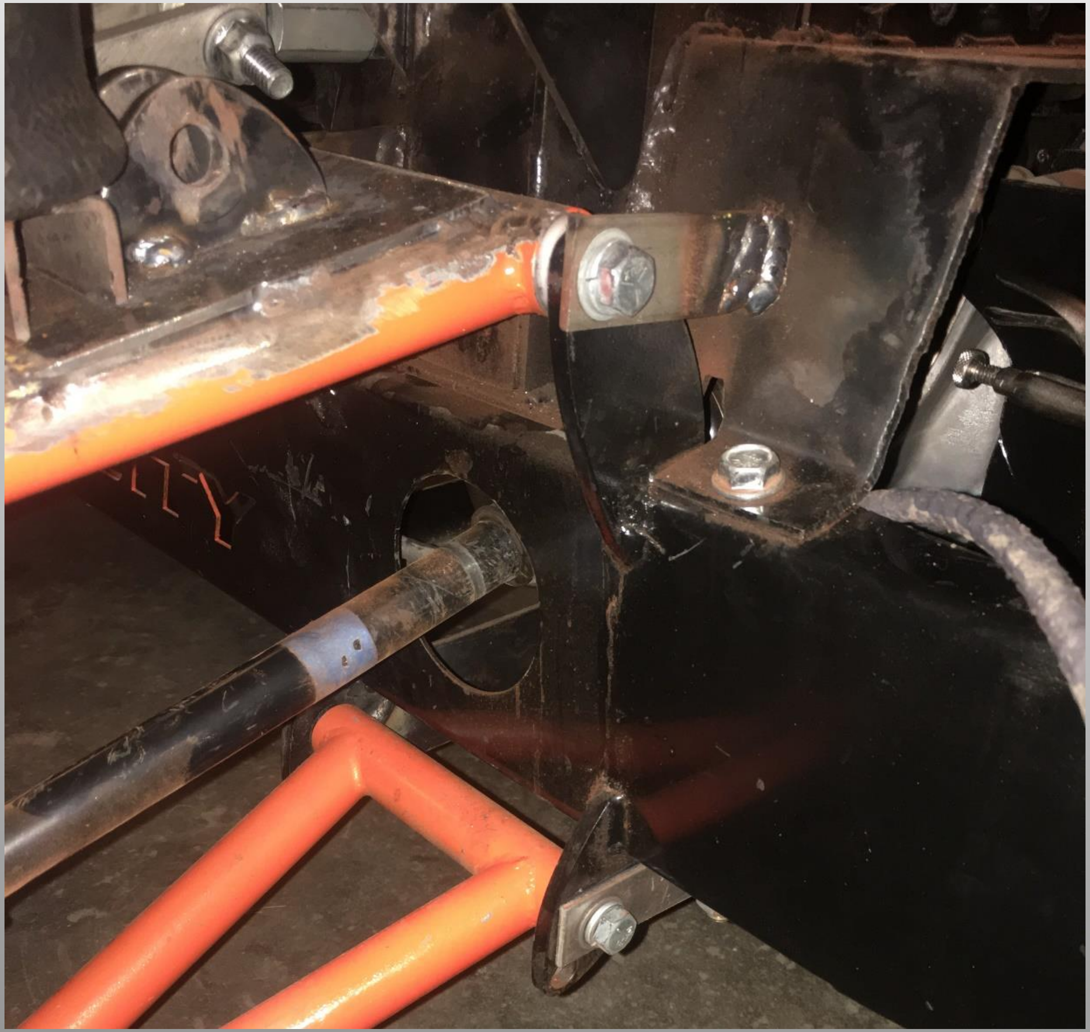
L (in)	O (in)	C (in)	W (Lbf)
11.64	5.64	2.5	700

