Rose Float Overheight Final Project Report

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Statement of Confidentiality

The complete senior project report was submitted to the project advisor and sponsor. The results of this project are of a confidential nature and will not be published at this time.

Statement of Disclaimer

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Table of Contents

Table of Figures

Table of Tables

Intro

The Rose Parade can be dated back to 1890 and has since become an American New Year celebration. It started as a small effort by Pasadena's Valley Hunt Club with the purpose of promoting the "Mediterranean of the West". The festival grew too large for the club to handle, so the Tournament of Roses Association was formed in 1895. Today, the Rose Parade consists of elaborate floats that feature computerized animations and beautiful botanical material from around the world.

The Cal Poly Universities jointly build and enter a floral entry, commonly known as a float, into the Pasadena Tournament of Roses Rose Parade. These two institutions have continuously participated in the parade since 1949 - winning the Award of Merit in their first year. The Cal Poly floats have introduced new technology to the Parade, including the use of hydraulics for animation in 1968, computer-controlled animation in 1978, fiber optics in 1982, and animated deco in 2014. As of January 1, 2017, the floats have won 57 awards. This program is one of the longest consecutive running self-built entries in the parade, as well as the only self built float designed and built entirely by students year-round on two campuses. They compete against professional float builders who manufacture entries for sponsors and companies, many of them with development budgets approaching $1 million. This tradition continues today and marks the partnership between the two campuses.

For the parade, floats drive down Colorado Boulevard. At the end of this route, there is a 16'6" bridge all floats must drive under. The scope of our project is to design and build a mechanism roughly described as an "overheight" mechanism, as its function is to raise and lower large heavy structures so the float is able to pass under the bridge. This mechanism is powered via the float animation system. For Cal Poly, our float has a chassis made of c-channel with specific bolt patterns, seen in Figure 1.

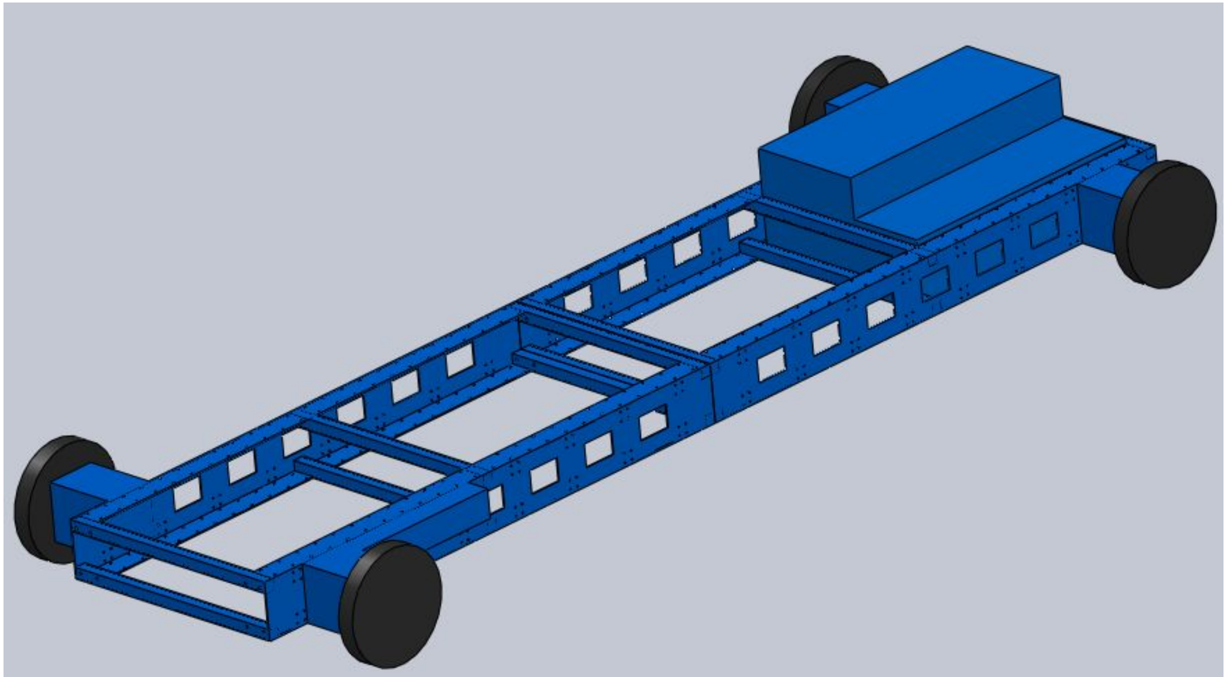

Figure 1. Chassis for Cal Poly Rose Parade Floats

These bolt patterns allow for the chassis to be split into smaller square segments. Thus, the parade float chassis is able to split into two halves, one for San Luis Obispo and one for Pomona campus. On each half of the chassis, there are two hydrostatic wheels and an engine. One half has an engine to power the drive system and the other section's engine powers the animation system. The drive engine powers the hydrostatic drive for the float while the animation engine powers the hydraulic pump and generator. The hydraulic pump is connected to a hydraulic distribution manifold which is then connected to several hydraulic proportional directional control valves. A schematic of our hydraulic animation system is provided in Figure 2 below.

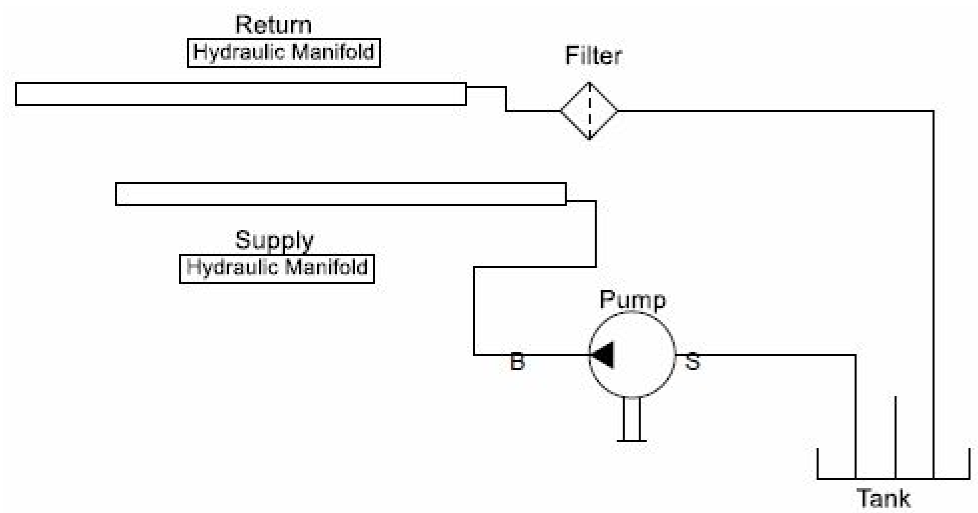

Figure 2. Schematic of Animation System

Each of these proportional directional control valves controls various programmable mechanisms on the float. The overheight mechanism is connected to a spool valve for manual control. Overheight mechanisms are manually controlled as a precaution for any emergencies that may arise during the parade to our electrical system.

During the parade, there are four operators within a float to operate both engines, navigate, and steer the float. Each operator has a separate compartment within the float called a crew compartment, and the engine operators often have no view to the outside world. The animation engine operator not only controls the programmable mechanisms, but also will have access to the manual valve for the overheight mechanism. Using a radio communication system, the operator is told when to raise and lower the overheight. At any time during the parade, the overheight must be able to raise and lower. Moreover, if our animation system loses pressure, the overheight must lower without any powered assistance. This safety feature is often referred to as the gravity drop of an overheight as it is the overheight's ability to fall under gravitational forces without system pressure.

Another aspect to overheight mechanisms besides their mechanical and safety features is their implementation into float designs. Each year, Cal Poly builds a vastly different float design with different characters and dimensions. As seen in Figure 3[1][2], the overheight structures being raised and lowered are vastly different. Thus, in order to create an overheight mechanism usable on 5 different floats it must have a few key features. Common among various overheight structures is their heaviness due to the size and weight of floral decorations. However, there is usually limited volumetric space for a lifting mechanism. Since the design of the float attempts to be artistically appealing, usually elements (such as the castle towers) on the float have a small cross-sectional area and are long (Figure 4[3]).

Figure 3. Variation in Float Designs

Figure 4. Cal Poly's Soaring Stories Float

Our sponsor for this project is the advisor of Cal Poly Rose Float, Josh D'Acquisto. He will review our design and is providing the financial means to complete the project. Additional financial aid may also be provided as needed from the Rose Float Alumni Association. By investing in this project, the customer expects the mechanism to provide use across multiple floats. If this project is used multiple years, it will help decrease the amount of work necessary for the team, as they build a new float each year.

Background

The Tournament of Roses provides a Float Manual, or a set of rules on how to build and decorate a parade float, to each builder of the Rose Parade.[4] Within the Float Manual, there are construction standards for the drive system, animation, safety, exhaust, overheight mechanisms, and more. Additionally, there is information on the inspection process, the float, and all mechanisms on the float. In the design considerations section, an overheight mechanism must retract to under a height of 16'-6" in a time frame of less than 60 seconds. These two specifications are the primary constraints any overheight must adhere to. A general design of the overheight mechanism must be described to the Design/Variance Committee with the float concept rendering in May at the first Design/Variance meeting. The committee will conditionally approve the design for the mechanism until further review on the inspection dates. Moreover, compelling design reasons may be presented to the Design/Variance Committee to obtain approval for a longer time to drop the overheight.

Every float has to pass three primary inspections by a group of Tournament of Roses inspectors: the Maneuverability Test, Technical Inspection 1, and Technical Inspection 2. At the first Technical Inspection, the inspectors will review the overheight mechanism, if ready, for structural stability and progress. After this inspection, if the inspectors have any safety concerns, those concerns will be brought up for the builder to address. During Technical Inspection 2, the overheight mechanism is fully inspected for complete functionality as well as a timed retraction to ensure the float will safely pass under the bridge.

Existing Overheight Mechanisms

One type of overheight mechanism that has been implemented in previous floats consists of a telescoping cylinder and a telescoping structure built around the cylinder. Shown in Figure 5 is the hydraulic telescoping cylinder; fluid is pumped into the cylinder, increasing the pressure and forcing it to extend. When it is time for the cylinder to retract, the pump is shut off, therefore, decreasing the pressure in the cylinder, and then gravity causes the sections of the cylinder to fall down. For the Tournament of Roses, the entire float must be covered in flowers; this requirement creates complications because the telescoping structure must be designed to leave enough clearance between sections so the flowers are not damaged when the device is retracted. Additionally, there are stress concentrations at each step, so most of the telescoping devices seen on floats are fairly vertical. However, the booms of telescoping cranes are usually set at an angle. In order to prevent the crane from tipping over, a counterweight is placed towards the base of the boom.

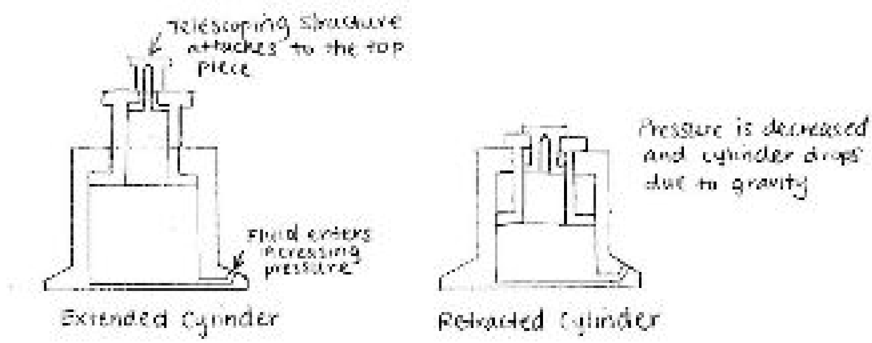

Figure 5. Telescoping Hydraulic Cylinder

Cal Poly Rose Float has used telescoping devices on many of their floats. The largest telescoping mechanism they have in their inventory has a volume of approximately 22.5 cubic feet and can support a weight of at least 2000 pounds. Figure 6 below shows the telescoping rocket in Cal Poly's Galactic Expedition float in 2011.[5]

Figure 6. Cal Poly's Galactic Expedition Float with Telescoping Spaceship

Winches and cables are also commonly used in floats because of their versatility and small volume. Since the cables are fairly small in diameter, they can be routed almost anywhere within the float with the use of some pulleys. Cal Poly Rose Float tends to stay away from winch and cable designs because of the immense amount of research that is required to choose the appropriate winch and cables as well as their costs. Additionally, the most recent time Cal Poly Rose Float used a winch and cables, both cables, holding the head of a giraffe up, snapped; the float is shown below in Figure 7.[6]

Figure 7. Cal Poly's Jungle Cuts Float with Winch and Cable

To further explain how this mechanism works and why it failed, the layout of the winch and cables in this structure are shown in Figure 8. For this float, the giraffe's head was held up when the winch had the cable reeled in to the minimum length. When the giraffe's head needed to be lowered, the winch released some of the cable allowing the neck to hinge at the break point. The cables snapped when the flowers were added to this portion of the float because the weight exceeded the capacity of the two cables. Overall, more research is required to create an overheight mechanism with winches and cables.

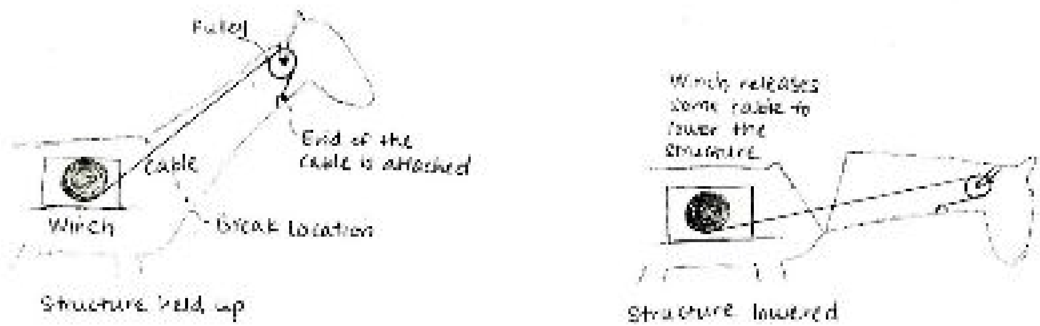

Figure 8. Schematic of the Winch and Cable

Hinges are another type of mechanism that are frequently implemented in the float world. A hinge is a movable joint or mechanism that can connect linkages; Rose Float uses hinges in a way that can be easily automated and can carry heavy loads of at least 1400 pounds. Usually, hinges are custom built by students to function for a single float year. They can be made from any type of material, but the strongest and cheapest material that Rose Float has access to is steel, which is used almost exclusively on the float because it can be easily welded. Cal Poly's hinges consist of a hydraulic cylinder and two rectangular pipes connected at a pivot point. By incorporating a hydraulic cylinder to actuate the motion required for the overheight, a very simple mechanism can be custom-built to fit any type of object. The movement of a hinge mechanism can be seen in Figure 9. In the lowered position, the hydraulic cylinder is completely retracted whereas in the up position, the cylinder is extended, which forces one of the pipes to rotate about the pivot point.

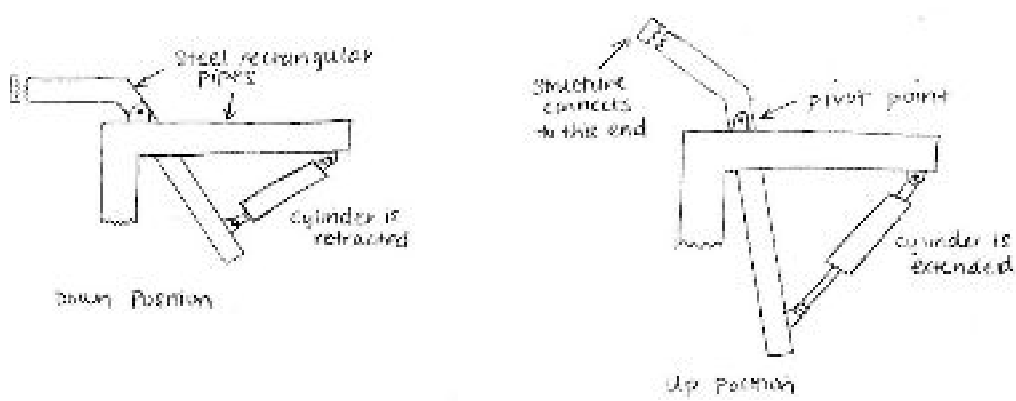

Figure 9. Hinge Mechanism Schematic

Hinges are the most widely used type of mechanism in the parade, as float builders can manufacture these mechanisms relatively quickly and for a very low cost with the materials they already have. In Figure 10, the winged creature in the back of the float is supported by a hydraulic cylinder in correlation with a hinge mechanism.[7] The hinge mechanism is invisible to the audience, as the main goal was to conceal the mechanism when it is in the up position.

Figure 10. Cal Poly's Guardians of Harmony Float with Hinge Mechanism

Another concept that is used in overheight mechanisms are linkages. Linkages are very versatile in how they can be set up and utilized, and they can be used to convert one type of motion to another type of motion. For example, a linkage can have rotation as an input and have a lifting motion as an output. The governing equations vary greatly between the different kinds of linkage applications. The simplest form of linkages are levers, which can utilize Archimedes' law of the levers. Links can also be formed in an X pattern which will cause a lifting motion when a force is applied to a supporting link. Another common formation of linkages are 4-bar linkages, which have four bodies and four joints. Examples of 4-bar linkages are crank sliders, parallel linkages, and planar quadrilateral linkages.[8] Since the set up for a 4-bar linkage can be versatile, they are used in multiple applications such as landing gears, suspensions, and pliers. Figure 11 below shows the 4-bar linkage mechanism involved in landing gears.[9] The curved lines represent the motion carried out by those four locations as the mechanism is tucked into the plane. The parts labeled 1 and 2 are linkages in this contraption. Linkages are not commonly used in floats because they require a great deal of precision, especially when the structure needs to be symmetric. In addition, the analysis is difficult since there are many more small parts that make up the entire mechanism.

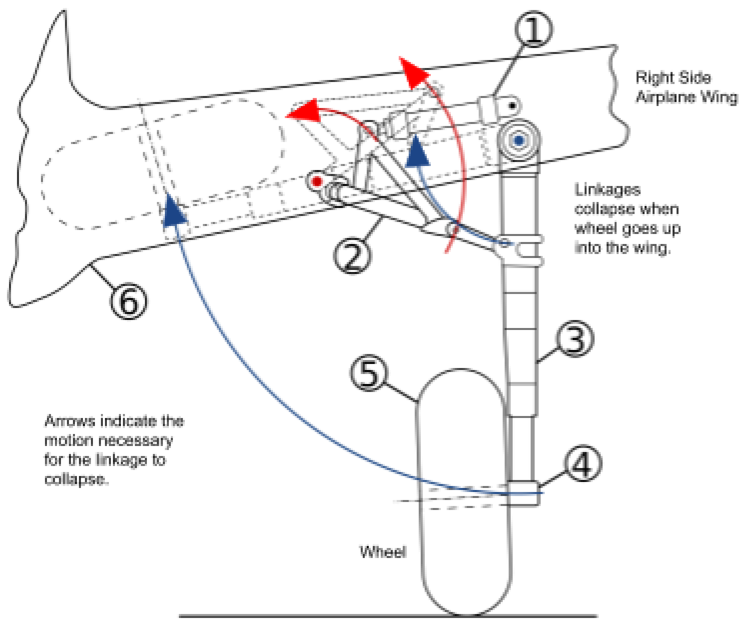

Figure 11. Aircraft Landing Gear Linkage

A method used less frequently in the float building industry is a rotational mechanism to lower overheight structures. Due to the high torque requirements, the rotation is often done using hydraulic motors over electric motors. Although this method is not used very often, rotating the character or element allows for a small seam line around that is fairly easy to hide from the viewer. Recently, rotational overheight mechanisms have been used effectively in the parade by Cal Poly on Sweet Shenanigans as well as on The Monkey King by Paradiso. These two floats with their overheight structures up are shown in Figure 12.[10]

Figure 12. Cal Poly's Sweet Shenanigans Float with Rotating Mechanisms

A rotational mechanism is more compact compared to other methods of lifting while still providing the necessary lifting and height capacities. Additionally, hydraulic motors are industry standard for heavy duty manufacturing and agriculture equipment with torques ranging from several hundred to several thousand foot pounds.[11] In order to achieve the appropriate torque and speed, we would need to combine a motor with a reduction gearbox. Ideally, we would use a hydraulic motor we have and only need to buy or make a gearbox. The industrial gearboxes available are relatively expensive, and if we needed to buy a hydraulic motor, the cost would increase even more. Although rotational mechanisms have many benefits with customization and compactness, the gears required would be sensitive to shaft bending which may pose a potential issue given our immense loading. There are a few industrial gearboxes that can handle upward of 1,300 ft-lbs, but the cost may be a limiting factor.[12] The primary reason this type of method is not used as frequently in the parade is due to the higher level of engineering and increased cost needed to produce an effective rotational mechanism. An image of the rotating mechanism used in the ice cream cones is shown below in Figure 13.

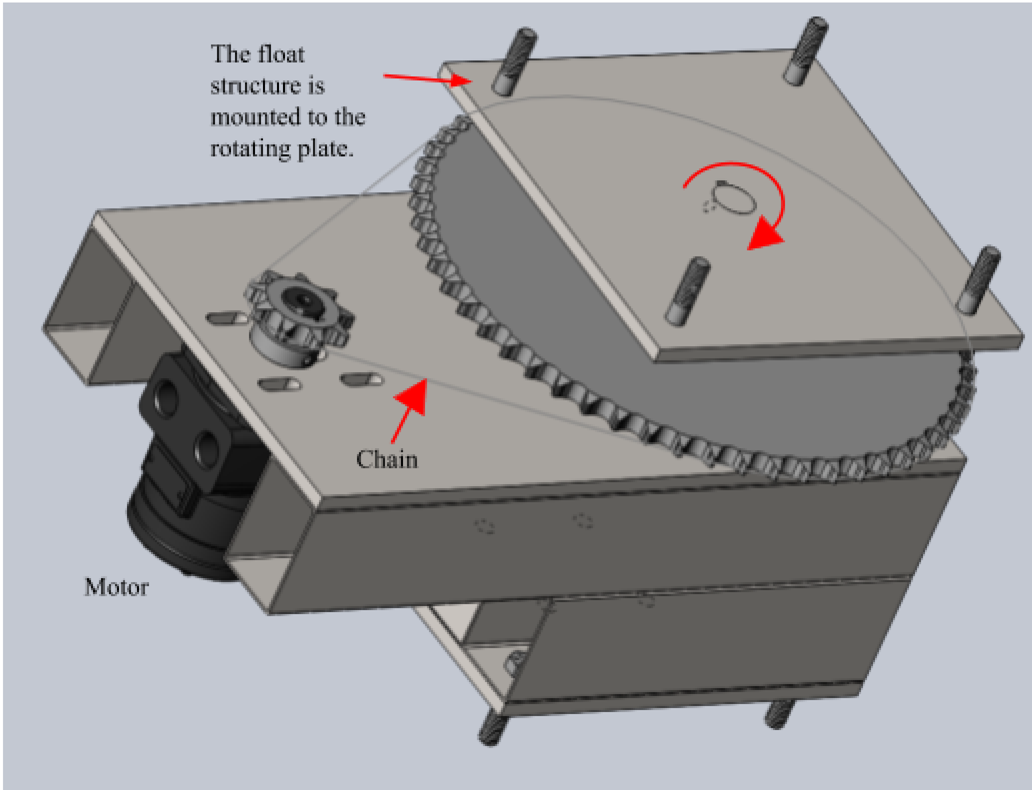

Figure 13. Rotational Mechanism Model

Competitor Overheight Mechanisms

As part of our research into overheight mechanisms, we toured the facilities of two prominent float builders: Phoenix Decorating Company and Paradiso Parade Floats. Although both of these float builders are professional float building companies, they had vastly different approaches to the same problem.

The first company we toured was Phoenix Decorating Company in Pasadena, California. Phoenix produces on average nineteen parade floats for every New Year's parade. Consequently, the way they design and manufacture floats is similar to a production line set-up whereas Cal Poly produces one off products. As we were touring their facility, we noticed how they approached the overheight mechanism is from a similar mindset. Most of their overheight mechanisms were similar mechanisms scaled according to each application. Moreover, many of the overheight mechanisms were simply forklift masts welded to the chassis of a float (Figure 14). Their reasoning for attaching forklift masts was to eliminate the need to engineer a new product. Since the engineering on the mast is already completed by an outside party, they simply have to implement the mast at the rated capacity.

Figure 14. Overheight Mechanism Using Forklift Mast

Another common mechanism used on many of their floats is a method we use often on our floats, hinge mechanisms. As described above, these hinge mechanisms are simple to manufacture. For these mechanisms, there is a main structure to hold the cylinder and pivot points and a main truss that is connected to the lever arm. As shown in Figure 15, Phoenix produces many of the same hinge mechanisms scaled for various load capacities. Unlike Cal Poly, they use the same hinge design across many floats. This hinge design consisted a constrained 90° travel and cylinder orientation. The only disadvantage to this design is the cylinder uses the smaller bore area to pull the overheight up instead of using the larger bore area to push it upward as shown previously in Figure 9. However, an advantage to their design of the hinge mechanism is decreased volume compared with our mechanisms. They achieved a smaller volume by positioning the cylinder horizontally mirrored from Figure 9 and leaving any support structure needed for mounting to be welded to the main support beam.

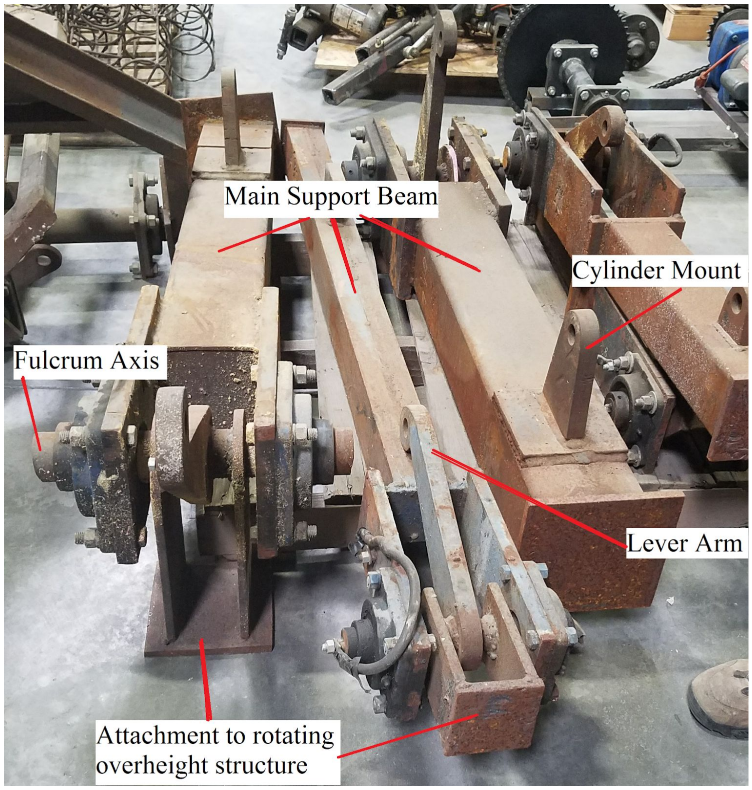

Figure 15. Phoenix Decorating Company's Hinge Mechanism

The second company we toured was Paradiso Parade Floats in Irwindale, California. For comparison, Paradiso produces on average 4 floats per year. Therefore, the way Paradiso approaches float design and construction is more alike to how Cal Poly Rose Float approaches parade floats. Each mechanism, including overheight is done specific to a particular character on a single float. Although the designs are similar to each other, they only use the same overheight mechanism on occasion. Also, almost all the overheight mechanisms we saw on their floats were variations of hinge mechanisms. Opposite to Phoenix they do not have the resources or extra time to allow for exploring different, more complex options. The hinge mechanism was a reliable option for their purpose, and possibly could be used on multiple floats if it was an adjustable mechanism.

From both of these float builders, even though very different, both confirmed the similarity between most overheight mechanisms. Thus, for this project our goal is to synthesize all this research into a robust and versatile overheight mechanism. Since most of the designs are similar, it is feasible to create a generic solution similar to Phoenix's hinge mechanism. However, we want to preserve the flexibility Paradiso has when designing float overheight structures by making the mechanism compact and adjustable.

Industry Equivalents

Researching in the industrial field for inspiration allowed us to find very similar concepts being implemented using various techniques. Some of these ideas came from the design of tree trimming/bucket utility trucks. These trucks employ different methods in order to lift a person several stories in the air to conduct work. Two large scale manufacturers of these types of trucks are Altec and Terex. Although these two companies are building virtually similar products, they are each design slightly different and that uniqueness helped us consider different methods for designing our mechanism. In Figure 16, it can be seen that the Altec LR 760 uses a heavy duty chain coupled by two hydraulic cylinders.[13] This technique allows a large degree of rotation available during operation. In Figure 17, the Terex XT Pro is equipped with a single cylinder attached to a half circle lever arm.[14] This method still allows for the operator to have plenty of freedom to rotate while also having the added benefit of a rigid connection from the cylinder to the lever arm. Both these ideas are very practical and are currently utilized throughout the world on similar style work trucks.

Figure 16. Altec LR760

Figure 17. Terex XT Pro

Objective

All floats in the Pasadena Rose Parade must pass under a 16' 6" tall bridge regardless of the float's height. Cal Poly designs and builds floats well over 17' tall which require an overheight mechanism that lowers tall structures to avoid collisions with the bridge. Rather than redesigning the overheight mechanism every year, this device will be reused on at least five different float designs, which would make it a justifiable investment for the program. To achieve this goal, the device must be able to be mounted at various positions and angles on the float. This means that it could be bolted to different match plates so that a level of reusability may be maintained. Allowing at least two configurations for the mechanism to attach contributes to the overall goal of implementing this device on multiple floats. Additionally, since the float design changes drastically every year, the device must have multiple modes for the float component to attach to the device. The height range of the mechanism is also an important factor in deciding whether or not it will work on numerous floats, so it will be designed to have a range of 9 feet to ensure it can be used on most float designs. In order for the operators of the float to be sure that the mechanism is in good working order, there needs to be an indicator that relays to the operator whether the overheight is either in the up or down position. The indicator is implemented as a safety feature and warning sign to the operators if something is amiss. Based on the available materials and the skill level of the students, the mechanism will be made of weldable steel for easy attachment and repairs in case an emergency occurs that requires welding. By using steel, the Rose Float Team can easily acquire a portable stick welder that can be used to repair any critical failures that may occur.

QFD

To begin the design process, we completed a Quality Function Deployment diagram, shown in Appendix A, in order to better understand how our product can satisfy the customer's needs. After speaking with our sponsor Josh D'Acquisto, advisor of Cal Poly Rose Float, we were able to pinpoint the customer's requirements and implemented them into the chart. There were three main categories these requirements fell into: functional performance, interface with the float, and manufacturing factors. Included in functional performance were the device's ability to lift heavy structures and its adjustable height range, while interface with the float included modular placement and easy implementation into various float designs. From the weighted percentages, these four traits were the most important to our customer. We then developed engineering specifications to address each of the customer requirements. When relating the customer requirements to our engineering specifications, we learned how each specification affects the customer. The engineering specifications that are most strongly related to the top four customer requirements were: lift capacity, height range, life, number of placements, number of floats can be used on, volume, and safety drop time. When designing our overheight mechanism, these engineering requirements will be the highest priority in our specifications table. Next in the QFD, we compared the competitors' products to our engineering and customer requirements. This data was accumulated from our visit to two different float builders, Phoenix and Paradiso. We toured their facilities and discussed how they built overheight mechanisms for their various floats. Overwhelmingly, they used some type of hinge mechanism to lift and Phoenix also used the lift off of a forklift. Thus, the hinge method was a top solution across our competitors and our floats.

Design Requirements

Table 1. Engineering Specifications Table

Parameter Description	Requirement Target	Tolerance	Risk	Compliance
Lift Capacity	1250 lbs	±250 lbs	H	A, T
Distance of Load from Pivot Point	8’	±2’	H	A, T
System Pressure	1400 psi	MAX	H	A, T
Gravity Drop Time	60 s	MAX	M	T
Height Difference	9 ft	±5 ft	H	A, T
Cost	$1300	MAX	L	A
Floats can be used on	5	MIN	M	S, I
Life	400 cycles	±50 cycles	M	A
Volume	15 ft^3	MAX	M	A, I
Indicator Light	Up/Down Position	1 Light Per Mech MIN	L	T ,I
Operators Required	1 Operator	MAX	H	T, I
Locking System	Must be able to lock in up position	1 MIN	H	T, I, A
Unpressurized position	Down Position	N/A	H	A, T, I
Safety Factor	3	-1/+2	M	A, T

In the design requirements presented above, we have established a set of guidelines and goals in order to be successful with this project. From reviewing overheight mechanisms from the past 8 years of Cal Poly floats, we determined the minimum lift capacity to be 1000 lbs for our mechanism. Although some of these floats did not have quite as high of a lifting capability, the trend of our overheight structures has been increasingly more weight. Therefore, we determined our lift capacity needed to suffice for a larger overheight structure in order to be usable for future floats. The target lifting capacity and tolerance was based on one of our larger overheight structures which was on the Cal Poly float “A New Leaf”. Another criteria we based on this research of past floats was the lever arm that this lifting load is placed. All overheight mechanisms have an axis of rotation by which it is lowered and raised with the actuator on one side and the moving structure on the other as shown in Figure 18. The load lever arm is the distance from this axis of rotation to the center of mass of the lifted structure where the lifting load is applied (Figure 18). We determined this lever arm to be on average 8 ft with a maximum of 10 ft by looking at the average of past overheights and last year's float, "A New Leaf".

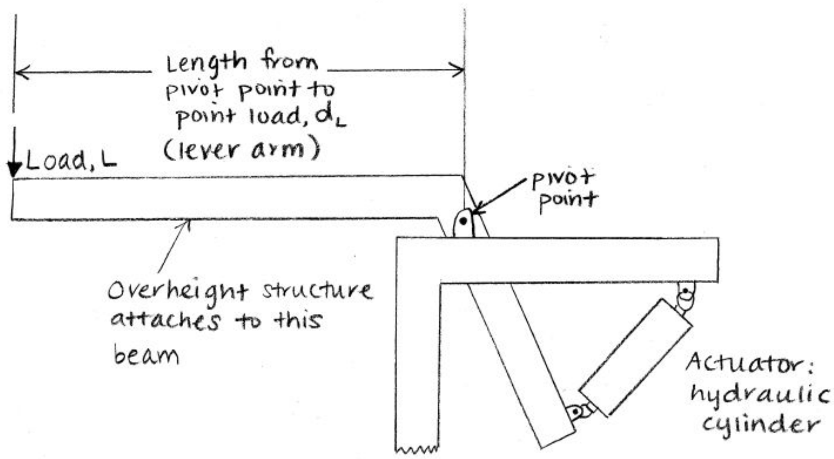

Figure 18. Lever Arm Diagram

Along with researching past overheight mechanisms, we also are implementing this mechanism into our animation system. The animation system in our float is run with a hydraulic pump driven with a Chevy 350 engine. Thus, our animation system has a set system pressure of 1400 psi our mechanism must operate using. Additionally, we referenced the Tournament of Roses' Float Manual for other requirements. This Manual is a set of rules every float builder must abide by if they are participating in the Rose Parade. According to the 2014 Float Construction Manual, "In no cases will a retraction time of more than 60 seconds be allowed." The retraction time they describe is, in the event our system losses power, the time our overheight mechanism must be able to retract due to gravity pulling it down. This excerpt helps us to define our desired maximum drop time due to gravity of 60 seconds. Furthermore, the Float Manual dictates a maximum height of 16'-6" of the float after the overheight has been retracted. Usually, our float is a maximum of 16 ft tall in order to ensure we are under this height requirement after adding flowers.

Our project sponsor, from a long tenure of advising different floats, set a minimum height range in order to justify an overheight mechanism. Due to the increased design and fabrication time an overheight mechanism demands, a float must pass a threshold height of 20 ft to pursue building overheight. From this criteria, and looking at maximum height ranges from floats in the past several years, we determined a height range of 9 ft with a 5 ft tolerance to allow for the prescribed minimum and maximum height ranges of floats. The project sponsor also imposed requirements for the mechanism to justify the capital investment. Based on an overheight mechanism reused in the past 10 years, the sponsor set a goal that this mechanism to be used on at least 5 different floats. Not only was this criteria based on a previous overheight used on 5 floats, but for the mechanism to be usable in future years it must be versatile to different designs. Our budget has been dictated by our project sponsor to be $1300 after considering annual budgeting of the float. This budget limit is only applicable for any new materials purchased for the project which are not in the Rose Float Lab. Thus, this budget is only limiting material and labor cost since the actuator will be from the Rose Float Lab.

Since our sponsor defined our total life to be 5 years for this mechanism we calculated the total life from this 5 year limit. In a year, we only run the overheight mechanism in the end stages of construction because we need the float almost completely assembled. Consequently, the overheight mechanism is operated during the work days between November and December. This leaves a maximum of 20 days of potential usage for the mechanism with an average usage of 4 cycles per day. In total, the mechanism may be used 400 up-down cycles in its lifetime. Another extrapolated design requirement was the volumetric constraint of 15 cubic feet for the mechanism when it is retracted. From our experience and knowledge of previous space allocations for overheight mechanisms, we know the less volume a mechanism takes the more likely it will be reused on different floats. Although some mechanisms were relatively small in past years or only a telescoping cylinder, we found the maximum volumetric space of the mechanism itself to be about 12 cubic feet from the float "A New Leaf". Ideally, we will minimize this requirement in our design.

The device will be used around crowds of people at the parade and our facilities, so we must adhere to additional safety requirements. For safety of the float, Cal Poly Rose float has four operators within the float during the parade day, one of which is the operator of the animation system. These operators are in closed compartments within the float, often separated from each other. Animation operators have all animated mechanism controls in their designated compartment including overheight mechanism controls. During the parade, the animation operator must be able to control overheight alone. Also, they usually cannot see outside of their compartment to the exterior of the float. Consequently, they have no indication for the position of the overheight. To assist the operator in telling when and if the overheight structure is fully up and down, we are requiring an indication light for our mechanism. This indication light will illuminate when the overheight is in a fully up or down position. Another safety requirement is to accommodate emergency situations during the parade. In the event an emergency causes our float to lose power, the float is towed under the 16'-6" bridge at the end of the parade route. In order to ensure no damage to the bridge, our mechanism must retract to a down position with an unpressurized system. Lastly, from OSHA standards on hoisting and rigging standard number 1926.753(e)(2), all components must have a safety factor between 1 and 5.[15] For all of our components, we are performing analysis for a target safety factor of 3 with a bilateral tolerance of 2. Due to the proximity of our mechanism with people, we will aim for the more conservative safety factors.

Idea Generation

At the start of the quarter, we brainstormed a list of concepts that could possibly be used as the solution to our problem. Practicality was not a limiting factor at this point since we wanted to keep all avenues open and prevent the risk of rejecting a good idea. To begin, we listed concepts that had been done by Cal Poly Rose Float before, including hinges, telescoping cylinders, rotating devices, and winch and cables. Next, were the ideas influenced by existing industrial lifting machines; these included scissor lifts, boom lifts, and elevators. For the more exotic ideas, we wrote down anything that came to mind even if we did not know how they worked. Among these concepts were boat locks, flexible hinges, and mechanisms that split down the middle. We iterated through two brainstorming phases to ensure we had exhausted all our options. All of these generated concepts were researched further in preparation for the concept evaluation phase of our project.

<u>Hinges</u>

As shown by Figure 9, a typical hinge mechanism manufactured by Cal Poly Rose Float is composed of steel rectangular tubes and a hydraulic cylinder. When the cylinder is retracted, the overheight is in the down position. As the cylinder extends, the overheight rotates about a pivot point and raises to a greater height. Our required height range could be achieved by using a cylinder with a stroke long enough or by altering the length of steel tube between the pivot point and the cylinder attachment point. However, this is only one configuration of a hinge mechanism; there are many others that involve different actuators. In general, a hinge mechanism involves a lever arm rotating about a pivot point with the help of an actuator.

<u>Telescoping Mechanisms</u>

The most common actuator for a telescoping mechanism on Cal Poly's floats is a hydraulic telescoping cylinder, shown in Figure 19.[16] To extend the cylinder, hydraulic fluid is pumped inside, which creates an upwards force on each telescoping section. As the sections of the cylinder extend, the telescoping structure around the cylinder extend as well. When it is time to retract the cylinder, the hydraulic fluid is drained out, and the sections collapse due to gravity. The motion of the hydraulic cylinder is described in Figure 5. The height of these type of mechanisms depends on the collapsed and extended length of the cylinder.

Figure 19. Hydraforce Telescoping Hydraulic Cylinder

Rotating Mechanisms

As described previously in the "Existing Overheight Mechanisms" section, the rotating mechanism contains a hydraulic motor and a gearbox. The overheight structure is securely attached to the rotating plate in the mechanism, as shown in Figure 13. When the overheight needs to be lowered, the hydraulic motor is shut off, and the structure will smoothly lower due to gravity; the hydraulic fluid prevents the structure from rotating down too quickly. To raise the overheight, the hydraulic motor produces a torque and rotates the structure back up to its original position. Since the torque required is typically very large, it is key that the appropriate motor is selected for this mechanism to operate well.

Winch and Cable

For winch and cable mechanisms, the tension in the cable determines whether the overheight is in the up or down position. A typical winch and cable configuration is shown in Figure 8. When the cable is taut, the structure is held up with the help of a pulley. The down position is achieved when the winch releases some of the cable, allowing for the structure to hinge downward at the break point. As explained previously, the tension in the cables is so great that they are at risk of snapping; therefore, selecting cables capable of large loads is extremely crucial. The winch is also an important component in the functioning of a winch and cable mechanism. With the required loads, a hydraulic winch, like one shown in Figure 20[17], is capable of providing the larger cable loads needed.

Figure 20. Ramsey Hydraulic Winch

Scissor Lifts

Scissor lifts are widely used in industry to safely lift people to greater heights. These machines are composed of large links configured into diamonds, as shown below in Figure 21.[18] Most scissor lifts are powered electrically and have a maximum capacity of about 500 pounds. Since our lifting requirement is 1250 pounds, the scissor lift would not fulfill this condition. When the scissor lift is extended, the diamonds become taller. However, when it is compressed, the diamonds become very wide; therefore, it would be difficult to implement a scissor lift into a float because of its large width. Scissor lifts are generally used solely in the vertical direction, so it would be difficult to satisfy the five float requirement if motion is restricted to the y-axis.

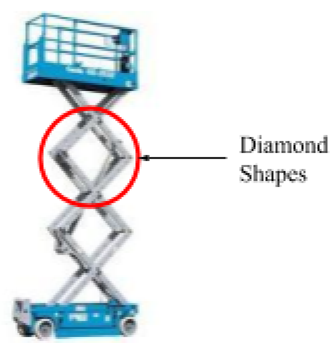

Figure 21. Scissor Lift

Boom Lifts

Boom lifts incorporate both telescoping square tubing and cylinders, shown in Figure 22[19], to lift people to heights greater than that of a scissor lift. The machines are often used to reach elevated objects when trimming trees and fixing power lines. However, the volume swept out by the motion of a boom lift is very large. This would make the mechanism difficult to conceal in a float. Additionally, the lifting capacity of boom lifts is usually 500 pounds, which does not meet our specified design requirement.

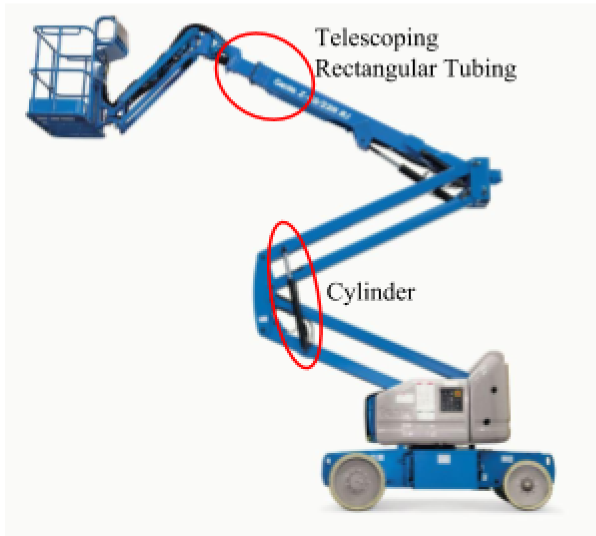

Figure 22. Boom Lift

Elevator Links

The compressed and extended configurations of the elevator links can be seen in Figure 23 below. When compressed, all the links are located next to one another. To extend the links, an actuator would need to provide a force that pushes the leftmost link, causing the links to slide and extend in a staircase motion. Our load requirement is 1250 pounds, so the links must be fairly thick to handle the loading with a safety factor of at least two. As a result, the links would take up a lot horizontal space and would weigh a significant amount.

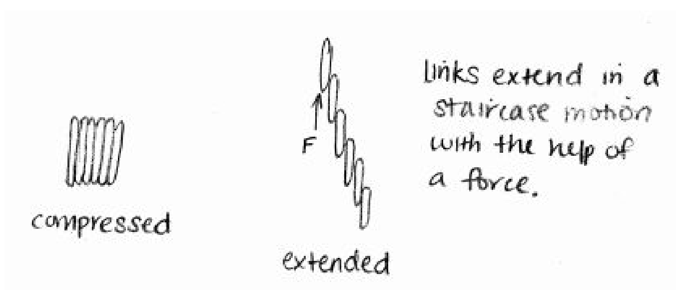

Figure 23. Elevator Linkages

Boat Locks

The boat lock shown in Figure 24 is a unique design found in Scotland. The purpose of the boat lock is to lift boats out of the water in a safe and controlled manner. A 30 horsepower electric motor is used for this boat lock to lift a group of boats in just four minutes.[20] The motor only rotates the structure 180 degrees at a time to bring the boats from the water to where the bridge connects. Again, the torque of the motor is what allows this device to work, so selecting the correct motor would be an important aspect of the design. In addition, the floats created by Cal Poly Rose Float contain many structures with little space between, so rotating a heavy overheight with a lever arm of 10 feet would likely result in collision.

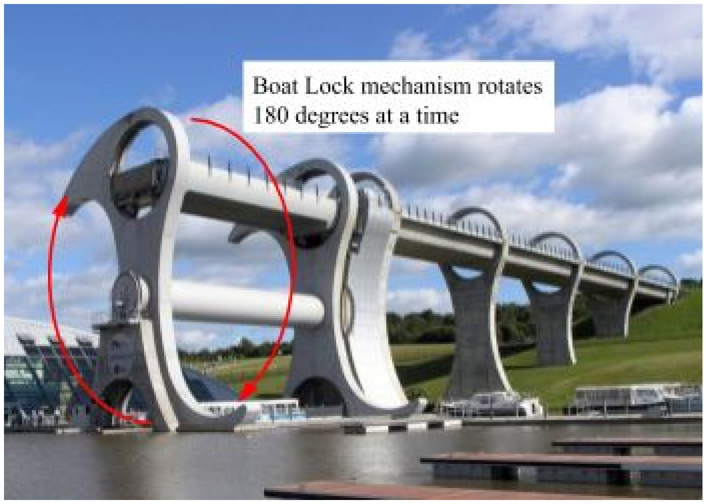

Figure 24. Boat Lock Mechanism

Flexible Hinges

One type of flexible hinge is composed of pieces of aluminum interlocked with polyurethane hinges, as shown in Figure 25.[21] There are small gaps between the aluminum pieces to provide clearance when the whole hinge is rolled up. Although these devices create an ideal flexible surface for a float element needing to hinge up and down, there is no effective way to simulate a flexible hinge for a large scale application. Moreover, these hinges do not provide enough structure or strength for the lifting capacity we are required to reach.

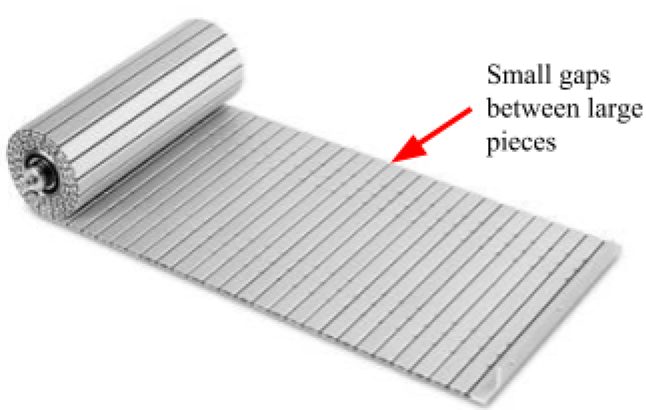

Figure 25. Flexible Hinge

Mechanisms that Split Down the Middle

A mechanism that splits down the middle would separate the overheight into two pieces. In the up position, the two halves of the overheight structure would be together; however, as the overheight is lowered, the two halves would separate. The two sides could be rotated downwards in opposite directions in a motion similar to the rotating mechanism, as shown in Figure 26. Although the load would be split into half on each lever arm, the device would need a large volume of clearance in order to lower the overheight, which would likely be an issue since the float is covered with various structures within close proximity to one another.

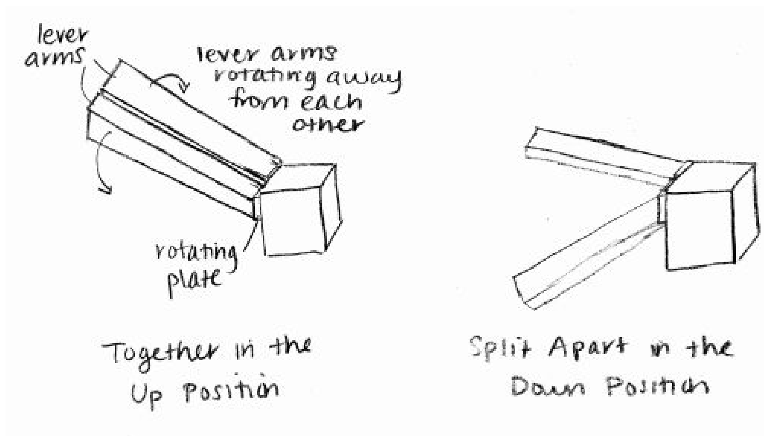

Figure 26. Splitting Mechanism with Rotation

Concept Evaluation

As seen in the idea generation section above, we did not limit ourselves when we were brainstorming. Allowing the brainstorming phase to be open lead to very different and creative ideas being presented, as we felt this could lead to a new answer we never expected. After accumulating a vast range of brainstormed ideas, we needed to narrow down our design solutions to converge towards a single idea. The first tool we used to reduce our concept selection was a feasibility study in which we judged ideas based on practically in terms of the design requirements of our project.

Feasibility Study

Throughout the first four weeks of the project, we were brainstorming concepts as well as developing our QFD and design requirements. After defining the problem and potential solutions, we moved straight into our feasibility study during week four. For our feasibility study, we assessed the overall practicality of every concept by discussing each and ranking them into the following categories: feasible, possibly feasible, or not feasible. As we discussed each concept, we covered mechanical feasibility in addition to referencing the most important customer requirements in the QFD. The top customer requirements considered included ability to lift heavy structures, height range capability, usability on at least five different floats, and compactness. We strived to keep these customer requirements at the forefront of our selection process because our design must not only be mechanically effective but also worthwhile to the Rose Float program.

As shown in Table 2, we decided to eliminate six concepts based on the infeasibility of meeting both our customer and design requirements. Due to our high lifting capacity, we eliminated flexible hinges as they are more effective in light loading applications. Moreover, we need to hide this mechanism with a character or element, so we excluded folding and coiling as they may cause interference issues with the floral decoration on the element. Although mechanically sound, the boat lock, sliding planes, and split down the middle mechanisms were removed as possible options due to potential issues with creating a compact enough envelop for our customer.

In addition to the six concepts removed, we decided four of our concepts were possibly feasible. The concepts under the might be feasible category shown below were determined to be conditionally feasible if we could create these motions in a compact volume. Usually, mechanisms such as elevators, scissor lifts, and boom lifts have a large volume in order to operate properly. Considering our customer asked to have this mechanism be used on five different float designs, we need to make the mechanism as compact as possible to achieve this requirement. Although many of these types of mechanisms are large, we could attempt to design a mechanism similar to one of these in a small package.

From performing this feasibility study, we reduced our range of concepts nearly in half with 14 concepts as feasible. These 14 concepts we subcategorized into four main top level ideas: hinge, rotation, telescoping, and linkages. Each of these top level motions had various actuators that could be used to achieve each type of motion. However, by categorizing each general motion, we were able to distribute the remaining concepts for further research among the group, so there would be an expert on each type of motion. By having each person specialize in a limited number of ideas, the amount of work necessary to research all the topics was greatly decreased. This technique tremendously aided our next steps in the decision making process as we were able to freely discuss questions or concerns with the resident experts in that specific field.

Table 2. Feasibility Study Summary

Feasibility Study		
Feasible	Might be Feasible	Not Feasible
Hinge w/ hydraulic cylinder	Detatchable motion	Sliding Planes
Hinge w/ hydraulic motor	Elevator	Folding
Hinge w/ winch	Scissor lift	Coiling
Telescoping w/ hydraulic cylinder	Boom Lift	Flexible Hinge
Telescoping w/ winch		Boat Lock
Rotating w/ hydraulic motor		Split down middle
Rotating w/ hydraulic cylinder		
Rotating w/ electric motor		
Rotating w/ winch		
Retract w/ winch		
Retract w/ hydraulic cylinder		
Linkage - double rocker		
Linkage - crank rocker		
Linkage - crank slider		

Go/No-Go

After conducting our feasibility study, we were able to move into an analysis based decision making tool. The Go/No-Go decision making process allowed our team to quantitatively compare each mechanism based on lifting capacity required. Throughout the next week, we started by dividing up the remaining concepts into four main topics along with their respective actuation methods (Table 3).

Table 3. Main 4 conceptual ideas with potential actuators

Type	Actuator
Hinge Mech	Hydraulic Cylinder
	Hydraulic Motor
	Winch & Pulley
Rotation Mech	Hydraulic Cylinder
	Hydraulic Motor
	Winch & Pulley
Linkage Mech	Double Rocker
	Crank Rocker
	Crank Slider
Telescoping Mech	Hydraulic Cylinder
	Winch & Pulley

Preliminary analysis was carried out on these categories as our Go/No-Go evaluation. The preliminary analysis consisted of using statics of a loaded system to determine the force required by the actuator. The set up for our loaded system was to use the maximum loading from our design requirements of 1500 lbs at a length of 10 ft from the mechanism. An example of such calculations are presented in Appendix B. Then, we researched actuators to see if there were actuators available capable of providing the force calculated given our system pressure design requirement. At this point, we ruled out using a hydraulic motor on a mechanism because it could not produce the necessary torque, given our system pressure, without using a gearbox. If we were to buy both a new motor and a gearbox, we would go over our allotted budget. Furthermore, it was highly unlikely we could get both a motor and gearbox donated. We also eliminated linkage systems from further pursuit since it would not be feasible due to the volume needed to achieve the target height range requirement. Additionally, a telescoping mechanisms was withdrawn as it did not meet the versatility requirement of being usable on five different float concepts.

We narrowed down the number of ideas to three after the Go/No-Go test, and they were hinges with a hydraulic cylinder, hinges with a winch and cable, and a rotational mechanism with a hydraulic cylinder. Also, we added an additional concept after discussing how bucket truck lifts in industry use hinges with a gear chain and hydraulic cylinders. These four ideas then continued to our final stages of concept evaluation.

Discussion on Top Concepts

The following four concepts we determined met a majority of our specifications. All four met lift capacity through our preliminary analysis when had a general setup of applying a max load of 1500 lbs of force with a max lever arm of 10 ft. Three of the four concepts utilize hydraulic cylinders, which we know will meet the life requirement as long as there is proper treatment to avoid any seals from failing. The unpressurized position requirement is also achieved in these designs; the designs with hydraulic cylinders will be in the down position when the cylinders are unpressurized, and the winch design is in the down position as long as the cable is not in tension.

<u>Hinge with one cylinder</u>

A hinge mechanism using a hydraulic cylinder is a reliable and proven method for an overheight mechanisms, as described previously. The lifting capacity is dependent on the leverage distance as well as the cylinder bore. The volume of this mechanism consists of its mounting, the cylinder, and the cylinder's swing as it extends and retracts. The overall volume satisfies our requirement of being less than 15 cubic feet. The height difference is achieved when the cylinder extends and retracts causing the connecting arm to rotate upward and downward. An example can be seen below in Figure 27.

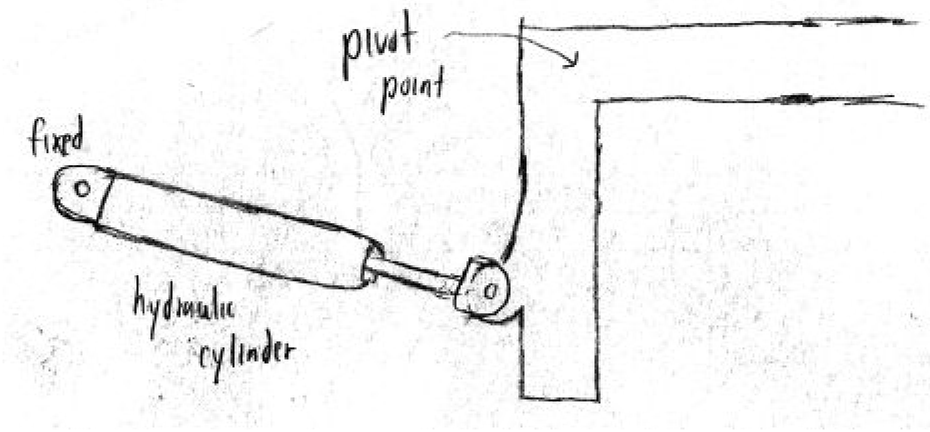

Figure 27. Hinge with Cylinder Concept

Hinge with a chain and cylinders

As seen below in Figure 28, this mechanism uses hydraulic cylinders to pull on roller chain which causes the sprocket to rotate. As the sprocket rotates, the horizontal beam will raise and lower, because the sprocket is welded to a shaft which is welded to the beam. The rotation is what causes the height difference to be achieved, and this design can be modified to fit the rotation angle needed for a desired height gain. Also, the volume of this mechanism is mostly contained in the area from its base to the sprocket, so the device is compact overall.

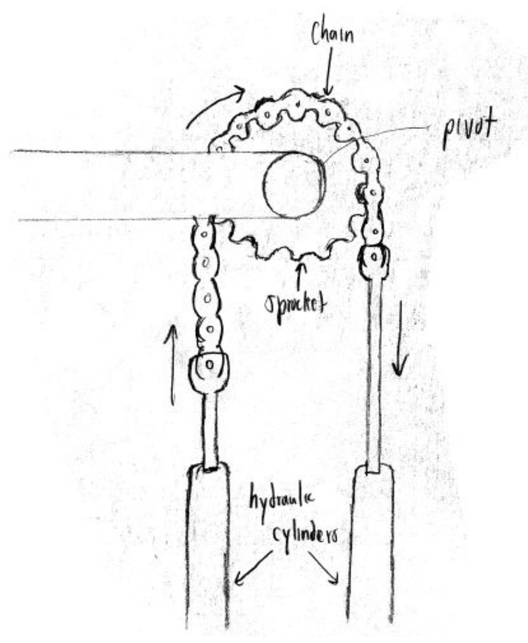

Figure 28. Hinge with Chain and Two Cylinders Concept

Hinge with a winch and cable

The size of this mechanism primarily depends on the pulley required to pull the overheight up, because the winch can be mounted on the chassis of the float. Since the winch does not have to be directly mounted near the pivot point, the space required can be minimized and meet the volume requirement well. When the cable is in tension, the overheight is raised, and when the tension is released, the overheight is lowered. The total height difference is accomplished by the angle of rotation and length of beam used. Thus the target height gain defined in our design requirements is easily achieved. For an example sketch please refer to Figure 29.

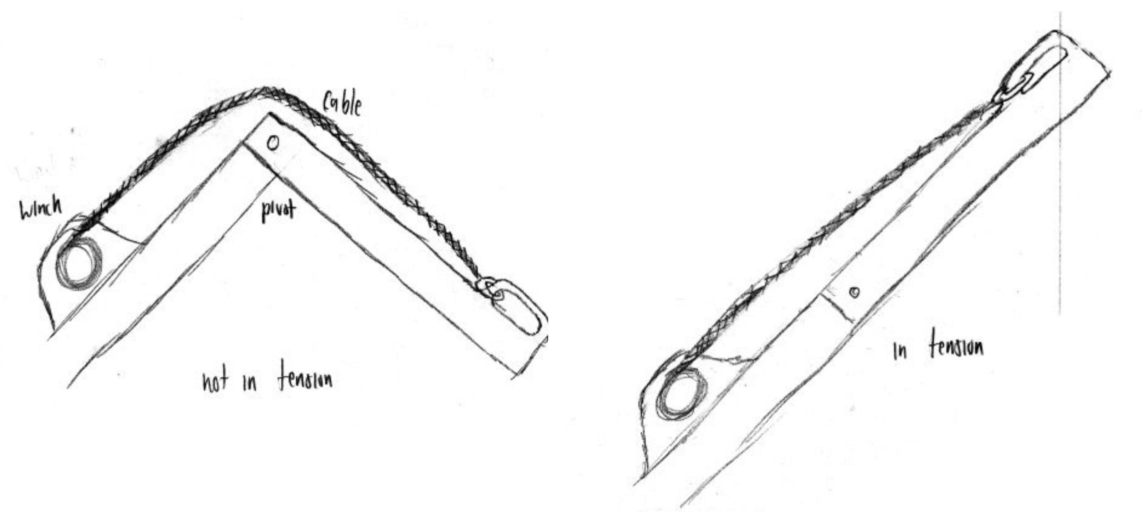

Figure 29. Hinge with a Winch and Cable Concept

Rotational mechanism with a cylinder

This concept was the only rotational mechanism deemed possible by having a hydraulic cylinder as the actuator (Figure 30). For overheight structures with a midrange height difference and lifting capacity, this rotational mechanism is a viable option. The cylinder is set up at an angle, so it can rotate a circular plate. This plate is where the overheight would attach, and as it rotates the structure rotates to the proper height needed.

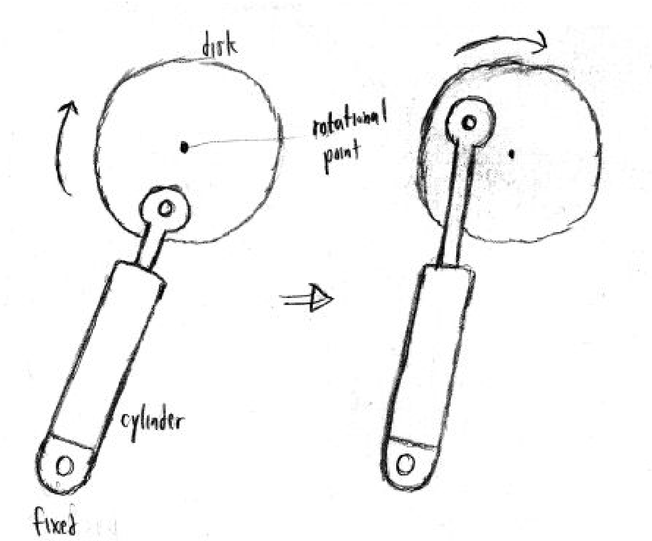

Figure 30. Rotation with Cylinder Concept

Pugh Matrices

We placed these concepts in a Pugh Matrix to compare them based on the most important design requirements (Appendix C). Referring to our QFD, we decided to include the highest weighted engineering requirements such as lifting capacity, height range, cost, volume, life, and reusability across float designs. Also, we include traits such as simplicity, reliability, and repairability. Although these were not critical in our design requirements, they are important traits in designing an overheight mechanisms that is reusable across many Rose Float teams. From previous weightings performed in the QFD and design requirements, we ranked all the criteria relative to one another. Then, we set the most common overheight mechanism, a hinge with one cylinder, as the datum for our matrix because it has been proven to meet all the criteria.

Starting with the first iteration, it can be seen that we included the rotation with a hydraulic motor to see if it was a good concept despite the cost (in case we could get an actuator donated). After our first run through the Pugh Matrix, this concept was eliminated as it did not adequately meet our other requirements. As expected, the hinge using cylinders and chain as well as the hinge using a winch were highly rated. We performed a second iteration with the remaining four concepts, but with hinge with winch and cable as a datum. From this iteration, we learned the hinge with winch and cable was a good option, but not as promising as the hinge using cylinders and chain. To verify this, we performed one last iteration with the datum as the hinge with cylinders and chain. Both iterations proved the hinge with cylinders and chain would be the highest rated option. After iterations two and three, the hinge using a winch and cable was a higher rated option than the hinge with one cylinder. However, we decided the winch and cable may not be feasible since we would have to rely on a donation. Thus, we decided to continue with analyzing our top two concepts: hinge with one cylinder and hinge with cylinder and chain.

Analysis for Decision Matrix

Once we confirmed our top two concepts to be the hinge mechanism with a single cylinder and a chain and sprocket set up with a cylinder, we performed preliminary analysis on these two systems. For both systems, we defined the load applied and distance the load was applied from the pivoting axis. Also, we set the height range and system pressure for each system as a given design parameter. We performed concurrent engineering analysis on both systems given these design parameters to solve for volume and cost of each system in order to compare them accurately in the decision matrix. To solve for volume and cost, we calculated approximate sizing for all major components and determined the safety factors to verify the size of each component was large enough. We developed governing equations on engineering paper, and then programmed these into a matrix calculator file for each system. Since all of the variables are dependent on one another, we needed to utilize a matrix calculator for its simultaneous equations solving capabilities.

For the hinge mechanism with one cylinder, we determined equations for sizing the beams, cylinder, and lever arm as shown in Appendix D. First, we wrote static equations for the upright and down position of the mechanism to relate the force applied by the cylinder to the angular position of the rotating overheight structure. From these equations, we also had several dependent variables such as the lever arm length as well as the angle between the lever arm and the cylinder. We also developed equations for the mounting distance for the cylinder as well as the height range achieved. Using these geometric variables, we were able to develop an accurate estimation of the volume of the mechanism. By using beam theory and distortion energy, we developed equations to solve for the stress and the safety factors for the necessary structural beams. All of these equations were then put into a single matrix calculator file in which we solved for all of the dependent variables given that the cylinder and beam sizes are provided (Appendix E). A parametric table was used to put in various beam sizes to determine the best beams to use for the safety factor range defined in our design requirements. Tables 4 and 5 are the two separate parametric tables used to solve for the different beams needed.

Table 4. Square Beam 1 sizes versus safety factors

1..10	a_1 [in]	b_1 [in]	I_1 [in4]	n_1 [-]
Run 1	4	4	8.22	3.309
Run 2	5	5	16.9	5.436
Run 3	6	6	30.3	8.115
Run 4	7	7	49.4	11.33
Run 5	4	6	9.34	3.776
Run 6	6	4	17.5	4.701
Run 7	5	3	11.3	3.625
Run 8	3	5	5.05	2.717
Run 9	5	4	14.1	4.531
Run 10	4	5	10	4.027

Table 5. Square Beam 2 sizes versus safety factors

1..10	a_2 [in]	b_2 [in]	I_2 [in4]	n_2 [-]
Run 1	4	4	8.22	1.233
Run 2	5	5	16.9	2.028
Run 3	6	6	30.3	3.03
Run 4	7	7	49.4	4.234
Run 5	4	6	9.34	1.401
Run 6	6	4	22.1	2.21
Run 7	5	3	11.3	1.356
Run 8	3	5	5.05	1.01
Run 9	5	4	14.1	1.692
Run 10	4	2	1.83	0.2745

Our results of the a matrix calculator analysis for our hinge mechanism with one cylinder are shown in Table 6 below. The dimensions that contributed to our total volume and cost estimates for the decision matrix are included. For our cost estimates, we found the price for our specified square beams.[22] Additionally, we added a 10% buffer in the cost for any extraneous parts required such as fasteners and small pieces of metal which is why there is an asterisk on this parameter. The safety factors included are to verify all components in our system meet the design requirement for safety factor.

Table 6. Hinge with cylinder analysis results

Hinge w/ Cylinder	
Parameter	Value
Lever arm (d)	8.5 in
Mounting distance (l)	36 in
Beam 1 dimension	4"x4"x1/4"
Beam 2 dimension	6"x6"x1/4"
Beam 1 safety factor	3.3
Beam 2 safety factor	3
Cost Estimate*	$822
Volume	1.2 ft^3

We performed a separate analysis on the design with a hinge mechanism using chain, sprocket, and a cylinder (Appendix F). For this analysis, we began in a similar manner with writing equations relating the force required by the cylinder to the sprocket size and load applied. Moreover, we solved for a bore size on the cylinder in relation to the force required and system pressure. Using distortion energy theory on the size of the steel pipe, we were able to solve for the stress and consequently the safety factor of the main rotational shaft. We also performed weldment and chain analysis on this system since both are critical to supplying the torque required to lift the overheight structure. From these analyses we determined the safety factor in both areas to verify they exceeded our minimum safety factor requirement. We solved for the stress in the structural beams of our design by utilizing beam theory to relate the force applied by the load and cylinder. Then we solved for the safety factor as we did before using distortion energy to determine the box steel size required. Finally, we developed a general equation to solve for this system's total volume. The results of this analysis are shown in Table 7 using the same cost estimation source and 10% buffer described above.[22] The associated a matrix calculator file of the analysis performed is given in Appendix G.

Table 7. Hinge with cylinder and chain analysis results

Hinge w/ Cylinder + Chain	
Parameter	Value
Chain Pitch	1.75 in
Sprocket Pitch Diameter	25.09 in
Sprocket Teeth Number	45
Beam 1 dimension	4"x4"x1/4"
Beam 2 dimension	6"x6"x1/4"
Chain safety factor	2.5
Beam 1 safety factor	3
Beam 2 safety factor	3
Cost Estimate*	$985
Volume	12 ft^3

Our analysis validated the feasibility of both systems to meet various design requirements such as lifting capacity, height range, safety factor, system pressure, cost, and volume. Additionally, the analysis done provided accurate quantitative values for the cost and total volume of each system. Since we analyzed each system given a set lifting load, distance from pivot to load, system pressure, and height range, the dependent design requirements being solved for were cost and volume. These two parameters for both systems was a main quantitative comparison in our decision matrix.

Decision Matrix

While developing the decision matrix, we referred to our design requirements as well as our QFD to determine the criteria we would compare for each design. We utilized the relative weight rating of each design requirement on the QFD to determine our most important design requirements. These design requirements are listed in Table 8 below as well as our definitions of them. These definitions and targets guided our evaluation of our top two concepts in the decision matrix.

Table 8. Decision matrix criteria

Criteria	Definition
Lift Capacity	Must be able to lift a range of weight of 1250 ±250 lbs. This capacity should be calculated based on the max stress conditions for the mechanism.
Height Range	The height range corresponds to how high the mechanism can lift the specified weight provided in lift capactiy 9 ± 5 ft.
Life	The neccesary operation time will correspond directly to a life of 400 ± 50 cycles
Number of Floats can be used on	The number of floats that this device must be usable on will be at leasst 5 minimum, this value will be verified by comparing to past floats and seeing if they'd fit in those constraints.
Fabrication Time	No more than 3 weeks of fabrication time (3 days/week at 4 hours/day per person)
Volume	Max 15 ft^3
Cost	$1300 Max

Once we decided on which criteria we would judge each concept based on, we weighted the relative importance of each criteria. From previous weightings on the QFD and Pugh matrices of these attributes, we ranked the criteria relative to one another. From this ranking of each criteria we then calculated the relative percent weighting. As a team, we evaluated and rated each concept on a scale of zero to five with zero as not meeting the criteria. Our complete decision matrix is included in Table 9 below.

Table 9. Decision matrix

Decision Factors		Hinge with Chain + 2 Cylinders	Hinge with 1 Cylinder
Criteria	**Wt.**	**1**	**2**
Lift Capacity	0.16	5	5
Height Range	0.13	5	5
Life	0.14	4	4
Number of Floats can be used on	0.18	5	5
Fabrication Time	0.11	3.5	4
Volume	0.16	3	5
Cost	0.13	4.5	5
Weighted Scores		4.36	**4.80**

We set both concepts as a five on lifting capacity and height range since we analyzed both systems using the same values for both of these parameters. Life, number of floats can be used on, and fabrication time were approximated using similarity tests. We estimated the life based on how easily each design would be to maintain over the five year period specified by our sponsor. We determined both designs were not perfectly maintainable, but components subject to wear such as the shaft rotating in a steel pipe sleeve would be fairly accessible for servicing. By surveying the past decade of floats, we determined how many floats each design could be used on. From this similarity analysis, we determined both designs could feasibly be used on many different float designs which is why we gave both designs a five. For the fabrication time rating, we approximated the fabrication time based on previous similar overheight mechanisms that have manufactured in Rose Float. Based on fabrication times for similar mechanisms, we estimated the chain and cylinder design to have a longer fabrication time due to the complexity of attaching the chain to the cylinder and added time for chain breaking. In comparison, we have fabricated a hinge mechanism numerous times in Rose Float and they have a fairly simple fabrication process. Added time and attention must be taken in both designs to ensure welding the mechanism does not distort critical dimensions. For these reasons, we scored the hinge with one cylinder slightly higher than the chain and sprocket design. We calculated ratings for volume and cost based on the two values derived from our previous analysis. We rated the hinge with cylinder as five for volume, because it had a significantly smaller volume than the chain and cylinder design. We rated the chain and cylinder design at a three since it still was under our specified requirement, but not as minimal as the hinge with one cylinder. The cost was comparable between both designs with the hinge with one cylinder lower than the chain and cylinder design. Consequently, we scored the hinge with one cylinder higher than the hinge with chain and cylinder. Fortunately the cylinders required for both designs are accessible in the Rose Float Lab, so the cost difference was mainly between the amount of steel required and chain components.

Combining all of the scores, the total weighted scores showed the hinge with one cylinder had a slightly higher score than the chain and cylinder design. However, we ultimately chose to continue with the chain and cylinder design. Since a design like the hinge with a cylinder and chain is more time intensive for the analysis and fabrication than the hinge mechanism, Rose Float does not often build overheight mechanisms such as these. We concluded there was a benefit to pursuing a more complex design with the additional time for analysis, fabrication, and testing provided in a senior project setting. Unlike a regular year in Rose Float, we can explore these more complex mechanism options. Therefore, we decided not to continue with the higher scored design because we wished to investigate a design not often feasible to pursue in a normal float building year.

Initial Selected Design

The design that we originally chose to go with most closely resembles the Altec LR 760 mechanism as seen in the background section of this report. Its basic outline starts with two cylinders that are hooked up in series so when one cylinder is going up, the other would be going down. Each cylinder is connected to a heavy duty chain that wraps around a large sprocket. This sprocket is then attached to a pivot point where the load lever arm is also attached. As either hydraulic cylinder pulls the chain, the sprocket will rotate which, through power transmission (splines or keyways), will cause the structure to pivot up or down.

The conceptual solid model presented below represents the concept we deemed most beneficial for the Rose Float program. In this model, the dimensions specified from analysis are provided as well as an overall design layout shown in Figure 31. Our mechanism requires in depth analysis that needs to be validated before we can move forward to the full design. Also the mechanism requires very precise alignment so that the cylinders can pull and push in their strongest positions. This idea has proven to be a challenge to illustrate, even in the 3D modeling software, as aligning components is not always the simplest task in CAD. As a result of this challenge, we were unable to fully wrap the roller chain around the sprocket, which results in the odd appearance of the chain interfering with the sprocket in Figure 32. The device is currently attached to the base with a thick-wall 4 inch by 4 inch square tube located between the two cylinders. This tube acts as the main structural beam that will support the mechanism and structure above it. This design will need to be modified in the future, as we believe that this 4 inch square pipe will not be enough to support the large loads.

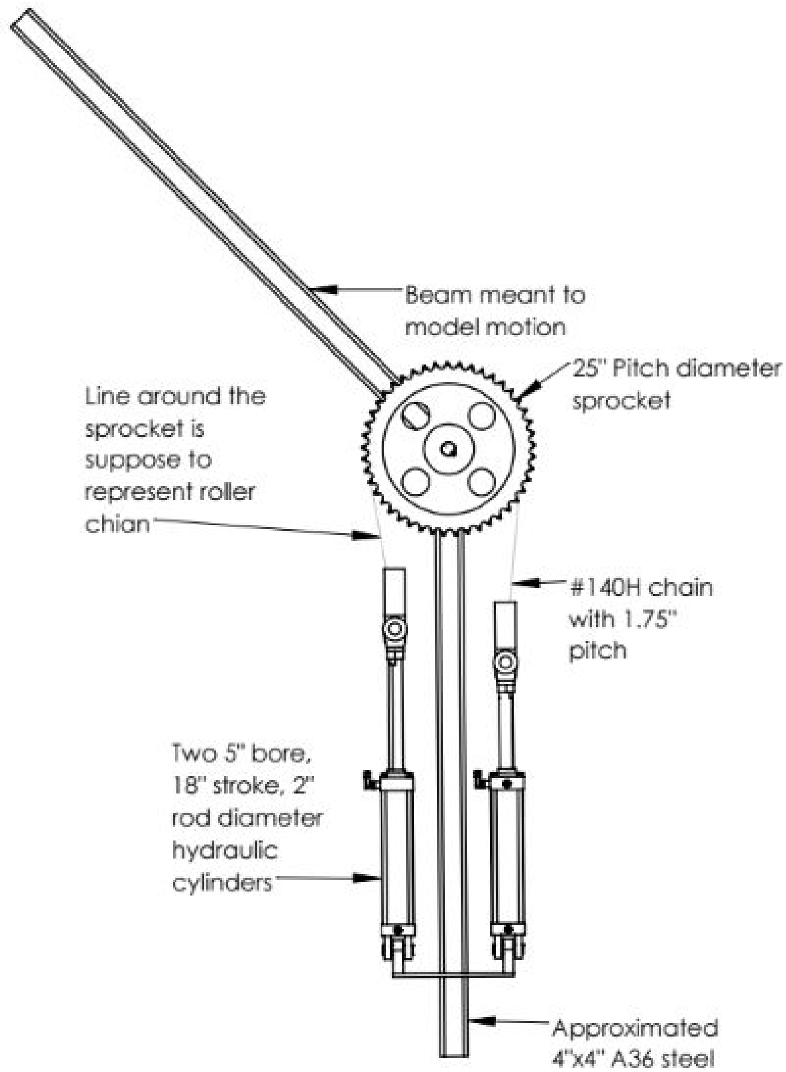

Figure 31. Top Level Concept Drawing

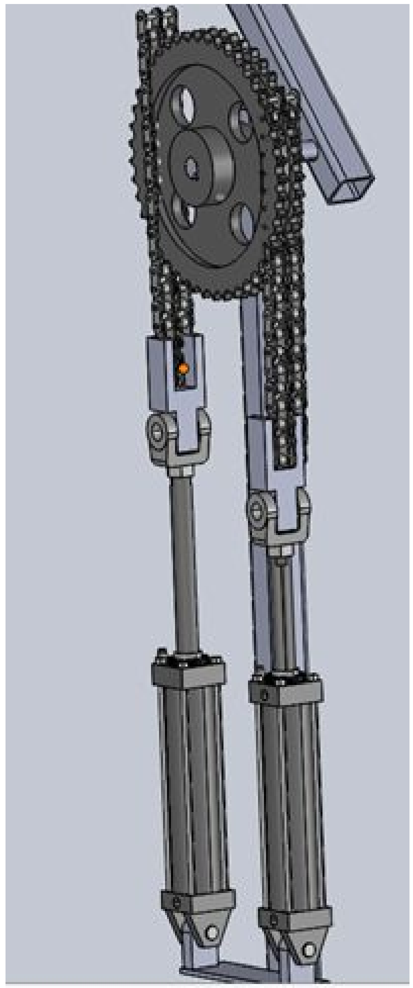

Figure 32. Solid Model of Initial Chosen Concept

We felt that this particular concept has great potential for the Rose Float program in terms of how it satisfies their needs. The main benefits to this design include multiple mechanism orientations, multi-year usability, and feasibility for manufacturing. Although it was not the highest ranked concept on our decision matrix, the team collectively agreed that this would be a more unique way to approach the problem that the Rose Float program might benefit from in the long run. If anything, this mechanism will allow the team to start exploring new and more creative ways of building overheight mechanisms as it steers away from traditional methods. With this type of mechanism, the Rose Float program will be able to focus more on the creative design side of the float and worry less about how things will fit together. This is only the beginning of the project itself; the next phase is the most crucial for the success of this project, as we will need to design this mechanism to satisfy each of our design requirements.

Final Selected Design

After presenting the Preliminary Design Review, the team members spoke to Doctor Zohns, a professor from the BRAE department at Cal Poly who is a registered Fluid Power Engineer, to gain some insight on the hydraulic system necessary for the initial selected design. Following hours of discussion, Doctor Zohns concluded it would not be possible to maintain tension in the chain once power was lost or if the engine were to fail. Therefore, the initial selected design would not meet the requirements necessary to be allowed for use in the Rose Parade. As a result, the team decided to return to the top ranked design in the design matrix, which was the hinge with one cylinder.

Description

The final selected design includes one cylinder for actuation that produces a hinging motion in the lever arm, shown in Figure 33. The cylinder connects to the frame at one end with a "C" mount and two sets of nuts and bolts. At the other end, a pin goes through the two "sandwich plates" with the clevis of the cylinder between, hence, the name "sandwich plate". Since the sandwich plates are welded to the main shaft, the extending cylinder will cause the sandwich plates to rotate, which causes the shaft to rotate. As a result, the lever arm rotates upwards and raises the large structure. This motion is displayed in Figure 33 with the red arrows. When the cylinder is retracted, the chain of events will cause the lever arm to lower the structure. According to the 3D model, the mechanism is estimated to weigh approximately 670 pounds. In order to lift and transport the overheight mechanism, four D-rings are welded onto the frame as attachment points for hooks. An isometric view of the final overheight mechanism can be seen below in Figure 34. The individual components that make up the final design will be described in further detail in the following section.

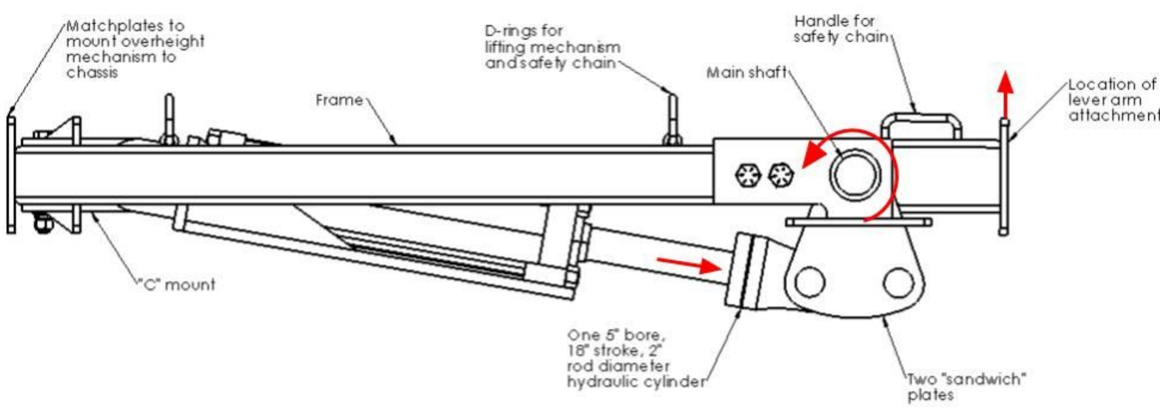

Figure 33. Top Level Concept Drawing

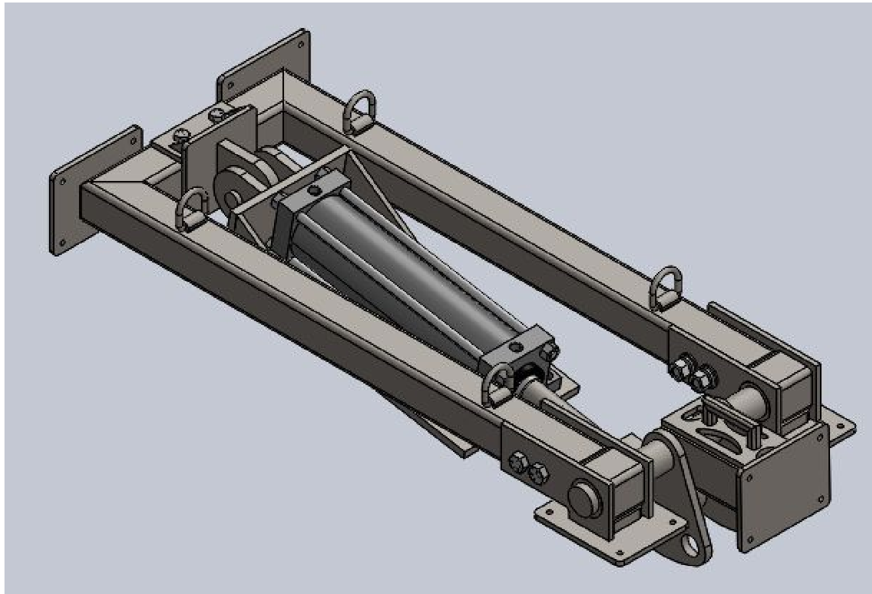

Figure 34. Solid Model of Final Chosen Concept

To make the mechanism more versatile for future float designs, we designed it to have three different mounting orientations. These include 0 degrees, 45 degrees, and 90 degrees, as shown in Figures 35, 36, and 37.

Figure 35. 0 Degree Orientation

Figure 36. 45 Degree Orientation

Figure 37. 90 Degree Orientation

Subassemblies and their Components

The final overheight mechanism is composed of multiple subassemblies with numerous parts in each. A few of these components are purchased while some of them need to be manufactured or cut to the required dimensions. The following subsections describe the purpose of each subassembly and its components.

Shaft Assembly

The shaft subassembly, shown in Figure 38, is composed of the main shaft, two cylinder lever arms, two truss mount plates, two truss braces, four side bearing plates, two pipe sleeves, two 4X4 pipe sleeves, two 4X4 pipe spacers, three main match plates, and a handle.

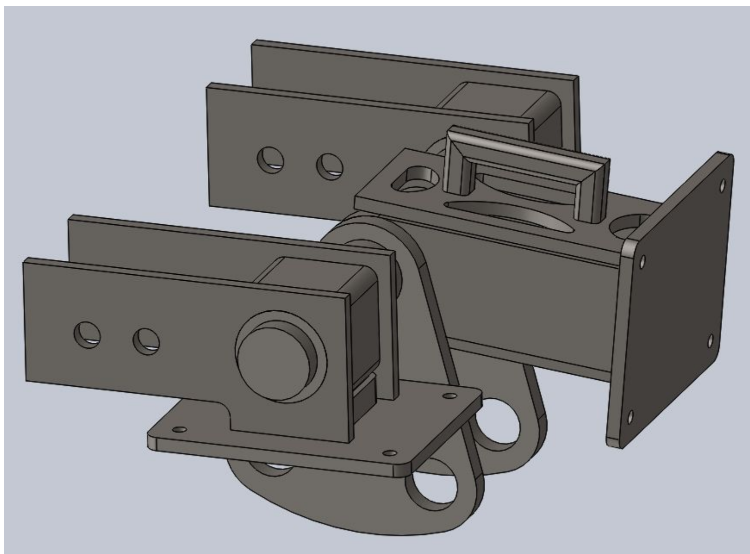

Figure 38. Solid Model of Shaft Subassembly

The cylinder lever arms, also referred to as "sandwich plates", rotate the shaft as the cylinder extends and retracts, as mentioned previously. As seen in Figure 39, these plates have an abnormal shape because of the height range requirement. There are a total of three holes in this plate: one for the main shaft and two for the pin. Because of our target height range, two of the two-inch diameter holes are necessary for the various mechanism orientations. When used in the 0 degree and 90 degree orientation, the clevis will be pinned with the 2" hole on the left, whereas for the 45 degree orientation, the 2" hole on the right will be used. The cylinder lever arm is created from ¾" thick plate and can be manufactured by using a water jet or a plasma cutter.

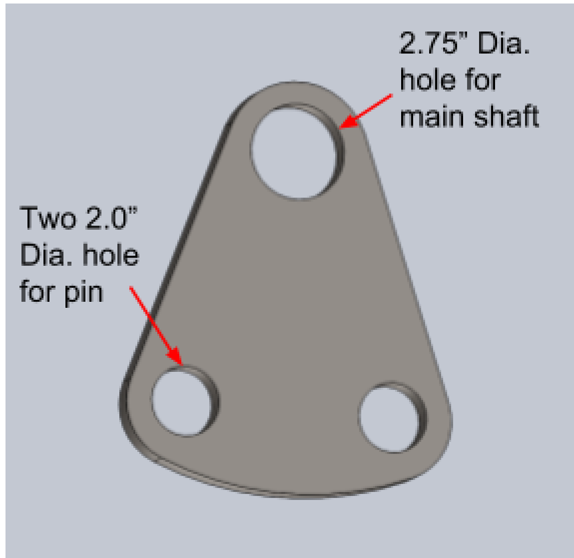

Figure 39. Solid Model of Cylinder Lever Arm

Next is the main shaft, shown in Figure 40, which is made of 2.75” diameter round stock. Due to the loads on this particular component, the shaft will be 1045 steel, which is stronger than the A36 steel used for most of the other parts. The shaft’s length was determined to be 25.75 inches to make the overheight mechanism as compact as possible while still allowing for proper maintenance.

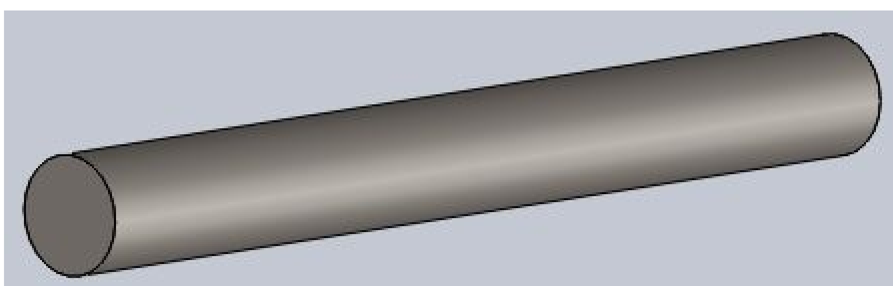

Figure 40. Solid Model of Main Shaft

The main shaft is supported at either end with the subassembly shown in Figure 41. The pipe sleeve, which has a 2.75” inner diameter holds the main shaft and act like a sleeve bearing. This circular pipe sleeve is inscribed into a 4” by 4” square pipe because the square tube offers a greater weld area necessary to adequately assemble the overheight mechanism. The two bearing side plates will slide over the circular pipe sleeve and be welded to the square pipe as well as the matchplate. To provide the necessary clearance for the bolts, the bearing side plates have a slight “L” shape as shown by Figure 41. A thin piece of 4” by 4” square pipe will be used as a spacer; this component will be welded to the square pipe sleeve and the matchplate. The matchplate provides a mounting area, so the overheight mechanism can be easily attached to the float.

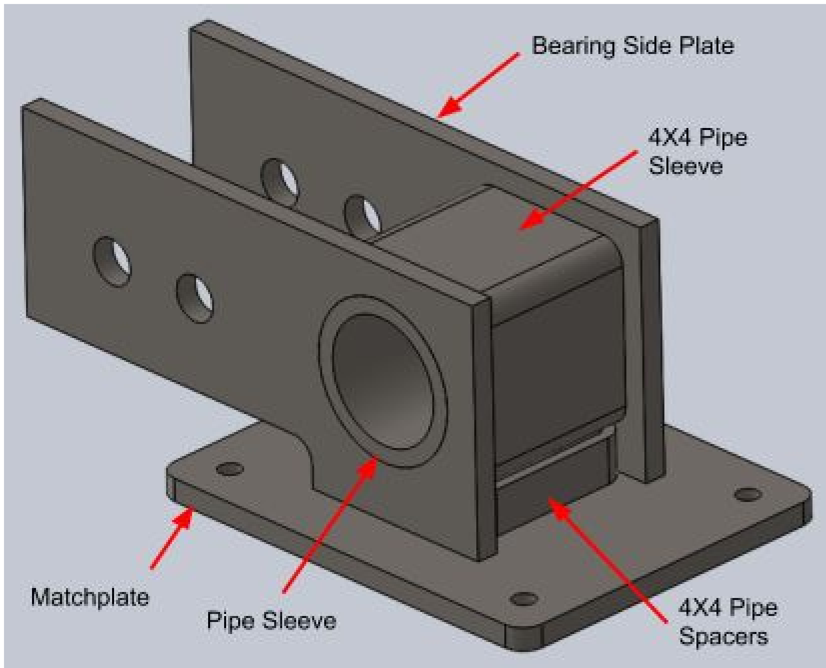

Figure 41. Solid Model of Bearing Support Assembly

In the center of the shaft subassembly, there is another structure where the lever arm will be attached to, shown in Figure 42. The two truss mount plates slide onto the main shaft with 2.75" diameter holes and are welded to the shaft. The two truss braces connect the truss mount plates and provide additional support to prevent the mount plates from twisting. The handle, which is welded to the top truss brace, is one of the safety features implemented into our design. Its role in safety will be explained later on in the report. The matchplate provides a connecting point between the lever arm where the large load is applied and the overheight mechanism.

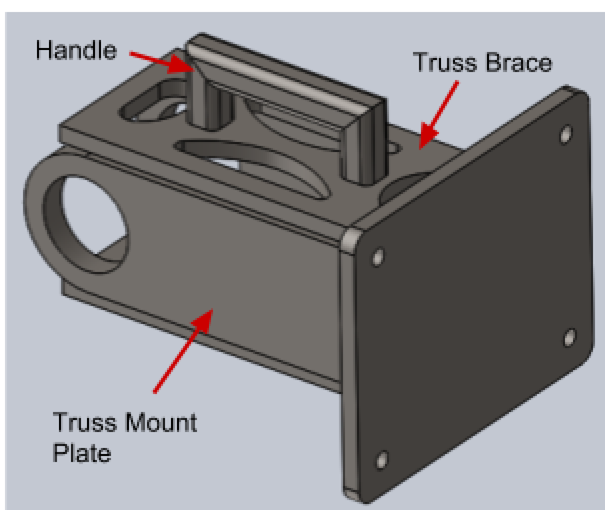

Figure 42. Solid Model of Lever Arm Attachment Assembly

Cylinder Mount Assembly

Our cylinder mount subassembly, seen below in Figure 43, is made up of eight components of A36 steel. Seven of the components are made of ½" thick steel plate. This includes the two 8" by 3.5" plate that each have ¾" diameter holes so the subassembly can be bolted onto the frame assembly. These two plates are welded onto a 8" by 8" steel plate that is also ½" inch thick. The remaining ½" thick components are the four steel gussets used as supports. Finally, the last component, which is made from 1" inch thick plate, is the fixed end pin mount that contains a 2" diameter hole, which is where we will pin the base of our hydraulic cylinder.

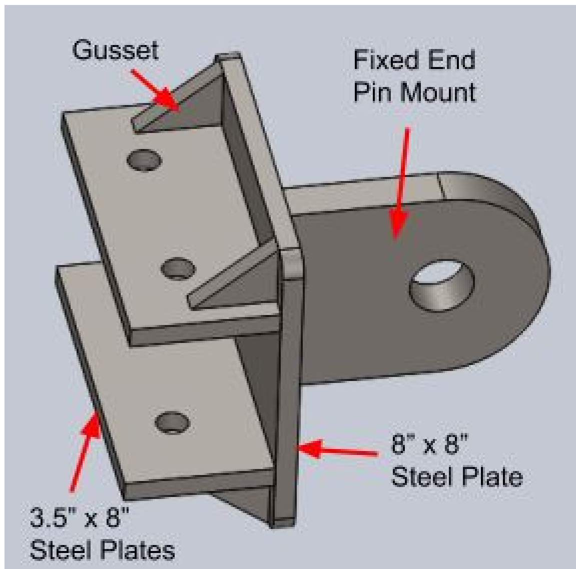

Figure 43. Solid Model of Cylinder Mount Subassembly

Frame Assembly

The frame subassembly consists of five components as seen below in Figure 44. Three components are 4" by 4" square tube steel with a wall thickness of ¼", and the other two components are generic Cal Poly Rose Float match plates on the back.