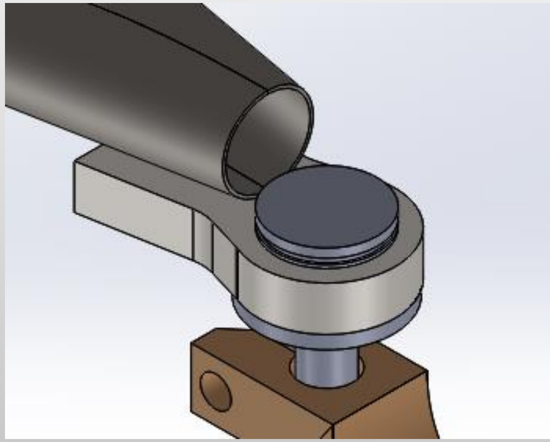
A-ARM DESIGN

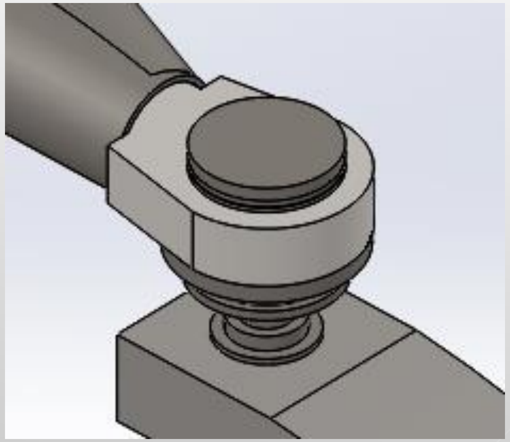
- 1in O.D. Chrome-moly tubing
- Right angle
- Double wishbone
- Improved serviceability
- Improved manufacturability