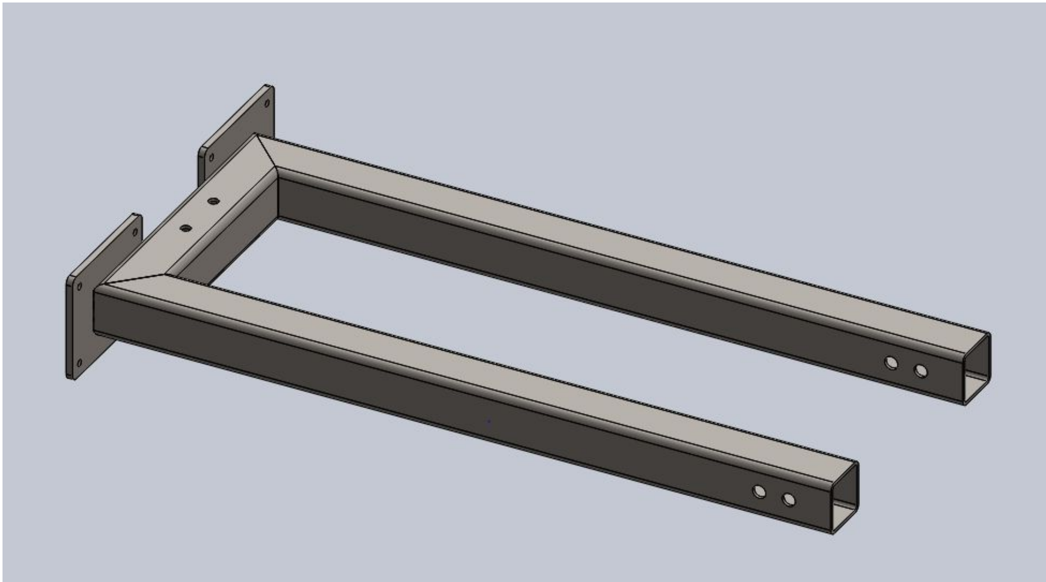

Figure 44. Solid Model of Frame Subassembly

The back square tube steel has both ends with 45° cuts and two ¾” diameter holes where the cylinder mount subassembly will be bolted onto. Two 10” by 8” by ½” match plates, each with four ½” diameter holes, will be welded onto the back piece to serve as connection points to the float chassis. These three components can be seen below in Figure 45.

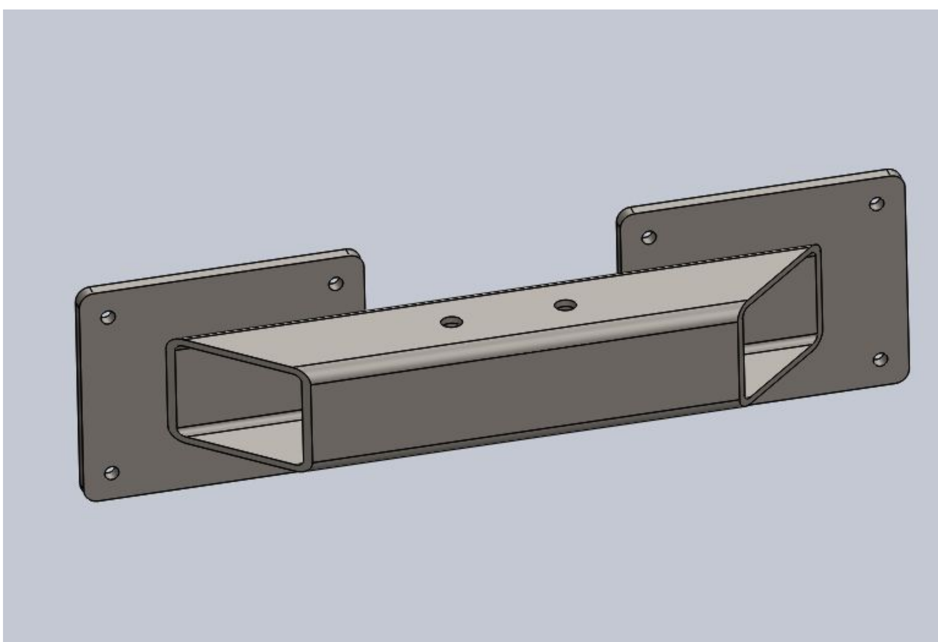

Figure 45. Solid Model of the Back of the Frame Subassembly

The remaining two side square tube steel, seen below in Figure 46, each have one end with a 45° cut where they will be welded to the 45° cut on the back piece. Each side square tube steel has two 1" diameter holes where the side bearing plates from the shaft subassembly can be bolted onto.

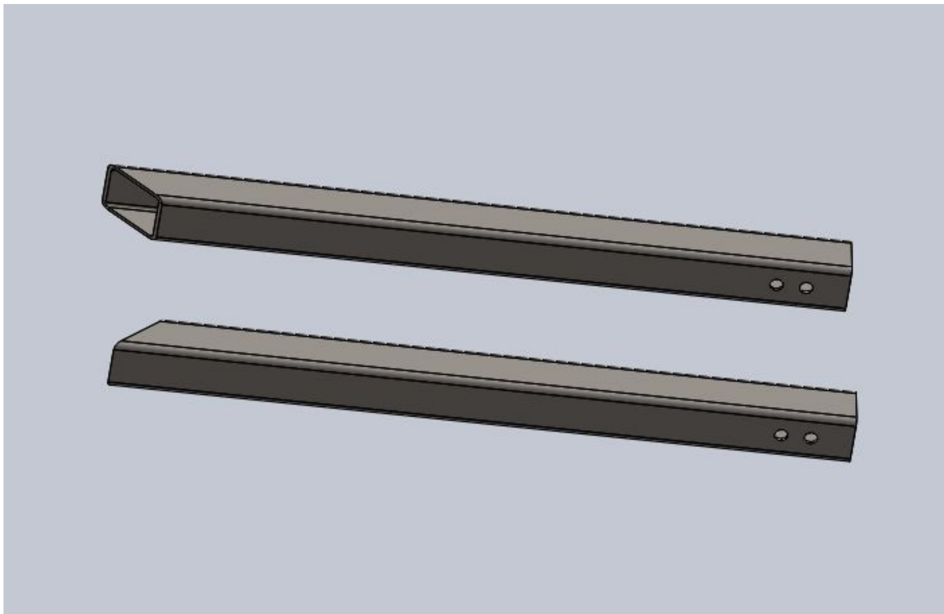

Figure 46. Solid Model of the sides of the Frame Subassembly

Main Assembly

Figure 47 below shows the full assembly of our mechanism. To assemble the overheight mechanism, the cylinder is pinned to the cylinder mount assembly through the 1.5" hole of the fixed end pin mount described previously. The clevis of the cylinder then is pinned through the "sandwich plates", found in the shaft subassembly, with a 2" diameter pin. Snap rings are placed on both ends of the pin to prevent the pin from accidentally falling out. To attach the shaft subassembly to the frame, the side bearing plates from the shaft subassembly can be bolted to the frame subassembly with 1" bolts that are 6.5" in length as well as flat washers, split washers, and hex nuts sized for a 1" bolt. The cylinder mount subassembly is bolted to the piece of square tube steel piece at the back of the frame subassembly using ¾" bolts with flat washers, split washers, and hex nuts for ¾" bolts. Two sets of clamp-on shaft collars plan to be installed onto the main shaft on the inside edges of the shaft bearing plate to prevent the shaft from sliding from side to side. Furthermore, four D-rings will be welded on top of the base frame and serve as dedicated lifting points. The two D-rings closest to the shaft subassembly are also part of a safety feature to prevent injuries when people are working around the mechanism. A heavy duty chain can be looped through the handle in the shaft subassembly and hooked onto those two D-rings; further explanation is located later on in the Assembly and Safety Enhancements section. All of the cut sheets for the purchased hardware can be found in Appendix H.

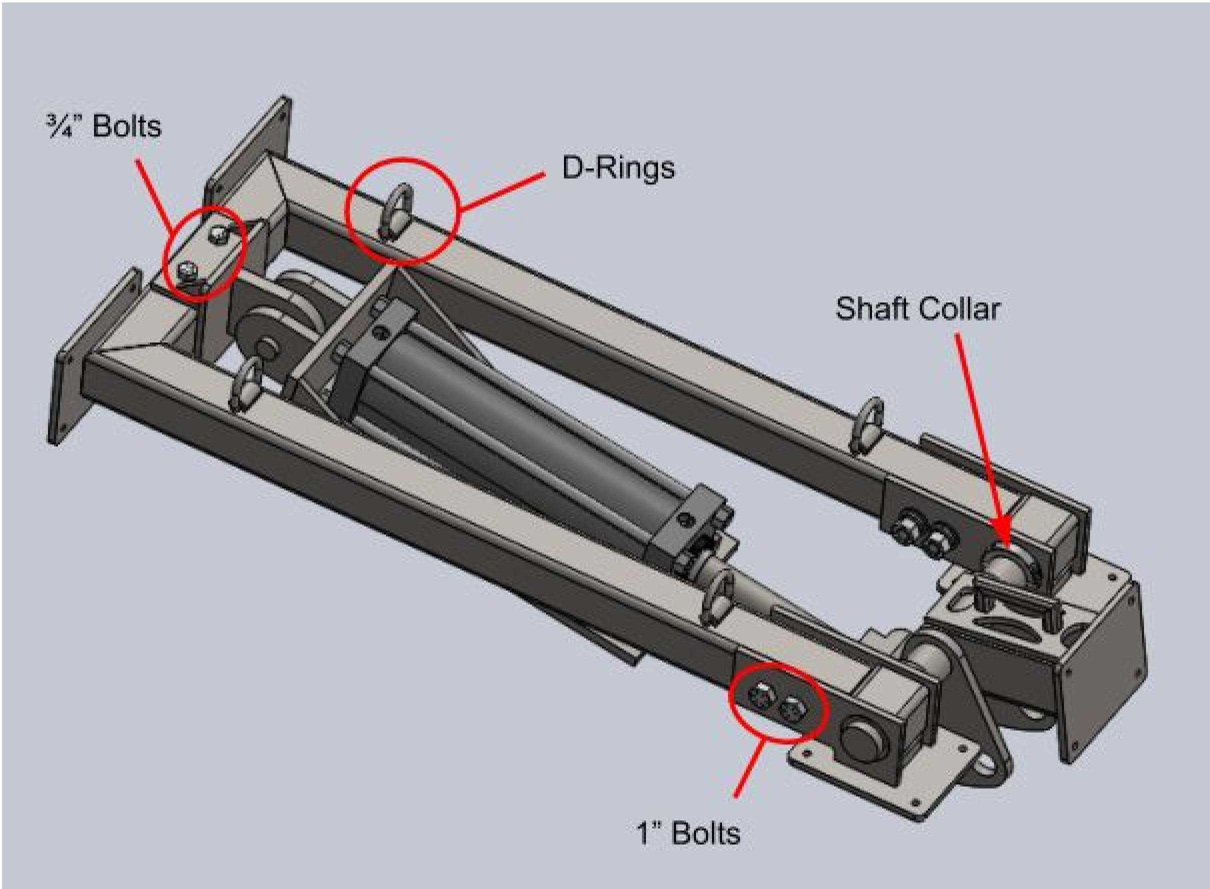

Figure 47. Solid Model of Main Assembly

Analysis

Due to the request by our sponsor to make this mechanism usable in three different mounting angles, we pursued efficiency throughout our analysis to fully analyze each mounting configuration. In order to improve the rate we performed analysis, we utilized various computational tools such as a spreadsheet calculator and a matrix calculator. We began our analysis on paper to develop governing equations for each loading situation. The analysis of each configuration followed the same analysis scheme: system level statics, shaft strength, shaft fatigue, welds, and fasteners.

During summer, we worked for several weeks figuring out the layout of our design as well as the general dimensions. Once we had a general design of the entire system, we used 3D modeling software to sketch out the geometry of our cylinder and the truss in order to find the specific geometric dimensions of our mechanism given our cylinder dimensions, maximum weight requirement, and height requirement. From this sketch, we were able to solve for dimensions including: lever arm length, distance between cylinder back mount and main shaft, angle of travel of the load, and angle of the cylinder relative to the lever arm.

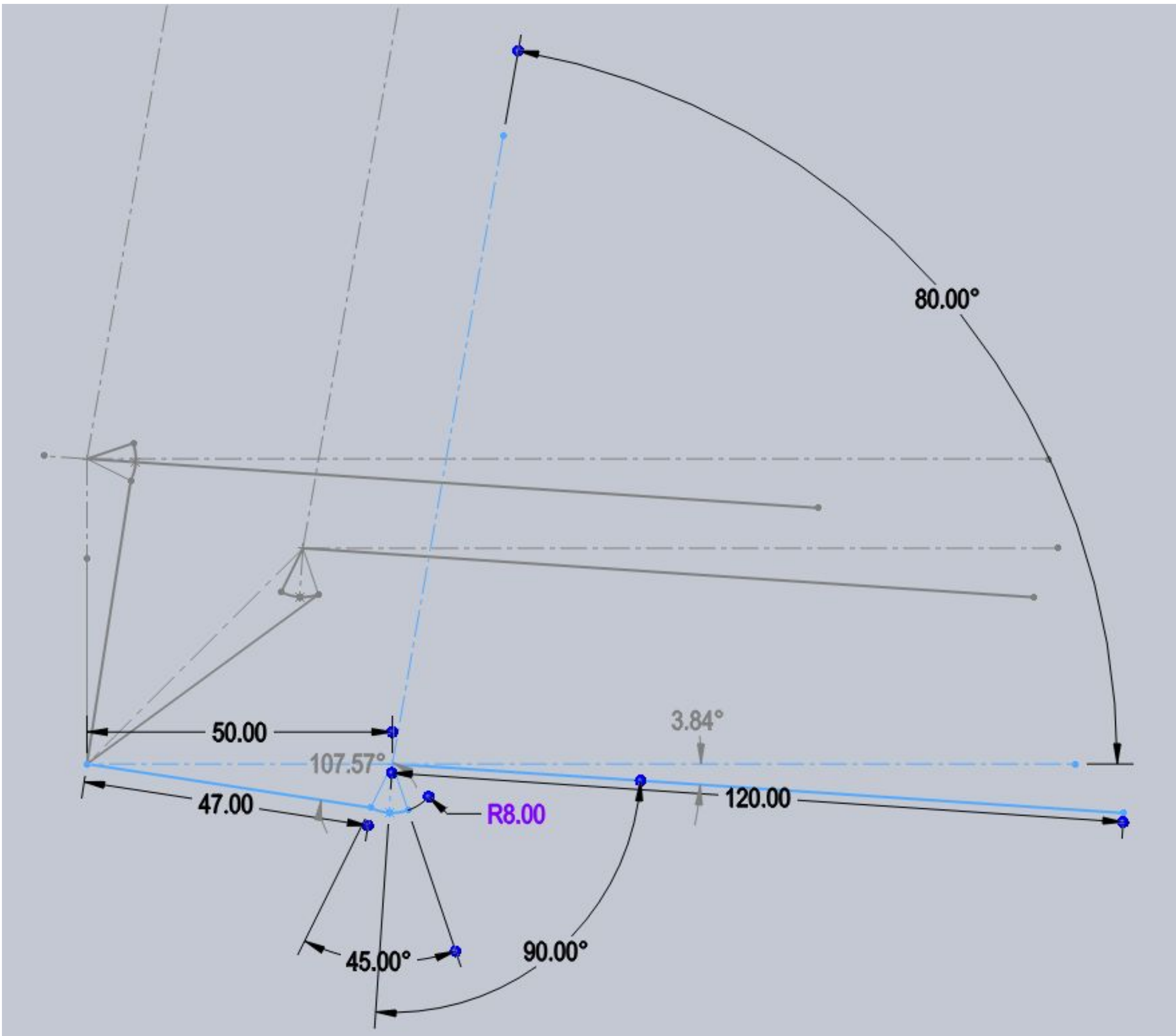

Figure 48. Geometric Sketch

Using these general dimensions, we performed a static analysis on our overheight system design in order to solve for the forces and moments at the shaft. These equations were developed by hand for all three configurations with a set of variables defined in the beginning of the analysis (Appendix I). The free body diagram for the 0 degree case can be seen in Figure 49. The derived governing equations were then transferred into a spreadsheet (Appendix J) to solve for the forces and moments at every angle the cylinder traverses. Utilizing a spreadsheet calculator to solve for these forces and moments for each configuration allowed us to input these values into various matrix analysis files for shaft, weld, and fastener analysis.

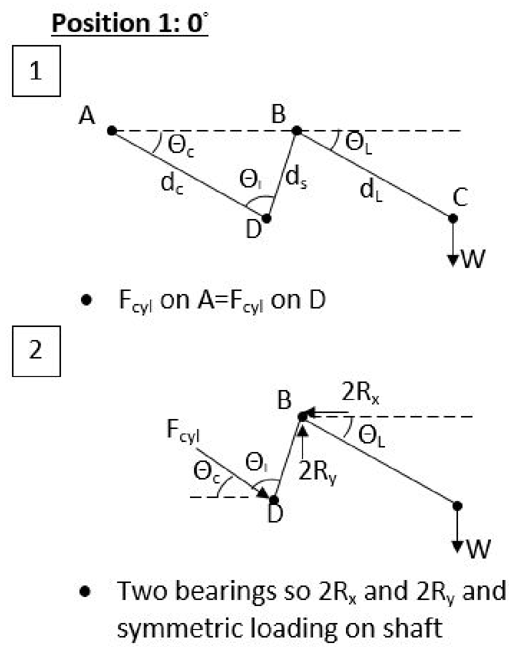

Figure 49. Static Analysis FBD

Using the shear and moment forces from the static analysis spreadsheet, we were able to solve for the stress in our shaft at every angle the load rotated. Unlike other common machinery, our shaft loading had varying moments and torques. Due to these conditions, we analyzed our shaft at every angle of travel in our shaft strength analysis. As seen in the figure below, we solved for the maximum stress at two points on our shaft with the highest combined loading (Point A and Point C). We then took the larger stress between these two points to solve for our lowest safety factor for shaft strength using the distortion energy theory (Appendix K).

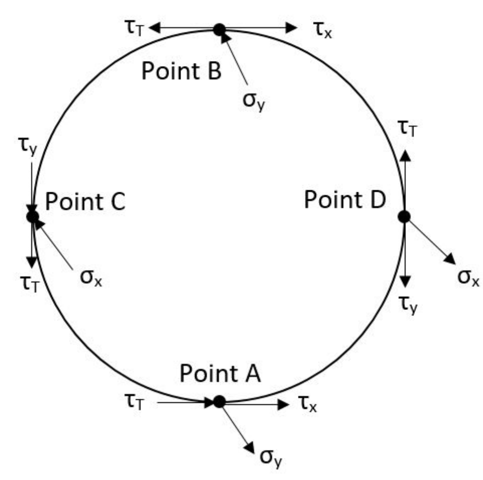

Figure 50. Shaft Combined Loading Diagram

After the strength analysis, we solved for the fatigue safety factor. Since our shaft does not rotate 180 degrees in a fatigue situation, we used the minimum and maximum stress a single point on our shaft will experience. From the figure above, the four main combined loading points on our shaft are listed as A, B, C, and D. For a fatigue case, the worst scenario is when point A in the down position traverses to point D in the up position. We can prove this to be true because the highest bending in the y direction is when the mechanism is fully down whereas the highest bending in the x direction is when the cylinder is keeping the load upright. Using this relation, we found the mean and alternating Von Mises stress. Once we solved for the corrected endurance strength, we calculated the fatigue strength since this is a low-cycle fatigue case. With all these values, we checked Langer static yield safety factor as well as the Modified Goodman safety factor. All of these safety factors were solved for symbolically as seen in Appendix K. Lastly, we transposed all this analysis into a matrix calculator file to allow us to parametrically solve for the shaft material and size needed to satisfy our safety factor requirement for each condition (Appendix L). Moreover, the matrix calculator allowed us to solve for each mounting configuration more easily, since we only needed to change the initial data called out at the top of the file. From our fully parametrized analysis with the matrix calculator, our resultant shaft diameter and material necessary to meet the safety factor requirement was a 1045 cold drawn steel shaft with a diameter of 2.75 inches. The safety factor results for distortion energy and fatigue of each mounting configuration are shown in Table 10 and Table 11, respectively.

Table 10. Safety factor results Distortion Energy

Shaft Analysis- Distortion Energy	
Design Parameter	Value
0 Degree SF	2.02
45 Degree SF	2.29
90 Degree SF	2.26

Table 11. Safety factor results for Fatigue

Shaft Analysis- Modified Goodman	
Design Parameter	Value
0 Degree SF	2.93
45 Degree SF	2.93
90 Degree SF	2.74

With the shaft size defined, we finalized our model of the mechanism in our 3D model. After all the dimensions were finalized for the beams and other structural steel, we continued to analyze welds on our mechanism. Most of our mechanism is welded together, and we could not feasibly analyze every weld on our mechanism, nor would it be reasonable to do so. Thus, we broke down our weld analysis to a few critical weld locations, which are on the main load path, as seen on the first page of Appendix M. From these weld locations, we analyzed each weld load condition (bending, torsional, and axial) for each mounting configuration (Appendix M). Once we developed equations for each weld location for the zero degree mounting configuration, we related those shear forces to the forces seen in the other two mounting configurations. This allowed us to use the same general equations for each weld location regardless of mounting configuration, and only change the shear forces in our analysis with the matrix calculator. The complete analysis for all the weld locations at every mounting angle are shown in Appendix N. From this analysis, we determined the weld height required to maintain our safety factors above the minimum requirement. The safety factor results from this analysis are shown below in Table 12, given a weld height of 0.375" using a 7018 electrode. Case 6, the back cylinder mount, and Case 9 which is the weld connecting the shaft to the truss attachment had the same safety factor for all configurations; consequently, they are shown at the top of this table. On the other hand, Case 1 varied for each mounting angle so the safety factors are described for each angle. Lastly, we checked the weld around our safety handle. Under the same weld conditions as the other analyzed welds, the safety handle has a safety factor of 3.11.

Table 12. Safety factor results for welds

Weld Analysis	
Design Parameter	Value
Case 6 SF	22.52
Case 9 SF	3.52
0 Degree	
Case 1 Torsional SF	4.02
Case 1 Bending SF	9.29
45 Degree	
Case 1 Torsional SF	19.56
Case 1 Bending SF	2.18
90 Degree	
Case 1 Torsional SF	4.30
Case 1 Bending SF	12.98

The last form of analysis we performed was fastener analysis. Although there are many fasteners, we only investigated two areas where we used fasteners. Since we are using Grade 8 bolts on the entire mechanism, the yield strength of the welds are lower than the yield strength of the bolts. Thus, at the joints where we used welds and fasteners to carry the loads, the welds would fail before the bolts sheared. Moreover, the back cylinder mount bolts were unlikely to shear because most of the load is transferred straight through the 4" by 4" beam. The two locations where there were no welds for additional support were where the shaft subassembly attaches to the frame and where the truss attaches to our mechanism. We followed a similar pattern of analysis where we developed governing equations we could parametrize in the matrix calculator and spreadsheet calculator. The hand analysis showing the load case and symbolic equations is shown in Appendix O. From our matrix and spreadsheet analysis, we were able to optimize the location, number, and size of these bolts. The results of these analyses are shown in Table 13 assuming partially threaded Grade 8 bolts with the bolt diameter at the top. The results for the analysis on these fasteners can be found in Appendix P. The bolts attaching the truss to our mechanism are also partially threaded Grade 8 bolts with a diameter of 0.75 inches. The safety factor for tension was lower than shear for these bolts, the analysis for which can be seen in Appendix P. The safety factor for tension loading of the eight 0.75 inch bolts attaching to the truss is 2.08 for fatigue.

Table 13. Safety factor and size results for bolts

Fastener Analysis	
Design Parameter	Value
Bolt diameter	1.00"
0 Degree	
Case 4 SF	7.70
45 Degree	
Case 4 SF	2.78
90 Degree	
Case 4 SF	9.06

Cost Analysis

We carried out a cost analysis based on materials necessary for our chosen concept. We had access to the materials in Cal Poly Rose Float's inventory. This saved us from having to buy material such as the required square tube steel, steel plate, Grade 8 hardware, match plates, and fittings. Most importantly, our mechanism design was based on an existing hydraulic cylinder that the Rose Float program already owns so that was also excluded from the cost analysis. Furthermore, there was no labor cost because we will be building this mechanism ourselves. The materials we needed included a 2 inch and 2-¾ inch diameter 1045 steel shaft, steel pipe and shaft collars that goes over the 2-¾ inch diameter shaft, snap rings for the 2 inch diameter shaft. We also needed to buy items for our safety chain lock system which included heavy duty chain, shackles, and D-rings. For the cost analysis, we picked out parts and added all their individual costs, which can be seen in Table 14. The cost came out to be $378.37. This excluded accounting for shipping and handling, but the total is well below our budget of $1300. Cut Sheets of the items in Table 14 can be found in Appendix H.

Table 14. Cost Analysis

Material	Specs	Vendor	Part #	Lead Time	Quantity	Cost [$]
1045 Steel Shaft	2-¾”Ø	McMaster-Carr	8924K74	2 days	3 ft	141.76
1045 Steel Shaft	2”Ø	McMaster-Carr	8924K58	2 days	1 ft	34.20
Steel Pipe	3-½” OD 2-¾” ID	Speedy Metals	N/A	4-6 days	1 ft	43.56
A36 Steel Rod	½”Ø	Metals Depot	R112	3-11 days	1 ft	3.96
Snap rings	For a 2”Ø shaft	McMaster-Carr	97633A420	2 days	10	6.23
D-rings	Grade 80	McMaster-Carr	3028T32	2 days	4	39.88
Shackles	2,000 lbs capacity	McMaster-Carr	8494T14	2 days	2	15.36
Shaft Clamping Collars	2-¾”Ø	Fastenal	34334	2-3 days	2	88.24
Heavy Duty Chain	Grade 70	McMaster-Carr	3363T93	2 days	2 ft	9.14
					Total [$]	378.37

Manufacturing Plan

This section provides a general overview of how we plan on building the mechanism as well as a general timeline of the total build time.

Initial Parts Acquisition

One of the biggest challenges that came with working with hydraulic components was their extremely long lead times when acquiring the parts. We anticipated it taking anywhere from three weeks to three months to acquire parts, and that is why getting an early start was very important for us. We had a full parts list by the end of week four of Fall Quarter so that we could order anything that would take a while to ship by week five. Rose Float’s collection of hydraulic components has grown tremendously over the years, and we found many of the parts we needed in their lab. As we designed our mechanism, we used websites like McMaster, Amazon, and Grainger to check if there are suppliers that stock the exact parts we needed, so we could decrease time spent on trying to fabricate every component ourselves. We tried to follow the 80/20 rule by trying to purchase at least 80% of our parts and manufacturing no more than 20%. We anticipated that we could stay within our budget by being flexible on the material we used and by incorporating what was readily available to us in the Rose Float Lab.

Final Parts Acquisition List

After CDR, the final purchase list was compiled and can be seen above in Table 14. The final critical parts that were built can be seen below in Table 15. These parts were made on campus using campus resources and consisted of utilizing various shops including but not limited to: Mustang 60/Hangar, BRAE Shops, Rose Float Lab, and possibly IT Shop’s Waterjet. When designing our final mechanism, we took into account all possible manufacturing method that we had access to and were comfortable using. We wanted to make sure everything was manufacturable without the use of special tools that were inaccessible or would take too much time to set up and use. Since our mechanism is a one-of-a-kind piece, we had to make sure to not make everything so custom that it would be too difficult to build. Even though most of our parts are custom, we made sure to use standard Rose Float parts including stock sizes, material, bolt sizes, and bolt patterns. One of the convenient things that we were able to do is to reuse one of our larger cylinders, while still making it removable for use in other mechanisms. This saved us time and money as we were able to design our mechanism from a solid starting point and we did not have to buy a $1200 cylinder.

Table 15. Final Subassembly Estimates and Processes

Part Numbers Needed	Part Name	Quantity	Estimated Difficulty	Estimated Build Time	Manufacture Process
009	Back Mount	1	Medium	14 Days	Plasma Torch+Weld
104	Base Struc.	1	Easy	4 Days	Bandsaw+Weld
005,013	Side Bearings	2	Medium	5 Days	Plasma+Weld
006,007, 008	Sleeve Bearings	2	Easy	2 Days	Bandsaw+Weld
002	Sandwich Plates	2	Hard	14 Days	CNC Plasma
003,004,013	Truss Attachment	1	Hard	14 Days	CNC Plasma+Weld

Initial Parts Manufacturing

In January of 2018, we began building our final mechanism. Before that, we planned to prototype our concept by creating a scaled down model. We used materials that we came across easily, such as cardboard or plastic. The prototyping phase happened after the CDR as a proof of concept and as a way to see what kind of issues we will need to address during our actual build. After the prototyping phase, we had a good idea of how we were going to build the actual mechanism. The actual mechanism was constructed in the Rose Float Lab as well as Mustang 60. Due to the shear size of the mechanism, it needed to be housed at the Rose Float Lab; thus, more precise parts were made in Mustang 60 while the heavy duty welding and metalwork happened at the lab. Manufacturing began as soon as we completed the CDR and were satisfied with moving forward in the process, given material was present. This timeline ranged anywhere from November or December 2017 to January 2018. According to our Gantt Chart seen in Appendix Q, we acquired materials, cut all structural steel, made mounting plates, and built the structure to hold the components starting in January. In February, we welded everything that needed to be welded and assembled the mechanism. By mid February, we began testing and troubleshooting any issues that may have arisen with the mechanism. An in depth outline can be found in the Gantt Chart in Appendix Q.

Final Parts Manufacturing Plan

As summarized in Table 15, our critical assemblies and parts took some time to build and assemble. As a result we needed to begin manufacturing the quarter of Fall 2017, so that we would have adequate time for assembly and testing. A simplified critical path list has been identified below in Figure 51.

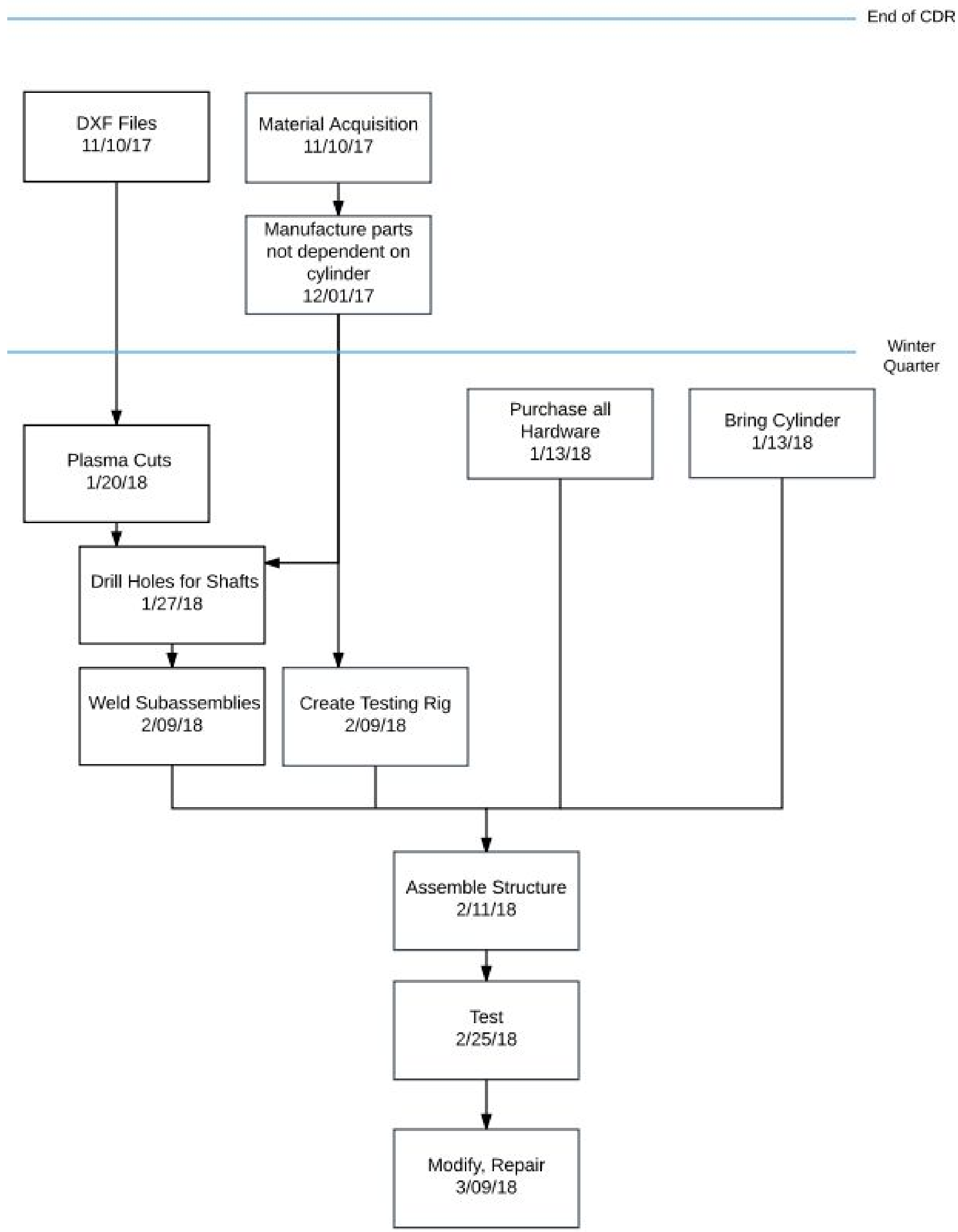

Figure 51. Critical Path Flowchart

Individual Parts Manufacturing
This portion of the report outlines the plan to manufacture each of the parts.

Main Frame:

The main mounting frame will be manufactured by cutting 4x4 A36 steel by cutting the two longer piece at a 45° angle on our Marvel vertical bandsaw. The back shorter piece will also be cut on the bandsaw at a 45° angle. We will then be drilling the holes on our radial arm drill press. The 2 back holes are going to be ¾" while the front 4 holes are 1" in diameter. When everything is cut and drilled The 3 pieces will be jigged and clamped to a flat table to be welded. We will take extra precautions to make sure the pieces stay in line and do not warp and cause misalignment. We will be running at minimum 2-3 passes with 7018 welding rod over any welding surfaces. The two match plates on the bottom will be premade and pulled from our stock, these will also be welded to the exact position so that they will mate with the bolt pattern on the float. The outline of the frame can be seen below in Figure 52. For detailed cut sheets and drawings please refer to Appendix H and R, respectively.

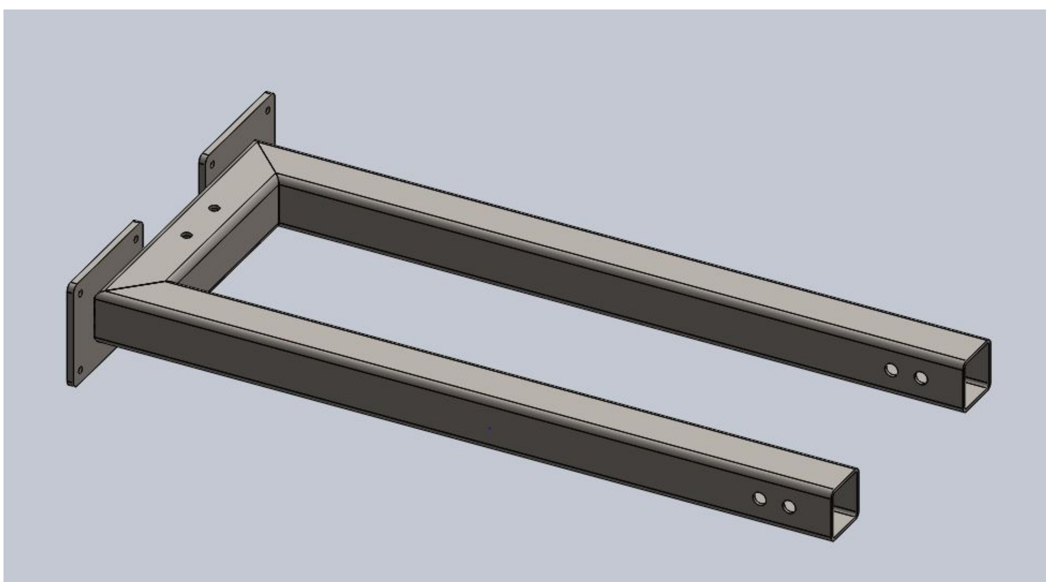

Figure 52. Main Frame Assembly

Main Frame Assembly:

The Back Mount, seen in Figure 53 will need to be fabricated in 8 separate pieces; however, there are only 4 different pieces, the other 4 will be copies of those parts. The first piece that will need to be made is the clevis, which is the rounded piece with a 1.5 in hole for the cylinder pin. This will be made out of 1 in plate steel and will be cnc plasma cut at the BRAE shop. In order to use the cnc plasma we will need to convert all the parts into a .dxf file. The piece will be cleaned up manually with grinders and flap disks. The next piece that will be made is the flat plate with no holes. This piece is the main piece that all the other pieces will be welded to. This piece will be made out of ½” plate also utilizing the BRAE plasma. The next 2 pieces are simple rectangular pieces that will have 2, ¾” holes in each one. These will be sheared from flat ½” stock plate using Rose Float’s Ironworker. The last 4 pieces are going to be gussets made from ½” plate, these will also be made using Rose Float’s Ironworker, specifically the notcher side. The entire setup will be clamped onto a 4x4 piece with minimal shimming to allow for some natural warpage for weld cooling. We will be using 7018 rod and running at least 2 passes over all welded joints. For accurate dimensions please refer to Appendix R. A backup for our plasma cut parts in the event that we are unable to use the plasma, is to either try the IT shop’s waterjet or hand cut the parts on a manual plasma and clean them up using the mill and hand power tools.

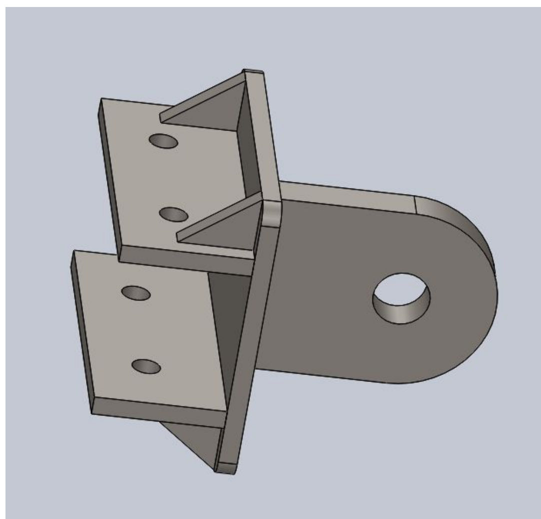

Figure 53. Solid Model of Cylinder Mount Subassembly

Bearing Support Assembly:

This assembly will be built twice as one is needed for each side of the mechanism and an illustration can be seen in Figure 54 below. This is arguably one of our most important parts in terms of alignment and making sure everything is square. The two bearing side plates will be manufactured on the cnc plasma in the BRAE Shop including the 3 holes that will be 3.5 in and 2 1 in holes. Then we will be cutting a 3.5 in OD pipe with a 2.75 in ID so that it can fit from outside surface of one side plate to the other. We will also have to bore the inside of the pipe in order to remove the weld bead left from initial OEM manufacturing, and for this we will be using the lathe in Mustang 60 as well as a deep boring tool. The 4x4 steel pieces will be cut on the Marvel vertical bandsaw in the Rose Float Lab which is also true for the pipe piece. The one 4x4 that the pipe will go inside will need to also have its internal weld bead milled out as it will cause interference with the pipe insert. The last piece we will need is one of Float’s standard Match Plates which is already premade, then once all the pieces are cut and fit together nicely, we will begin welding everything together. We will need to take great precaution in this step as there needs to be the least amount of warpage possible to make sure our shaft is not skewed in any direction. The pipe sleeve will also have a zerk fitting drilled and tapped into it so that the system could be greased to maintain its life. For more detailed drawings for each piece please refer to Appendix R.

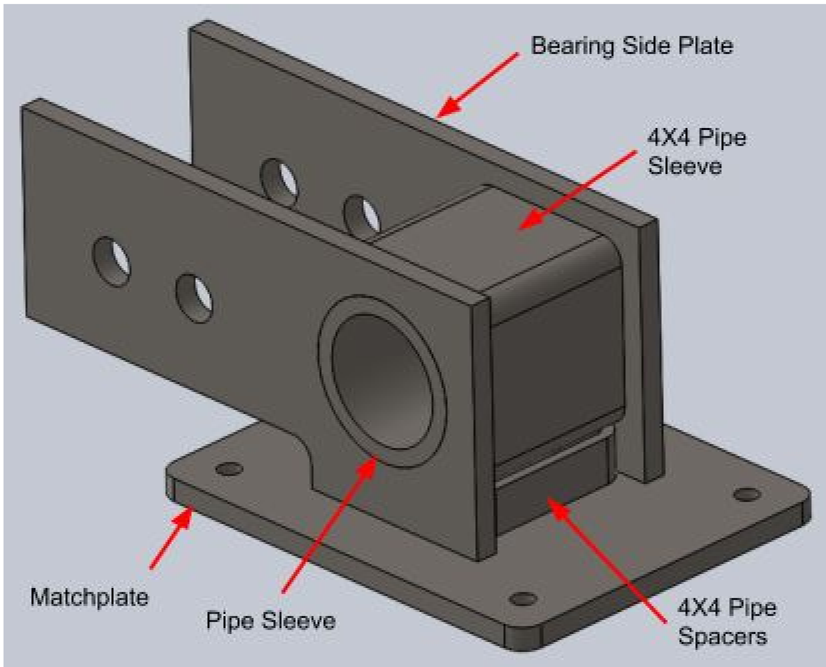

Figure 54. Solid Model of Bearing Support Assembly

Truss Support System:

For the Truss Support System, we will need to 6 separate pieces. The first two pieces are the Truss Mount Plates, these plates will be plasma cut from ¾” plate and they contain a 2.75 in hole and a radial on one end with a flat side on the other end. The next two plates are the Truss Braces and theses are more to create a beam than to take much load. These plates will also be plasma cut, but they will be cut out of a ½” plate. The next piece is the safety handle, this piece will be made out of 1x1 box steel which will be cut on a metal chop saw at 45° angle and welded together. The final piece will be a standard Rose Float match plate which will be pulled off our shelf. These 6 pieces will be welded together using 7018 rod in the orientation shown in Figure 55. These pieces will be jigged so that they cannot move during welding. For detailed drawings please refer to Appendix R.

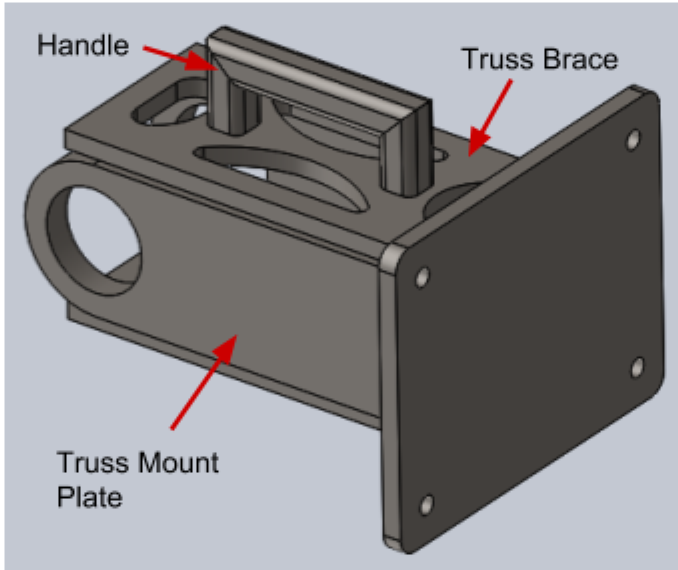

Figure 55. Solid Model of Lever Arm Attachment Assembly

Assembly and Safety Enhancements

Since the assembly phase will take place during the month of February, the half of the Float that belongs to Cal Poly should be in SLO with the Rose Parade finished. When the float is back, we will be able to assemble and test using the float as a solid mounting structure. When we begin assembling, we will need to take into account the accessibility of the mechanism so that maintenance can easily be conducted. At first, the mechanism will have a myriad of pinch points and safety hazards, but we intend to make warning stickers to indicate hazards. We also would like to allow for the implementation of sensors and warning lights that will help the users know when the mechanism is in the up or down position; an example of one of these parts is a limit switch, shown below in Figure 56.[23]

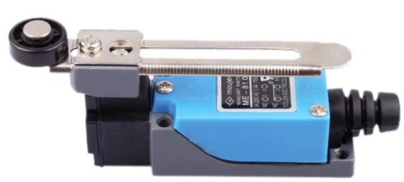

Figure 56. Limit Switch

Two limit switches will be placed in the back of the base frame where it would be triggered by the frame around the cylinder when the cylinder is fully retracted or extended. Furthermore, we will have a string potentiometer similar to the one shown below in Figure 57.[24] One end will be on the rod mount of the hydraulic cylinder while the base will be placed on the frame that is around the cylinder. This is so we can know the position of the cylinder in-between being fully retracted and fully extended as well as to verify the limit switches are activating.

Figure 57. String Potentiometer

One end will be on the rod mount of the hydraulic cylinder while the base will be placed on the frame that is around the cylinder. This is so we can know the position of the cylinder in-between being fully retracted and fully extended as well as to verify the limit switches are activating. The locations of the string pot and the limit switches can be seen below in Figure 58.

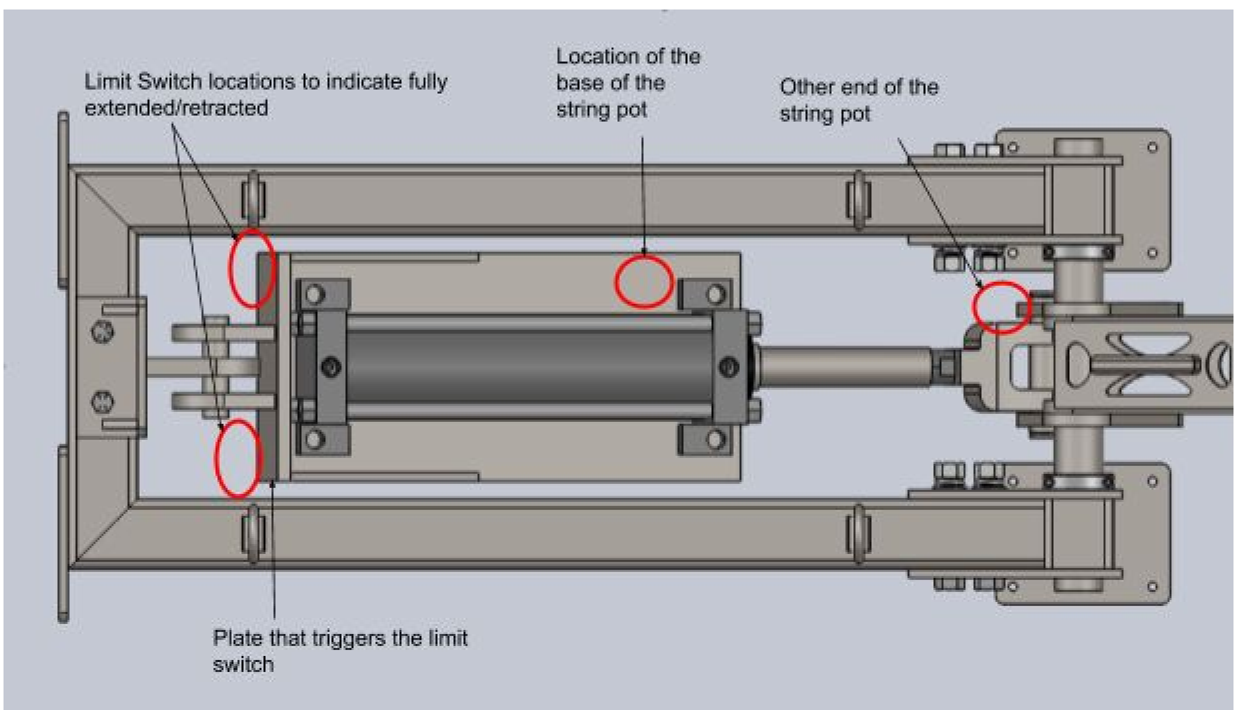

Figure 58. Location of String Potentiometer and Limit Switches

The two D-rings near the lever arm will have shackles connecting them to heavy duty chain that will loop around the handle on the truss attachment in the up position. This chain will help to safely retain the truss in its upright position while the system pressure is turned off. A depiction of this set up can be better represented below in Figure 59.

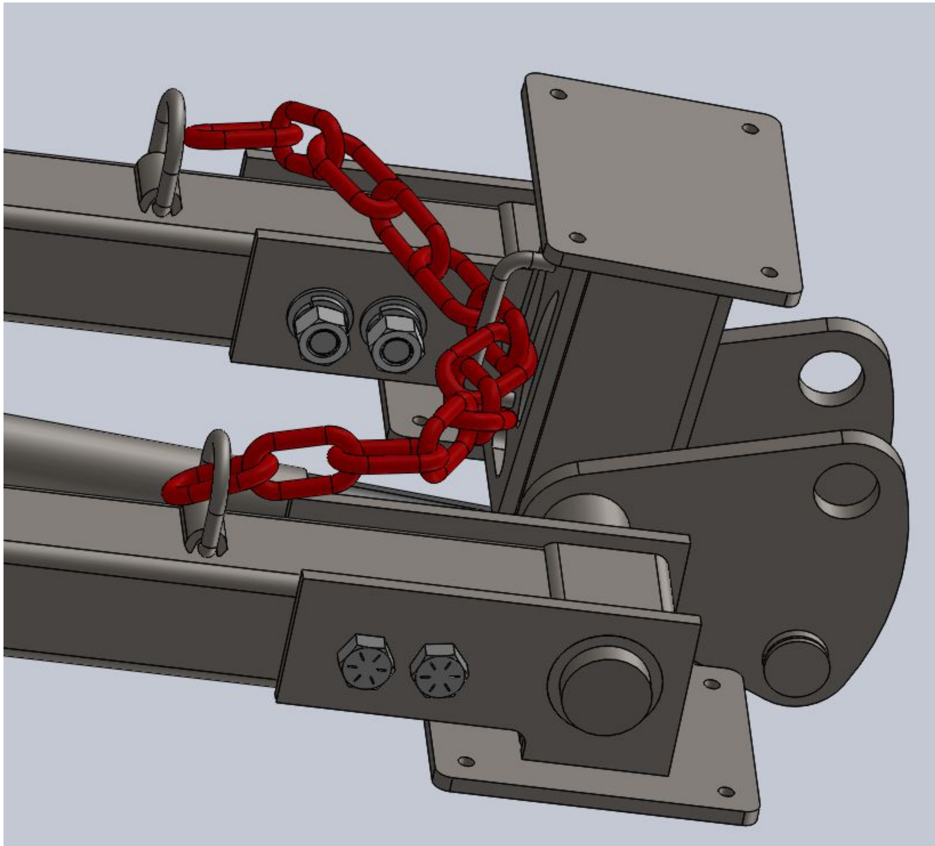

Figure 59. Safety Chain-Lock (Shackles not included)

Final Manufacturing and Assembly

This section provides an overview of our manufacturing process for our parts and assemblies. Moreover, we describe modifications made to parts and assembly processes throughout the process.

Mechanism Manufacturing and Assembly

Our team began manufacturing parts in November by cutting the box steel for the frame subassembly and the shaft subassembly. By the end of November, we had the box steel pieces cut and milled, the pipe sleeves cut and all the main match plates acquired (Figure 60). Also, we created the DXF files for water jetting parts early in January.

Figure 60. Box steel cut for frame and shaft sub assembly

Throughout January, we proceeded to water jet various plates for our assemblies. One of these plates is shown on the ITP water jet in Figure 61, and there are more water jet plates shown in Figure 62. For these plates, we increased the hole sizes to allow for clearance regardless of the taper on the water jet. The hole sizes were calculated using an estimated taper of 3 degrees from the shop technicians in the ITP department.

Figure 61. Plate on ITP water jet

Figure 62. Completed water jet plates

As we were testing the fit of the side bearing plates on the shaft assembly, we noticed we would have an easier weld if we elongated the pipe sleeves inside the 4x4 box steel such that there was ¼" extended past the plate. The side bearing assembly with the longer pipe sleeves are shown in Figure 63 below, and with this adjustment it would be a simpler fillet weld around the pipe circumference to plate.

Figure 63. Box steeland pipe sleeve manufactured for shaft assembly

The first pieces we leveled and tack welded together were the frame pieces as shown in Figure 64. After we had the base frame pieces tack welded, we began to assemble and tack weld the shaft assembly as shown in Figure 65. As we were assembling the jig for tack welding, we were careful to keep both subassemblies level and in the correct location relative to one another. The shaft and pipe clamps provided additional alignment to the jig set up.

Figure 64. Frame assembly leveled and tack welded

Figure 65. Shaft assembly leveled and tacked onto base frame assembly

Once the shaft assembly was tack welded, we then manufactured the pieces for the back cylinder mount. We tack welded two pieces of the back cylinder mount together, Figure 66, to use for marking where the holes needed to be drilled on the frame. We then leveled our entire assembly on I-beams to begin marking holes for drilling (Figure 67).

Figure 66. Back cylinder mount tack welded for marking holes

Figure 67. Leveled assembly for marking holes

After drilling, we assembled the shaft assembly to the base frame in order to weld the two subassemblies as one unit. We also assembled the back cylinder mount assembly on the back of the frame to tack the rest of the pieces together. For the welding process, we tacked struts of box steel to keep pieces aligned as well as welding in 1” sections on different pieces of subassemblies. This process worked well to keep our elements aligned so we did not have as much post machining. Figure 68 shows the base and shaft assembly welded and post machined. After we welded these sub assemblies and took out the strut, we had to dremel the 1” holes connecting the base frame to the shaft assembly due to warping which caused the shaft to no longer be concentric.

Figure 68. Fully welded shaft and base frame assemblies

As we were welding the mechanism, we were also welding together a truss for testing. Since the truss was made of a few struts of 2x4 and 1x1 box steel, we were able to manufacture the pieces quickly and weld while machining parts for the main assembly. The welded truss along with the matchplate used to attach to the mechanism is shown below in Figure 69.

Figure 69. Testing truss with matchplate

The last assembly we welded together was the truss attachment assembly. The truss attachment assembly consisted of the two sandwich plates welded to a box structure made out of four water jet plates (Figure 70). We tack welded 1x1 struts between the two sandwich plates to keep the cylinder pin and shaft holes concentric.

Figure 70. Truss attachment assembly in progress

We modified the truss attachment assembly by welding the gusset originally meant for the back cylinder mount. The gussets were welded from the box structure to the matchplate connecting the testing truss. We modified the assembly in this manner, because we wanted to ensure the matchplate connecting to the truss did not bend as we were lifting our load. Lastly, we added the matchplates, pipe bushings with zerk fittings, and lifting rings. The finalized mechanism with all the subassemblies welded and components attached is shown in Figure 71. For more detailed drawings on the final assembly, please refer to Appendix R.

Figure 71. Finalized overheight mechanism

Testing Fixture

In order to prepare for testing we not only built a testing truss, but we also manufactured a testing fixture. Our testing fixture was a long ¾” plate we utilized as a base to connect four upright posts to hold our mechanism. After welding the match plates on our mechanism, we bolted on another set of match plates to get their location on each post. We then leveled the mechanism and tack welded each match plate to their respective post. Once each post and match plate were welded we attached them back onto the mechanism to then weld them to the plate. The final testing fixture is depicted in Figure 72 below.

Figure 72. Truss attachment assembly in progress

For our specific testing plan, see the Preliminary Testing Plan section that follows. Moreover, the fully detailed description of our testing procedure and results are in the Testing section. For the use after testing, Appendix X is an operator and safety manual that outlines how to use and repair the mechanism. The manual will be very important for the Rose Float program since it will aid in keeping the mechanism maintained and in operating condition for years to come.

FMEA

For our Failure Mode Effects Analysis, we defined the system to be the overheight mechanism and structure attaching the mechanism to the float. The elements of the system have been defined to be the hydraulic cylinder, shaft-in-pipe bearings, mounts, main shaft, fasteners, and welds.

Failure Modes

For the FMEA, we determined the possible failure modes for each element. Table 16 below shows failure modes corresponding to each element of the overheight mechanism.

Table 16. Failure Modes to their Corresponding Element

Element	Failure Mode
Hydraulic cylinder	Seal breaking, rod fracture, insufficient pressure
Shaft-in pipe-bearing	Grease stiffens up
Mounts	Shear
Main Shaft	Buckling
Fasteners	Shear
Welds	Welds breaking

Classification of Severity Failure Scale

We ranked each failure mode on how severe the effects would be if they occurred. Table 17 defines the scale that we used.

Table 17. Classification of Severity Failure Scale

Scale	Definition
Negligible: 1-2	Less than minor injury, occupational illness, or system damage.
Marginal: 3-5	Minor injury, occupational illness, or system damage
Critical: 6-9	Severe injury, occupational illness, or system damage.
Catastrophic: 10	Death or system loss.

Probability of Each Mode

Similarly, we also judged each failure on the likelihood it would happen, and the scale for this aspect is shown in Table 18.

Table 18. Probability of Failure Scale

Scale	Definition
Extremely Remote: 1-2	Unlikely to occur.
Remote: 3-6	Possible to occur in time.
Reasonably Probable: 7-8	Probably will occur in time.
Probable: 9	Likely to occur immediately or within a short period of time.
Guaranteed: 10	It will occur.

The severity and probability of all the failure modes can be seen below in Table 19.

Table 19. Classification of Severity and Probability of Failure for each mode

Element	Failure Mode	Severity	Probability	Risk = Severity*Probability
Hydraulic Cylinder	Seal Breaking	5	3	15
	Rod Fracture	8	2	16
	Insufficient Pressure	4	3	12
Shaft-in-pipe bearing	Grease Stiffens Up	6	2	12
Mounts	Mount Breaking	6	3	18
Main Shaft	Buckling	9	2	18
Fasteners	Shear	8	3	24
Welds	Welds breaking	7	3	21

The severity for most of the failure modes lie in the critical range since our mechanism is designed to carry a large load high off the ground. Furthermore, most failure modes will cause damage to the overall mechanism that would be tough to repair especially since we might not be able to have access to the mechanism back down at that given time. Failure modes like insufficient pressure are not considered as severe because it would not cause any damage; however, the mechanism would not operate the way we designed it to. The failure mode for the main shaft was given a value of 9 since the main shaft is what we consider to be the most critical part of the mechanism. It would also cause the most damage if it were to failure.

As for the probability values, most of the failure modes have a greater chance of occurring in high speed applications. Since we are dealing with a very low speed application, our concern mainly has to do with loading. However, we are very aware of the load hazards and are going to design with high safety factors to minimize the probability of failure. Therefore, we decided that the probability values for all failure modes would be in the remote range. This led to low risk values with the highest risks being with fasteners shearing and welds breaking.

Cause and Effect

We assessed all the failure modes to determine their causes and the effects if they fail. The results are outlined below in Table 20.

Table 20. Failure Mode Cause and Effects

Element	Failure Mode	Cause of Failure	Effect of Failure
Hydraulic Cylinders	Seal Breaking	Dirt, mud, etc. getting in the lines	Cylinder will not work properly; cylinder needs to be repaired or replaced
	Rod Fracture	Incorrect loading on rod	Cylinder needs to be replaced; system design needs changing
	Insufficient Pressure	Force required exceeds the capability of the pressure we can supply	Cylinder will not push/pull; cylinder sizing needs change since pressure cannot change
Pipe Sleeve Bearing	Grease Stiffens Up	Improper grounding when welding	Rotation about the pivot will be more difficult or not occur at all
Mounts	Mount Breaking	Improper welding or too much stress on welds	Cylinder will be out of place; lever arm will fall if both mounts break
Main Shaft	Buckling	Selected sizing or material of shaft is incapable of handling the loading	Design mechanism with a shaft that is capable of handling the loading
Fasteners	Shear	Too much shear force on the fasteners from the weight	System damage, misalignment. Undesired & uncontrolled motion
Welds	Welds break	Incorrect welding procedure	Depends on location, can increase the force that other components feel which can cause further failures

Designing for Safety

For safety, we came up with some recommended actions in order to reduce the probability or improve the detection of the failure mode. Table 21 below shows the recommended actions we came up with.

Table 21. Recommended Actions to Curb Failure

Element	Failure Mode	Recommended Actions
Hydraulic Cylinders	Seal Breaking	Inspect the lines, improve storage and treatment of lines, clean lines
	Rod Fracture	Proper set up of cylinder
	Insufficient Pressure	Proper sizing of cylinder so the pressure supplied will produce the proper force
Pipe Sleeve bearing	Grease Stiffens Up	Don't ground connecting material
Mounts	Mount Breaking	Proper welding, inspecting the welds
Main Shaft	Buckling	Design mechanism with a shaft that is capable of handling the loading
Fasteners	Shear	Using high grade fasteners. Calculate the safety factor of the bolts in shear
Welds	Welds breaking	Proper welding procedure and calculate safety factor on the welds

A complete overview of the FMEA can be seen in Appendix S.

Hazard Identification Checklist

To identify and address some of the safety hazards encompassed in the overheight mechanism design, a Concept Design Hazard Identification Checklist was completed (Appendix T). While answering the questions about our project, we realized there were many more hazards than we initially expected. Some of these included exposing the device to extreme environmental conditions and having stored energy in the system in the form of pressurized fluid. The hazards we expected included: the pinch points in the mechanism, the large mass being lifted, the possibility of the user being injured by the device if it fell under gravity, danger of inhaling welding fumes, and consuming hydraulic fluid.

To address all these dangers, corrective actions were discussed to minimize the possibility of injury while operating the device. For the pinch point, pinch point caution stickers will be placed in a visible location near the hazard on the overheight mechanism. Also, a chain implemented, as described in the safety section of the report, to keep the mechanism in the up position when people are near it, such as when the flowers are placed on the float. This chain will prevent the overheight structure from falling on people in the case the hydraulic system fails. When it is time for the parade, the chain will be removed, so the mechanism will drop down if the hydraulic system loses pressure in order to have the float clear the bridge. To warn users of the pressurized fluid, a pressurized fluid sticker will be placed on the device in a clearly visible location. In the instructions manual, we will emphasize that our device is for use as an overheight mechanism only to prevent misuse. Also, the manual will state that the hydraulic fluid is hazardous when consumed, and that breaks for fresh air should be taken when welding during the manufacturing process.

Design Verification Plan

When it came time to test our mechanism for the specified requirements, we needed to have a plan to test each and every one; the plan is outlined in this section, and the results are in the Testing Results section. Our testing plans are outlined in the Design Verification Plan seen in Appendix U. For each specification, a test description was written and the quantity of samples were defined. The Test Report section was completed when testing was carried out in March 2018.

Lift Capacity

In the design requirements, the accepted lift capacity was set to be between 1000 and 1500 pounds. With our design, the mechanism will be able to withstand a load of 1500 pounds with a safety factor of at least two. Therefore, we are going to load our overheight mechanism with a weights of 500, 1000, and 1500 pounds borrowed from the BRAE department to ensure it can meet this requirement. Additionally, we will test the lift capacity through nine cycles, three for each weight, to gather a good amount of data while ensuring the mechanism can withstand weights less than the design weight. There will be three trials with 500 pounds, three with 1000 pounds, and three with 1500 pounds. Since the 0 degree case is the worst loading case, we will only be testing this orientation. In addition, the chassis did not come back to San Luis Obispo in time for this final quarter of Senior Project, so if desired, the 90 degree and 45 degree cases will be tested upon its return. Theoretically, we expect our mechanism to operate with ease under the 1500 pounds because of our relatively high safety factor. We expect to complete the testing for the lift capacity in one day on March 11, 2018.

Distance of Load from Pivot Point

Since the moment created by the point load is maximized when the load is placed at the very tip of the load lever arm, we will measure the distance from the pivot point to the end of the lever arm as the distance of the load from the pivot point. The acceptable criteria for this specification is 8 ± 2 ft. The length of the load lever arm will be measured using a tape measure, and the measurement will only be taken twice since this distance is a fixed value. The second measurement is to ensure that the first one is correct. A one-day window was allowed for this test since one trial can be completed in a few minutes; the testing for this requirement will be conducted on March 11, 2018.

System Pressure

The system pressure should be at a maximum of 1400 pounds per square inch. To ensure that the pressure is constant and below the specified value, we will use a pressure gage to measure the pressure of the hydraulic fluid at two different points in time. More specifically, one data point will be collected when the system is starting to go into the up position and one when it is fully in the up position. We allowed one day, March 11, 2018, to test this requirement since it is fairly easy to test.

Gravity Drop Time

According to the Float Manual distributed by the Tournament of Roses, the maximum gravity drop time is 60 seconds. The gravity drop time will be tested after three lift capacity tests; this is because the lift capacity test involves bringing the mechanism fully into the raised position with a load on it. Therefore, the gravity drop time will be measured three times per orientation. When the mechanism is in the raised position, it will be released, and the time it takes for the arm to fall into the down position due to gravity will be recorded with a stopwatch.

Height Difference

The requirement for our height difference ranges from 4 to 14 feet. To test this specification, the vertical distance from the ground to the tip of the lever arm will be measured in the down and up position, and the difference between the two values will be calculated. This process will be completed only two times because the height range should also be a constant value in each case. One day is allowed for the height difference testing because it is fairly simple to test, but the mechanism needs to be raised and lowered to collect data. Therefore, the samples for this specification will be collected the same days as the lift capacity test.

Operators Required

As stated earlier, it should be possible for one person to operate the mechanism easily because only one person will be available to operate it during the Rose Parade. To check that this requirement is satisfied, each member of Team HighRise will operate the mechanism on their own as well as a few members of the Rose Float Team. If everyone succeeds, then the requirement is fulfilled. These trials will be conducted towards the end of the first testing phase, which is on March 11 of 2018.

Locking System

As a safety precaution, a locking system will be incorporated into the mechanism to prevent injuries from happening in the case the hydraulic system shuts down unexpectedly when people are working on the float. To test this, the mechanism will be raised to the up position, and the hydraulic system will be shut off. If the lever arm stays up when the hydraulic system is shut down, that means the system is preventing gravity from lowering the arm, so the locking system is working correctly. The mechanism will still lower if the valve in the hydraulic system is opened. This process will be completed three times with max weight conditions to ensure that the locking mechanism works before it is used on an actual float. The testing date for the locking system are March 17 in 2018.

Unpressurized Position

When the system is unpressurized, the lever arm should default to the down position. This test will be conducted along with the gravity drop test, at least three times, since the hydraulic system will be shut off when the device is in the up position. If the mechanism goes into the down position due to gravity, then the unpressurized position requirement is satisfied.

Safety Factor

Since the required safety factor for each element ranges from two to five, the overheight mechanism was designed to meet the minimum safety factor with a great amount of analysis on the shaft, as mentioned previously. In addition, the lift capacity test is testing the safety factor in a way because if the safety factor is actually less than one on any component, the overheight mechanism will catastrophically fail.

Floats Can Be Used On

It is required that our mechanism be usable on at least five different float designs. Since there is no way for us to know what the design of future floats will be, the test for this specification will be conducted by looking at past float layouts. Fifteen significantly diverse floats will be chosen from those constructed the previous twenty years; the past floats will be analyzed to determine whether our overheight mechanism could be implemented. If our device could be used on five out of the ten chosen floats, then our mechanism satisfies this requirement. Testing for the number of floats the mechanism can be used will be conducted this upcoming March 17 to 18.

Volume

From our design requirements, the volume of our overheight mechanism cannot exceed 15 cubic feet. To calculate the actual volume of our manufactured device, the necessary dimensions will be measured with a tape measure, and the volume will be calculated. In the case the volume of our mechanism cannot be easily calculated due to its shape, the volume may be slightly overestimated with the shape simplified. The volume will be calculated once because it is a constant value. Since this requirement only involves measuring the overheight dimensions, the testing will be completed March 11 in 2018.

Preliminary Testing Plan

This section outlines our plans on how we will test our mechanism to see if it holds up against the standards and requirements we set at the start of the project. Our detailed testing plan is available in our DVP section and the Gantt Chart.

Plan of Action

Some of the most important requirements that needed to be tested after this mechanism was built included lift capacity, gravity drop time, volume in the down position, operation by one operator, and a safety lock position. When the mechanism was fully functional, we needed to also incorporate building a testing rig since we did not have access to use the float chassis. We conducted testing 2-3 weeks before we presented the mechanism to the Rose Float Program. We wanted to make sure that this device could serve the needs of the program for future years, and, thus, it needed to be rigorously tested to make sure it could withstand any scenario it may encounter while in use by Cal Poly Rose Float. The testing plan also incorporated time for modifications to the mechanism as those may be important if requirements were not met. Our plan was to test the mechanism in every extreme case possible, which included overloading the mechanism within reason. For our testing to be truly accurate, we needed to document all results and have a member from the Rose Float Team present, so they could learn the capabilities and limitations of the mechanism.

Testing Conditions

In order to test the outlined requirements, certain procedures were established. For instance, when we were testing the lift capacity of the mechanism, we loaded it at its maximum rated capacity in order to test our safety factor conditions. This was very important because having a low safety factor could be catastrophic. We loaded the device with a weight greater than what we felt would ever be utilized in the program, but this was also important so that we may gain confidence that the device would not fail during operation or during the parade. Another important requirement that we tested is gravity drop time, which was done by loading the mechanism, raising it to its maximum height, and taking down the time it takes to lower to its lowest height without assistance from an outside power source. In order to test the mechanism to see if it would work on at least 5 other float designs, we took our design and cross checked it with the past 20 years to see if it would work in those scenarios.

Necessary Materials Needed for Testing

The mechanism needed to be hydraulically controlled; thus, it needed to have a hydraulic power source so that it could function. Normally on the float, there is a Chevy 350 Engine that powers all the hydraulic components; however, this engine is on the Pomona half of the float, so we did not have access to that source of power. For any sort of testing to take place, we needed to utilize a different power source; there were several options available including: the SLO engine, the hydraulic test bench, and the Upright hydraulic motors. Each option had its own pros and cons, and what we ultimately used depended on the situation at hand. Other necessary equipment included a forklift to assist in lifting the large weights onto the mechanism for the lift capacity test. Fortunately, on our campus we had access to many different sized forklifts as well as operators who would be willing to aid us in our efforts. In order to test our lift capacity, we used a tractor pull sled weight, which can weigh upwards of 2000 pounds. This item was easily borrowed from the BRAE department, and it looked like a small cube made of steel with side lengths of 2 feet. This weight can represent a point load for our largest loading condition in order to test for failure of any of our parts and structures. Therefore, our apparatus was tested with a weight approximately 1.5 times the rated maximum lift capacity.

Testing Results

Upon the completion of manufacturing, tests were conducted to ensure the overheight mechanism met all the requirements established at the start of the project. The results can be seen in the DVP in Appendix U. Due to some delays in the return of the float chassis, a test rig was constructed to begin the testing procedures outlined in the Design Verification Plan.

Lift Capacity

To test the lift capacity, the overheight mechanism was installed into the constructed test rig and connected to the hydraulic test bench. The hydraulic system was cycled, meaning the mechanism was raised and lowered numerous times, in order to remove air from the hydraulic lines. The test weight was attached to the truss with a wood pallet and four tow straps, as shown in Figure 73. The weight consisted of angle iron borrowed from the BRAE Department; with each piece of angle iron weighing approximately 25 pounds, the first three runs were completed with 20 pieces on the pallet for a total of 500 pounds. The run consisted of raising the loaded weight to the maximum height from the ground and lowering it back down. The overheight mechanism was able to lift 500 pounds with ease. An additional 20 pieces of angle iron was added to the pallet to test the 1000 pound weight. Three runs were completed with 1000 pounds with no problems with the test truss or the mechanism itself. With 1500 pounds loaded, three more runs were completed. Since this was the maximum design weight, we paid extra attention to ensure the mechanism wasn't failing in any way. Under such a high load, the pin connecting the tow straps to the test truss was experiencing a great amount of friction, so it did not rotate as smoothly as it did in the previous six runs. This caused the weight to swing while the overheight mechanism was raising and lowering, so we performed these runs slower.

Figure 73. Test Setup

Distance of Load from Pivot Point

Since we decided to test our mechanism at maximum capacity, we had our testing truss longer than the maximum distance from the pivot point. Our testing truss was manufactured such that the load was approximately 11 feet from the pivot point. Accounting for the weight of the truss and the added load weight, our overall distance away from the pivot point was calculated as approximately 10 feet from our pivot axis. We measured this using a combination of knowing the loaded weight and measuring the weight of the truss from CAD. Using these weights and the two distances of each center of gravity, we calculated the accumulated center of gravity distance from the pivot point.

System Pressure

Our system pressure was measured off a gauge on the hydraulic test bench we used to power our mechanism. We watched the pressure gauge as we were lifting the truss to ensure our maximum system pressure was 1400 pounds per square inch. For each run, we measured the system pressure at the start of raising the load. While measuring the system pressure, our highest value was while raising the maximum load where it reached 1400 psi for a short period when it was first raising the load. For the majority of the lifting time for each run, our system pressure was around 1100 to 1200 pounds per square inch.

Gravity Drop Time

According to the Float Manual distributed by the Tournament of Roses, the maximum gravity drop time for a float overheight is 60 seconds. The gravity drop time for our mechanism was tested by lifting the truss up to maximum height and then releasing the valve on the test bench to allow hydraulic fluid to flow freely. We performed this test three times using maximum loading to calculate an average gravity drop time. From our testing, the average gravity drop time was 25.4 seconds.

Height Difference

During testing, we calculated the height difference achieved by measuring the height of the truss when fully lifted and in the horizontal position. After measuring this value twice, our actual height difference measured was 10 feet 9 inches. Thus, the height difference of our fully loaded mechanism is well within our range of 4 to 14 feet.

Operators Required

For each run of the lifting tests, there was one person operating the valve on the hydraulic test bench powering the mechanism. Since our system needed to be simple enough for a single operator, we tested this requirement by having each member of Team HighRise operate the mechanism for at least one of the test runs. Every operator easily controlled the system in lifting and lowering the load, and passed this requirement.

Locking System

In order to test the locking capacity of our system, we raised a fully loaded truss to the maximum height and shut down the hydraulic power. With the hydraulic power off, we then ensured our mechanism stayed in the upright position due to static pressure in the system resisting gravity. Even though the truss did not fall due to gravity, we were able to lower the truss if we opened the valve to allow free flow. After testing this requirement three times at max loading, we confirmed our system successfully locked in the upright position without a powered system.

Unpressurized Position

As described above, the system must still lower to the down position if the system is unpressurized. We conducted this test three times while the system was loaded and unloaded to confirm our system lowered to a default down position. Our mechanism lowered each run once we released the hydraulic pressure.

Safety Factor

In order to check for the safety factor, we tested the overheight mechanism under a load of 1500 pounds, which was our design load. Under this load, the overheight mechanism did not have any failures, so the runs were a success. It was not physically possible to test for the exact safety factor of the mechanism because we would have had to keep loading it until it failed catastrophically.

Floats Can Be Used On

As stated in the Design Verification Plan, we looked at significantly different float designs over the past years to see if our overheight mechanism could be implemented. Specifically, we looked at the Cal Poly float designs from 2008 to 2018, excluding 2009 because of its similarity to another float. When looking through these ten designs, there were only three years where our overheight could not be incorporated, which were 2010, 2011, and 2016. Our overheight exceeded the requirement that five out of ten designs should be able to include our mechanism. Thus, this overheight mechanism is very likely to be used in upcoming years.

Volume

The volume of the overheight mechanism was calculated to be 14.5 cubic feet from the dimensions measured. This value meets the design requirement that the volume must be less than 15 cubic feet. In addition, the calculated volume is the volume of a prism that can hold the mechanism within it, so the dimensions were measured in areas where they would be maximized. The dimensions of the overheight mechanism were measured to be 69” by 24” by 15.2”. Due to accessibility, our design includes an external cylinder collar rather than an internal one. If we had decided to implement an internal cylinder collar, the length of the mechanism would be approximately 9.8 inches shorter, which would make the volume 12.5 cubic feet.

Management Plan

Since this is a team project, all members must work in unison to produce a finished product that meets the needs of the sponsor. Furthermore, all members must take on responsibilities to meet the specified deadlines and prevent procrastination. In the following subsections, the responsibilities of each member is outlined, and the timeline of the project is discussed.

Member Responsibilities

Each of the team members was assigned a general role in the team contract. As the Communications Officer, Ali Harake will be the main point of contact with the sponsor, Josh D'Acquisto from Cal Poly Rose Float. In addition, he will facilitate the meetings with the sponsor and arrange sponsor-related trips. Morgan Montalvo, who is the Scheduling Officer, created the Gantt chart and will keep track of due dates from the Gantt chart. She will also coordinate the team meeting schedule and continually revise the Gantt chart in order to meet the goals of the team. As the Accounting Officer, Breanna Tran will collect a bill of materials and track all purchases made for the project to ensure the project is within budget. To make sure the Google Drive folder for our Senior Project stays organized, Sergio Gutierrez was assigned the title Organization Officer.

Some additional roles for each subsystem were necessary for the completion of this project. With these job assignments comes the responsibility of making sure each team member is doing his or her part for these subsystems. Ali Harake will be in charge of the 3D modeling of the overheight mechanism. The analysis, including stress and fatigue calculations, will be lead by Morgan Montalvo. Breanna Tran will be responsible for prototype fabrication and manufacturing for the project. Finally, Sergio Gutierrez will ensure each report meets all of the specified criteria and testing procedures are completed before testing is expected to occur. Moreover, beside these general project roles, we have assigned specific responsibilities for the manufacturing phase. Sergio Gutierrez will be in charge of completing the main frame subassembly. Morgan Montalvo will be in charge of the shaft subassembly and Ali Harake will lead the assembly of the truss attachment on the shaft. Lastly, Breanna Tran will be in charge of the truss assembly. We will also each be leading 3 tests out of all the testing done on our mechanism. The details of who is in charge of which test is detailed in our Design Verification Plan (Appendix U).

Action Plan/Timeline

At the start of the first quarter, we identified our problem by talking amongst ourselves and with our sponsor about past overheight mechanisms. From these conversations, we learned what types of mechanisms were reliable and common across various floats. With the knowledge of these past mechanisms, we also went through a round of researching and brainstorming ideas for our project. After speaking with our sponsor again, we refined our problem statement and created a requirements table. Also, we made a QFD to help us define our problem further by considering our customer's needs and engineering requirements to meet those needs. Once our problem was defined, we then created a Design Process Flowchart (Appendix V) and a Gantt chart (Appendix Q) to organize our project in the coming months. The flowchart allows us to layout a general plan of how we will proceed through the next quarters in terms of prototypes, designing, and testing. From this flowchart, we then developed a more detailed plan using the Gantt Chart. Using our Gantt chart we were then able to determine a timeline of events for the first quarter and future quarters. The first quarter was mainly all of the development and planning for our mechanism and project. For the previous quarter, we divided it into three main categories: research and brainstorming, concept evaluation, and preliminary design evaluation.

In the first month of last quarter, we completed the first and second round of researching lifting methods to gather a large collection of ideas we can use for this project. For the following four weeks, we went into the concept evaluation stage to decide which concepts will be under consideration for our final design. The general timeline for brainstorming and concept evaluations is summarized in the task table below from our Gantt chart.

Table 22. Task table for concept evaluation

7	**◢ Concept Evaluation**	**27 days**	**Tue 4/18/17**	**Sun 5/14/17**	
8	QFD	8 days	Tue 4/18/17	Tue 4/25/17	
9	Requirements Table	10 days	Tue 4/25/17	Thu 5/4/17	
10	Feasibility Study	1 day	Mon 5/8/17	Mon 5/8/17	6
11	◢ Analysis of concepts: Phase 1	5 days	Tue 5/9/17	Sat 5/13/17	9,10
12	Static analysis of structure	5 days	Tue 5/9/17	Sat 5/13/17	
13	Force required by actuator	5 days	Tue 5/9/17	Sat 5/13/17	
14	Specify possible actuators	5 days	Tue 5/9/17	Sat 5/13/17	
15	Go/No-Go	1 day	Sun 5/14/17	Sun 5/14/17	12

Initially, we performed a feasibility study on all of the various concepts we brainstormed to reduce our range of concepts to ones that worked for our given application. Many of these feasible concepts were similar designs with different actuators, so we grouped them into categories to perform preliminary analysis. The first phase of analysis was to calculate the force required by the actuator in a given design from the statics of a maximally loaded system. From this force required, we researched actuators available able to provide the force required given our limited hydraulic pressure provided. Although some concepts were feasible, from this analysis we learned that not all the concepts met the design requirements. Therefore, when we performed a Go/No-Go evaluation, the rotating mechanism using a hydraulic motor did not meet both lifting capacity and cost as well as the linkage system not meeting lifting and size restraints. After this evaluation, we had our top concepts to perform analysis on and decide between for another two weeks. Upon deciding our design matrix criteria and weighting, we then analyzed each of the top concepts. This second phase of analysis consisted of analyzing: stress in critical components, safety factors, and general sizing for structural beams and components. From all of this analysis, we were able to accurately gage the relative cost and volume of each design which were one of the distinguishing differences between the designs. After performing the design matrix, we chose our top concept to continue pursuing.

Concurrent with the concept evaluation stage of the first quarter, we were planning the rest of the quarters by creating preliminary analysis, manufacturing, and testing plans. From the analysis performed for the design matrix, we also decided what analysis is needed to develop a system to meet our design requirements and prove safe operation for any given design. We also developed a Design Verification Plan in order to detail how we will test and verify that our final design has met the requirements previously established. Additionally, the FMEA we initially created helped to understand the hazards of manufacturing, testing, and operating our mechanism. With these safety hazards understood, we then were able to design our mechanism with safety measures implemented to address these hazards. At the end of the first quarter, we presented our Primarily Design Report to the advisor and then our sponsor. From these presentations, we received crucial feedback on our chosen design.

As discussed previously, we were advised and then decided at the end of the first quarter to change our chosen design to the hinge with a cylinder design. Due to this design change, we updated our management plan for the subsequent quarters to reflect the design changes. Moreover, we decided to meet throughout the summer in order to develop the general design of the mechanism. From our work throughout the summer on the initial design allowed us to begin the second quarter primarily focused on analyzing our system to finalize sizing. The second quarter of this project is divided into several subcategories: analysis of chosen design, solid model of design, manufacturing & testing plans, and manufacturing. For the analysis, we created an expanded list from the analysis done the previous quarter. As shown in our Gantt chart and provided in the table below, the next phase of analysis is broken up into critical elements.

Table 23. Task table for analysis

51	**Analysis of Chosen Design**	**24 days**	**Thu 9/14/17**	**Sat 10/7/17**	
52	Static analysis of finalized structure	8 days	Thu 9/14/17	Thu 9/21/17	
53	Steel thickness and size calculations	6 days	Mon 10/2/17	Sat 10/7/17	52,55
54	**Shaft calculations**	**10 days**	**Fri 9/22/17**	**Sun 10/1/17**	
55	Shaft strength calculation	10 days	Fri 9/22/17	Sun 10/1/17	52
56	Shaft fatigue calculation	10 days	Fri 9/22/17	Sun 10/1/17	52
57	**Bearing calculations**	**11 days**	**Fri 9/22/17**	**Mon 10/2/1**	
58	Radial and Shear load calculations	7 days	Fri 9/22/17	Thu 9/28/17	52
59	Specify bearing requirements (bought or made)	4 days	Fri 9/29/17	Mon 10/2/17	58
60	Fastener/Weldment strength calculations	4 days	Tue 10/3/17	Fri 10/6/17	59

The critical elements of our design include the shaft, welded joints, and fastening joints. From the preliminary analysis done on our chosen concept, we know the structural beams would likely be 3x3 or 4x4 steel beams. Also, we know from the preliminary analysis performed in the first quarter that the critical point in the design is the shaft. Thus, the limiting safety factor for our design will be the safety factor in the shaft since this dictates the size of the rest of the system. The complete analysis of our system was strongly dependant on the efficiency we analyzed the shaft. In order to mitigate this potential schedule delay, performed most of our analysis concurrently and parametrically using engineering tools such as a spreadsheet calculator and matrix calculator. Thus, from our overall static analysis, we were able to efficiently adjust many design parameters in the various analyses.

In order to maintain an appropriate pace so we could achieve all of our needed tasks by Critical Design Review, we were creating the solid model of our design simultaneously with the analysis phase. By concurrently creating the solid model of our system as we were analyzing, we created a more streamlined and divided workflow. This divided the work more evenly among all our team members. In addition to modeling our design, we developed all assembly and part drawings to display each component with specified dimensions. The tasks necessary for this phase of our project are illustrated in the task table for modeling presented below.

Table 24. Task table for modeling

62	**SolidWorks Model: Phase 1**	**26 days**	**Thu 9/14/17**	**Mon 10/9/1**	
63	Structure	10 days	Thu 9/14/17	Sat 9/23/17	
64	Actuator and transmission components	6 days	Thu 9/14/17	Tue 9/19/17	
65	Main rotation shaft	7 days	Mon 10/2/1'	Sun 10/8/17	56
66	Bearings and fasteners	3 days	Sat 10/7/17	Mon 10/9/1'	59,60
67	Assembly	15 days	Thu 9/14/17	Thu 9/28/17	63SS
68	Top Level Assembly Drawing	4 days	Fri 9/29/17	Mon 10/2/1'	67
69	**Drawings**	**10 days**	**Fri 9/29/17**	**Sun 10/8/17**	
70	Cut sheet for structural steel	10 days	Fri 9/29/17	Sun 10/8/17	67
71	Match plates	10 days	Fri 9/29/17	Sun 10/8/17	67
72	Shaft layout	10 days	Fri 9/29/17	Sun 10/8/17	67
73	Specialty parts	10 days	Fri 9/29/17	Sun 10/8/17	67

As we were analyzing and developing our design, we updated our Preliminary Design Report to reflect the progress we made. Since our design changed from the previous report, the main sections we updated were the Design Verification Plan, Hazard Identification Checklist, FMEA, Manufacturing Plan, and the Testing Plan. Many of these shared similarities with the chain and cylinder design, but there were a few changes we implemented in terms of safety considerations and adjustments to the plans. Moreover, as we were modeling the final design we were developing our analysis section as well as the Bill of Materials and cost estimation for our project.

After presenting our final design to our peers, advisor, and sponsor, we will synthesize all the feedback into design improvements. We will then finalize the model for manufacturing using the feedback provided. After we finalize the solid model, we will finalize cut lists and part drawings. The redesigning and drafting time is approximately two weeks long. Concurrent with the completion of the solid model, we will be finalizing our manufacturing and testing plans. As the manufacturing plan is being completed, we anticipate utilizing the last 4 weeks of this quarter to begin purchasing parts and manufacturing the structural steel parts. The timeline of the last 4 weeks of second quarter are shown in the task table below.

Table 25. Task table for finalizing design and manufacturing

87	**SolidWorks Model: Phase 2**	**14 days**	**Tue 10/17/1**	**Mon 10/30/**	
88	Revisions from CDR	7 days	Tue 10/17/1	Mon 10/23/:	74
89	Finalize Model	7 days	Tue 10/24/1	Mon 10/30/:	88
90	Finalize Drawings	7 days	Tue 10/24/1	Mon 10/30/:	88
91	Manufacturing Plan	14 days	Tue 10/24/1	Mon 11/6/1'	88
92	Testing Plan	14 days	Tue 10/24/1	Mon 11/6/1'	88
93	**Manufacturing**	**32 days**	**Tue 10/31/1**	**Fri 12/1/17**	
94	Acquire materials	11 days	Tue 10/31/1	Fri 11/10/17	90
95	Make all DXF files for plasma cut parts	11 days	Tue 10/31/17	Fri 11/10/17	90
96	Cut all structural steel	32 days	Tue 10/31/1	Fri 12/1/17	90

In the last quarter, we are primarily assembling and testing our mechanism. The manufacturing process in this quarter will include drilling parts in the Rose Float Lab on campus and laser cutting plate in the BRAE shops. Since we have access to a large fabrication and construction space allocated for Rose Float use only, we will be meeting as a group throughout the first month at set times during the weekend to complete all part manufacturing in the allotted time. Furthermore, this is why we will be making our design such that it can be manufactured using the tools in the Rose Float Lab. Although we will most likely be building subassemblies as we are manufacturing, we have planned for a two week period dedicated to full system assembly. Most of the assembly period is dedicated to welding all of the components together, because set up time for welding our mechanism correctly will take a majority of the total welding time. Moreover, we only have a few subassemblies we are attaching to the frame of our mechanism. As we weld segments together, we will be able to combine subassemblies of our mechanism together into a top level assembly. The allotted time for top level assembly is concurrent with our construction and welding timeline, with an extended deadline shortly after we are done welding. The task table for the manufacturing and assembly period of our project is shown in Table 26.

Table 26. Task Table for manufacturing and assembly

98	**Manufacturing**	**20 days**	**Mon 1/8/18**	**Sat 1/27/18**	
99	Purchase all hardware	6 days	Mon 1/8/18	Sat 1/13/18	
100	Bring cylinder back from Pomona	1 day	Sat 1/13/18	Sat 1/13/18	
101	Fabricate all plates and suplementary pieces	13 days	Mon 1/8/18	Sat 1/20/18	94
102	Cut al structual truss pieces	20 days	Mon 1/8/18	Sat 1/27/18	
103	Drill all fastener holes into 4x4	7 days	Sun 1/21/18	Sat 1/27/18	101
104	**Assembly**	**35 days**	**Mon 1/8/18**	**Sun 2/11/18**	
105	Weld mechanism structure	33 days	Mon 1/8/18	Fri 2/9/18	96
106	Assemble mechansim	21 days	Mon 1/22/1	Sun 2/11/18	105SS-
107	Weld truss structure	13 days	Sun 1/28/18	Fri 2/9/18	102

Upon completing all manufacturing and assembly, we will then transition into our testing phase for the last quarter of this project. The testing is divided into two main components, acquiring all testing materials and performing testing. Since a main component of our testing is verifying the lifting capacity at the target load distance, we will need time to confirm we have the weights necessary reserved for our testing days. Our planned time for acquisition of testing materials and running testing is fifteen days. Most of this time is also for us attaching the weights to our truss and mounting the mechanism on the chassis for testing. A benefit to having the Rose Float Lab is we plan to perform all testing there since we have immediate access to equipment such as a hydraulic system, forklift, and measurement tools. Ideally, when we test, our system will meet all specified testing design requirements. However, realistically, we have anticipated a need to modify our system after testing. We have planned for eight days after completing testing for any needed modifications to be designed and manufactured. Moreover, we have left approximately a week buffer period of excess time for needed time extensions and unforeseen circumstances. For an overview of all the tasks and general timeline, please refer to Appendix Q.

Conclusion

The project met all critical requirements that we sought out to achieve. The design is compact and can be implemented into multiple float designs. It can successively carry a 1500 lb load within our system pressure constraints. Should be noted we are not expecting for the mechanism to experience a point load of 1500 lb at a 10 ft level arm. The max loading this mechanism would experience is 1500 lb distributed load so we are satisfied with our results.

There were some unforeseen issues that we came across. For starters we were not able to retrieve the hydraulic cylinder when we originally planned due to mudslides closing down highway 101. During this time, we focused on completing any task that did not require the cylinder. There was also the issue of the float chassis not being back in time for testing. This resulted in us having to makeshift a testing rig for our mechanism.

The main change we made to our design for improvement during the manufacturing phase was moving the location of the zerk fittings. Instead of incorporating zerk fittings to the bearing support assembly, it was instead decided to weld pipe to the cylinder lever arm and incorporate zerk fittings to this added pipe.

We hope that this mechanism becomes an asset to the Cal Poly Rose Float program. An overheight mechanism takes a lot of time and resources to manufacture and typically has to be redesigned and built from scratch with every new float.. With our design, we are aiming to have this mechanism be used for multiple years regardless of what designs the float may have. This will save time and resources that can be utilized for other creative aspects of the float.