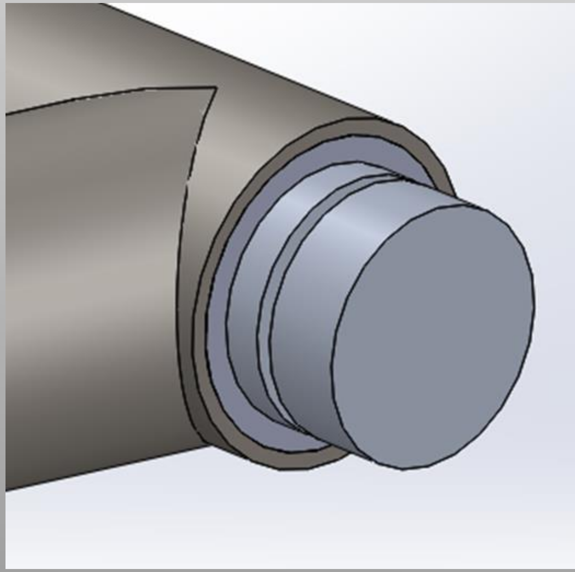


A-ARM DESIGN CONTINUED



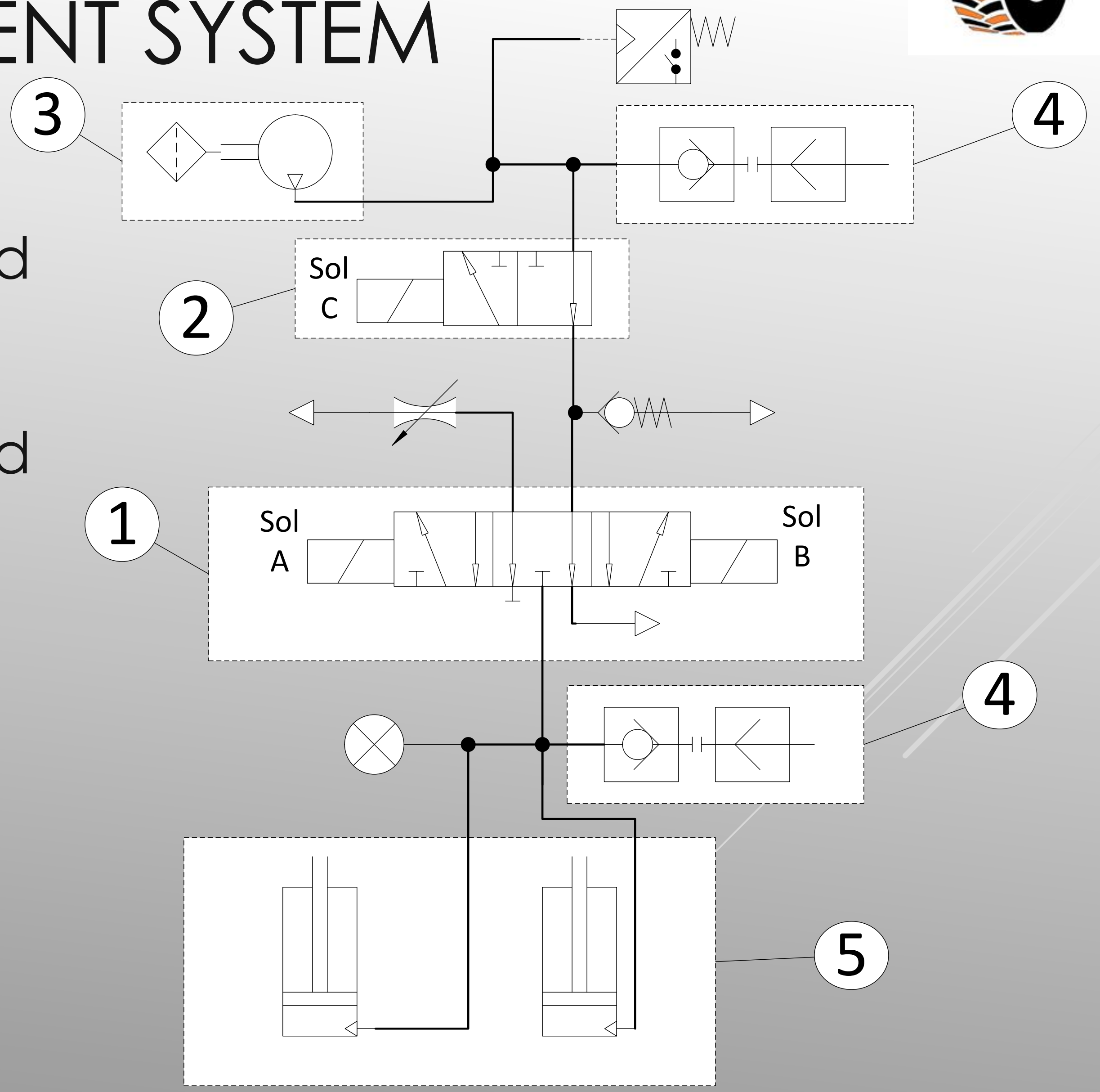






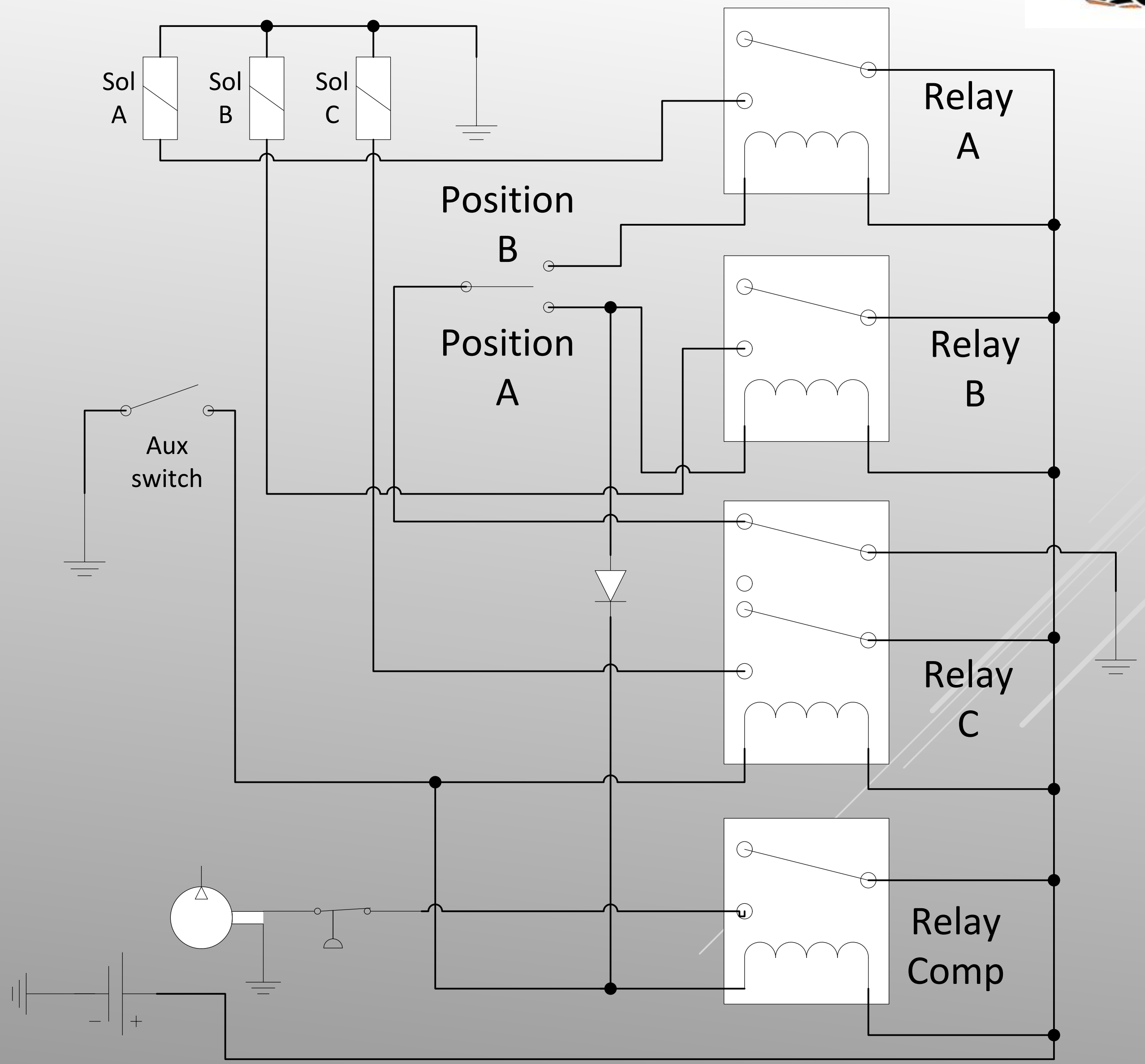
PNEUMATIC MANAGEMENT SYSTEM

- 1: 5 port, 3 way, solenoid controlled pneumatic valve
- 2: 3 port, 2 way, solenoid controlled pneumatic valve
- 3: 200 psi max air compressor
- 4: Auxiliary quick disconnect
- 5: Dual air springs