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Appendices

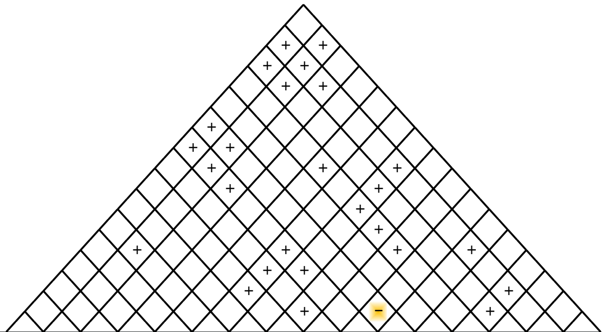
QFD: House of Quality

Project: HighRise
Revision:
Date: 4/27/2017

Correlations	
Positive	+
Negative	−
No Correlation	

Relationships	
Strong	●
Moderate	○
Weak	▽

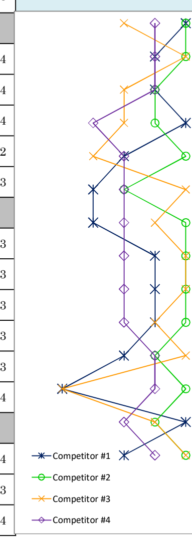
Direction of Improvement	
Maximize	▲
Target	◇
Minimize	▼



WHO: Customers

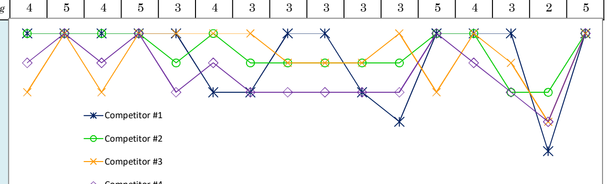
NOW: Current Product Assesment - Customer Requirements

Row #	Weight Chart	Relative Weight	Construction Team	Design Team		Maximum Relationship	WHAT: Customer Requirements (explicit & implicit) / HOW: Engineering Specifications	Lift Capacity	Height Range	Life	Indication Light	Time to Disassemble	Number of Placements	Modes of Attachment to mechanism	Steps to attach	Steps to detach	Number of Floats can be used on	Volume	Steel	Number of Parts	Fabrication Time	Cost	Safety Drop Time	Competitor #1: *Telescoping Cylinder*	Competitor #2: *Hinge*	Competitor #3: *Winch and Cables*	Competitor #4: *Rotating*	Row #
							Column #	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16					
							Direction of Improvement	▲	▲	▲	◇	▼	◇	▲	▼	▼	▲	▼	◇	▼	▼	▼	▼					
1							Functional Performance																					1
2		11%	14	9		9	Ability to lift heavy structures*	●	○	●			○	●	▽	▽	○	●		○	○	●	●	5	5	3	4	2
3		10%	8	13		9	Adjustable height range	○	●	▽			●				●	●		○	○	○	●	4	5	5	4	3
4		6%	7	5		9	Reusable	●	●	●			●	○			●	○				○		4	4	3	4	4
5		5%	3	8		9	Easy control system		○		●						○					○	○	5	4	3	2	5
6		7%	12	3		9	Ease of maintenance/fixability	○		●		●	○	○			●			○	▽	▽		3	5	2	3	6
7							Interface with Float																					7
8		10%	11	11		9	Modular placement	○	○	▽			●		○	○	●	●		▽	○	○		2	3	5	3	8
9		9%	9	10		9	Versatile attachment to mechanism	○						●			●			▽	○	▽		2	5	4	3	9
10		7%	10	4		9	Easy to attach to float						○		●	●	○	▽	▽					4	5	5	3	10
11		1%	2	1		9	Easy to detach from float						○		●	●	○	▽	▽					4	5	5	3	11
12		13%	13	14		9	Easy to implement in various float designs	●	●				●	●	○	○	●	●	○					4	5	4	3	12
13		9%	6	12		9	Compact	○	○			○	●	○			●	●		●	○	▽		3	4	5	4	13
14							Manufacturing Factors																					14
15		1%	1	2		9	Weldable					○							●					1	5	1	4	15
16		5%	5	6		9	Design simplicity			▽		●	▽	▽	●	●	●	○	▽	●	●	○		5	4	4	3	16
17		5%	4	7		9	Simple to duplicate					▽	▽	▽			▽	▽		●	●	●		3	5	5	4	17



	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
HOW MUCH: Target	1250 ±250 lbs	9 ±5 ft	100 ±15 hrs	Included	4 hours	4 Placements Min	2 Modes Min	5 Steps Max	5 Steps Max	5 Floats Min	64 ft^3 Max	Included	30 pieces Max	60 hrs Max	$1300 Max	60 s Max
Max Relationship	9	9	9	9	9	9	9	9	9	9	9	9	9	9	9	9
Technical Importance Rating	401.43	362.86	240	47.143	146.67	517.62	370.48	200.95	200.95	699.52	521.9	64.762	275.24	248.57	280.48	204.29
Relative Weight	8%	8%	5%	1%	3%	11%	8%	4%	4%	15%	11%	1%	6%	5%	6%	4%
Weight Chart																
Competitor #1: *Telescoping Cylinder*	5	5	5	5	5	3	3	5	5	3	2	5	5	5	1	5
Competitor #2: *Hinge*	5	5	5	5	4	5	4	4	4	4	4	5	5	3	3	5
Competitor #3: *Winch and Cables*	3	5	3	5	5	5	5	4	4	4	5	3	5	4	2	5
Competitor #4: *Rotating*	4	5	4	5	3	4	3	3	3	3	3	5	4	3	2	5

Current Product Assesment - Engineering Specifications

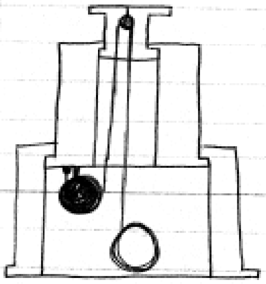


Column #	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16

Template Revision: 0.9 Date: 4/23/2010
Christopher Battles

Preliminary Analysis

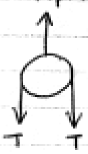
5/11/2017



Telescoping with Winch + Cable

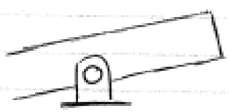
$\Sigma F_y: T = 1250 \text{ lbs}$

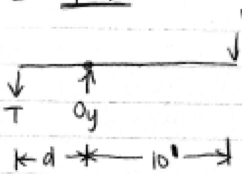
$T = W$



$\Sigma F_y: R_{pulley} = 2T$

Hinge with Winch + Cable





$\Sigma M_o = 0$

$-W(10') + T(d) = 0$

$T = \frac{W(10')}{d} = \frac{(1250 \text{ lbs})(10')}{d}$

$\Sigma F_y: O_y = T + W$

$\begin{cases} 1 \text{ ft} < d < 10 \text{ ft} \\ 1250 \text{ lbs} < T < 12500 \text{ lbs} \end{cases}$

20

Pugh Matrix - 1st Iteration

Concept Selection Legend	
Better	+
Same	S
Worse	-

Key Criteria	Importance Rating	Hinge with 1 Cylinder	Hinge with Winch + Cable	Hinge with Chain + 2 Cylinders	Rotational Mechanism with Hydraulic Motor	Rotational Mechanism with Cylinder
			Solution Alternatives			
Lift Capacity	9	N/A	-	S	-	S
Reusable on Multiple Float Designs	10	N/A	+	+	S	S
Cost	7	N/A	-	-	-	S
Lifespan	8	N/A	S	S	-	S
Volume	9	N/A	+	+	+	+
Can be built with material accesible to us	5	N/A	-	-	-	+
Height Range	7	N/A	+	+	-	-
Replicable	5	N/A	-	S	-	S
Simplicity	6	N/A	S	-	-	-
Can be taken Apart to repair or fix	4	N/A	+	-	+	S
Sum of Positives			4	3	2	2
Sum of Negatives			4	4	7	2
Sum of Sames			2	3	1	6
Weighted Sum of Positives			30	26	13	14
Weighted Sum of Negatives			26	22	47	13
TOTALS			4	4	-34	1

Pugh Matrix - 2nd Iteration

Concept Selection Legend

Better	+
Same	S
Worse	-

Key Criteria	*Importance Rating*	Hinge with Winch + Cable	Hinge with 1 Cylinder	Hinge with Chain + 2 Cylinders	Rotational Mechanism with Cylinder
		Solution Alternatives			
Lift Capacity	9	N/A	+	+	-
Reusable on Multiple Float Designs	10	N/A	-	-	-
Cost	7	N/A	+	+	+
Lifespan	8	N/A	S	+	S
Volume	9	N/A	-	-	S
Can be built with material accesible to us	5	N/A	+	+	+
Height Range	7	N/A	-	+	-
Replicable	5	N/A	+	S	+
Simplicity	6	N/A	S	-	-
Can be taken Apart to repair or fix	4	N/A	-	-	S
Sum of Positives			4	5	3
Sum of Negatives			4	4	4
Sum of Sames			2	1	3
Weighted Sum of Positives			26	36	17
Weighted Sum of Negatives			30	29	32
TOTALS			-4	7	-15

Pugh Matrix - 3rd Iteration					
		Solution Alternatives			
Concept Selection Legend Better + Same S Worse - **Key Criteria**	***Importance Rating***	Hinge with Chain + 2 Cylinders	Hinge with 1 Cylinder	Hinge with Winch + Cable	Rotational Mechanism with Cylinder
Lift Capacity	9	N/A	S	-	-
Reusable on Multiple Float Designs	10	N/A	-	+	S
Cost	7	N/A	+	-	+
Lifespan	8	N/A	S	-	-
Volume	9	N/A	-	+	S
Can be built with material accesible to us	5	N/A	+	-	+
Height Range	7	N/A	-	-	-
Replicable	5	N/A	S	S	+
Simplicity	6	N/A	+	+	-
Can be taken Apart to repair or fix	4	N/A	+	+	S
		Sum of Positives	4	4	3
		Sum of Negatives	3	5	4
		Sum of Sames	3	1	3
		Weighted Sum of Positives	22	29	17
		Weighted Sum of Negatives	26	36	30
		TOTALS	-4	-7	-13

Analysis: Hinge with Cylinder

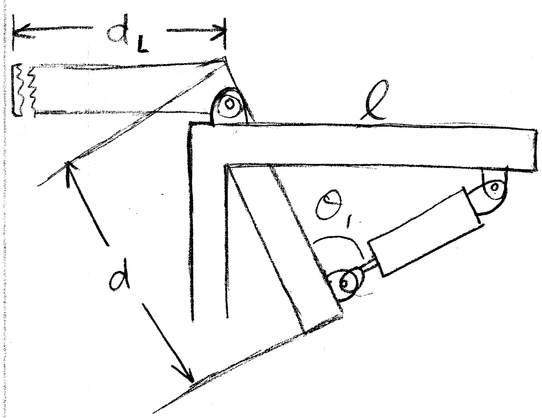
Defined Variables:

- Angle of travel: $\theta = 80°$
  Cylinder Force Angle, Down: θ_1
  Cylinder Force Angle, Up: θ_2
- Assumed point load: $L = 1500$ lbs
  Cylinder Lever Arm: d
- System Pressure: $P_{sys} = 1.1$ ksi
  Cylinder Force: F_{cyl}
  Bearing Reaction Force: R (kips)
- Cylinder Diameter: $D = 5$ in
  Cylinder Stroke: $S = 19$ in
- Load lever Arm: $d_L = 10$ ft

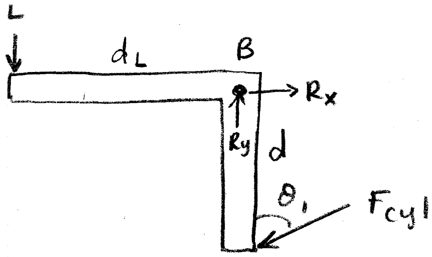
Material: 1018 CD Steel

→ $S_y = 54$ kpsi

Sketch:



Down Position:



$\overset{+}{\rightarrow} \Sigma F_x: -F_{cyl} \sin\theta_1 + R_x = 0$

$R_x = F_{cyl} \sin\theta_1$

$+\uparrow \Sigma F_y: -L + R_y - F_{cyl} \cos\theta_1 = 0$

$R_y = L + F_{cyl} \cos\theta_1$

$$+\circlearrowleft \Sigma M_B = 0$$

$$d_L L - F_{cyl} \sin\theta_1 \cdot d = 0$$

$$F_{cyl} = \frac{d_L L}{d \sin\theta_1}$$

$$= \frac{M_1}{d \sin\theta_1} \quad \text{where } \underline{M_1 = d_L L}$$

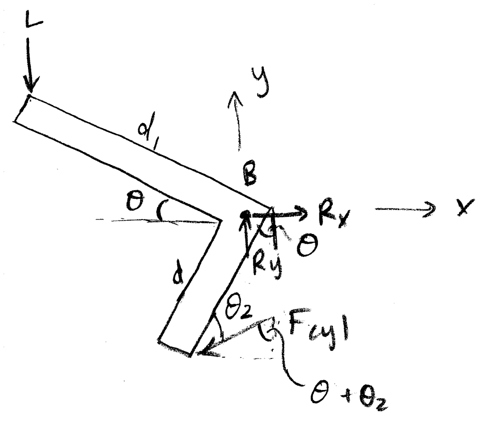
$$F_{cyl} = P_{sys} \cdot A_{cyl}$$

$$= P_{sys} \frac{\pi}{4} D^2$$

$$① \Rightarrow \frac{M_1}{d \sin\theta_1} = P_{sys} \frac{\pi}{4} D^2$$

unknowns: d, θ_1

Up Position



$$\xrightarrow{+} \Sigma F_x : R_x - F_{cyl} \sin(\theta + \theta_2) = 0$$

$$R_x = F_{cyl} \sin(\theta + \theta_2)$$

$$+\uparrow \Sigma F_y : R_y - L - F_{cyl} \cos(\theta + \theta_2) = 0$$

$$R_y = L + F_{cyl} \cos(\theta + \theta_2)$$

$$+\circlearrowleft \Sigma M_B : d_L \cos\theta\, L - d\, F_{cyl} \sin\theta_2 = 0$$

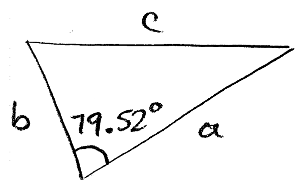
$$F_{cyl} = \frac{d_L \cos\theta \cdot L}{d \sin\theta_2}$$

$$F_{cyl} = P_{sys} \frac{\pi}{4} D^2$$

$$② \Rightarrow P_{sys} \frac{\pi}{4} D^2 = \frac{d_L \cos\theta\, L}{d \sin\theta_2}$$

Unknowns: θ_2

Solve for Retracted Cylinder Length:



$b = 19.05$ in

$c = 36.68$ in

$a \equiv$ retracted cylinder length

$$c^2 = a^2 + b^2 - 2ab\cos C$$

$$(36.68\text{ in})^2 = a^2 + (19.05\text{ in})^2 - 2a(19.05\text{ in})\cos 79.52^\circ$$

$$a^2 - 6.93a - 982.52 = 0$$

$$a = \frac{6.93 + \sqrt{(-6.93)^2 - 4(1)(-982.52)}}{2(1)}$$

$$a = 35.0\text{ in}$$

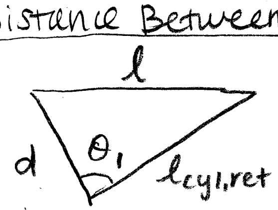
$$\boxed{l_{cyl,ret} = 35.0\text{ in}}$$

$$l_{cyl,ext} = l_{cyl,ret} + S$$

$$= 35.0\text{ in} + 19\text{ in}$$

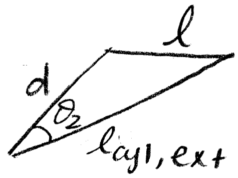
$$\boxed{l_{cyl,ext} = 54.0\text{ in}}$$

Distance Between bearing and cylinder Attachment



$$③ \Rightarrow l^2 = d^2 + l_{cyl,ret}^2 - 2(d)(l_{cyl,ret})\cos\theta_1$$

Unknown: l



④ ⇒ $l^2 = d^2 + l_{cyl,ext}^2 - 2(d)(l_{cyl,ext})\cos\theta_2$

Unknown: $l_{cyl,ext}$

* compare to calculated $l_{cyl,ext}$

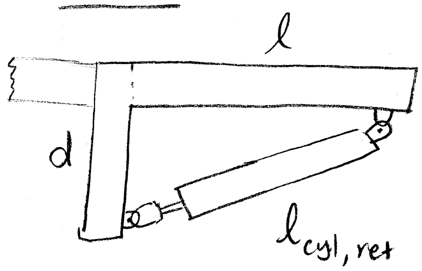
Height Range

$h = d_1 \sin\theta$

$= (10\text{ ft}) \sin 80^\circ$

$\boxed{h = 9.85\text{ ft}}$

Volume



some width, w

$V = Aw$

$= \frac{1}{2}(dl)w$

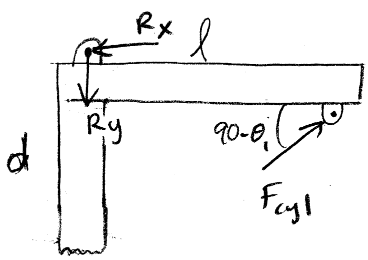
⑤ ⇒ $V = \frac{1}{2} dlw$

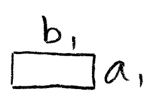
Unknowns: $\forall$, w

$w = 2b_1 + b_2$

Stress Analysis

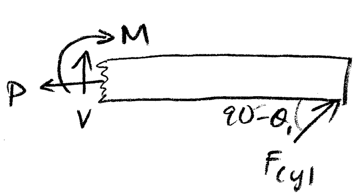
Bar 1: Down Position





$\sigma_{max} < \frac{S_y}{n}$

$\sigma' < \frac{S_y}{n}$



$\Sigma F_x: -P + F_{cyl}\cos(90-\theta_1) = 0$

$P = F_{cyl}\cos(90-\theta_1)$

$\Sigma F_y: V + F_{cyl}\sin(90-\theta_1) = 0$

$V = -F_{cyl}\sin(90-\theta_1)$

$$\Sigma M_{cut}: -M + \ell\, F_{cyl} \sin(90-\theta_1) = 0$$

$$M = \ell\, F_{cyl} \sin(90-\theta_1)$$

Unknowns: σ_1, $I_1 \rightarrow a_1$

$$⑥ \Longrightarrow \sigma_1 = \frac{Mc}{I} = \frac{\ell F_{cyl} \sin(90-\theta_1)(\frac{1}{2}a_1)}{I_1}$$

Unknowns: τ_1, b_1 (I, a)

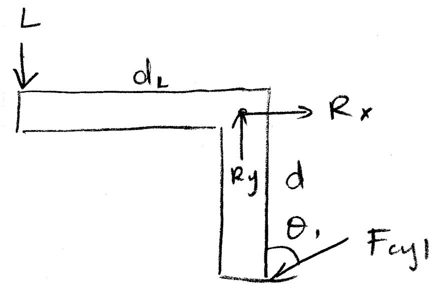
$$⑦ \longrightarrow \tau_1 = \frac{P}{A} = \frac{F_{cyl} \cos(90-\theta_1)}{a_1 b_1}$$

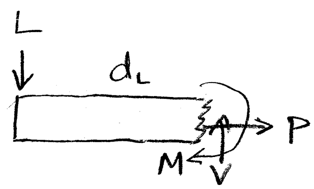
$$\sigma_1' = \frac{S_y}{n_1}$$

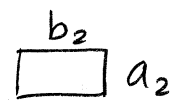
$$⑧ \Longrightarrow \sqrt{\sigma_1^2 + 3\tau_1^2} = \frac{S_y}{n_1}$$

Unknowns: n_1

Bar 2: Down Position







$$\Sigma F_y: -L + V = 0$$
$$V = L$$

$$\Sigma F_x: P = 0$$

$$\Sigma M_{cut}: -M + d_L L = 0$$
$$M = d_L L$$

Unknowns: σ_2, $a_2 \rightarrow I_2$

$$⑨ \Longrightarrow \sigma_2 = \frac{d_L L(\frac{1}{2}a_2)}{I_2}$$

$$\tau_2 = 0$$

$$\sigma_2' = \frac{S_y}{n_2}$$

unknown: n_2

(10) $\Rightarrow \sigma_2 = \dfrac{S_y}{n_2}$

EES:

Metals online

Bar 1: 4"x4" square pipe 1/4" thick 4ft ⇒ $141.78

Bar 2: 6"x6" square pipe 1/4" thick 12ft ⇒ $605.28

Metals Depot

Defined Variables from Design Requirements

$\theta = 80$ [deg] *Angle of travel*

$L = 1500$ [lb_f] *Point Load*

$P_{sys} = 1100$ [psi] *System Pressure*

$D_{cyl} = 5$ [in] *Cylinder Diameter*

$d_L = 10$ [ft] *Distance to Point Load*

$S_y = 54000$ [psi] *1018 CD Yield Strength*

$l_{cyl,ret} = 35$ [in] *Retracted Cylinder Length*

Defined for Testing

$\theta_1 = 85$ [deg]

$a_1 = 4$ [in] *Height of Beam 1*

$b_1 = 4$ [in] *Width of Beam 1*

$I_1 = 8.22$ [in^4] *Moment of inertia of Beam 1*

$a_2 = 6$ [in] *Height of Beam 2*

$b_2 = 6$ [in] *Width of Beam 2*

$I_2 = 30.3$ [in^4] *Moment of inertia of Beam 2*

Equation for the Force the Cylinder Exerts

$$F_{cyl} = P_{sys} \cdot \frac{\pi}{4} \cdot D_{cyl}^{\ 2}$$

Derived Equations

Equation 1

$$d_L \cdot \left| 12 \cdot \frac{in}{ft} \right| \cdot \frac{L}{d \cdot \mathbf{sin}\left[\theta_1\right]} = P_{sys} \cdot \frac{\pi}{4} \cdot D_{cyl}^{\ 2}$$

Equation 2

$$d_L \cdot \left| 12 \cdot \frac{in}{ft} \right| \cdot \mathbf{cos}\left[\theta\right] \cdot \frac{L}{d \cdot \mathbf{sin}\left[\theta_2\right]} = P_{sys} \cdot \frac{\pi}{4} \cdot D_{cyl}^{\ 2}$$

Equation 3

$$length^2 = d^2 + l_{cyl,ret}^{\ 2} - 2 \cdot d \cdot l_{cyl,ret} \cdot \mathbf{cos}\left[\theta_1\right]$$

Equation 4

$$length^2 = d^2 + l_{cyl,ext}^2 - 2 \cdot d \cdot l_{cyl,ext} \cdot \mathbf{cos}\left[\theta_2\right]$$

Equation 5: Volume

$$V = 1 / 2 \cdot d \cdot length \cdot w \cdot \left[\left| 0.083333333 \cdot \frac{ft}{in} \right| \right]^3$$

$$w = 2 \cdot b_1 + b_2$$

Beam Analysis

Beam 1

Equation 6

$$\sigma_1 = length \cdot F_{cyl} \cdot \mathbf{sin}\left[90 \ [deg] - \theta_1 \right] \cdot 1 / 2 \cdot \frac{a_1}{I_1}$$ *Normal stress of Beam 1*

Equation 7

$$\tau_1 = F_{cyl} \cdot \frac{\mathbf{cos}\left[90 \ [deg] - \theta_1 \right]}{a_1 \cdot b_1}$$ *Shear stress of Beam 1*

Equation 8

$$\sqrt{\sigma_1^2 + 3 \cdot \tau_1^2} = \frac{S_y}{n_1}$$ *Von Mises stress to calculate SF of Beam 1*

Beam 2

Equation 9

$$\sigma_2 = d_L \cdot \left| 12 \cdot \frac{in}{ft} \right| \cdot L \cdot 1 / 2 \cdot \frac{a_2}{I_2}$$ *Normal stress of Beam 2*

Shear Stress is zero for Beam 2

$$\tau_2 = \frac{0 \ [lb_f/in]}{b_2}$$ *Shear stress of Beam 2*

Equation 10

$$\sigma_2 = \frac{S_y}{n_2}$$ *Safety Factor of Beam 2*

SOLUTION

Unit Settings: Eng F psia mass deg

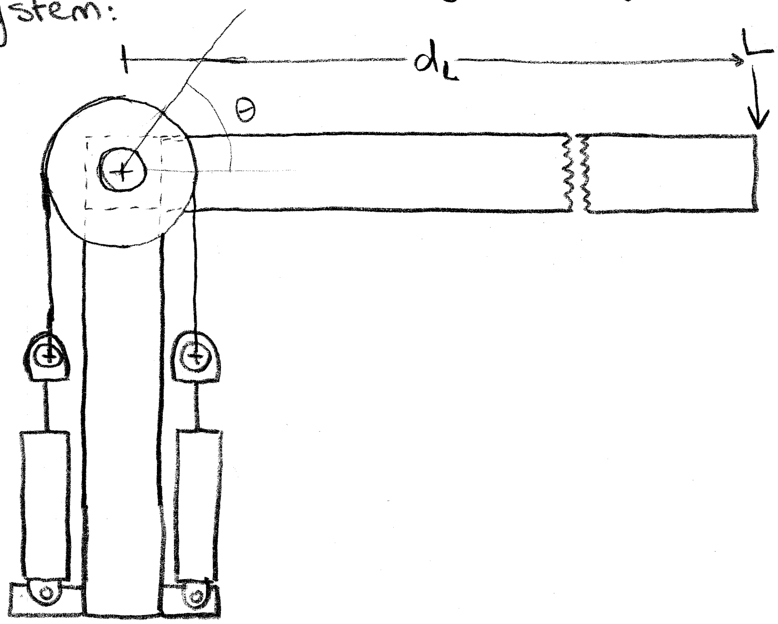
a_1 = 4 [in]	a_2 = 6 [in]
b_1 = 4 [in]	b_2 = 6 [in]
d = 8.366 [in]	D_{cyl} = 5 [in]
d_L = 10 [ft]	F_{cyl} = 21598 [lb_f]
I_1 = 8.22 [in^4]	I_2 = 30.3 [in^4]

$L = 1500$ [lb_f]	length = 35.27 [in]
$l_{cyl,ext} = 43.48$ [in]	$l_{cyl,ret} = 35$ [in]
$n_1 = 3.309$ [-]	$n_2 = 3.03$ [-]
$P_{sys} = 1100$ [psi]	$\sigma_1 = 16154$ [psi]
$\sigma_2 = 17822$ [psi]	$S_y = 54000$ [psi]
$\tau_1 = 1345$ [psi]	$\tau_2 = 0$ [psi]
$\theta = 80$ [deg]	$\theta_1 = 85$ [deg]
$\theta_2 = 9.962$ [deg]	$V = 1.195$ [ft^3]
$w = 14$ [in]	

No unit problems were detected.

Analysis - Hinge w/ Cylinder + Chain

System:



Givens:

$L = 1500\,lb$

$d_L = 10\,ft$

Angle of Travel

$\theta = 80°$

System Pressure

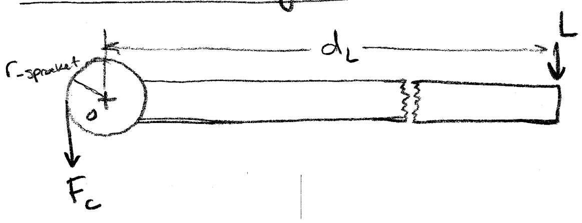
$P_{sys} = 1100\,psi$

Metal: 1018 CD steel

Cylinder bore max

$\varnothing_{cyl\text{-}bore} = 5''$

Cylinder Analysis



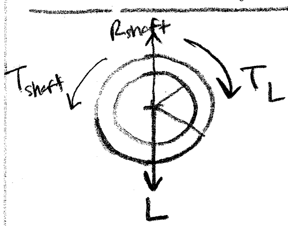
$\circlearrowright^{+} \Sigma M_o \quad -F_c r_{sprocket} + L d_L = 0$

$F_{c\,max} = P_{sys}\left(\frac{\pi}{4} 5^2\right) = 86.4\,kip$

$r_{sprocket} \geq \dfrac{L d_L}{F_{c\,max}} \Rightarrow r_{sprocket} \; 2.1''$

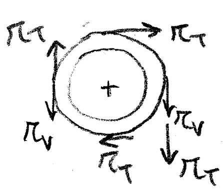
$$\boxed{F_c = \frac{L d_L}{r_{sprocket}}} \qquad \boxed{F_c = P_{sys}\left(\frac{\pi}{4}\right)\varnothing^2_{cyl\,bore}}$$

Main Shaft Analysis



$\circlearrowleft^{+} \Sigma T_o \quad -T_{shaft} + T_L = 0 \qquad T_{shaft} = T_L = L d_L$

$\Sigma F_y \quad R_{shaft} - L = 0 \qquad R_{shaft} = L$



$\tau = \tau_V + \tau_T$

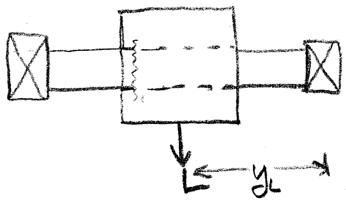
$\tau_V = \dfrac{VQ}{It} \qquad Q = \dfrac{4(d_o - d_i)A}{6\pi} \qquad I = \frac{\pi}{64}(d_o^4 - d_i^4)$

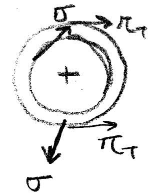
$\tau_T = \dfrac{T(d_o - t)\frac{1}{2}}{J} \qquad J = 2I$

→ Set d_o, t

$d_i = d_o - 2t$

Main Shaft Analysis Cont.





→ Get ~y_L from beam analysis

$$\sigma = \frac{(L y_L)(d_o - t)\frac{1}{2}}{I}$$

DE

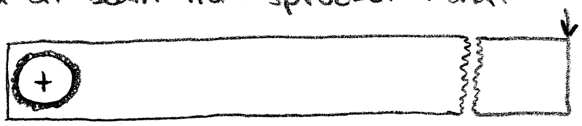
$$\sigma' = \sqrt{\sigma^2 + 3\tau_T^2}$$

$$\sigma_{max} = \max(\tau, \sigma')$$

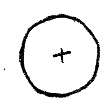
$$\boxed{\sigma_{max} \leq \frac{S_y}{n_{shaft}}}$$

Weldment Analysis

Weld on beam from sprocket + shaft



Weld Area



$T_{weld} = \frac{L d_L}{6}$

$h = .25$

$V = L$

$$\tau' = \frac{V_{weld}}{A_{weld}}$$

Shigley's Table 9-1

$A = 1.414 \pi h \frac{d_o}{2}$

$J_u = 2\pi (d_o/2)^3$

$$\tau'' = \frac{T(\frac{d_o}{2})}{J}$$

$J = .707 h J_u$

$$\tau_{weld} = \sqrt{\tau'^2 + \tau''^2}$$

$$\boxed{n_{weld} = \frac{S_y(.577)}{\tau_{weld}}}$$

Roller Chain Analysis

Set p, N $$n_{rpm} = \frac{\theta}{(60s)}\left(\frac{\pi\ rad}{180 deg}\right)\left(\frac{1 rev}{2\pi\ rad}\right)\left(\frac{60s}{1 min}\right)$$

$$H_1 = .004\, N_{teeth}^{1.08}\, n_{rpm}^{.9}\, p^{(3-.07p)}$$ pre extreme

$$H_2 = \frac{1000\, K_r\, N_{teeth}^{1.5}\, p^{.8}}{n_{rpm}^{1.5}}$$ post extreme $K_r = 17$ for chain above 40 pitch

$$H_{tab} = \min(H_1, H_2)$$

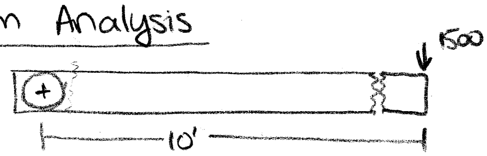
$$H_a = K_1 K_2 H_{tab}$$ $K_1 = \left(\frac{N_{teeth}}{17}\right)^{1.08}$ K_2 - # of strands

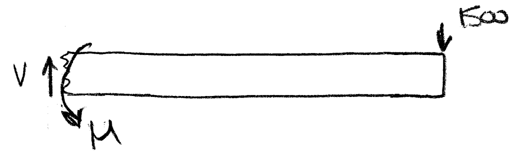
$$H_d = K_s H_{nom}$$ $K_s = 1.15$ light shock

$$H_{nom} = \frac{F_c (r_{sprocket})}{2}\, n_{rpm} \left(\frac{2\pi}{60}\right)\left(\frac{1hp}{550 ftlb}\right)$$

$$\boxed{n_{chain} = \frac{H_a}{H_d}}$$

Beam Analysis



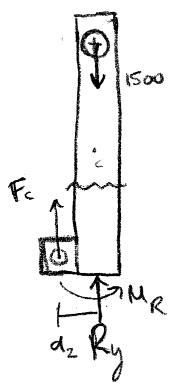


$V = 1500\text{ lb}$ $\quad$ $M = 15000\text{ ft lb}$

$\sigma_{max} = \frac{Mc}{I}$ $\quad$ Plug in I+c for different square steel tube

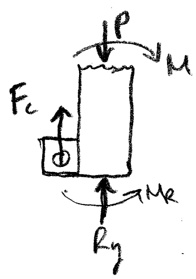
DE

$$\boxed{\sigma_{max} \leq \frac{S_y}{n}}$$



ΣF_y: $R_y + F_c - 1500 = 0$ $\quad$ $R_y = 1500 - F_c$

@ ΣM_c $\quad$ $F_c d_2 - M_R = 0$ $\quad$ $M_R = F_c d_2$



$P = R_y + F_c = 1500$

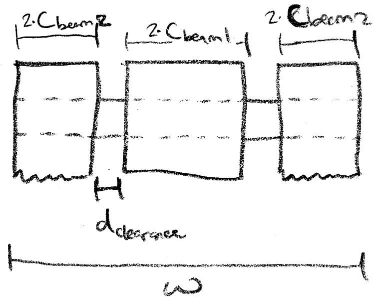
$M = M_R = F_c d_2$

$\sigma_{max} = \frac{Mc}{I} + \frac{P}{A}$ $\quad$ Plug in I+c+A for square steel tubes

DE

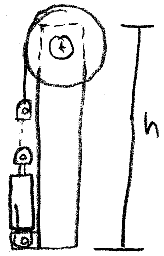
$$\boxed{\sigma_{max} \leq \frac{S_y}{n}}$$

Total Volume



$d_{clearance} \to 0$

$\omega = 2C_{beam1} + 4(C_{beam2})$



$S_{cylinder} = r_{sprocket}(\theta)\left(\frac{\pi}{180}\right)$

$h \approx 2 \cdot S_{cyl} + 8'' + r_{sprocket} + d_o$

$V \approx 2r_{sprocket} \cdot \omega h$

Defined Variables from Design Requirements

$\theta = 80$ [deg] *Angle of travel*

$L = 1500$ [lb$_f$] *Point Load*

$d_L = 10 \text{ [ft]} \cdot \left| 12 \cdot \frac{\text{in}}{\text{ft}} \right|$ *Distance to Point Load*

$P_{sys} = 1200$ [psi] *System Pressure*

$S_y = 54000$ [psi] *1018 CD Yield Strength*

phi_{bore} = 5 [in]

Diameter of Hydraulic Cylinder

Cylinder Analysis

$$F_{cyl} = L \cdot \frac{d_L}{r_{sprocket}}$$

$$F_{cyl} = P_{sys} \cdot \frac{\pi}{4} \cdot \phi_{bore}^{\ 2}$$

Main Shaft Analysis

$T_{shaft} = L \cdot d_L$ *Torque on Shaft*

$V_{shaft} = L$ *Shear Force on Shaft*

$d_o = 4$ [in] *Outer diameter of Shaft*

$t = 0.375$ [in] *Thickness of Pipe*

$d_i = d_o - 2 \cdot t$ *Inner diameter of Shaft*

$\tau_{shaft} = \tau_v + \tau_t$ *Shear stress of shaft*

$\tau_v = V_{shaft} \cdot \frac{Q}{I_{shaft} \cdot t}$ *Shear stress due to shear force*

$Q = 4 \cdot \left[\frac{d_o - d_i}{6 \cdot \pi} \right] \cdot \frac{\pi}{4} \cdot \left[d_o^{\ 2} - d_i^{\ 2} \right]$ *Area times y-bar*

$I_{shaft} = \frac{\pi}{64} \cdot \left[d_o^{\ 4} - d_i^{\ 4} \right]$ *Moment of inertia of hollow shaft*

$\tau_t = T_{shaft} \cdot \left[d_o - t \right] \cdot \frac{0.5}{J_{shaft}}$ *Shear stress due to torque*

$J_{shaft} = 2 \cdot I_{shaft}$ *Polar moment of inertia*

$\sigma_{shaft} = L \cdot y_L \cdot \left[d_o - t \right] \cdot \frac{0.5}{I_{shaft}}$ *Normal stress of shaft*

$von_{mises} = \sqrt{\sigma_{shaft}^2 + \tau_t^2}$ *Von mises stress of shaft*

$\sigma_{shaftmax} = \mathbf{Max}\left[von_{mises}, \tau_{shaft}\right]$ *Maximum normal stress*

$SF_{shaft} = \dfrac{S_y}{\sigma_{shaftmax}}$ *Safety Factor for shaft*

Weldment Analysis

$T_{weld} = L \cdot \dfrac{d_L}{6}$ *Torque on weld*

$h_{weld} = 0.25$ [in] *Weld height*

$V_{weld} = L$ *Shear force on weld*

$\tau' = \dfrac{V_{weld}}{A_{weld}}$ *Shear stress components*

$\tau'' = T_{weld} \cdot d_o \cdot \dfrac{0.5}{J_{weld}}$

$A_{weld} = 1.414 \cdot \pi \cdot h_{weld} \cdot d_o \cdot 0.5$ *Area of weld*

$J_u = 2 \cdot \pi \cdot \left[\dfrac{d_o}{2}\right]^3$

$J_{weld} = 0.707 \cdot h_{weld} \cdot J_u$

$\tau_{weld} = \sqrt{\tau'^2 + \tau''^2}$ *Shear stress on weld*

$SF_{weld} = S_y \cdot \dfrac{0.577}{\tau_{weld}}$ *Safety factor for weld*

Roller Chain Analysis

$p = 1.75$ [in] *Pitch [in]*

$chain_{num} = p \cdot 80$ [1/in] *Pitch #*

$N_{teeth} = 36$
$r_{sprocket} = .5*20.080$ [in]

$N_{teeth} = 45$

$r_{sprocket} = 0.5 \cdot 25.088$ [in]

$n_{rpm} = \dfrac{\theta}{60\ [s]} \cdot 1 / 6 \cdot 1$ [rev-s/deg-min] *Speed of chain [rpm]*

$K_r = 17$ *17 for chain above 40 pitch*

$H_1 = 0.004 \cdot N_{teeth}^{1.08} \cdot n_{rpm}^{0.9} \cdot p^{\left[3 - 0.07\ [1/in] \cdot p\right]}$ *Equation from Shigley's; units produce hp*

$$H_2 = 1000 \cdot K_r \cdot N_{teeth}^{\ 1.5} \cdot \frac{p^{\ 0.8}}{n_{rpm}^{\ 1.5}}$$ *Equation from Shigley's; units produce hp*

$$H_{tab} = \textbf{Min}\left[H_1, H_2\right]$$ *Equation from Shigley's; units produce hp*

$$H_a = K_1 \cdot K_2 \cdot H_{tab}$$

$$K_1 = \left[\frac{N_{teeth}}{17}\right]^{1.08}$$

$$K_2 = 1$$

$$H_d = K_s \cdot H_{nom}$$

$$K_s = 1.15$$ *For light shock*

$$H_{nom} = F_{cyl} \cdot r_{sprocket} \cdot \frac{\left|0.083333333 \cdot \frac{ft}{in}\right|}{2} \cdot n_{rpm} \cdot 2 \cdot \frac{\pi}{60} \cdot \frac{1 \ \ [hp]}{550 \ \ [ft\text{-}lb_f]}$$

$$SF_{chain} = \frac{H_a}{H_d}$$

Beam Analysis

Beam 1

$$V_{beam1} = L$$ *Shear force on beam 1*

$$M_{beam1} = L \cdot d_L$$ *Moment of beam 1*

$$I_{beam1} = 30.3 \ \ [in4]$$ *Moment of inertia for beam 1*

$$c_{beam1} = 3 \ \ [in]$$

$$y_L = c_{beam1} + c_{beam2} + 1 \ \ [in]$$

$$\sigma_{beam1} = M_{beam1} \cdot \frac{c_{beam1}}{I_{beam1}}$$ *Normal stress for Beam 1*

$$SF_{beam1} = \frac{S_y}{\sigma_{beam1}}$$ *Safety factor for Beam 1*

Beam 2

$$P_{beam2} = L$$ *Shear force on beam 2*

$$M_{beam2} = F_{cyl} \cdot d_2$$ *Moment of beam 2*

$$d_2 = c_{beam2} + 3 \ \ [in]$$

$$I_{beam2} = 8.22 \ \ [in4]$$ *Moment of inertia for beam 2*

$$c_{beam2} = 2 \ \ [in]$$

$A_{beam2} = 3.59$ [in2] *Area of beam 2*

$\sigma_{beam2} = M_{beam2} \cdot \dfrac{c_{beam2}}{I_{beam2}} + \dfrac{P_{beam2}}{A_{beam2}}$ *Normal stress of Beam 2*

$SF_{beam2} = \dfrac{S_y}{\sigma_{beam2}}$ *Safety factor for Beam 2*

Volume Analysis

$w = 2 \cdot c_{beam1} + 4 \cdot c_{beam2}$ *Total width of mechanism*

$s_{cyl} = r_{sprocket} \cdot \theta \cdot \dfrac{\pi}{180 \ [deg]}$ *Stroke of the cylinder*

$h = 2 \cdot s_{cyl} + 8 \ [in] + r_{sprocket} + d_o$ *Total height of the mechanism*

$V = 2 \cdot r_{sprocket} \cdot w \cdot h \cdot \left| 0.000578704 \cdot \dfrac{ft3}{in3} \right|$ *Total volume of the mechanism*

SOLUTION

Unit Settings: SI C kPa kJ mass deg

$A_{beam2} = 3.59$ [in^2]	$A_{weld} = 2.221$ [in^2]
$chain_{num} = 140$ [-]	$c_{beam1} = 3$ [in]
$c_{beam2} = 2$ [in]	$d_2 = 5$ [in]
$d_i = 3.25$ [in]	$d_L = 120$ [in]
$d_o = 4$ [in]	$F_{cyl} = 14349$ [lb$_f$]
$h = 59.57$ [in]	$H_1 = 0.3155$ [hp]
$H_2 = 7.665E+07$ [hp]	$H_a = 0.9027$ [hp]
$H_d = 0.3649$ [hp]	$H_{nom} = 0.3173$ [hp]
$H_{tab} = 0.3155$ [hp]	$h_{weld} = 0.25$ [in]
$I_{beam1} = 30.3$ [in^4]	$I_{beam2} = 8.22$ [in^4]
$I_{shaft} = 7.09$ [in^4]	$J_{shaft} = 14.18$ [in^4]
$J_u = 50.27$ [in^3]	$J_{weld} = 8.884$ [in^4]
$K_1 = 2.861$ [-]	$K_2 = 1$ [-]
$K_r = 17$ [-]	$K_s = 1.15$ [-]
$L = 1500$ [lb$_f$]	$M_{beam1} = 180000$ [lb$_f$-in]
$M_{beam2} = 71747$ [lb$_f$-in]	$n_{rpm} = 0.2222$ [rev/min]
$N_{teeth} = 45$ [-]	$p = 1.75$ [in]
$\phi_{bore} = 3.902$ [in]	$P_{beam2} = 1500$ [lb$_f$]
$P_{sys} = 1200$ [psi]	$Q = 0.6797$ [in^3]
$r_{sprocket} = 12.54$ [in]	$SF_{beam1} = 3.03$ [-]
$SF_{beam2} = 3.021$ [-]	$SF_{chain} = 2.474$ [-]
$SF_{shaft} = 2.309$ [-]	$SF_{weld} = 4.591$ [-]
$\sigma_{beam1} = 17822$ [psi]	$\sigma_{beam2} = 17875$ [psi]
$\sigma_{shaft} = 2301$ [psi]	$\sigma_{shaftmax} = 23392$ [psi]
$s_{cyl} = 17.51$ [in]	$S_y = 54000$ [psi]
$t = 0.375$ [in]	$\tau'' = 6753$ [psi]
$\tau' = 675.3$ [psi]	$\tau_{shaft} = 23392$ [psi]
$\tau_t = 23008$ [psi]	$\tau_v = 383.5$ [psi]
$\tau_{weld} = 6787$ [psi]	$\theta = 80$ [deg]
$T_{shaft} = 180000$ [lb$_f$-in]	$T_{weld} = 30000$ [lb$_f$-in]
$V = 12.11$ [ft^3]	$von_{mises} = 23123$ [psi]
$V_{beam1} = 1500$ [lb$_f$]	$V_{shaft} = 1500$ [lb$_f$]

V_{weld} = 1500 [lb_f]
w = 14 [in]
y_L = 6 [in]

3 potential unit problems were detected.

McMASTER-CARR®

High-Strength 1045 Carbon Steel

Rod, 2" Diameter

8924K58

Material	1045 Carbon Steel
Cross Section Shape	Round
Construction	Solid
Appearance	Plain
Diameter	2"
Diameter Tolerance	-0.006" to 0"
Tolerance Rating	Undersized
Yield Strength	77,000 psi
Fabrication	Cold Worked
Temper Rating	Hardened
Hardness	Rockwell B90
Hardness Rating	Medium
Heat Treatable	Yes
Maximum Hardness after Heat Treatment	Not Rated
Temperature Range	Not Rated
Specifications Met	ASTM A108
Straightness Tolerance	1/8"
Density	0.28 lbs./cu. in.
Surface Resistivity	16.2 microhm-cm @ 32° F
Thermal Conductivity	350 Btu/hr. × in./sq. ft./°F @ 212° F
Coefficient of Thermal Expansion	7.4×10^{6}
Elongation	19%
Material Composition	
Iron	98.21-98.85%
Carbon	0.43-0.50%
Manganese	0.60-0.90%

Phosphorus	0-0.04%
Silicon	0.15-0.30%
Sulfur	0-0.05%
Length	1 ft., 3 ft., 6 ft.
RoHS	Compliant

Stronger than low-carbon steel with equally good machinability, 1045 carbon steel is commonly used for bolts, studs, and shafts.

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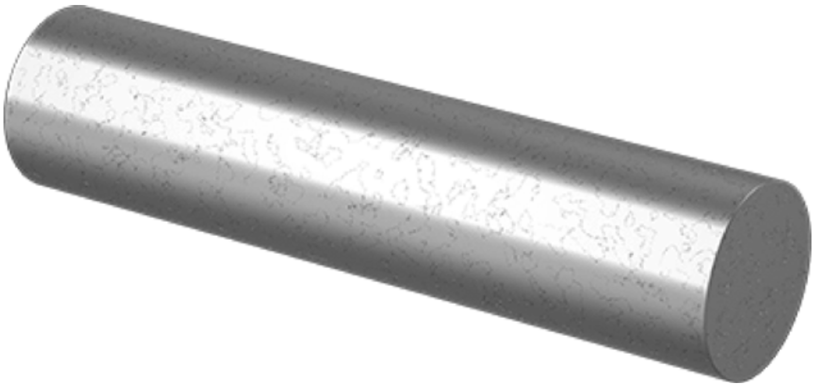
MCMASTER-CARR®

High-Strength 1045 Carbon Steel

Rod, 2-3/4" Diameter

8924K74



Material	1045 Carbon Steel
Cross Section Shape	Round
Construction	Solid
Appearance	Plain
Diameter	2 3/4"
Diameter Tolerance	-0.006" to 0"
Tolerance Rating	Undersized
Yield Strength	77,000 psi
Fabrication	Cold Worked
Temper Rating	Hardened
Hardness	Rockwell B90
Hardness Rating	Medium
Heat Treatable	Yes
Maximum Hardness after Heat Treatment	Not Rated
Temperature Range	Not Rated
Specifications Met	ASTM A108
Straightness Tolerance	1/8"
Density	0.28 lbs./cu. in.
Surface Resistivity	16.2 microhm-cm @ 32° F
Thermal Conductivity	350 Btu/hr. × in./sq. ft./°F @ 212° F
Coefficient of Thermal Expansion	7.4 x 10^6
Elongation	19%
Material Composition	
Iron	98.21-98.85%
Carbon	0.43-0.50%
Manganese	0.60-0.90%

Phosphorus	0-0.04%
Silicon	0.15-0.30%
Sulfur	0-0.05%
Length	1 ft., 3 ft., 6 ft.
RoHS	Compliant

Stronger than low-carbon steel with equally good machinability, 1045 carbon steel is commonly used for bolts, studs, and shafts.

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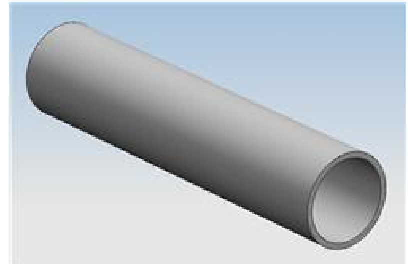
3-1/2" OD {A} x 2.750" ID {B} x .375" Wall {C} DOM Steel Tube

E-mail this product to a friend

Click here for material description, specification sheets and typical uses.



View larger image

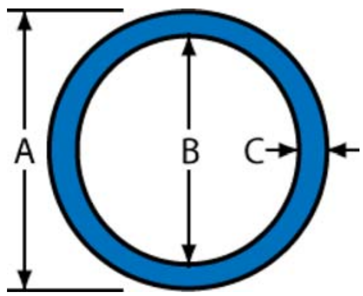
Click here for important information about telescoping.

DOM Steel Tube

Dimensions:
A: 3-1/2" OD
B: 2.750" ID
C: .375" Wall

Material: Steel

Shape: Round Tube



Length	Cutting Tolerance	Weight	Price	Order
By the Inch	Plus 1/8", Minus 0	1.0433 lbs	**Base Price: $4.26**	Enter Custom Lengths Here Quantity: 1 Length (in): 1 - 0" Max Length for UPS shipping 105". Call 866-938-6061 for lengths up to 240". Add to Cart Add to Wish List
12"	Plus or Minus 1/2"	12.5196 lbs	**Price: $43.56**	Quantity: 1 Add to Cart Add to Wish List
18"	Plus or Minus 1/2"	18.7794 lbs	**Price: $64.72**	Quantity: 1 Add to Cart Add to Wish List
24"	Plus or Minus 3/4"	25.0392 lbs	**Price: $85.07**	Quantity: 1 Add to Cart Add to Wish List
36"	Plus or Minus 3/4"	37.5588 lbs	**Price: $126.38**	Quantity: 1 Add to Cart Add to Wish List
48"	Plus or Minus 1"	50.0784 lbs	**Price: $166.87**	Quantity: 1 Add to Cart Add to Wish List
60"	Plus or Minus 1"	62.5980 lbs	**Price: $206.54**	Quantity: 1 Add to Cart Add to Wish List

McMASTER-CARR®

Pivoting Weld-on Tie-Down Ring

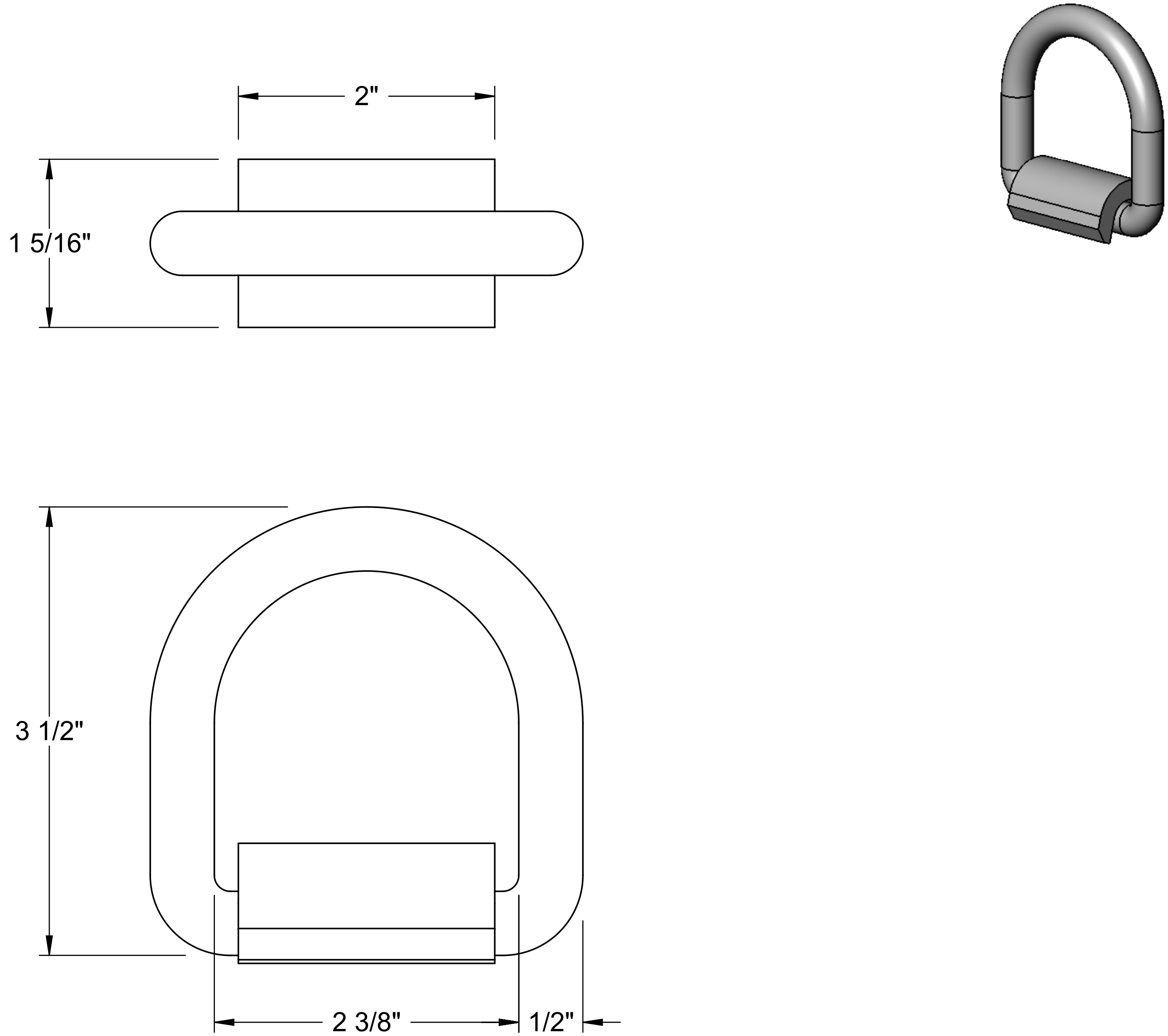
Uncoated Steel, 4080 lbs. Capacity

In stock
$9.97 Each
3028T32

Mount Type	Weld On
Load Capacity	4,080 lbs.
Ring	
Number of	1
Inner Width	2 3/8"
Height	3 1/2"
Thickness	1/2"
Style	D-Ring
Material	Uncoated Steel
Base	
Length	2"
Width	1 5/16"
Thickness	11/16"
Material	Uncoated Steel
Movement	180° Pivot

Also known as lashing rings, these rings weld to the mounting surface for a more secure connection than other tie-down rings. The ring pivots 180° to secure loads in various directions.

Warning: Never use for lifting applications.



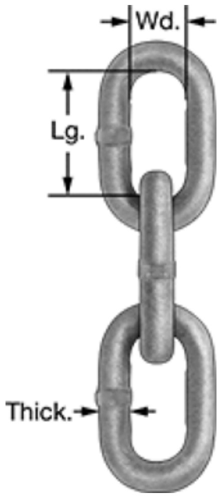
MCMASTER-CARR® CAD	PART NUMBER	3028T32
http://www.mcmaster.com © 2015 McMaster-Carr Supply Company Information in this drawing is provided for reference only.	Weld-On Tie-Down Ring	

McMASTER-CARR®

Grade 70 Chain-Not for Lifting

Extra-High-Strength, 3/8 Trade Size

In stock
$9.14 Each
3363T93



Chain Style	Straight Link
Application	Not for Lifting
Grade	70
Material	Zinc-Plated Steel
Trade Size	3/8
Metric Trade Size	10
Thickness	0.38"
Inside	
Width	0.60"
Length	1.30"
Capacity	6,600 lbs.
Weight per ft.	1.4 lbs.
Approximate Number of Links per Foot	9
For Fitting Thickness	9/16"
Maximum Temperature	400° F
For Use Outdoors	No
Specifications Met	ASTM A413, D.O.T. 49 CFR. 393.102, NACM (National Association of Chain Manufacturers)
Length	2 ft.
RoHS	Not Compliant

This chain is approximately 20% stronger than Grade 40/43 chain and 60% stronger than Grade 30 chain. Use it for binding loads, tie downs, and towing applications. It is stamped with the grade and manufacturer. When using fittings with chain, you must match the trade size and meet or exceed the chain's grade.

McMASTER-CARR®

External Retaining Ring

Black-Phosphate Steel, for 2" OD

In stock
$6.23 per pack of 10
97633A420

Ring Type	Retaining
For Shaft End Type	With Retaining Ring Grooves
Retaining Ring Style	Standard
Retaining Ring Type	External
Material	Black-Phosphate Steel
For OD	2"
For Groove	
Diameter	1.886"
Width	0.068"
Ring	
ID	1.85"
Thickness	0.062"
Min. Hardness	Rockwell C40
Specifications Met	ASME B18.27.1
Magnetic Properties	Magnetic
RoHS	Compliant

Pull rings open to pass over the end of a shaft and release to spring into the groove. Ring ID is measured with the ring uninstalled. Use retaining ring pliers (sold separately) to install and remove rings around the shaft.

Steel rings are an economical choice with good strength. Black-phosphate finish creates a dark appearance and provides mild rust resistance.

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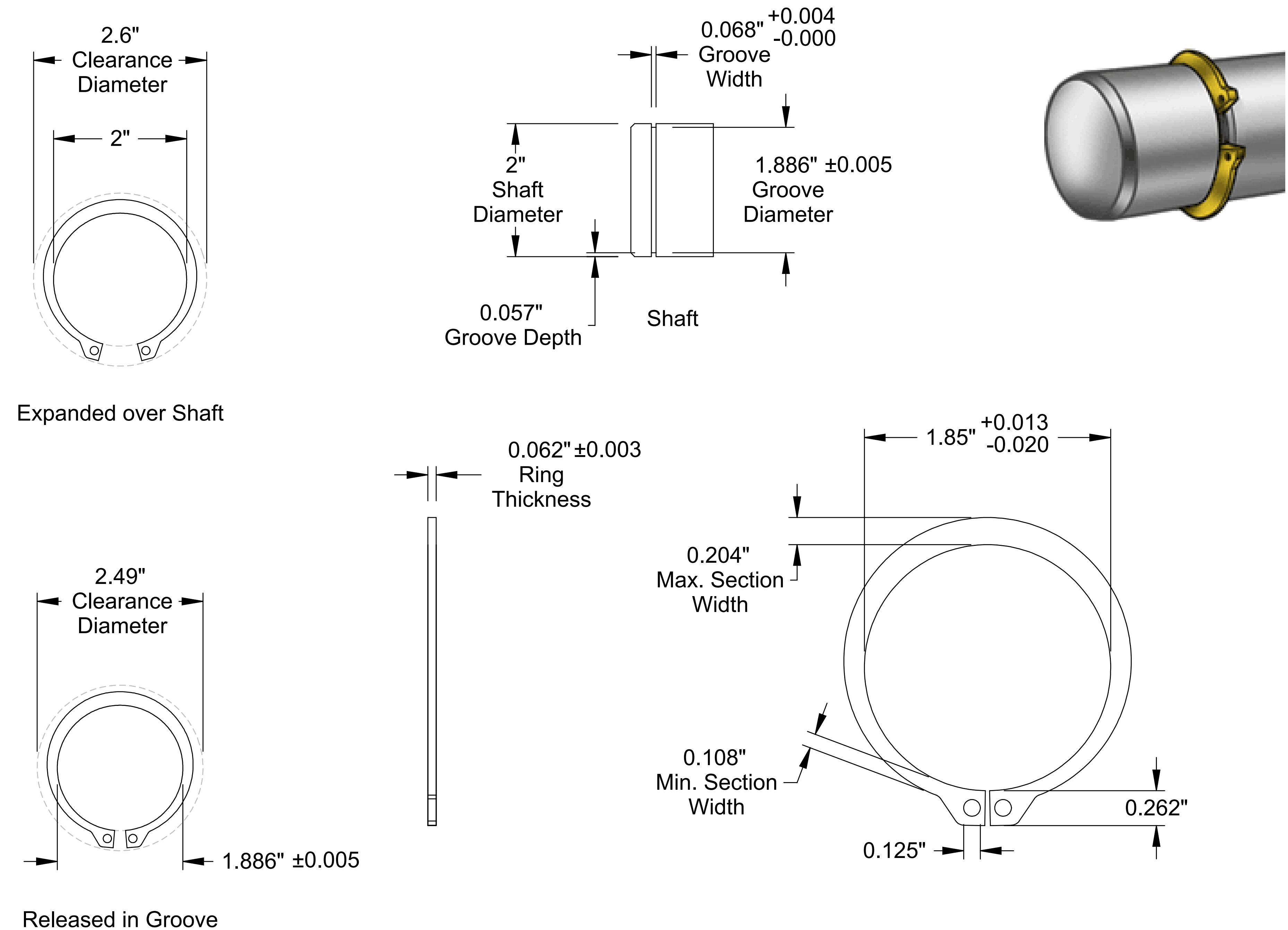


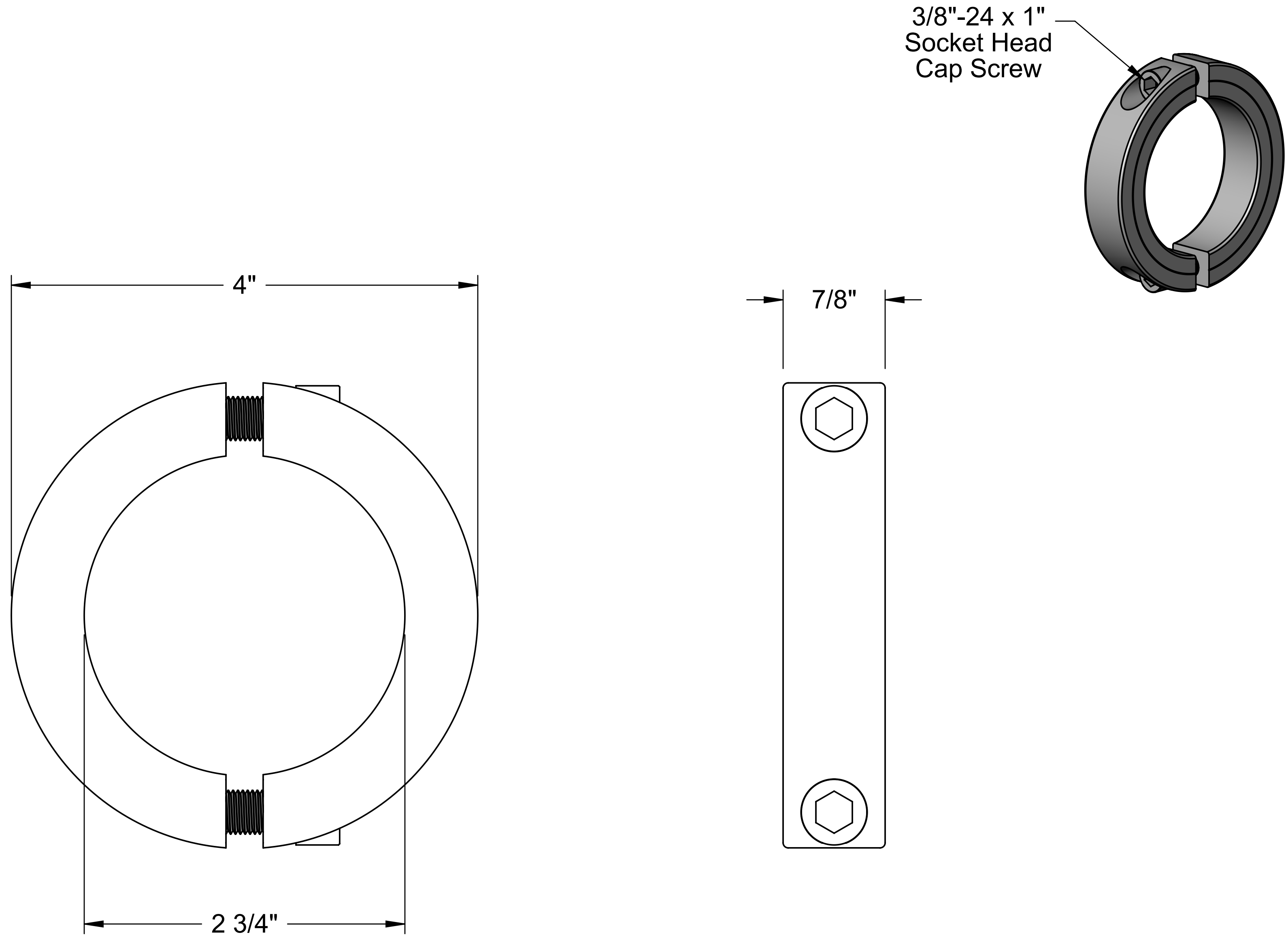
Note: Clearance diameter is the diameter of a housing that can pass freely over the ring.

McMASTER-CARR® CAD

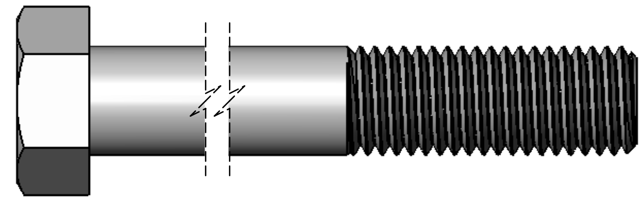
PART NUMBER 97633A420

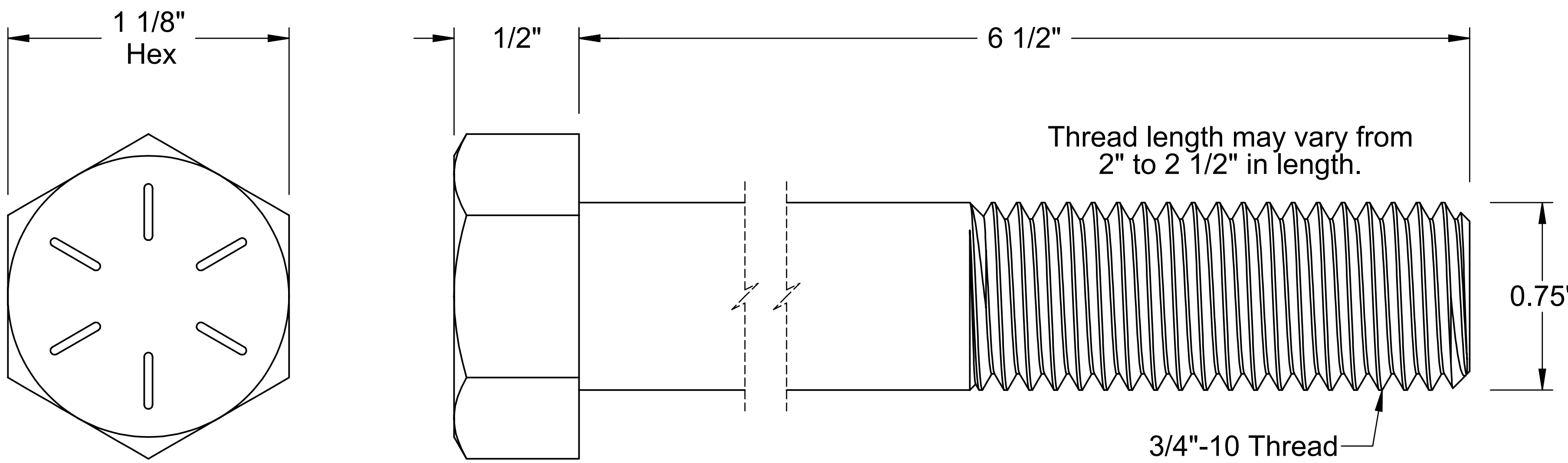
http://www.mcmaster.com

Black-Finish Steel
External Retaining Ring



McMASTER-CARR® CAD	PART NUMBER	6436K85
http://www.mcmaster.com © 2013 McMaster-Carr Supply Company Information in this drawing is provided for reference only.	Two-Piece Clamp-On Shaft Collar	





McMASTER-CARR® CAD	PART NUMBER 91257A860
http://www.mcmaster.com © 2014 McMaster-Carr Supply Company	High-Strength Steel Cap Screw-Grade 8
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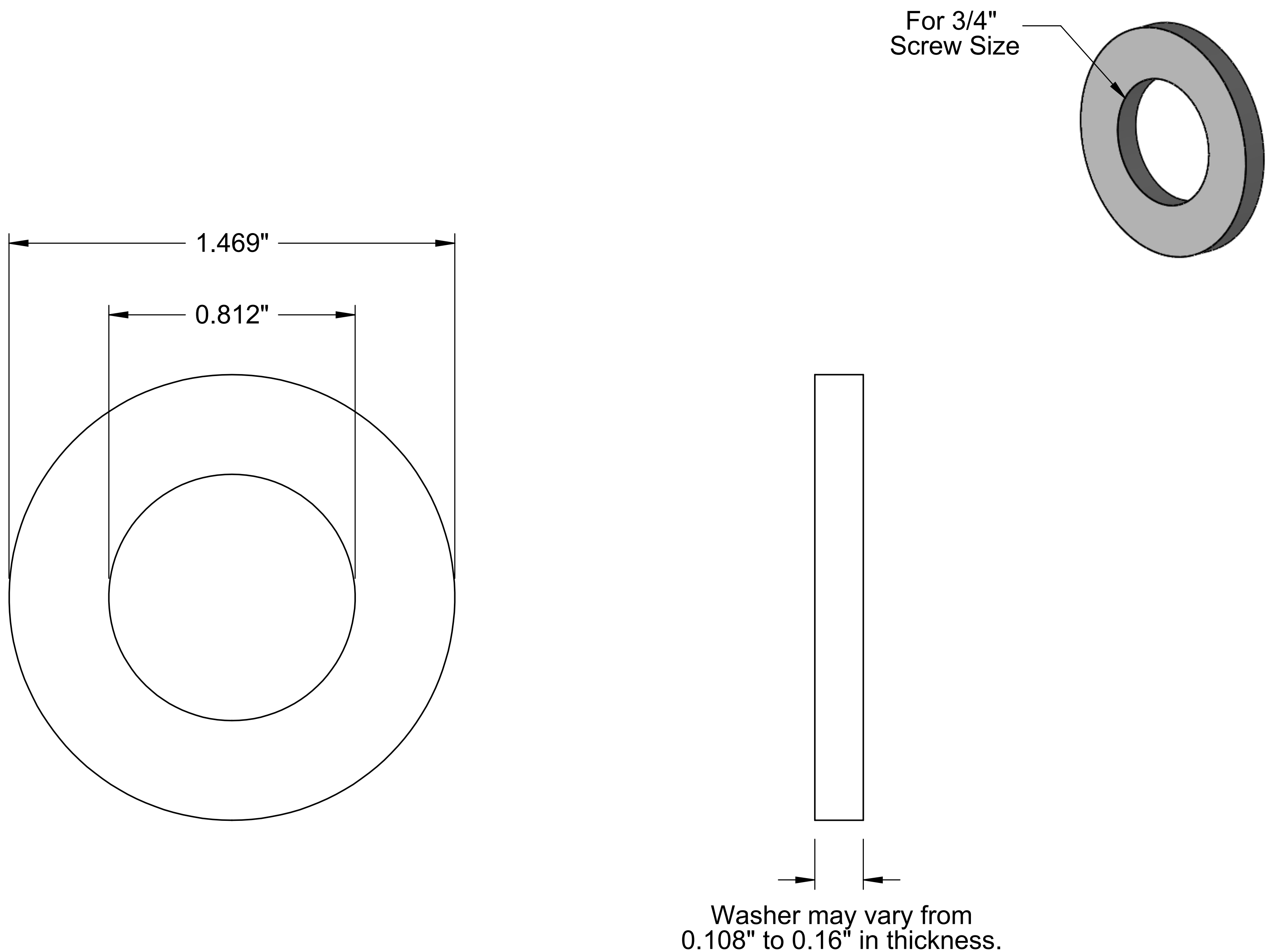
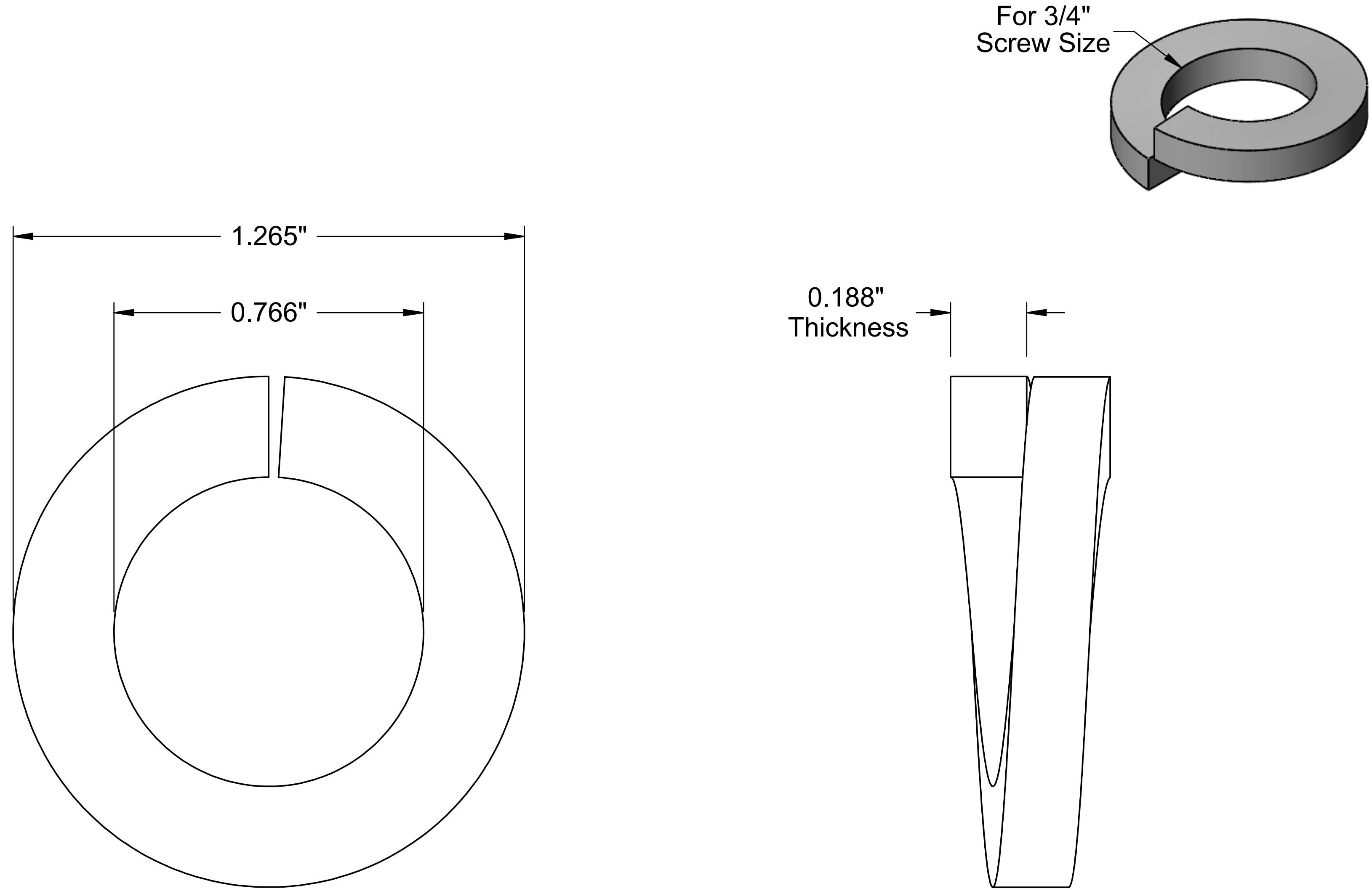
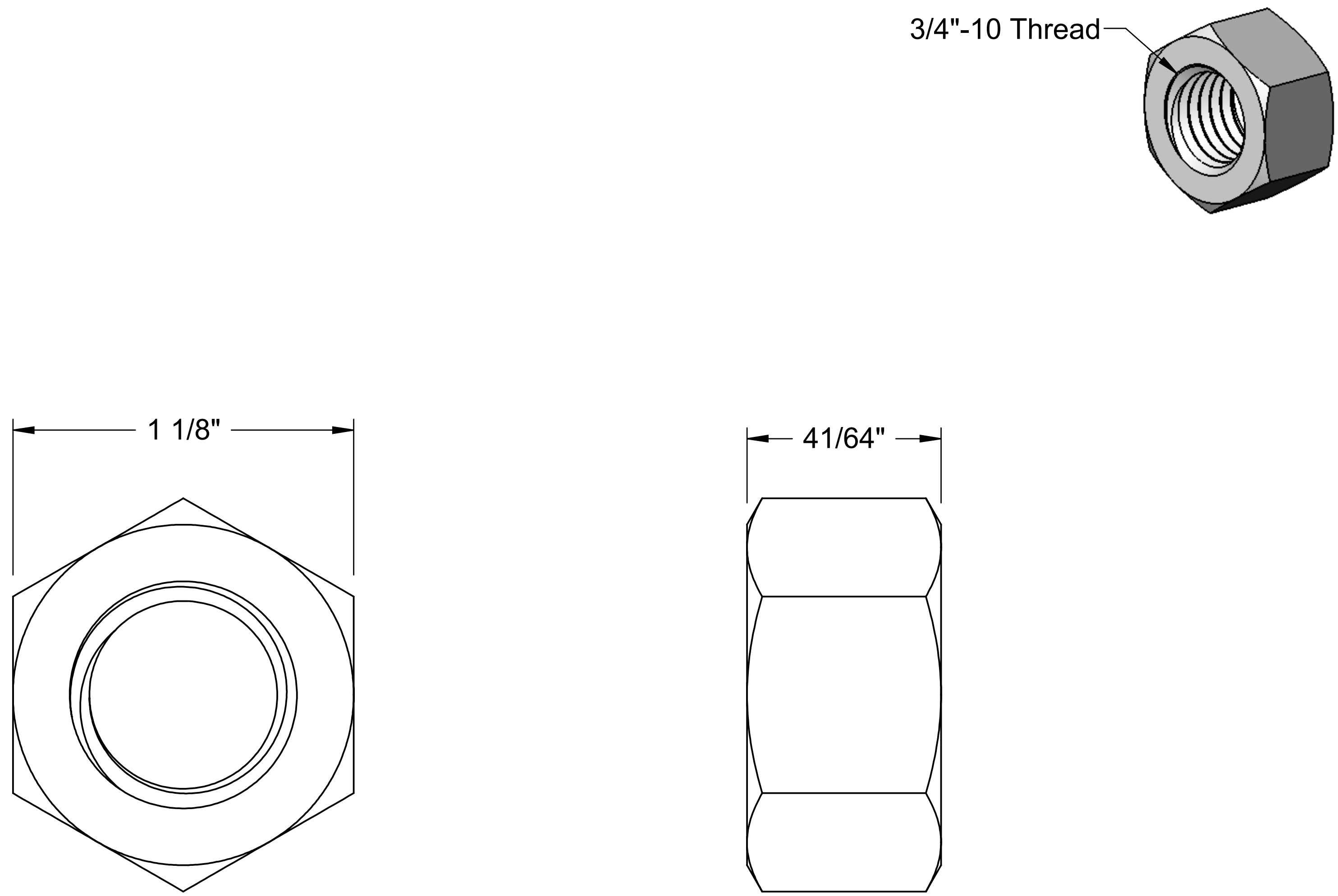


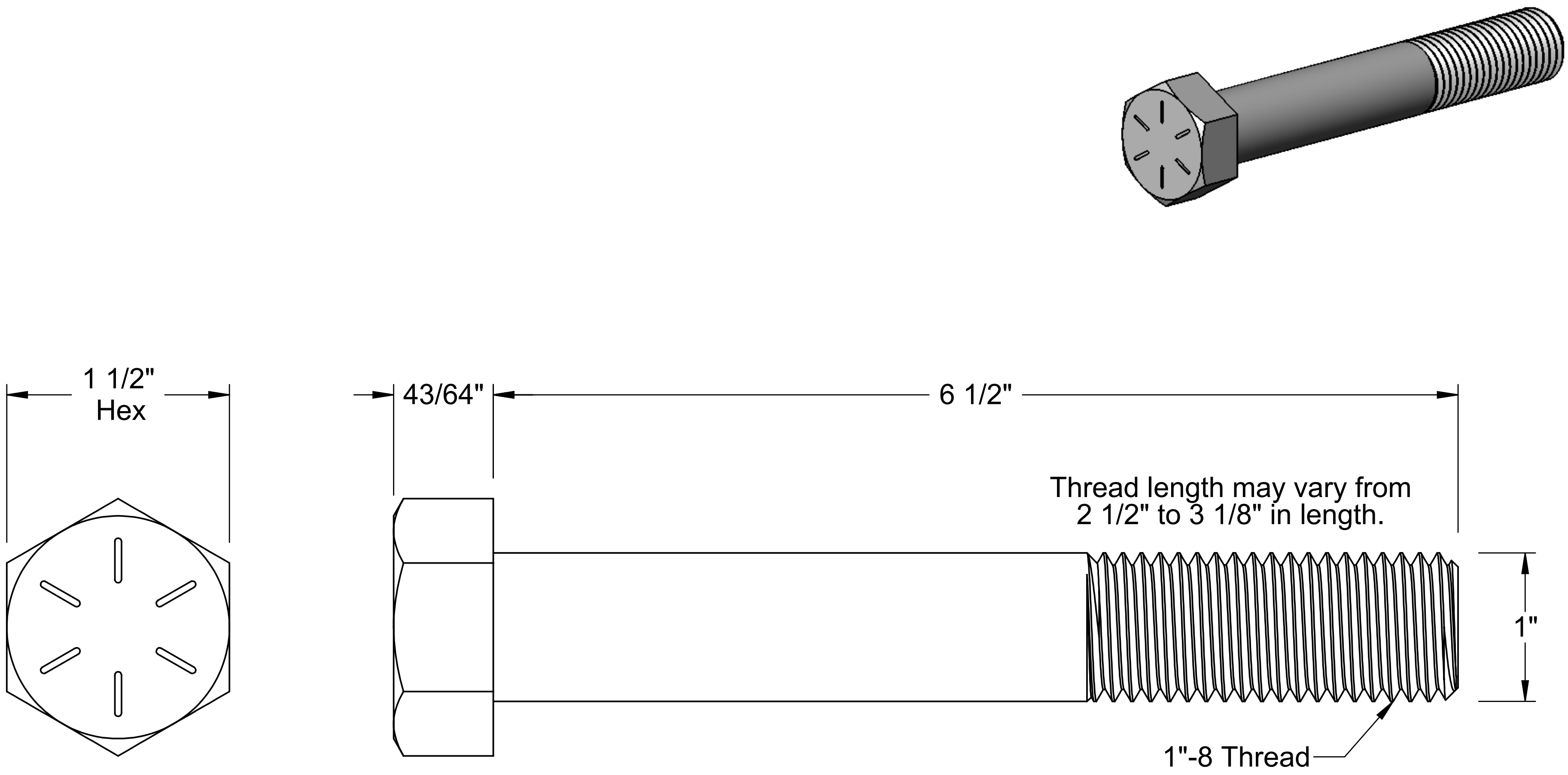
McMASTER-CARR® CAD	PART NUMBER	98023A036
http://www.mcmaster.com © 2014 McMaster-Carr Supply Company	General Purpose Washer	
Information in this drawing is provided for reference only.		



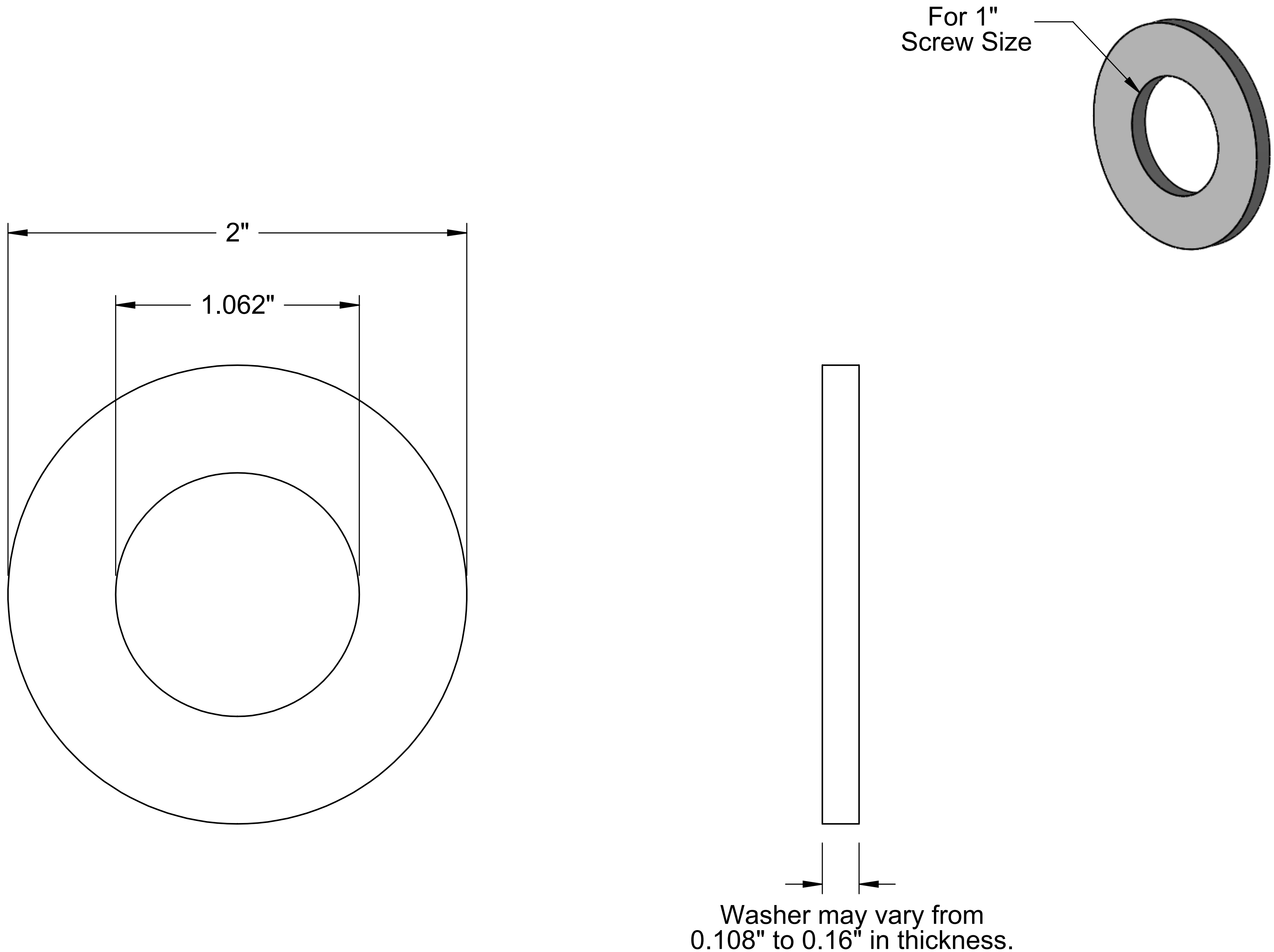
McMASTER-CARR® CAD	PART NUMBER	91104A047
http://www.mcmaster.com © 2014 McMaster-Carr Supply Company Information in this drawing is provided for reference only.	Split Lock Washer	



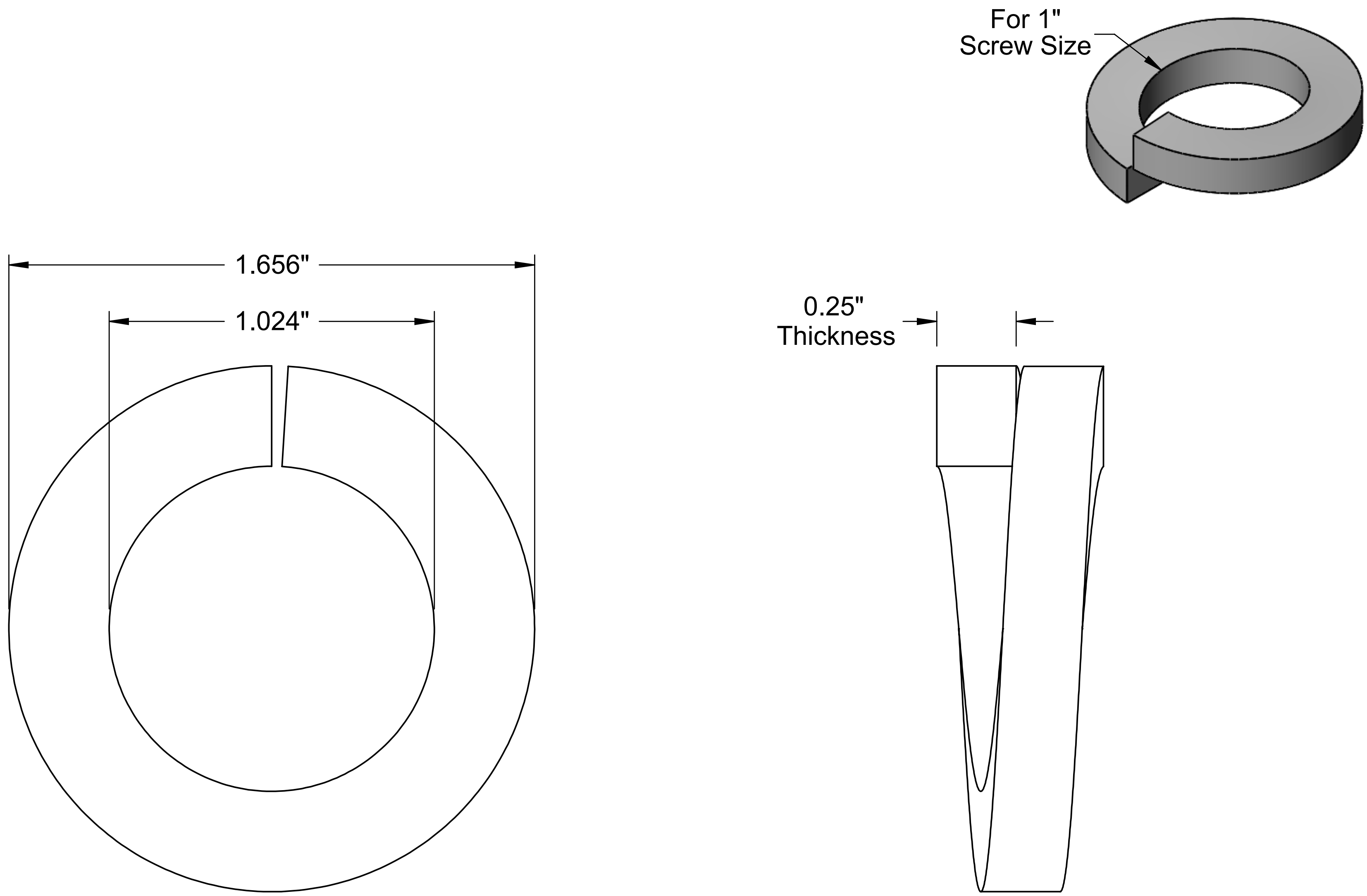
McMASTER-CARR® CAD	PART NUMBER	94895A036
http://www.mcmaster.com © 2015 McMaster-Carr Supply Company Information in this drawing is provided for reference only.	Hex Nut	



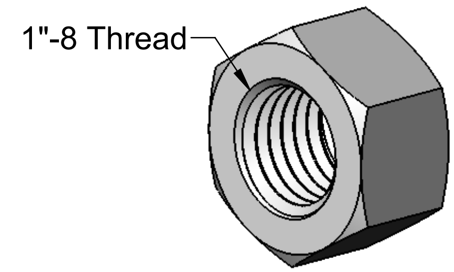
McMASTER-CARR® CAD	PART NUMBER 91257A970
http://www.mcmaster.com © 2014 McMaster-Carr Supply Company	High-Strength Steel Cap Screw-Grade 8
Information in this drawing is provided for reference only.	