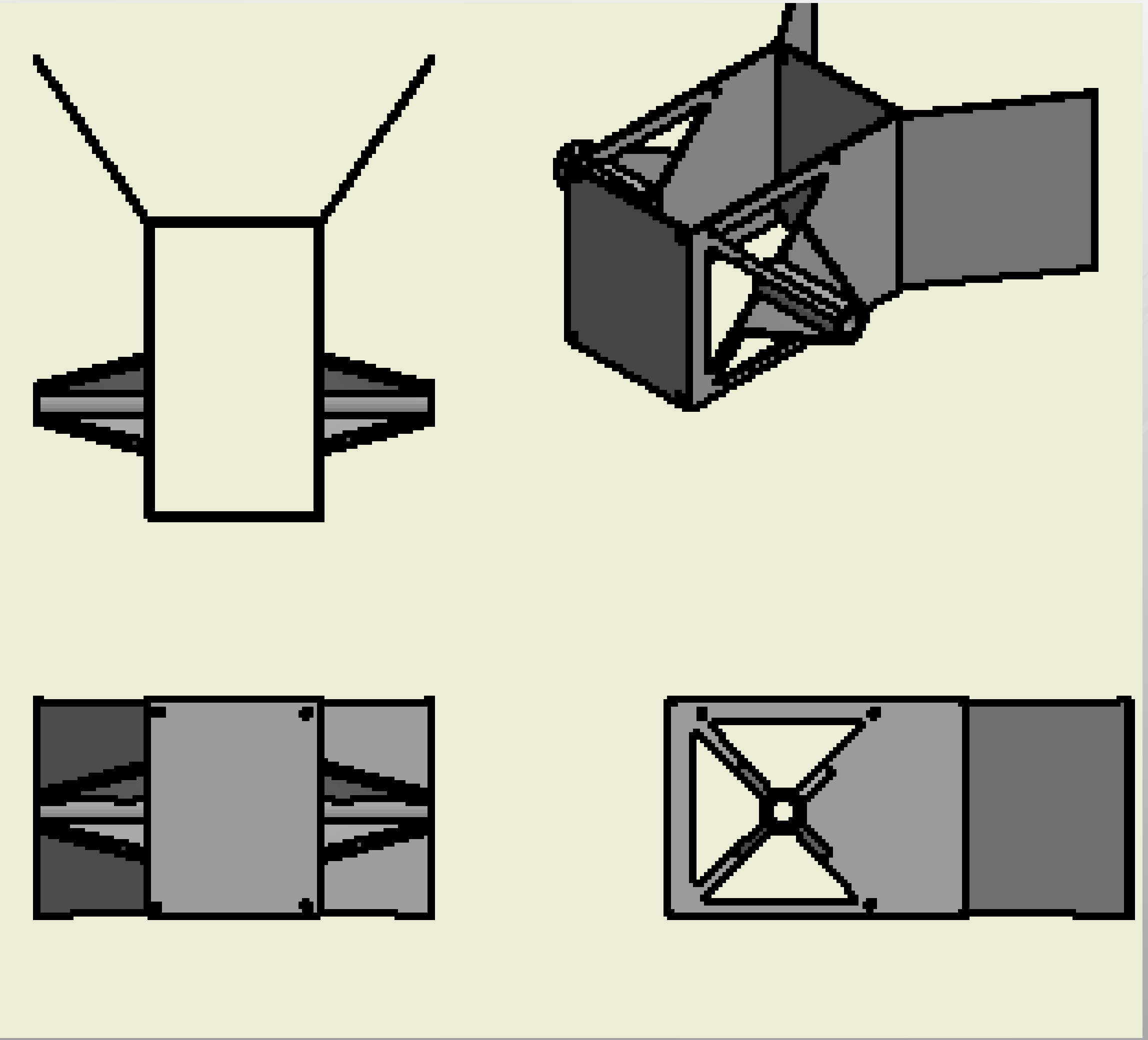
PNEUMATIC MANAGEMENT SYSTEM CONTINUED

- Inflate air springs
  - Switch position A
- Deflate air springs
  - Switch position B
- Fill aux reservoir
  - Activate Aux switch



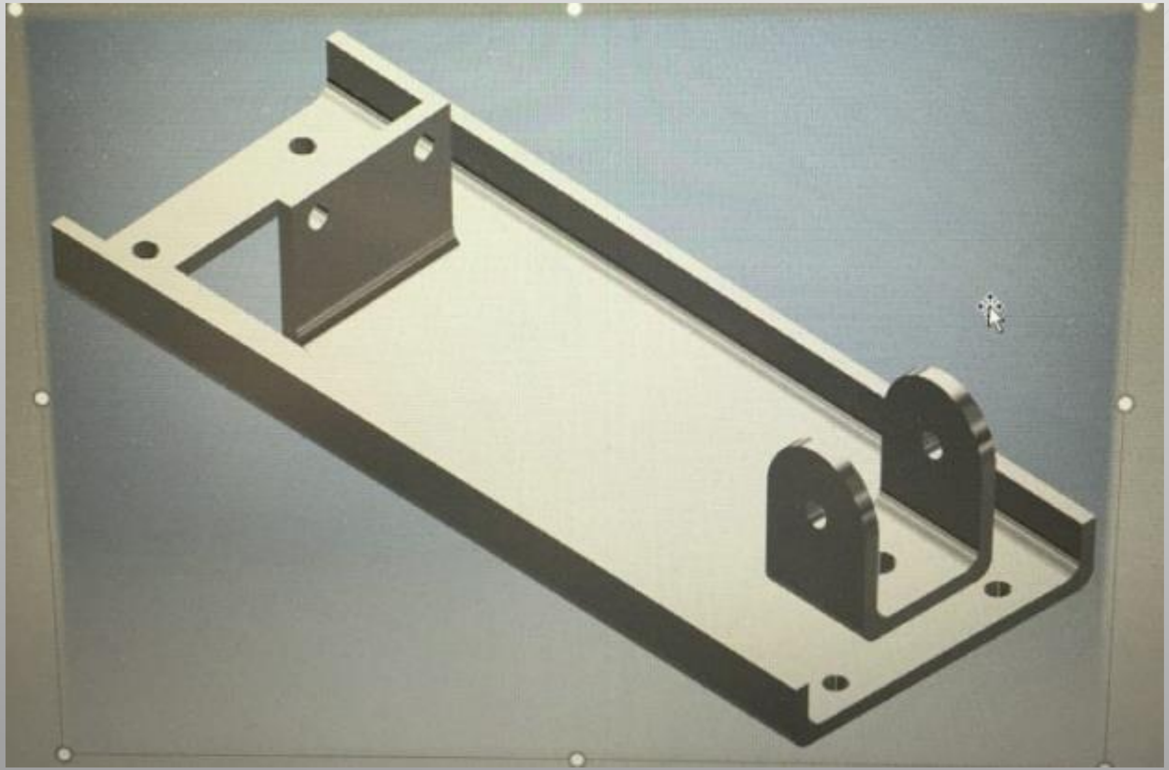
FRESHMAN INTERACTION

- Rear differential mount
  - Micah Arthaud, Shyanna Hansen, Michael Leiterman, Nick Liegerot, Heath Moorman



FRESHMAN INTERACTION CONTINUED

- Transmission mount
  - Jeremiah Foster, Brent Gwinn, Creston Moore, Austin Pickering, Ross Ruark



SPRING SEMESTER

- Finish Solidworks model
- Send parts to be manufactured
- Assemble prototype
- Test

THANK YOU FOR YOUR TIME

QUESTIONS?

SOURCES

- Auto Dimensions Inc. (2016, Septermber 23). *Wheel Alignment Explained*. Retrieved from Anewtoronto.com: http://www.anewtoronto.com/wheel%20alignment.html
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