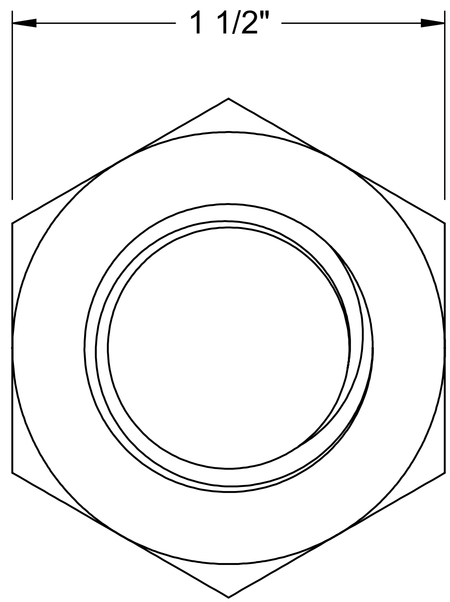


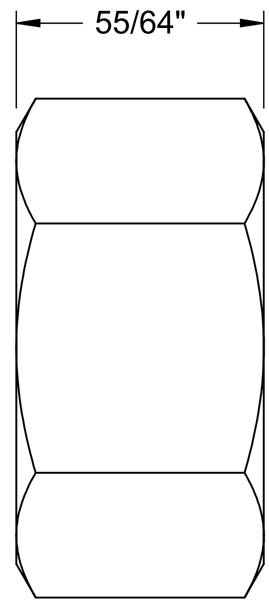
McMASTER-CARR® CAD	PART NUMBER 98023A038
http://www.mcmaster.com © 2014 McMaster-Carr Supply Company Information in this drawing is provided for reference only.	General Purpose Washer



McMASTER-CARR® CAD	PART NUMBER	91104A049
http://www.mcmaster.com © 2014 McMaster-Carr Supply Company Information in this drawing is provided for reference only.	Split Lock Washer	



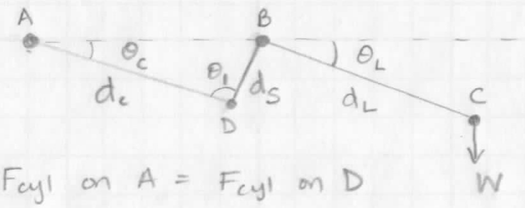




McMASTER-CARR® CAD	PART NUMBER 94895A038
http://www.mcmaster.com © 2015 McMaster-Carr Supply Company Information in this drawing is provided for reference only.	Hex Nut

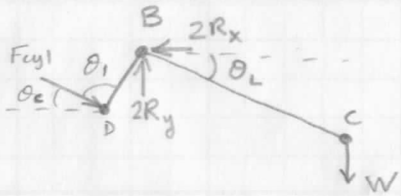
Position 1: 0°

①



- F_{cyl} on A = F_{cyl} on D

②



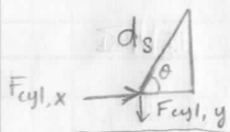
- Two bearings so $2R_x$ and $2R_y$, and symmetric loading on shaft

③ $\Sigma F_y: -F_{cyl}\sin\theta_c + 2R_y - W = 0$

$$\boxed{R_y = \frac{F_{cyl}\sin\theta_c + W}{2}}$$

$\Sigma F_x: -2R_x + F_{cyl}\cos\theta_c = 0$

$$\boxed{R_x = \frac{F_{cyl}\cos\theta_c}{2}}$$



$\theta = 180° - \theta_c - \theta_1$

$F_{cyl,x} = F_{cyl} \cdot \cos\theta_c$

$F_{cyl,y} = F_{cyl} \cdot \sin\theta_c$

$$\boxed{\begin{aligned}\Sigma M_B = &-d_L\cos\theta_L \cdot W \\ &+ d_s\sin(180°-\theta_c-\theta_1)\cdot F_{cyl}\cdot\cos\theta_c \\ &+ d_s\cos(180°-\theta_c-\theta_1)\cdot F_{cyl}\cdot\sin\theta_c\end{aligned}}$$

- ΣM_B is the torque on the shaft
- $\Sigma M_B = 0$ when static and constant accel

$F_{cyl} = P_{sys} \cdot A_{cyl}$

$= P_{sys} \cdot \frac{\pi}{4} D^2$

Variables

Point A: cylinder connection

Point B: Sandwich plate connection

Point C: load point

θ_c: angle between cylinder and frame

θ_1: angle between cylinder and sandwich plate

θ_L: angle between load lever arm and frame

d_s: distance between connection points on sandwich plate

d_L: length of load lever arm

- d_c: length of cylinder (between retracted and extended lengths)

Point D: connection point between cylinder and sandwich plate

F_{cyl}: Force exerted by cylinder

W: weight of float overheight

R_x: Reaction force in the x-dir of one bearing

R_y: Reaction force in the y-dir of one bearing

$F_{cyl,x}$: F_{cyl} in x-dir

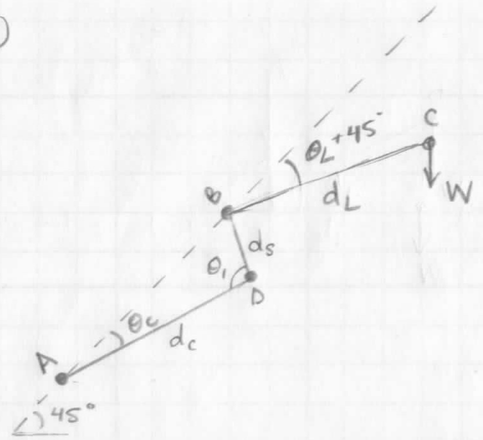
$F_{cyl,y}$: F_{cyl} in y-dir

P_{sys}: system pressure

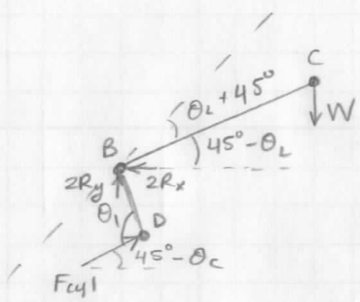
D: cylinder bore diameter

Position 2 : 45°

①



②

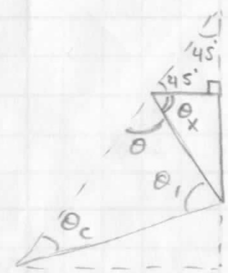


③ ΣF_y: $F_{cyl} \sin(45° - \theta_c) + 2R_y - W = 0$

$$\boxed{R_y = \frac{W - F_{cyl} \sin(45° - \theta_c)}{2}}$$

ΣF_x: $F_{cyl} \cos(45° - \theta_c) - 2R_x = 0$

$$\boxed{R_x = \frac{F_{cyl} \cos(45° - \theta_c)}{2}}$$



$\theta = 180 - \theta_1 - \theta_c$

$\theta_x = 180 - \theta - 45°$

$= 180 - (180 - \theta_1 - \theta_c) - 45°$

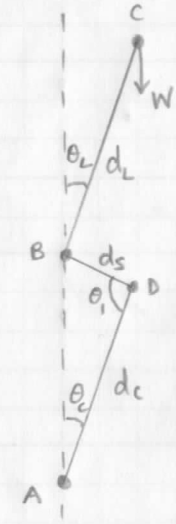
$= \cancel{180} - \cancel{180} + \theta_1 + \theta_c - 45°$

$= \theta_1 + \theta_c - 45°$

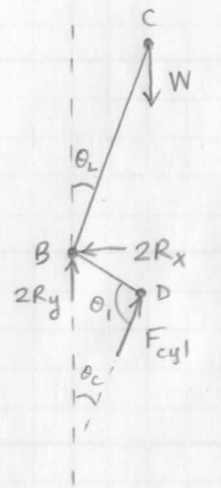
$$\boxed{\begin{aligned}\Sigma M_B = & - d_L \cos(\theta_L) \cdot W \\ & + d_s \sin(\theta_1 + \theta_c - 45°) \cdot F_{cyl} \cos(45° - \theta_c) \\ & + d_s \cos(\theta_1 + \theta_c - 45°) \cdot F_{cyl} \sin(45° - \theta_c)\end{aligned}}$$

Position 3: 90°

①



②



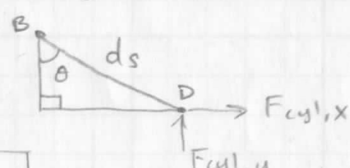
③ $\Sigma F_y: \; 2R_y + F_{cyl}\cos\theta_c - W = 0$

$$\boxed{R_y = \frac{W - F_{cyl}\cos\theta_c}{2}}$$

$\Sigma F_x: \; -2R_x + F_{cyl}\sin\theta_c = 0$

$$\boxed{R_x = \frac{F_{cyl}\sin\theta_c}{2}}$$

$\theta = 180 - \theta_1 - \theta_c$

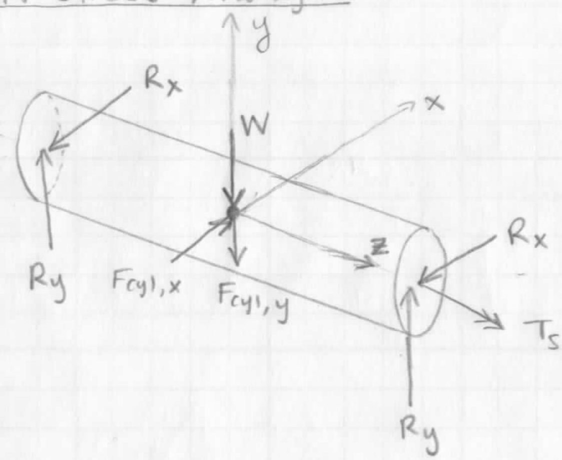


$F_{cyl,x} = F_{cyl}\sin\theta_c$

$F_{cyl,y} = F_{cyl}\cos\theta_c$

$$\boxed{\begin{aligned}\Sigma M_B = &- d_L \sin\theta_L \cdot W \\ &+ d_s \cos(180 - \theta_1 - \theta_c) \cdot F_{cyl}\sin\theta_c \\ &+ d_s \sin(180 - \theta_1 - \theta_c) \cdot F_{cyl}\cos\theta_c\end{aligned}}$$

Shaft Stress Analysis

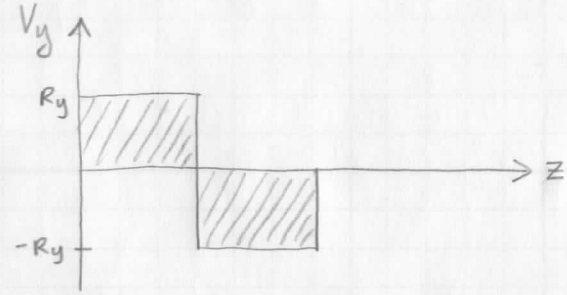


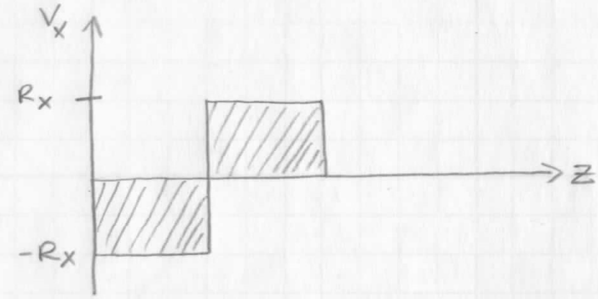
Variables:

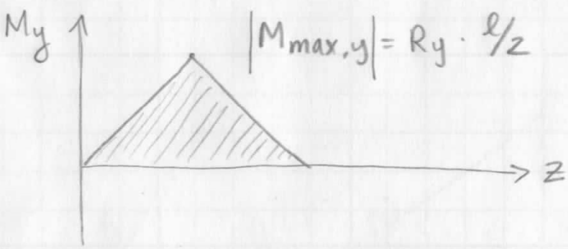
T_s: Torque on the shaft (ΣM_B)

ℓ: length of shaft

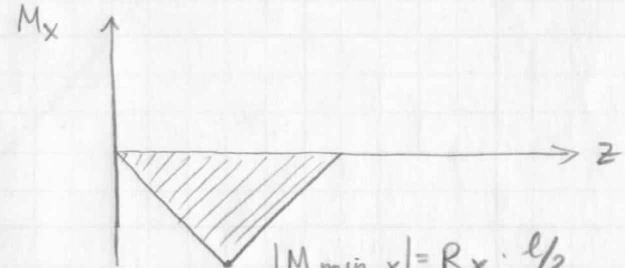
$\Sigma F_y = 0$, $\Sigma F_x = 0$, and $\Sigma M_z = 0$ from analysis







$|M_{max,y}| = R_y \cdot \ell/2$



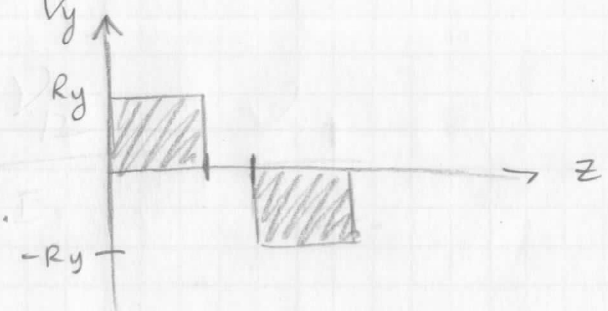
$|M_{min,x}| = R_x \cdot \ell/2$

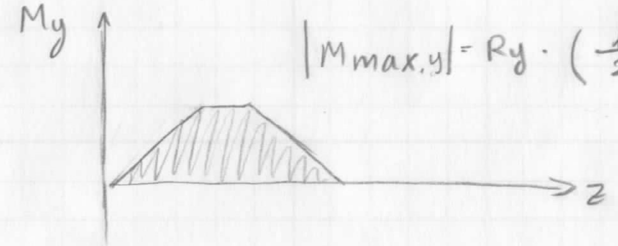
$$M_{max} = \sqrt{|M_{max,y}|^2 + |M_{min,x}|^2}$$

$$= \sqrt{R_y^2\left(\frac{\ell^2}{4}\right) + R_x^2\left(\frac{\ell^2}{4}\right)}$$

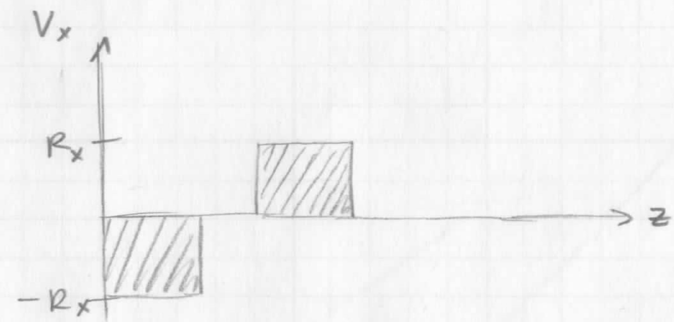
$$\boxed{M_{max} = \frac{\ell}{2}\sqrt{R_y^2 + R_x^2}}$$

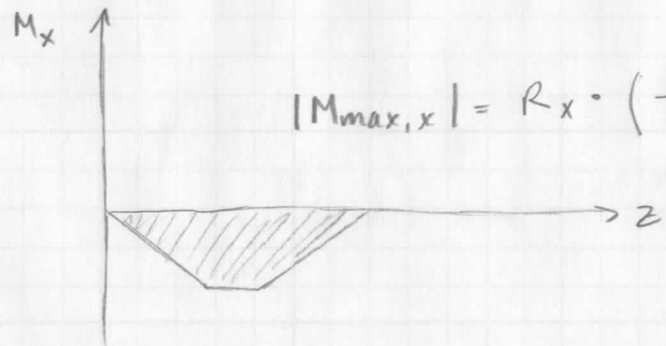
Two load points





$$|M_{max,y}| = R_y \cdot \left(\frac{l}{2} - \frac{6.25 \text{ in}}{2}\right)$$





$$|M_{max,x}| = R_x \cdot \left(\frac{l}{2} - \frac{6.25 \text{ in}}{2}\right)$$

0 Degree Configuration

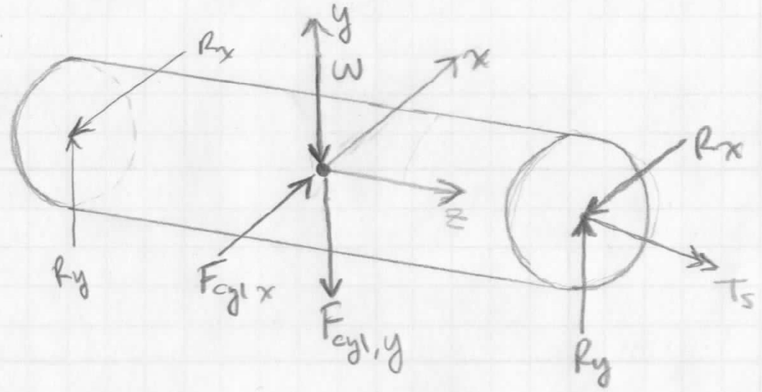
theta_1 [deg]	theta_C [deg]	theta_L [deg]	theta_1 [rad]	theta_C [rad]	theta_L [rad]	R_y [lb]	R_x [lb]	M_B [lb-in]	V_max,y [lb]	M_max,y [lb-in]	V_max,x [lb]	M_max,x [lb-in]	M_max [lb-in]
107.57	8.77	3.84	1.877	0.153	0.067	2546.226	11643.233	106.134	2546.226	11139.738	11643.233	50939.146	52142.980
105.57	8.87	1.94	1.843	0.155	0.034	2566.544	11640.081	1681.554	2566.544	11228.631	11640.081	50925.353	52148.574
103.57	8.95	0.02	1.808	0.156	0.000	2582.795	11637.533	3233.526	2582.795	11299.729	11637.533	50914.207	52153.047
101.57	9.02	1.91	1.773	0.157	0.033	2597.012	11635.285	4765.410	2597.012	11361.926	11635.285	50904.372	52156.960
99.57	9.08	3.85	1.738	0.158	0.067	2609.195	11633.345	6278.523	2609.195	11415.229	11633.345	50895.882	52160.313
97.57	9.13	5.8	1.703	0.159	0.101	2619.346	11631.718	7774.228	2619.346	11459.641	11631.718	50888.765	52163.107
95.57	9.16	7.77	1.668	0.160	0.136	2625.437	11630.737	9258.173	2625.437	11486.285	11630.737	50884.476	52164.783
93.57	9.19	9.74	1.633	0.160	0.170	2631.526	11629.754	10724.369	2631.526	11512.927	11629.754	50880.172	52166.459
91.57	9.2	11.73	1.598	0.161	0.205	2633.556	11629.425	12183.828	2633.556	11521.807	11629.425	50878.735	52167.017
89.57	9.21	13.72	1.563	0.161	0.239	2635.586	11629.096	13626.301	2635.586	11530.687	11629.096	50877.296	52167.576
87.57	9.2	15.73	1.528	0.161	0.275	2633.556	11629.425	15067.071	2633.556	11521.807	11629.425	50878.735	52167.017
85.57	9.18	17.75	1.493	0.160	0.310	2629.496	11630.082	16501.175	2629.496	11504.046	11630.082	50881.608	52165.900
83.57	9.15	19.78	1.459	0.160	0.345	2623.407	11631.064	17930.001	2623.407	11477.404	11631.064	50885.907	52164.224
81.57	9.11	21.82	1.424	0.159	0.381	2615.286	11632.369	19354.893	2615.286	11441.877	11632.369	50891.616	52161.990
79.57	9.05	23.88	1.389	0.158	0.417	2603.104	11634.316	20789.855	2603.104	11388.579	11634.316	50900.134	52158.637
77.57	8.99	25.94	1.354	0.157	0.453	2590.919	11636.251	22211.698	2590.919	11335.272	11636.251	50908.596	52155.283
75.57	8.91	28.02	1.319	0.156	0.489	2574.670	11638.810	23647.996	2574.670	11264.182	11638.810	50919.792	52150.811
73.57	8.83	30.1	1.284	0.154	0.525	2558.418	11641.346	25071.278	2558.418	11193.077	11641.346	50930.889	52146.337
71.57	8.73	32.2	1.249	0.152	0.562	2538.097	11644.485	26512.967	2538.097	11104.174	11644.485	50944.620	52140.742
69.57	8.62	34.31	1.214	0.150	0.599	2515.738	11647.896	27959.079	2515.738	11006.353	11647.896	50959.545	52134.586
67.57	8.5	36.43	1.179	0.148	0.636	2491.339	11651.569	29410.252	2491.339	10899.606	11651.569	50975.613	52127.867
65.57	8.38	38.55	1.144	0.146	0.673	2466.932	11655.190	30847.400	2466.932	10792.827	11655.190	50991.457	52121.145
63.57	8.24	40.69	1.110	0.144	0.710	2438.448	11659.351	32309.118	2438.448	10668.209	11659.351	51009.659	52113.299
61.57	8.09	42.84	1.075	0.141	0.748	2407.918	11663.731	33776.922	2407.918	10534.641	11663.731	51028.823	52104.889
59.57	7.93	45	1.040	0.138	0.785	2375.340	11668.315	35250.810	2375.340	10392.113	11668.315	51048.879	52095.912
57.57	7.76	47.17	1.005	0.135	0.823	2340.712	11673.086	36730.588	2340.712	10240.617	11673.086	51069.753	52086.370
55.57	7.58	49.35	0.970	0.132	0.861	2304.033	11678.026	38215.868	2304.033	10080.143	11678.026	51091.364	52076.259
53.57	7.4	51.53	0.935	0.129	0.899	2267.337	11682.851	39681.457	2267.337	9919.601	11682.851	51112.472	52066.143
51.57	7.2	53.73	0.900	0.126	0.938	2226.547	11688.076	41175.011	2226.547	9741.145	11688.076	51135.332	52054.895
49.57	7	55.93	0.865	0.122	0.976	2185.739	11693.159	42645.640	2185.739	9562.610	11693.159	51157.570	52043.640
47.57	6.78	58.15	0.830	0.118	1.015	2140.830	11698.585	44143.474	2140.830	9366.133	11698.585	51181.312	52031.251
45.57	6.56	60.37	0.795	0.114	1.054	2095.901	11703.840	45614.383	2095.901	9169.566	11703.840	51204.298	52018.854
43.57	6.33	62.6	0.760	0.110	1.093	2048.908	11709.148	47082.770	2048.908	8963.972	11709.148	51227.523	52005.884
41.57	6.09	64.84	0.726	0.106	1.132	1999.850	11714.486	48546.771	1999.850	8749.342	11714.486	51250.877	51992.340
39.57	5.85	67.08	0.691	0.102	1.171	1950.769	11719.619	49975.342	1950.769	8534.615	11719.619	51273.332	51978.787
37.57	5.6	69.33	0.656	0.098	1.210	1899.621	11724.747	51394.143	1899.621	8310.844	11724.747	51295.766	51964.659
35.57	5.34	71.59	0.621	0.093	1.249	1846.405	11729.843	52800.694	1846.405	8078.020	11729.843	51318.062	51949.955
33.57	5.08	73.85	0.586	0.089	1.289	1793.165	11734.697	54162.092	1793.165	7845.098	11734.697	51339.300	51935.241
31.57	4.81	76.12	0.551	0.084	1.329	1737.855	11739.483	55504.888	1737.855	7603.117	11739.483	51360.237	51919.951
29.57	4.53	78.4	0.516	0.079	1.368	1680.474	11744.170	56825.994	1680.474	7352.073	11744.170	51380.744	51904.083
29.34	4.5	78.66	0.512	0.079	1.373	1674.324	11744.656	56967.631	1674.324	7325.169	11744.656	51382.868	51902.382

45 Degree Configuration

theta_C [deg]	theta_L [deg]	theta_1 [rad]	theta_C [rad]	theta_L [rad]	R_y [lb]	R_x [lb]	M_B [lb-in]	V_max,y [lb]	M_max,y [lb-in]	V_max,x [lb]	M_max,x [lb-in]	M_max [lb-in]
8.77	3.84	1.877	0.153	0.067	-6212.886	9503.133	106.134	-6212.886	-27181.376	9503.133	41576.206	49673.012
8.87	1.94	1.843	0.155	0.034	-6196.289	9515.271	1681.554	-6196.289	-27108.765	9515.271	41629.310	49677.808
8.95	0.02	1.808	0.156	0.000	-6182.997	9524.960	3233.526	-6182.997	-27050.610	9524.960	41671.702	49681.649
9.02	1.91	1.773	0.157	0.033	-6171.354	9533.424	4765.410	-6171.354	-26999.676	9533.424	41708.728	49685.013
9.08	3.85	1.738	0.158	0.067	-6161.367	9540.666	6278.523	-6161.367	-26955.982	9540.666	41740.415	49687.898
9.13	5.8	1.703	0.159	0.101	-6153.039	9546.694	7774.228	-6153.039	-26919.545	9546.694	41766.786	49690.304
9.16	7.77	1.668	0.160	0.136	-6148.039	9550.307	9258.173	-6148.039	-26897.672	9550.307	41782.594	49691.749
9.19	9.74	1.633	0.160	0.170	-6143.038	9553.918	10724.369	-6143.038	-26875.791	9553.918	41798.389	49693.193
9.2	11.73	1.598	0.161	0.205	-6141.370	9555.121	12183.828	-6141.370	-26868.495	9555.121	41803.652	49693.675
9.21	13.72	1.563	0.161	0.239	-6139.702	9556.323	13626.301	-6139.702	-26861.198	9556.323	41808.914	49694.157
9.2	15.73	1.528	0.161	0.275	-6141.370	9555.121	15067.071	-6141.370	-26868.495	9555.121	41803.652	49693.675
9.18	17.75	1.493	0.160	0.310	-6144.705	9552.714	16501.175	-6144.705	-26883.085	9552.714	41793.125	49692.712
9.15	19.78	1.459	0.160	0.345	-6149.706	9549.103	17930.001	-6149.706	-26904.964	9549.103	41777.326	49691.267
9.11	21.82	1.424	0.159	0.381	-6156.371	9544.284	19354.893	-6156.371	-26934.123	9544.284	41756.242	49689.342
9.05	23.88	1.389	0.158	0.417	-6166.362	9537.046	20789.855	-6166.362	-26977.833	9537.046	41724.577	49686.455
8.99	25.94	1.354	0.157	0.453	-6176.345	9529.798	22211.698	-6176.345	-27021.510	9529.798	41692.867	49683.571
8.91	28.02	1.319	0.156	0.489	-6189.645	9520.118	23647.996	-6189.645	-27079.695	9520.118	41650.516	49679.728
8.83	30.1	1.284	0.154	0.525	-6202.930	9510.419	25071.278	-6202.930	-27137.821	9510.419	41608.083	49675.889
8.73	32.2	1.249	0.152	0.562	-6219.519	9498.269	26512.967	-6219.519	-27210.394	9498.269	41554.929	49671.095
8.62	34.31	1.214	0.150	0.599	-6237.741	9484.871	27959.079	-6237.741	-27290.117	9484.871	41496.312	49665.828
8.5	36.43	1.179	0.148	0.636	-6257.591	9470.216	29410.252	-6257.591	-27376.960	9470.216	41432.193	49660.090
8.38	38.55	1.144	0.146	0.673	-6277.410	9455.518	30847.400	-6277.410	-27463.668	9455.518	41367.892	49654.361
8.24	40.69	1.110	0.144	0.710	-6300.493	9438.319	32309.118	-6300.493	-27564.657	9438.319	41292.644	49647.687
8.09	42.84	1.075	0.141	0.748	-6325.178	9419.828	33776.922	-6325.178	-27672.655	9419.828	41211.748	49640.548
7.93	45	1.040	0.138	0.785	-6351.456	9400.034	35250.810	-6351.456	-27787.619	9400.034	41125.148	49632.949
7.76	47.17	1.005	0.135	0.823	-6379.315	9378.922	36730.588	-6379.315	-27909.503	9378.922	41032.784	49624.890
7.58	49.35	0.970	0.132	0.861	-6408.744	9356.478	38215.868	-6408.744	-28038.257	9356.478	40934.593	49616.376
7.4	51.53	0.935	0.129	0.899	-6438.103	9333.942	39681.457	-6438.103	-28166.702	9333.942	40835.998	49607.881
7.2	53.73	0.900	0.126	0.938	-6470.641	9308.794	41175.011	-6470.641	-28309.055	9308.794	40725.976	49598.464
7	55.93	0.865	0.122	0.976	-6503.091	9283.533	42645.640	-6503.091	-28451.023	9283.533	40615.457	49589.072
6.78	58.15	0.830	0.118	1.015	-6538.684	9255.615	44143.474	-6538.684	-28606.740	9255.615	40493.314	49578.767
6.56	60.37	0.795	0.114	1.054	-6574.169	9227.560	45614.383	-6574.169	-28761.988	9227.560	40370.575	49568.491
6.33	62.6	0.760	0.110	1.093	-6611.151	9198.085	47082.770	-6611.151	-28923.787	9198.085	40241.620	49557.779
6.09	64.84	0.726	0.106	1.132	-6649.616	9167.170	48546.771	-6649.616	-29092.068	9167.170	40106.368	49546.636
5.85	67.08	0.691	0.102	1.171	-6687.950	9136.094	49975.342	-6687.950	-29259.781	9136.094	39970.411	49535.528
5.6	69.33	0.656	0.098	1.210	-6727.743	9103.553	51394.143	-6727.743	-29433.874	9103.553	39828.044	49523.995
5.34	71.59	0.621	0.093	1.249	-6768.976	9069.526	52800.694	-6768.976	-29614.271	9069.526	39679.178	49512.041
5.08	73.85	0.586	0.089	1.289	-6810.055	9035.313	54162.092	-6810.055	-29793.990	9035.313	39529.495	49500.129
4.81	76.12	0.551	0.084	1.329	-6852.549	8999.587	55504.888	-6852.549	-29979.900	8999.587	39373.193	49487.804
4.53	78.4	0.516	0.079	1.368	-6896.438	8962.327	56825.994	-6896.438	-30171.916	8962.327	39210.179	49475.071
4.5	78.66	0.512	0.079	1.373	-6901.130	8958.322	56967.631	-6901.130	-30192.442	8958.322	39192.658	49473.710

90 Degree Configuration

theta_1 [deg]	theta_C [deg]	theta_L [deg]	theta_1 [rad]	theta_C [rad]	theta_L [rad]	R_y [lb]	R_x [lb]	M_B [lb-in]	V_max,y [lb]	M_max,y [lb-in]	V_max,x [lb]	M_max,x [lb-in]	M_max [lb-in]
107.57	8.77	3.84	1.877	0.153	0.067	-10893.233	1796.226	106.134	-10893.233	-47657.896	1796.226	7858.488	48301.459
105.57	8.87	1.94	1.843	0.155	0.034	-10890.081	1816.544	1681.554	-10890.081	-47644.103	1816.544	7947.381	48302.396
103.57	8.95	0.02	1.808	0.156	0.000	-10887.533	1832.795	3233.526	-10887.533	-47632.957	1832.795	8018.479	48303.153
101.57	9.02	1.91	1.773	0.157	0.033	-10885.285	1847.012	4765.410	-10885.285	-47623.122	1847.012	8080.676	48303.821
99.57	9.08	3.85	1.738	0.158	0.067	-10883.345	1859.195	6278.523	-10883.345	-47614.632	1859.195	8133.979	48304.398
97.57	9.13	5.8	1.703	0.159	0.101	-10881.718	1869.346	7774.228	-10881.718	-47607.515	1869.346	8178.391	48304.881
95.57	9.16	7.77	1.668	0.160	0.136	-10880.737	1875.437	9258.173	-10880.737	-47603.226	1875.437	8205.035	48305.172
93.57	9.19	9.74	1.633	0.160	0.170	-10879.754	1881.526	10724.369	-10879.754	-47598.922	1881.526	8231.677	48305.465
91.57	9.2	11.73	1.598	0.161	0.205	-10879.425	1883.556	12183.828	-10879.425	-47597.485	1883.556	8240.557	48305.562
89.57	9.21	13.72	1.563	0.161	0.239	-10879.096	1885.586	13626.301	-10879.096	-47596.046	1885.586	8249.437	48305.660
87.57	9.2	15.73	1.528	0.161	0.275	-10879.425	1883.556	15067.071	-10879.425	-47597.485	1883.556	8240.557	48305.562
85.57	9.18	17.75	1.493	0.160	0.310	-10880.082	1879.496	16501.175	-10880.082	-47600.358	1879.496	8222.796	48305.367
83.57	9.15	19.78	1.459	0.160	0.345	-10881.064	1873.407	17930.001	-10881.064	-47604.657	1873.407	8196.154	48305.075
81.57	9.11	21.82	1.424	0.159	0.381	-10882.369	1865.286	19354.893	-10882.369	-47610.366	1865.286	8160.627	48304.687
79.57	9.05	23.88	1.389	0.158	0.417	-10884.316	1853.104	20789.855	-10884.316	-47618.884	1853.104	8107.329	48304.109
77.57	8.99	25.94	1.354	0.157	0.453	-10886.251	1840.919	22211.698	-10886.251	-47627.346	1840.919	8054.022	48303.534
75.57	8.91	28.02	1.319	0.156	0.489	-10888.810	1824.670	23647.996	-10888.810	-47638.542	1824.670	7982.932	48302.773
73.57	8.83	30.1	1.284	0.154	0.525	-10891.346	1808.418	25071.278	-10891.346	-47649.639	1808.418	7911.827	48302.020
71.57	8.73	32.2	1.249	0.152	0.562	-10894.485	1788.097	26512.967	-10894.485	-47663.370	1788.097	7822.924	48301.087
69.57	8.62	34.31	1.214	0.150	0.599	-10897.896	1765.738	27959.079	-10897.896	-47678.295	1765.738	7725.103	48300.073
67.57	8.5	36.43	1.179	0.148	0.636	-10901.569	1741.339	29410.252	-10901.569	-47694.363	1741.339	7618.356	48298.981
65.57	8.38	38.55	1.144	0.146	0.673	-10905.190	1716.932	30847.400	-10905.190	-47710.207	1716.932	7511.577	48297.905
63.57	8.24	40.69	1.110	0.144	0.710	-10909.351	1688.448	32309.118	-10909.351	-47728.409	1688.448	7386.959	48296.668
61.57	8.09	42.84	1.075	0.141	0.748	-10913.731	1657.918	33776.922	-10913.731	-47747.573	1657.918	7253.391	48295.366
59.57	7.93	45	1.040	0.138	0.785	-10918.315	1625.340	35250.810	-10918.315	-47767.629	1625.340	7110.863	48294.004
57.57	7.76	47.17	1.005	0.135	0.823	-10923.086	1590.712	36730.588	-10923.086	-47788.503	1590.712	6959.367	48292.585
55.57	7.58	49.35	0.970	0.132	0.861	-10928.026	1554.033	38215.868	-10928.026	-47810.114	1554.033	6798.893	48291.117
53.57	7.4	51.53	0.935	0.129	0.899	-10932.851	1517.337	39681.457	-10932.851	-47831.222	1517.337	6638.351	48289.683
51.57	7.2	53.73	0.900	0.126	0.938	-10938.076	1476.547	41175.011	-10938.076	-47854.082	1476.547	6459.895	48288.129
49.57	7	55.93	0.865	0.122	0.976	-10943.159	1435.739	42645.640	-10943.159	-47876.320	1435.739	6281.360	48286.618
47.57	6.78	58.15	0.830	0.118	1.015	-10948.585	1390.830	44143.474	-10948.585	-47900.062	1390.830	6084.883	48285.005
45.57	6.56	60.37	0.795	0.114	1.054	-10953.840	1345.901	45614.383	-10953.840	-47923.048	1345.901	5888.316	48283.443
43.57	6.33	62.6	0.760	0.110	1.093	-10959.148	1298.908	47082.770	-10959.148	-47946.273	1298.908	5682.722	48281.864
41.57	6.09	64.84	0.726	0.106	1.132	-10964.486	1249.850	48546.771	-10964.486	-47969.627	1249.850	5468.092	48280.277
39.57	5.85	67.08	0.691	0.102	1.171	-10969.619	1200.769	49975.342	-10969.619	-47992.082	1200.769	5253.365	48278.751
37.57	5.6	69.33	0.656	0.098	1.210	-10974.747	1149.621	51394.143	-10974.747	-48014.516	1149.621	5029.594	48277.226
35.57	5.34	71.59	0.621	0.093	1.249	-10979.843	1096.405	52800.694	-10979.843	-48036.812	1096.405	4796.770	48275.711
33.57	5.08	73.85	0.586	0.089	1.289	-10984.697	1043.165	54162.092	-10984.697	-48058.050	1043.165	4563.848	48274.267
31.57	4.81	76.12	0.551	0.084	1.329	-10989.483	987.855	55504.888	-10989.483	-48078.987	987.855	4321.867	48272.844
29.57	4.53	78.4	0.516	0.079	1.368	-10994.170	930.474	56825.994	-10994.170	-48099.494	930.474	4070.823	48271.450
29.34	4.5	78.66	0.512	0.079	1.373	-10994.656	924.324	56967.631	-10994.656	-48101.618	924.324	4043.919	48271.306



Givens: (from Static Analysis)

W - weight of load

R_x - reaction force at the bearing in x

R_y - reaction force @ the bearing in y

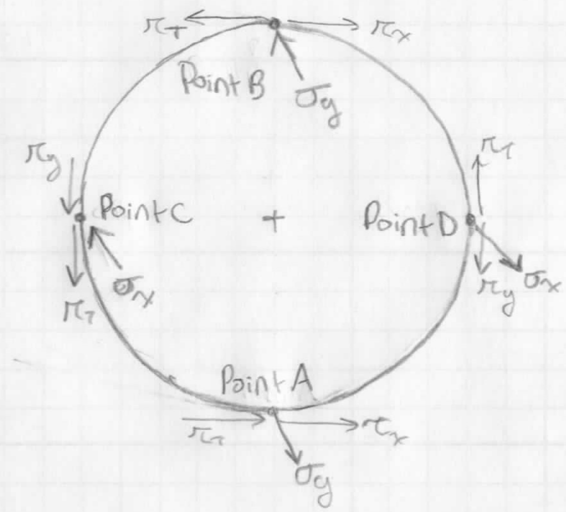
$F_{cyl\,x}$ - force of cylinder on shaft in x

$F_{cyl\,y}$ - force of cylinder on shaft in y

T_s - torque on shaft (ΣM_B)

d - diameter of shaft

Combined Loading on Shaft:



$$\tau_T = \frac{16\,T_s}{\pi d^3}$$

$$\tau_x = \frac{4V_x}{3A}$$

$$\tau_y = \frac{4V_y}{3A}$$

$$\sigma_x = \frac{32\,M_x}{\pi D^3}$$

$$\sigma_y = \frac{32\,M_y}{\pi D^3}$$

Static Case:

(Check pt. A + C @ all theta for static passing)

Point A:

$\tau_{ptA} = \tau_T + \tau_x$

$\sigma_{ptA} = \sigma_y$

Point C:

$\tau_{ptC} = \tau_y + \tau_T$

$\sigma_{ptC} = \sigma_x$

$\sigma'_{max\,ptA} = \max\left(\sqrt{\sigma_{ptA}^2 + 3(\tau_{ptA})^2}\right)$ $\quad \sigma'_{max\,ptC} = \max\left(\sqrt{\sigma_{ptC}^2 + 3\tau_{ptC}^2}\right)$

$\sigma'_{max} = \max(\sigma'_{max\,ptA}, \sigma'_{max\,ptC})$

DE criteria

$$n_{-DE} = \frac{S_y}{\sigma'_{max}}$$

Fatigue Case:

- Worst case is when pt. A goes to pt D in up

$\sigma_{max} = \sigma_{pt D}$

$\sigma_{min} = \sigma_{pt A_0}$

$\tau_{max} = \tau_{pt D}$

$\tau_{max} = \tau_{pt A_0}$

$$\sigma_a = \left| \frac{\sigma_{max} - \sigma_{min}}{2} \right|$$

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2}$$

$$\tau_a = \left| \frac{\tau_{max} - \tau_{min}}{2} \right|$$

$$\tau_m = \frac{\tau_{max} + \tau_{min}}{2}$$

$K_t = K_{ts} = 1$ (Table 7-1 Shigley's)

$K_f = 1 + q(K_t - 1) = 1$ (Fig 6-20)

$K_{fs} = 1 + q_s(K_{ts} - 1) = 1$ (Fig 6-21)

$$\sigma_a' = \left[(K_f \sigma_a)^2 + 3(K_{fs} \tau_a)^2\right]^{1/2}$$

$$\sigma_m' = \left[(K_f \sigma_m)^2 + 3(K_{fs} \tau_m)^2\right]^{1/2}$$

$S_e = K_a K_b K_c K_d K_e K_f S_e'$ (Eqn 6-18)

$S_e' = .5 S_{ut}$

$K_a = a S_{ut}^b$ (Table 6-2)

$K_b = .879 d^{-.107}$ $\quad .11 \leq d \leq 2 in$

$\quad .91 d^{-.157}$ $\quad 2 < d \leq 10 in$

$K_c = 1$

$K_d = 1$

$K_e = .897$ (Table 6-5)

$K_f = 1$

Langer Static Yield

$$n_y = \frac{S_y}{\sigma_a' + \sigma_m'} \qquad (n_y > 1)$$

Fatigue Strength

$$a = \frac{(f S_{ut})^2}{S_e}$$

$$b = -\frac{1}{3}\log\left(\frac{f S_{ut}}{S_e}\right)$$

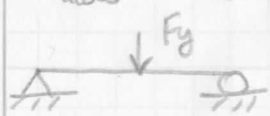
$$S_f = a N^b \qquad (N - \text{\# of cycles})$$

Goodman Criteria

$$n_{GM} = \frac{1}{\frac{\sigma_a'}{S_f} + \frac{\sigma_m'}{S_{ut}}}$$

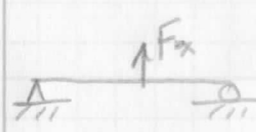
Stiffness (* Not used in this design, but kept for reference)

θ_{allow} (Table 7-2)



$F_y = 2R_y$

$\theta_{bearing\,y} = \left|\frac{F_y l^2}{16EI}\right|$



$F_x = 2R_x$

$\theta_{bearing\,x} = \left|\frac{F_x l^2}{16EI}\right|$

$\theta_{bearing} = \sqrt{\theta_{bearing\,x}^2 + \theta_{bearing\,y}^2}$

$\theta_{bearing\,max} = max(\theta_{bearing})$

Check $\theta_{bearing\,max} < \theta_{allow}$

if not

$d_{new} = d\left|\frac{n_d\,\theta_{bearing}}{\theta_{allow}}\right|^{1/4}$

Retrieve Static Analysis Data

0 Degree Configuration

data :=
Forces and Moments 0 Degree Position.xlsx

*for all direction and sign conventions refer to static analysis sheets

Parse Data into Variables

$V\_x := data^{\langle 3 \rangle} \cdot lbf$ Shear in x-direction

$V\_y := data^{\langle 1 \rangle} \cdot lbf$ Shear in y-direction

$M\_x := data^{\langle 4 \rangle} \cdot lbf \cdot in$ Moment in x-direction

$M\_y := data^{\langle 2 \rangle} \cdot lbf \cdot in$ Moment in y-direction

$T\_tot := data^{\langle 0 \rangle} \cdot lbf \cdot in$ Torque on shaft

Shaft Given Values

$d\_shaft := 2.75in$ Shaft Diameter

$A\_shaft := \frac{\pi}{4} \cdot d\_shaft^2 = 5.94\,in^2$ Shaft Area

$S\_ut := 99000psi$ Ultimate Strength (ie.1045)

$S\_y := 77000psi$ Yield Strength (ie 1045)

$E\_shaft := 29000000psi$ Youngs Modulus

$I\_shaft := \frac{\pi\, d\_shaft^4}{64}$ Shaft Moment of Inertia

Stresses in Shaft

$\tau\_t := \frac{16 \cdot T\_tot}{\pi \cdot d\_shaft^3}$ Torsional Shear

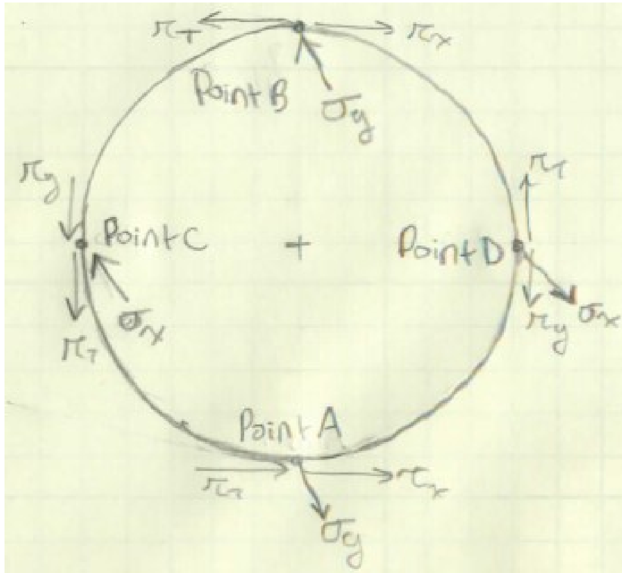
$\tau\_x := \frac{4 \cdot V\_x}{3 \cdot A\_shaft}$ Shear X-direction

$\tau\_y := \frac{4 \cdot V\_x}{3 \cdot A\_shaft}$ Shear Y-direction

$\sigma\_x := \frac{32 \cdot M\_x}{\pi \cdot d\_shaft^3}$ Bending X-direction

$\sigma\_y := \frac{32 \cdot M\_y}{\pi \cdot d\_shaft^3}$ Bending Y-direction

Combined Stresses at Critical Points



Point A:

$\tau\_ptA := \tau\_t + \tau\_x$

$\sigma\_ptA := \sigma\_y$

Find Maximum Von Mesis Stress

$$\sigma'\_ptA := \max\left[\sqrt{(\sigma\_ptA)^2 + 3\cdot(\tau\_ptA)^2}\right] = 2.895\times 10^4\ \text{psi}$$

Point C:

$\tau\_ptC := \tau\_t + \tau\_y$

$\sigma\_ptC := \sigma\_x$

Find Maximum Von Mesis Stress

$$\sigma'\_ptC := \max\left[\sqrt{(\sigma\_ptC)^2 + 3\cdot(\tau\_ptC)^2}\right] = 3.819\times 10^4\ \text{psi}$$

Distortion Energy Safety Factor

$$\sigma'\_max := \max(\sigma'\_ptC, \sigma'\_ptA) = 3.819\times 10^4\ \text{psi}$$

$$\boxed{SF\_de := \frac{S\_y}{\sigma'\_max} = 2.016}$$

**This is checking the static case safety factor on the maximum von mesis stress at each value of theta*

Calculate the Alternating and Mean Stress

****Assume worst case in fatigue is when pt.A in down position rotates to pt. D in up position****

Point D:

$\tau\_ptD := \tau\_t - \tau\_y$

$\sigma\_ptD := \sigma\_x$

Maximum/Minimum Bending Stress

$\sigma\_max := \max\left(\sigma\_ptD_{40}\right) = 2.517 \times 10^{4}\ \text{psi}$

$\sigma\_min := \min\left(\sigma\_ptA_{0}\right) = 5.456 \times 10^{3}\ \text{psi}$

*Point A for lowest down position associates with σ_ptA_{0}

*Point D for highest up position associates with σ_ptD_{40}

Find Mean/Alternating Bending Stress

$$\sigma\_m := \frac{\sigma\_max + \sigma\_min}{2} = 1.531 \times 10^{4}\ \text{psi}$$

$$\sigma\_a := \left|\frac{\sigma\_max - \sigma\_min}{2}\right| = 9.855 \times 10^{3}\ \text{psi}$$

Find Maximum/Minimum Torsional Stress

$\tau\_max := \max\left(\tau\_ptD_{40}\right) = 1.131 \times 10^{4}\ \text{psi}$

$\tau\_min := \min\left(\tau\_ptA_{0}\right) = 2.64 \times 10^{3}\ \text{psi}$

Find Mean/Alternating Torsional Stress

$$\tau\_m := \frac{\tau\_max + \tau\_min}{2} = 6.977 \times 10^{3}\ \text{psi}$$

$$\tau\_a := \left|\frac{\tau\_max - \tau\_min}{2}\right| = 4.337 \times 10^{3}\ \text{psi}$$

Determine Fatigue Stress Concentration Factors

Table 7–1

First Iteration Estimates for Stress-Concentration Factors K_t and K_{ts}.

Warning: These factors are only estimates for use when actual dimensions are not yet determined. Do *not* use these once actual dimensions are available.

	Bending	Torsional	Axial
Shoulder fillet—sharp ($r/d = 0.02$)	2.7	2.2	3.0
Shoulder fillet—well rounded ($r/d = 0.1$)	1.7	1.5	1.9
End-mill keyseat ($r/d = 0.02$)	2.14	3.0	—
Sled runner keyseat	1.7	—	—
Retaining ring groove	5.0	3.0	5.0

$K_t := 1$

$K_{ts} := 1$

*Equals 1 since we do not have step downs or grooves

$q := 0$

$q_s := 0$

Look up q & qs values from Shigley graphs 6-20 & 6-21

$K_f := 1 + q\cdot(K_t - 1) = 1$ Bending Stress Concentration Factor (Shigley Eqn 6-32)

$K_{fs} := 1 + q_s\cdot(K_{ts} - 1) = 1$ Shear Stress Concentration Factor (Shigley Eqn 6-32)

Calculate Von Mesis Mean/Alternating Stress at Critical Points

$$\sigma'_a := \sqrt{(K_f\cdot\sigma_a)^2 + 3\cdot(K_{fs}\cdot\tau_a)^2} = 1.239 \times 10^4\ \text{psi}$$

$$\sigma'_m := \sqrt{(K_f\cdot\sigma_m)^2 + 3\cdot(K_{fs}\cdot\tau_m)^2} = 1.951 \times 10^4\ \text{psi}$$

Calculate Endurance Limit

$$S_e = k_a k_b k_c k_d k_e k_f S'_e \qquad (6\text{–}18)$$

where k_a = surface condition modification factor
k_b = size modification factor
k_c = load modification factor
k_d = temperature modification factor
k_e = reliability factor[13]
k_f = miscellaneous-effects modification factor
S'_e = rotary-beam test specimen endurance limit
S_e = endurance limit at the critical location of a machine part in the geometry and condition of use

$S'e := .5 \cdot S\_ut = 4.95 \times 10^4\ psi$ Shigleys Eqn 6-8 *If Sut >200kspi S'e=100kpsi

Calculate Marin Factors

Table 6–2
Parameters for Marin Surface Modification Factor, Eq. (6–19)

Surface Finish	Factor a S_{ut}, kpsi	Factor a S_{ut}, MPa	Exponent b
Ground	1.34	1.58	−0.085
Machined or cold-drawn	2.70	4.51	−0.265
Hot-rolled	14.4	57.7	−0.718
As-forged	39.9	272.	−0.995

$a := 2.7$

$b := -.265$

$k\_a := a \cdot \left(S\_ut \cdot psi^{-1}\right)^b = 0.128$

$:= .879 \cdot \left(d\ \ shaft \cdot in^{-1}\right)^{-.107}$ for .11in < d_shaft < 2in

$k\_b := .91 \cdot \left(d\_shaft \cdot in^{-1}\right)^{-.157}$ for 2in < d_shaft < 10in

$k\_c := 1$ Equals 1 for combined loading case

$k\_d := 1$

$k\_e := .897$ 90% Reliability (Shigleys Table 6-5)

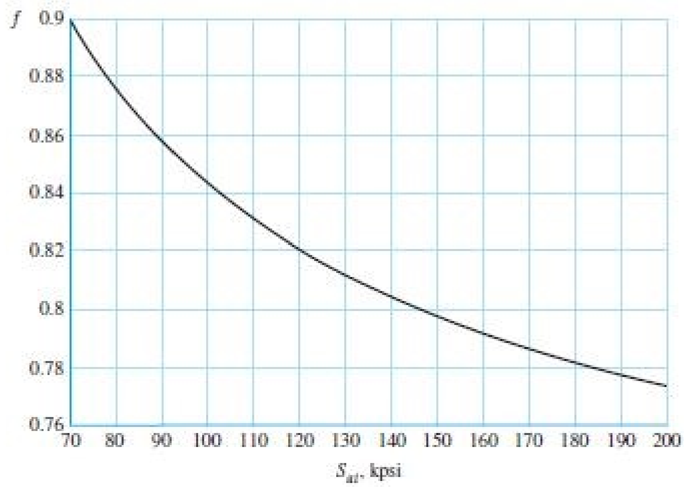
$k\_f := 1$

$$Se := k\_a \cdot k\_b \cdot k\_c \cdot k\_d \cdot k\_e \cdot k\_f \cdot S'e = 4.416 \times 10^{3}\ \text{psi}$$

Calculate Fatigue Strength

Figure 6–18

Fatigue strength fraction, f, of S_{ut} at 10^3 cycles for $S_e = S_e' = 0.5 S_{ut}$ at 10^6 cycles.



$$f := .845$$

$$a\_2 := \frac{(f \cdot S\_ut)^2}{Se} = 1.585 \times 10^{6}\ \text{psi}$$

$$b\_2 := \frac{-1}{3} \cdot \log\left(\frac{f \cdot S\_ut}{Se}\right) = -0.426$$

$$N\_cycles := 400$$

$$S\_f := S\_ut \cdot N\_cycles^{\frac{\log(f)}{3}} = 8.554 \times 10^{4}\ \text{psi}$$

Check Langer Static Yield

$$SF\_y := \frac{S\_y}{\sigma'\_a + \sigma'\_m} = 2.414$$

Modified Goodman Safety Factor

$$\boxed{SF\_mg := \frac{1}{\left(\frac{\sigma'\_a}{S\_f}\right) + \left(\frac{\sigma'\_m}{S\_ut}\right)} = 2.925}$$

Check Stiffness Criteria

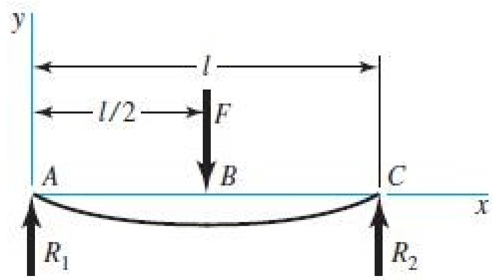
**Not used in our design, purely for future reference*

Table 7-2

Typical Maximum Ranges for Slopes and Transverse Deflections

Slopes	
Tapered roller	0.0005–0.0012 rad
Cylindrical roller	0.0008–0.0012 rad
Deep-groove ball	0.001–0.003 rad
Spherical ball	0.026–0.052 rad
Self-align ball	0.026–0.052 rad
Uncrowned spur gear	<0.0005 rad

5 Simple supports—center load



$l\_bearing := 11.5in$ Distance between bearing supports

$R\_x := V\_x$

$F\_x := 2 \cdot R\_x$

$$slope\_bearing\_x := \left| \frac{F\_x \cdot l\_bearing^2}{16 \cdot E\_shaft \cdot I\_shaft} \right|$$

$R\_y := V\_y$

$F\_y := 2 \cdot R\_y$

$$slope\_bearing\_y := \left| \frac{F\_y \cdot l\_bearing^2}{16 \cdot E\_shaft \cdot I\_shaft} \right|$$

$$slope\_bearing := \sqrt{slope\_bearing\_x^2 + slope\_bearing\_y^2}$$

$slope\_bearing\_max := max(slope\_bearing) = 0.015$

Retrieve Static Analysis Data

45 Degree Configuration

data :=
Forces and Moments - 45 Degree Position.xlsx

*for all direction and sign conventions refer to static analysis sheets

Parse Data into Variables

$V\_x := data^{\langle 3 \rangle} \cdot lbf$ — Shear in x-direction

$V\_y := data^{\langle 1 \rangle} \cdot lbf$ — Shear in y-direction

$M\_x := data^{\langle 4 \rangle} \cdot lbf \cdot in$ — Moment in x-direction

$M\_y := data^{\langle 2 \rangle} \cdot lbf \cdot in$ — Moment in y-direction

$T\_tot := data^{\langle 0 \rangle} \cdot lbf \cdot in$ — Torque on shaft

Shaft Given Values

$d\_shaft := 2.75in$ — Shaft Diameter

$A\_shaft := \frac{\pi}{4} \cdot d\_shaft^2 = 5.94\, in^2$ — Shaft Area

$S\_ut := 99000psi$ — Ultimate Strength (ie.1045)

$S\_y := 77000psi$ — Yield Strength (ie 1045)

$E\_shaft := 29000000psi$ — Youngs Modulus

$I\_shaft := \frac{\pi\, d\_shaft^4}{64}$ — Shaft Moment of Inertia

Stresses in Shaft

$\tau\_t := \frac{16 \cdot T\_tot}{\pi \cdot d\_shaft^3}$ — Torsional Shear

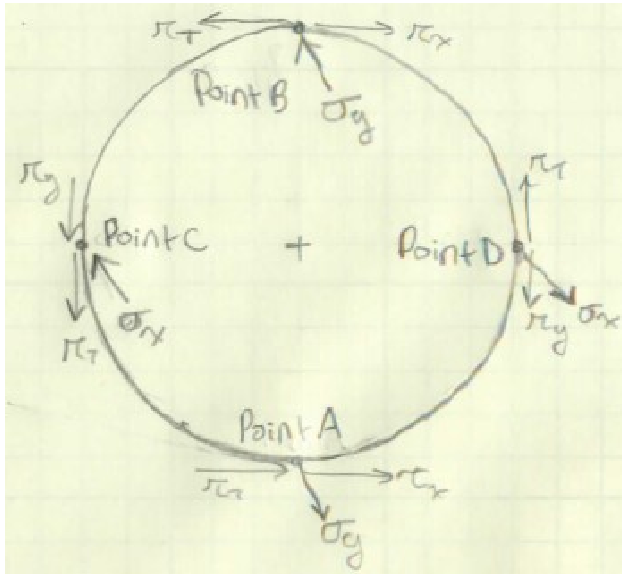
$\tau\_x := \frac{4 \cdot V\_x}{3 \cdot A\_shaft}$ — Shear X-direction

$\tau\_y := \frac{4 \cdot V\_x}{3 \cdot A\_shaft}$ — Shear Y-direction

$\sigma\_x := \frac{32 \cdot M\_x}{\pi \cdot d\_shaft^3}$ — Bending X-direction

$\sigma\_y := \frac{32 \cdot M\_y}{\pi \cdot d\_shaft^3}$ — Bending Y-direction

Combined Stresses at Critical Points



Point A:

$\tau\_ptA := \tau\_t + \tau\_x$

$\sigma\_ptA := \sigma\_y$

<u>Find Maximum Von Mesis Stress</u>

$$\sigma'\_ptA := \max\left[\sqrt{(\sigma\_ptA)^2 + 3\cdot(\tau\_ptA)^2}\right] = 3.135 \times 10^4 \text{ psi}$$

Point C:

$\tau\_ptC := \tau\_t + \tau\_y$

$\sigma\_ptC := \sigma\_x$

<u>Find Maximum Von Mesis Stress</u>

$$\sigma'\_ptC := \max\left[\sqrt{(\sigma\_ptC)^2 + 3\cdot(\tau\_ptC)^2}\right] = 3.366 \times 10^4 \text{ psi}$$

Distortion Energy Safety Factor

$$\sigma'\_max := \max(\sigma'\_ptC, \sigma'\_ptA) = 3.366 \times 10^4 \text{ psi}$$

$$\boxed{SF\_de := \frac{S\_y}{\sigma'\_max} = 2.288}$$

**This is checking the static case safety factor on the maximum von mesis stress at each value of theta*

Calculate the Alternating and Mean Stress

****Assume worst case in fatigue is when pt.A in down position rotates to pt. D in up position****

Point D:

$\tau\_ptD := \tau\_t - \tau\_y$

$\sigma\_ptD := \sigma\_x$

Maximum/Minimum Bending Stress

$\sigma\_max := \max(\sigma\_ptD_{40}) = 1.92 \times 10^4 \text{ psi}$

$\sigma\_min := \min(\sigma\_ptA_0) = -1.331 \times 10^4 \text{ psi}$

*Point A for lowest down position associates with σ_ptA_0

*Point D for highest up position associates with σ_ptD_{40}

Find Mean/Alternating Bending Stress

$$\sigma\_m := \frac{\sigma\_max + \sigma\_min}{2} = 2.941 \times 10^3 \text{ psi}$$

$$\sigma\_a := \left|\frac{\sigma\_max - \sigma\_min}{2}\right| = 1.625 \times 10^4 \text{ psi}$$

Find Maximum/Minimum Torsional Stress

$\tau\_max := \max(\tau\_ptD_{40}) = 1.194 \times 10^4 \text{ psi}$

$\tau\_min := \min(\tau\_ptA_0) = 2.159 \times 10^3 \text{ psi}$

Find Mean/Alternating Torsional Stress

$$\tau\_m := \frac{\tau\_max + \tau\_min}{2} = 7.05 \times 10^3 \text{ psi}$$

$$\tau\_a := \left|\frac{\tau\_max - \tau\_min}{2}\right| = 4.89 \times 10^3 \text{ psi}$$

Determine Fatigue Stress Concentration Factors

Table 7-1

First Iteration Estimates for Stress-Concentration Factors K_t and K_{ts}.

Warning: These factors are only estimates for use when actual dimensions are not yet determined. Do *not* use these once actual dimensions are available.

	Bending	Torsional	Axial
Shoulder fillet—sharp ($r/d = 0.02$)	2.7	2.2	3.0
Shoulder fillet—well rounded ($r/d = 0.1$)	1.7	1.5	1.9
End-mill keyseat ($r/d = 0.02$)	2.14	3.0	—
Sled runner keyseat	1.7	—	—
Retaining ring groove	5.0	3.0	5.0

$K_t := 1$

$K_{ts} := 1$

*Equals 1 since we do not have step downs or grooves

$q := 0$

$q_s := 0$

Look up q & qs values from Shigley graphs 6-20 & 6-21

$K_f := 1 + q\cdot(K_t - 1) = 1$ Bending Stress Concentration Factor (Shigley Eqn 6-32)

$K_{fs} := 1 + q_s\cdot(K_{ts} - 1) = 1$ Shear Stress Concentration Factor (Shigley Eqn 6-32)

Calculate Von Mesis Mean/Alternating Stress at Critical Points

$$\sigma'_a := \sqrt{(K_f\cdot\sigma_a)^2 + 3\cdot(K_{fs}\cdot\tau_a)^2} = 1.833\times 10^4\ \text{psi}$$

$$\sigma'_m := \sqrt{(K_f\cdot\sigma_m)^2 + 3\cdot(K_{fs}\cdot\tau_m)^2} = 1.256\times 10^4\ \text{psi}$$

Calculate Endurance Limit

$$S_e = k_a k_b k_c k_d k_e k_f S'_e \qquad (6\text{–}18)$$

where k_a = surface condition modification factor
k_b = size modification factor
k_c = load modification factor
k_d = temperature modification factor
k_e = reliability factor[13]
k_f = miscellaneous-effects modification factor
S'_e = rotary-beam test specimen endurance limit
S_e = endurance limit at the critical location of a machine part in the geometry and condition of use

$S'e := .5\cdot S\_ut = 4.95 \times 10^4\,psi$ Shigleys Eqn 6-8 *If Sut >200kspi S'e=100kpsi

Calculate Marin Factors

Table 6–2

Parameters for Marin Surface Modification Factor, Eq. (6–19)

Surface Finish	Factor a S_{ut}, kpsi	Factor a S_{ut}, MPa	Exponent b
Ground	1.34	1.58	−0.085
Machined or cold-drawn	2.70	4.51	−0.265
Hot-rolled	14.4	57.7	−0.718
As-forged	39.9	272.	−0.995

$a := 2.7$

$b := -.265$

$k\_a := a\cdot\left(S\_ut\cdot psi^{-1}\right)^{b} = 0.128$

$:= .879\cdot\left(d\ \ shaft\cdot in^{-1}\right)^{-.107}$ for .11in < d_shaft < 2in

$k\_b := .91\cdot\left(d\_shaft\cdot in^{-1}\right)^{-.157}$ for 2in < d_shaft < 10in

$k\_c := 1$ Equals 1 for combined loading case

$k\_d := 1$

$k\_e := .897$ 90% Reliability (Shigleys Table 6-5)

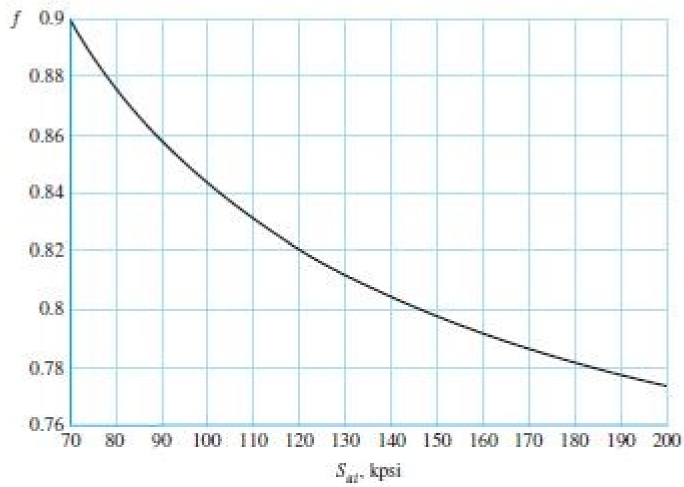
$k\_f := 1$

$Se := k\_a \cdot k\_b \cdot k\_c \cdot k\_d \cdot k\_e \cdot k\_f \cdot S'e = 4.416 \times 10^{3}\ psi$

Calculate Fatigue Strength

Figure 6–18

Fatigue strength fraction, f, of S_{ut} at 10^3 cycles for $S_e = S'_e = 0.5S_{ut}$ at 10^6 cycles.



$f := .845$

$$a\_2 := \frac{(f \cdot S\_ut)^2}{Se} = 1.585 \times 10^{6}\ psi$$

$$b\_2 := \frac{-1}{3} \cdot \log\left(\frac{f \cdot S\_ut}{Se}\right) = -0.426$$

$N\_cycles := 400$

$$S\_f := S\_ut \cdot N\_cycles^{\frac{\log(f)}{3}} = 8.554 \times 10^{4}\ psi$$

Check Langer Static Yield

$$SF\_y := \frac{S\_y}{\sigma'\_a + \sigma'\_m} = 2.493$$

Modified Goodman Safety Factor

$$\boxed{SF\_mg := \frac{1}{\left(\frac{\sigma'\_a}{S\_f}\right) + \left(\frac{\sigma'\_m}{S\_ut}\right)} = 2.931}$$

Check Stiffness Criteria

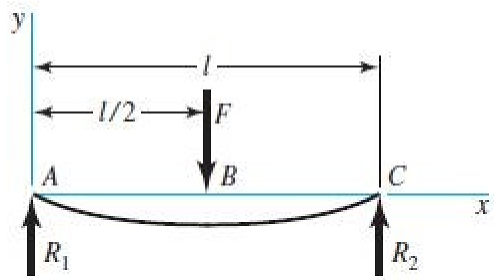
**Not used in our design, purely for future reference*

Table 7–2

Typical Maximum Ranges for Slopes and Transverse Deflections

Slopes	
Tapered roller	0.0005–0.0012 rad
Cylindrical roller	0.0008–0.0012 rad
Deep-groove ball	0.001–0.003 rad
Spherical ball	0.026–0.052 rad
Self-align ball	0.026–0.052 rad
Uncrowned spur gear	<0.0005 rad

5 Simple supports—center load



$l\_bearing := 11.5in$ Distance between bearing supports

$R\_x := V\_x$

$F\_x := 2 \cdot R\_x$

$$slope\_bearing\_x := \left| \frac{F\_x \cdot l\_bearing^2}{16 \cdot E\_shaft \cdot I\_shaft} \right|$$

$R\_y := V\_y$

$F\_y := 2 \cdot R\_y$

$$slope\_bearing\_y := \left| \frac{F\_y \cdot l\_bearing^2}{16 \cdot E\_shaft \cdot I\_shaft} \right|$$

$$slope\_bearing := \sqrt{slope\_bearing\_x^2 + slope\_bearing\_y^2}$$

$slope\_bearing\_max := max(slope\_bearing) = 0.015$

Retrieve Static Analysis Data

90 Degree Configuration

data :=

Forces and Moments - 90 Degree Position.xlsx

*for all direction and sign conventions refer to static analysis sheets

Parse Data into Variables

$V\_x := data^{\langle 3 \rangle} \cdot lbf$ Shear in x-direction

$V\_y := data^{\langle 1 \rangle} \cdot lbf$ Shear in y-direction

$M\_x := data^{\langle 4 \rangle} \cdot lbf \cdot in$ Moment in x-direction

$M\_y := data^{\langle 2 \rangle} \cdot lbf \cdot in$ Moment in y-direction

$T\_tot := data^{\langle 0 \rangle} \cdot lbf \cdot in$ Torque on shaft

Shaft Given Values

$d\_shaft := 2.75in$ Shaft Diameter

$A\_shaft := \frac{\pi}{4} \cdot d\_shaft^2 = 5.94\,in^2$ Shaft Area

$S\_ut := 99000psi$ Ultimate Strength (ie.1045)

$S\_y := 77000psi$ Yield Strength (ie 1045)

$E\_shaft := 29000000psi$ Youngs Modulus

$I\_shaft := \frac{\pi\, d\_shaft^4}{64}$ Shaft Moment of Inertia

Stresses in Shaft

$\tau\_t := \frac{16 \cdot T\_tot}{\pi \cdot d\_shaft^3}$ Torsional Shear

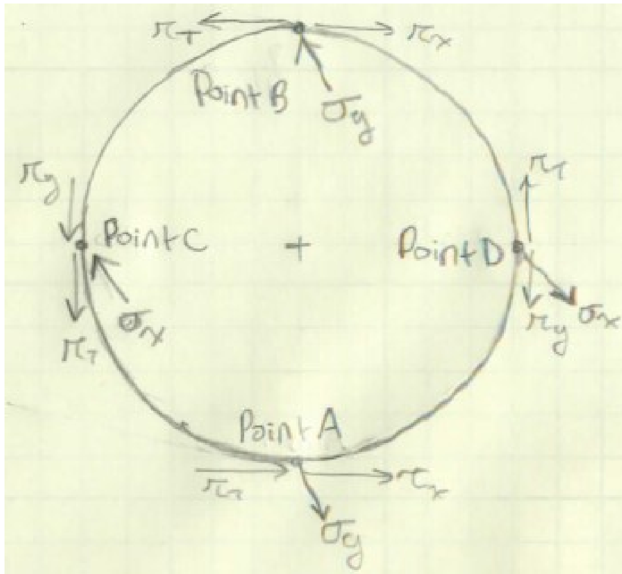
$\tau\_x := \frac{4 \cdot V\_x}{3 \cdot A\_shaft}$ Shear X-direction

$\tau\_y := \frac{4 \cdot V\_x}{3 \cdot A\_shaft}$ Shear Y-direction

$\sigma\_x := \frac{32 \cdot M\_x}{\pi \cdot d\_shaft^3}$ Bending X-direction

$\sigma\_y := \frac{32 \cdot M\_y}{\pi \cdot d\_shaft^3}$ Bending Y-direction

Combined Stresses at Critical Points



Point A:

$\tau\_ptA := \tau\_t + \tau\_x$

$\sigma\_ptA := \sigma\_y$

<u>Find Maximum Von Mesis Stress</u>

$$\sigma'\_ptA := \max\left[\sqrt{(\sigma\_ptA)^2 + 3\cdot(\tau\_ptA)^2}\right] = 3.401 \times 10^4\ \text{psi}$$

Point C:

$\tau\_ptC := \tau\_t + \tau\_y$

$\sigma\_ptC := \sigma\_x$

<u>Find Maximum Von Mesis Stress</u>

$$\sigma'\_ptC := \max\left[\sqrt{(\sigma\_ptC)^2 + 3\cdot(\tau\_ptC)^2}\right] = 2.46 \times 10^4\ \text{psi}$$

Distortion Energy Safety Factor

$$\sigma'\_max := \max(\sigma'\_ptC, \sigma'\_ptA) = 3.401 \times 10^4\ \text{psi}$$

**This is checking the static case safety factor on the maximum von mesis stress at each value of theta*

$$\boxed{SF\_de := \frac{S\_y}{\sigma'\_max} = 2.264}$$

Calculate the Alternating and Mean Stress

****Assume worst case in fatigue is when pt.A in down position rotates to pt. D in up position****

Point D:

$\tau\_ptD := \tau\_t - \tau\_y$

$\sigma\_ptD := \sigma\_x$

Maximum/Minimum Bending Stress

$\sigma\_max := max(\sigma\_ptD_{40}) = 1.981 \times 10^3\ psi$

$\sigma\_min := min(\sigma\_ptA_0) = -2.334 \times 10^4\ psi$

*Point A for lowest down position associates with σ_ptA_0

*Point D for highest up position associates with σ_ptD_{40}

Find Mean/Alternating Bending Stress

$$\sigma\_m := \frac{\sigma\_max + \sigma\_min}{2} = -1.068 \times 10^4\ psi$$

$$\sigma\_a := \left|\frac{\sigma\_max - \sigma\_min}{2}\right| = 1.266 \times 10^4\ psi$$

Find Maximum/Minimum Torsional Stress

$\tau\_max := max(\tau\_ptD_{40}) = 1.374 \times 10^4\ psi$

$\tau\_min := min(\tau\_ptA_0) = 429.213\ psi$

Find Mean/Alternating Torsional Stress

$$\tau\_m := \frac{\tau\_max + \tau\_min}{2} = 7.086 \times 10^3\ psi$$

$$\tau\_a := \left|\frac{\tau\_max - \tau\_min}{2}\right| = 6.657 \times 10^3\ psi$$

Determine Fatigue Stress Concentration Factors

Table 7-1

First Iteration Estimates for Stress-Concentration Factors K_t and K_{ts}.

Warning: These factors are only estimates for use when actual dimensions are not yet determined. Do *not* use these once actual dimensions are available.

	Bending	Torsional	Axial
Shoulder fillet—sharp ($r/d = 0.02$)	2.7	2.2	3.0
Shoulder fillet—well rounded ($r/d = 0.1$)	1.7	1.5	1.9
End-mill keyseat ($r/d = 0.02$)	2.14	3.0	—
Sled runner keyseat	1.7	—	—
Retaining ring groove	5.0	3.0	5.0

$K\_t := 1$

$K\_ts := 1$

*Equals 1 since we do not have step downs or grooves

$q := 0$

$q\_s := 0$

Look up q & qs values from Shigley graphs 6-20 & 6-21

$K\_f := 1 + q\cdot(K\_t - 1) = 1$ Bending Stress Concentration Factor (Shigley Eqn 6-32)

$K\_fs := 1 + q\_s\cdot(K\_ts - 1) = 1$ Shear Stress Concentration Factor (Shigley Eqn 6-32)

Calculate Von Mesis Mean/Alternating Stress at Critical Points

$$\sigma'\_a := \sqrt{(K\_f\cdot\sigma\_a)^2 + 3\cdot(K\_fs\cdot\tau\_a)^2} = 1.712\times 10^4\,\text{psi}$$

$$\sigma'\_m := \sqrt{(K\_f\cdot\sigma\_m)^2 + 3\cdot(K\_fs\cdot\tau\_m)^2} = 1.627\times 10^4\,\text{psi}$$

Calculate Endurance Limit

$$S_e = k_a k_b k_c k_d k_e k_f S'_e \qquad (6\text{–}18)$$

where k_a = surface condition modification factor
k_b = size modification factor
k_c = load modification factor
k_d = temperature modification factor
k_e = reliability factor[13]
k_f = miscellaneous-effects modification factor
S'_e = rotary-beam test specimen endurance limit
S_e = endurance limit at the critical location of a machine part in the geometry and condition of use

$S'e := .5 \cdot S\_ut = 4.95 \times 10^{4}\,\text{psi}$ Shigleys Eqn 6-8   *If Sut >200kspi S'e=100kpsi

Calculate Marin Factors

Table 6–2
Parameters for Marin Surface Modification Factor, Eq. (6–19)

Surface Finish	Factor a S_{ut}, kpsi	Factor a S_{ut}, MPa	Exponent b
Ground	1.34	1.58	−0.085
Machined or cold-drawn	2.70	4.51	−0.265
Hot-rolled	14.4	57.7	−0.718
As-forged	39.9	272.	−0.995

$a := 2.7$

$b := -.265$

$k\_a := a \cdot \left(S\_ut \cdot psi^{-1}\right)^{b} = 0.128$

$:= .879 \cdot \left(d\ shaft \cdot in^{-1}\right)^{-.107}$ for .11in < d_shaft < 2in

$k\_b := .91 \cdot \left(d\_shaft \cdot in^{-1}\right)^{-.157}$ for 2in < d_shaft < 10in

$k\_c := 1$ Equals 1 for combined loading case

$k\_d := 1$

$k\_e := .897$ 90% Reliability (Shigleys Table 6-5)

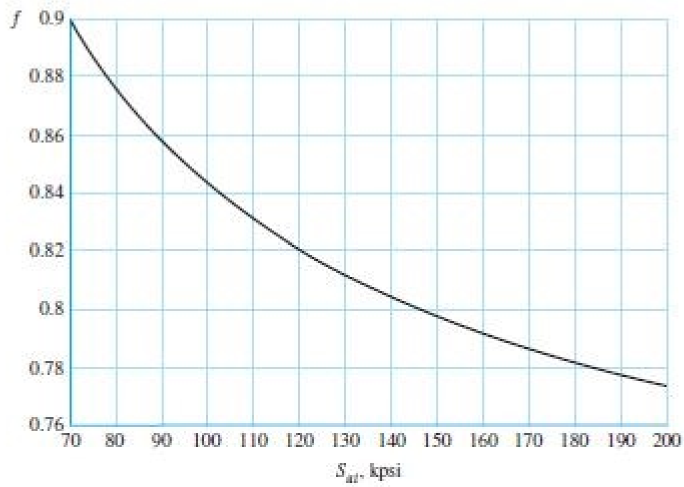
$k\_f := 1$

$$Se := k\_a \cdot k\_b \cdot k\_c \cdot k\_d \cdot k\_e \cdot k\_f \cdot S'e = 4.416 \times 10^{3}\ psi$$

Calculate Fatigue Strength

Figure 6–18

Fatigue strength fraction, f, of S_{ut} at 10^3 cycles for $S_e = S'_e = 0.5S_{ut}$ at 10^6 cycles.



$$f := .845$$

$$a\_2 := \frac{(f \cdot S\_ut)^2}{Se} = 1.585 \times 10^{6}\ psi$$

$$b\_2 := \frac{-1}{3} \cdot \log\left(\frac{f \cdot S\_ut}{Se}\right) = -0.426$$

$$N\_cycles := 400$$

$$S\_f := S\_ut \cdot N\_cycles^{\frac{\log(f)}{3}} = 8.554 \times 10^{4}\ psi$$

Check Langer Static Yield

$$SF\_y := \frac{S\_y}{\sigma'\_a + \sigma'\_m} = 2.306$$

Modified Goodman Safety Factor

$$\boxed{SF\_mg := \frac{1}{\left(\frac{\sigma'\_a}{S\_f}\right) + \left(\frac{\sigma'\_m}{S\_ut}\right)} = 2.743}$$

Check Stiffness Criteria

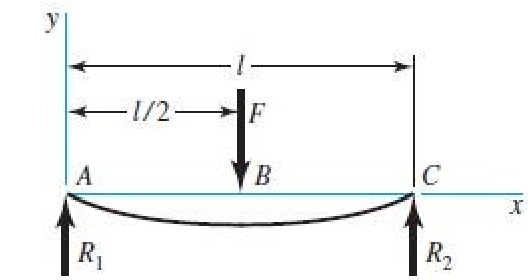
**Not used in our design, purely for future reference*

Table 7–2

Typical Maximum Ranges for Slopes and Transverse Deflections

Slopes	
Tapered roller	0.0005–0.0012 rad
Cylindrical roller	0.0008–0.0012 rad
Deep-groove ball	0.001–0.003 rad
Spherical ball	0.026–0.052 rad
Self-align ball	0.026–0.052 rad
Uncrowned spur gear	<0.0005 rad

5 Simple supports—center load



l_bearing := 11.5in   Distance between bearing supports

R_x := V_x

F_x := 2·R_x

$$slope\_bearing\_x := \left|\frac{F\_x \cdot l\_bearing^2}{16 \cdot E\_shaft \cdot I\_shaft}\right|$$

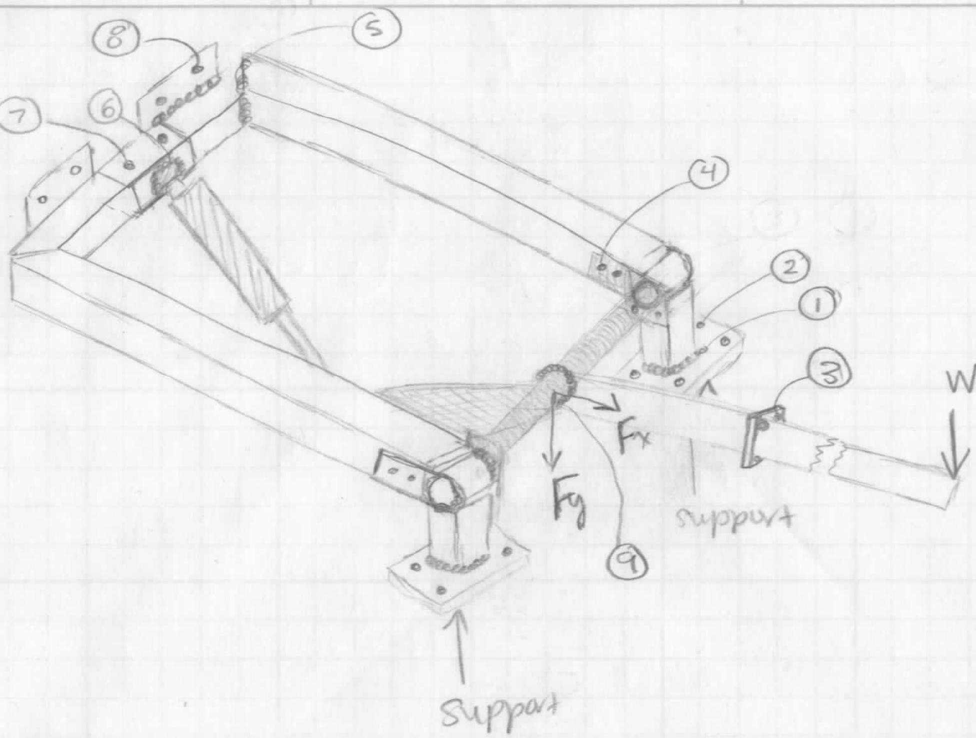
R_y := V_y

F_y := 2·R_y

$$slope\_bearing\_y := \left|\frac{F\_y \cdot l\_bearing^2}{16 \cdot E\_shaft \cdot I\_shaft}\right|$$

$$slope\_bearing := \sqrt{slope\_bearing\_x^2 + slope\_bearing\_y^2}$$

slope_bearing_max := max(slope_bearing) = 0.014

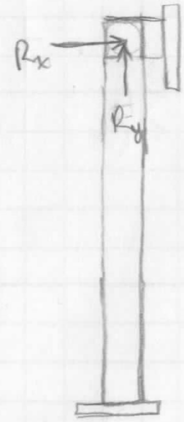


① weld between matchplate and leg
③ weld between pipe and plate
④ Bolts holding plate to frame
⑥ weld holding cylinder to U-shape
⑨ weld between frame and matchplate

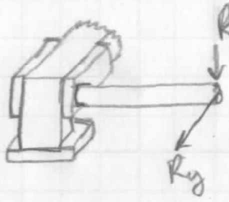
Critical Points to Analyze

0° Position		45° Position		90° Position	
3	9	3	✓1	3	1
✓4	1	✓4	✓9	✓4	9
✓6		✓6		✓6	

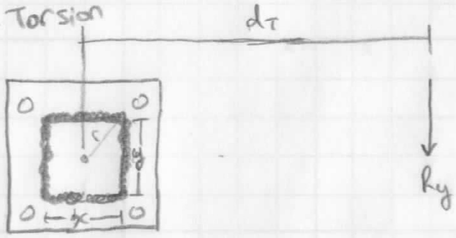
Case ① 90°



Case ① 0°
★ switch Rx + Ry



Ry - Torsion



$b = x$
$a = y$
$V_T = R_y$
$M_T = V_T d_T$

Table 9-1
$$\left\{ \begin{array}{l} A = 1.414\,h(b+d) \\ J_u = \dfrac{(b+d)^3}{6} \end{array} \right.$$

$J = .707\,h\,J_u$

Input:
r = distance from center of weld to weld outer
h = height of weld
b = length of side 1
a = length of side 2

$$\tau'_T = \frac{V_T}{A}$$

$$\tau''_T = \frac{M_T r}{J}$$

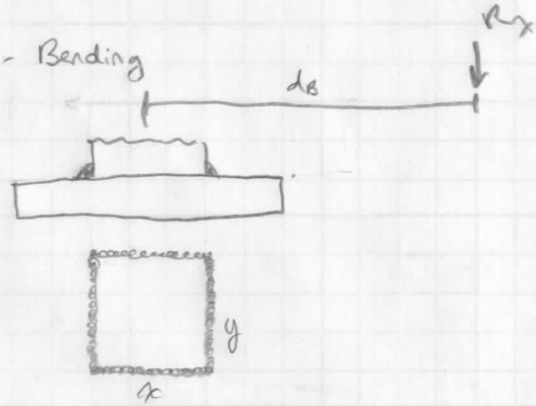
$$\tau_T = \sqrt{\tau'^2_T + \tau''^2_T}$$

$$S_{sy} = .577\,S_y$$

$$n_T = \frac{S_{sy}}{\tau_T}$$

$x = 4$ in
$y = 4$ in
$d_T = d_B = 9.5$ in

Rx - Bending



$b = y \quad d = x$

$V_B = R_x$
$M_B = R_x d_B$

$A_B = 1.414\,h\,(d+b)$

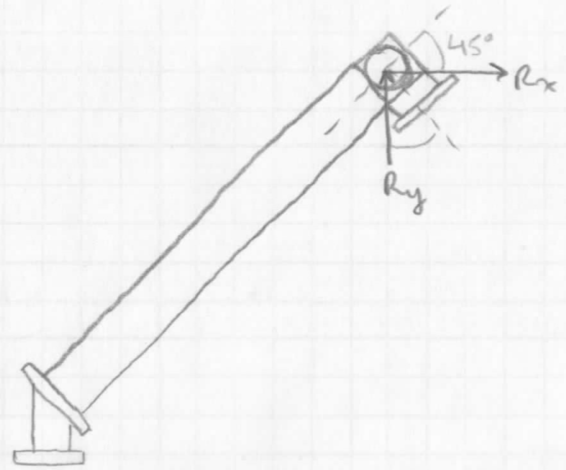
$$I_u = \frac{d^2}{6}(3b+d)$$

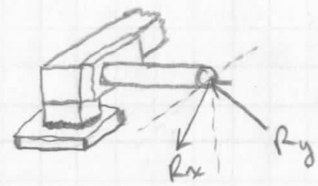
$I = .707\,h\,I_u$

$$\tau' = \frac{V_B}{A_B} \qquad \tau'' = \frac{1.414\,M_B}{bdh}$$

$$\tau_B = \sqrt{\tau'^2 + \tau''^2} \qquad n = \frac{S_{sy}}{\tau_B}$$

Case ① 45°





Torsion:

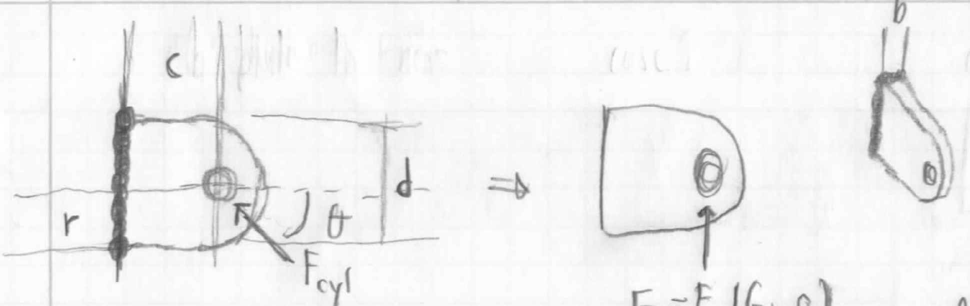
$V_t = R_x \cos 45 + R_y \sin 45$

Follow same steps as Case ① 90° Ry Torsion

Bending:

$V_B = R_y \cos 45 - R_x \sin 45$

Follow same steps as Case ① 90° Rx Bending



$F = F_{cyl}(\sin\theta)$

$M = Fc$

$l_u = \frac{a^2}{6}(3b + a)$

$l = .707 h l_u$

$A = 1.414 h (b + a)$

$S_{sy} = .577 S_y$

case 1

$$\tau' = \frac{F}{A} \qquad \tau'' = \frac{Mr}{l} \qquad \tau = \left(\tau'^2 + \tau''^2\right)^{1/2}$$

$$n = \frac{S_{sy}}{\tau}$$

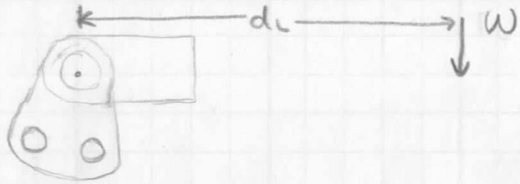
$$\Rightarrow n = \frac{.577 S_y}{\left(\left(\frac{F_{cyl}(\sin\theta)}{1.414h(b+d)}\right)^2 + \left(\frac{F_{cyl}(\sin\theta)\,cr}{.707h\frac{d^2}{6}(3b+d)}\right)^2\right)^{1/2}}$$

c = 4.00 in

r = 2.50 in

a = 5.00 in

b = 1.00 in



$W = 1500\,lbf$
$d_L = 10'$
$r = 1.7''$
$S_{sy} = .577 S_y$

$S_y = 57000$ psi

Torsional

$V_t = W$
$M_t = d_L W$

$A_t = 1.414\, h \pi r$

$J_u = 2\pi r^3$

$J_t = .707\, h\, J_u$

$\tau'_t = \frac{V_t}{A_t}$

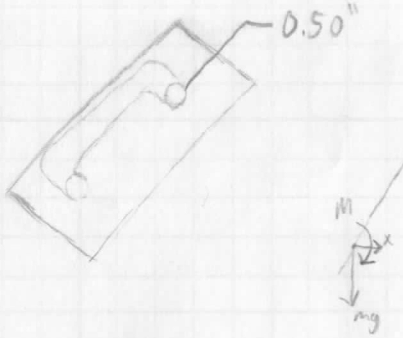
$\tau''_t = \frac{M_{-t} \cdot r}{J_t}$

$\tau = \sqrt{\tau'^2 + \tau''^2}$

$\tau_{max} = max(\tau)$

$SF_{-t} = \frac{S_{sy}}{\tau_{max}}$

Drawing:



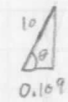
Given:

MTL: ASTM 4041 Steel

Weld: 3/8" Fillet Weld All around x 2, 7018 Rod

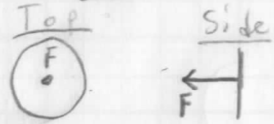
$S_{yt} = 70{,}000$ psi

$\theta = \cos^{-1}\left(\frac{.109}{10}\right)$

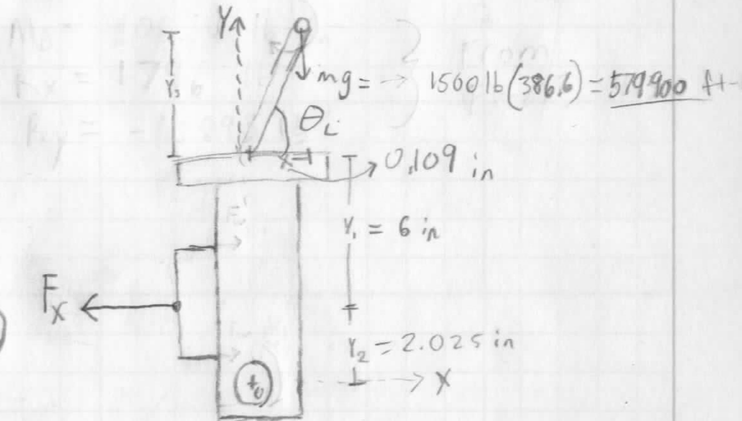


$\theta = 89.4°$

Rod



Structure:



$\Sigma M_o = 0;$

$0 = -(mg)(x) + F_x(Y_2)$

$0 = -\underset{[lb]}{(579900)}\underset{[in]}{(0.109)} + F_x\underset{[in]}{(2.025)}$

$\underline{F_x = 31214\ lb}$

Solve:

Largest Von Mises: $\sigma' = \dfrac{2.16\,F}{h\,L}$

$L = 2 \cdot 2\pi r$

$= 4\pi(.25)$

$L = \underline{3.14\ in}$

$h = 3/8" = \underline{0.375\ in}$

$F = \underline{31214\ lb}$

$\sigma' = \dfrac{2.16\,(31214\ lb)}{(0.375\ in)(3.14\ in)}$

$\sigma' = 57258\ lb/in^2$

$= 58000$ psi

< 70 K psi from 7018 Rod

✓ 0.5 in Safety Rod is okay in axial tension

Retrieve Static Analysis Data

0 Degree Configuration

static_data_0deg :=

Welds - 0 Degree Position.xlsx

*for all direction and sign conventions refer to static analysis sheets

Parse Data into Variables

$R\_x := static\_data\_0deg^{\langle 2\rangle} \cdot lbf$

$R\_y := static\_data\_0deg^{\langle 1\rangle} \cdot lbf$

$theta\_c := static\_data\_0deg^{\langle 0\rangle} \cdot rad$

$F\_cyl := 23561.9449 \cdot lbf$

Retrieve Welds Analysis Data

welds_data_0deg :=

Welds - 0 Degree Position.xlsx

Parse Data into Variables

$d\_b := welds\_data\_0deg^{\langle 0\rangle} \cdot in$

$d\_t := welds\_data\_0deg^{\langle 1\rangle} \cdot in$

$b := welds\_data\_0deg^{\langle 2\rangle} \cdot in$

$a := welds\_data\_0deg^{\langle 3\rangle} \cdot in$

$r\_w := welds\_data\_0deg^{\langle 4\rangle} \cdot in$

$h := .375in$

Strength of Weld Material

$S\_y := 57000 \cdot psi$ Electrode Yield Strength (E70xx)

$S\_sy := .577 \cdot S\_y = 3.289 \times 10^{4} psi$

Case 1 Torsional Stress

$V\_t\_1 := R\_x$

$M\_t\_1 := V\_t\_1 \cdot d\_t_0$

$A\_t\_1 := 1.414 \cdot h \cdot \left(b_0 + a_0\right)$

$$J\_u\_1 := \frac{\left(b_0 + a_0\right)^3}{6}$$

$J\_t\_1 := .707 \cdot h\, J\_u\_1$

$$\tau'\_t\_1 := \frac{V\_t\_1}{A\_t\_1}$$

$$\tau''\_t\_1 := \frac{M\_t\_1 \cdot r\_w_0}{J\_t\_1}$$

$$\tau\_t\_1 := \sqrt{\tau'\_t\_1^2 + \tau''\_t\_1^2}$$

$\tau\_t\_max\_1 := \max(\tau\_t\_1)$

$$\boxed{SF\_t\_1 := \frac{S\_sy}{\tau\_t\_max\_1} = 4.024}$$

Case 1 Bending Stress

$V\_b\_1 := R\_y$

$M\_b\_1 := V\_b\_1 \cdot d\_b_1$

$A\_b\_1 := 1.414 \cdot h \cdot \left(b_1 + a_1\right)$

$$I\_u\_1 := \frac{\left(a_1\right)^2 \left(3 \cdot b_1 + a_1\right)}{6}$$

$I\_b\_1 := .707h \cdot I\_u\_1$

$$\tau'\_b\_1 := \frac{V\_b\_1}{A\_b\_1}$$

$$\tau''\_b\_1 := \frac{M\_b\_1 \cdot r\_w_1}{I\_b\_1}$$

$$\tau\_b\_1 := \sqrt{\tau'\_b\_1^2 + \tau''\_b\_1^2}$$

$\tau\_b\_1\_max := max(\tau\_b\_1)$

$$\boxed{SF\_b\_1 := \frac{S\_sy}{\tau\_b\_1\_max} = 9.288}$$

Case 6 Bending

$V\_b\_6 := F\_cyl \cdot sin(theta\_c)$

$M\_b\_6 := V\_b\_6 \cdot d\_b_2$

$A\_b\_6 := 1.414 \cdot h \cdot \left(b_2 + a_2\right)$

$$I\_u\_6 := \frac{\left(a_2\right)^2 \left(3 \cdot b_2 + a_2\right)}{6}$$

$I\_b\_6 := .707h \cdot I\_u\_6$

$$\tau'\_b\_6 := \frac{V\_b\_6}{A\_b\_6}$$

$$\tau''\_b\_6 := \frac{M\_b\_6 \cdot r\_w_2}{I\_b\_6}$$

$$\tau\_b\_6 := \sqrt{\tau'\_b\_6^2 + \tau''\_b\_6^2}$$

$\tau\_b\_max\_6 := max(\tau\_b\_6)$

$$\boxed{SF\_6 := \frac{S\_sy}{\tau\_b\_max\_6} = 22.517}$$

Retrieve Static Analysis Data

45 Degree Configuration

static_data_0deg :=

Welds - 45 Degree Position.xlsx

*for all direction and sign conventions refer to static analysis sheets

Parse Data into Variables

$R\_x := static\_data\_0deg^{\langle 2 \rangle} \cdot lbf$

$R\_y := static\_data\_0deg^{\langle 1 \rangle} \cdot lbf$

$theta\_c := static\_data\_0deg^{\langle 0 \rangle} \cdot rad$

$F\_cyl := 23561.9449 \cdot lbf$

Retrieve Welds Analysis Data

welds_data_0deg :=

Welds - 45 Degree Position.xlsx

Parse Data into Variables

$d\_b := welds\_data\_0deg^{\langle 0 \rangle} \cdot in$

$d\_t := welds\_data\_0deg^{\langle 1 \rangle} \cdot in$

$b := welds\_data\_0deg^{\langle 2 \rangle} \cdot in$

$a := welds\_data\_0deg^{\langle 3 \rangle} \cdot in$

$r\_w := welds\_data\_0deg^{\langle 4 \rangle} \cdot in$

$h := .375in$

Strength of Weld Material

$S\_y := 57000 \cdot psi$ Electrode Yield Strength (E70xx)

$S\_sy := .577 \cdot S\_y = 3.289 \times 10^{4} psi$

Case 1 Torsional Stress

$$V\_t\_1 := R\_x \cdot \cos\left(\frac{\pi}{4}\right) + R\_y \cdot \sin\left(\frac{\pi}{4}\right)$$

$$M\_t\_1 := V\_t\_1 \cdot d\_t_0$$

$$A\_t\_1 := 1.414 \cdot h \cdot \left(b_0 + a_0\right)$$

$$J\_u\_1 := \frac{\left(b_0 + a_0\right)^3}{6}$$

$$J\_t\_1 := .707 \cdot h\, J\_u\_1$$

$$\tau'\_t\_1 := \frac{V\_t\_1}{A\_t\_1}$$

$$\tau''\_t\_1 := \frac{M\_t\_1 \cdot r\_w_0}{J\_t\_1}$$

$$\tau\_t\_1 := \sqrt{\tau'\_t\_1^2 + \tau''\_t\_1^2}$$

$$\tau\_t\_max\_1 := \max(\tau\_t\_1)$$

$$\boxed{SF\_t\_1 := \frac{S\_sy}{\tau\_t\_max\_1} = 19.564}$$

Case 1 Bending Stress

$$V\_b\_1 := R\_y \cdot \cos\left(\frac{\pi}{4}\right) - R\_x \cdot \sin\left(\frac{\pi}{4}\right)$$

$$M\_b\_1 := V\_b\_1 \cdot d\_b_1$$

$$A\_b\_1 := 1.414 \cdot h \cdot \left(b_1 + a_1\right)$$

$$I\_u\_1 := \frac{\left(a_1\right)^2 \left(3 \cdot b_1 + a_1\right)}{6}$$

$$I\_b\_1 := .707h \cdot I\_u\_1$$

$$\tau'\_b\_1 := \frac{V\_b\_1}{A\_b\_1}$$

$$\tau''\_b\_1 := \frac{M\_b\_1 \cdot r\_w_1}{I\_b\_1}$$

$$\tau\_b\_1 := \sqrt{\tau'\_b\_1^2 + \tau''\_b\_1^2}$$

$$\tau\_b\_1\_max := \max(\tau\_b\_1)$$

$$\boxed{SF\_b\_1 := \frac{S\_sy}{\tau\_b\_1\_max} = 2.183}$$

Case 6 Bending

$$V\_b\_6 := F\_cyl \cdot \sin(theta\_c)$$

$$M\_b\_6 := V\_b\_6 \cdot d\_b_2$$

$$A\_b\_6 := 1.414 \cdot h \cdot \left(b_2 + a_2\right)$$

$$I\_u\_6 := \frac{\left(a_2\right)^2 \left(3 \cdot b_2 + a_2\right)}{6}$$

$$I\_b\_6 := .707h \cdot I\_u\_6$$

$$\tau'\_b\_6 := \frac{V\_b\_6}{A\_b\_6}$$

$$\tau''\_b\_6 := \frac{M\_b\_6 \cdot r\_w_2}{I\_b\_6}$$

$$\tau\_b\_6 := \sqrt{\tau'\_b\_6^2 + \tau''\_b\_6^2}$$

$$\tau\_b\_max\_6 := \max(\tau\_b\_6)$$

$$\boxed{SF\_6 := \frac{S\_sy}{\tau\_b\_max\_6} = 22.517}$$

Retrieve Static Analysis Data

90 Degree Configuration

static_data_0deg :=

Welds - 90 Degree Position.xlsx

*for all direction and sign conventions refer to static analysis sheets

Parse Data into Variables

$R\_x := static\_data\_0deg^{\langle 2 \rangle} \cdot lbf$

$R\_y := static\_data\_0deg^{\langle 1 \rangle} \cdot lbf$

$theta\_c := static\_data\_0deg^{\langle 0 \rangle} \cdot rad$

$F\_cyl := 23561.9449 \cdot lbf$

Retrieve Welds Analysis Data

welds_data_0deg :=

Welds - 90 Degree Position.xlsx

Parse Data into Variables

$d\_b := welds\_data\_0deg^{\langle 0 \rangle} \cdot in$

$d\_t := welds\_data\_0deg^{\langle 1 \rangle} \cdot in$

$b := welds\_data\_0deg^{\langle 2 \rangle} \cdot in$

$a := welds\_data\_0deg^{\langle 3 \rangle} \cdot in$

$r\_w := welds\_data\_0deg^{\langle 4 \rangle} \cdot in$

$h := .375in$

Strength of Weld Material

$S\_y := 57000 \cdot psi$ Electrode Yield Strength (E70xx)

$S\_sy := .577 \cdot S\_y = 3.289 \times 10^4 psi$

Case 1 Torsional Stress

$V\_t\_1 := R\_y$

$M\_t\_1 := V\_t\_1 \cdot d\_t_0$

$A\_t\_1 := 1.414 \cdot h \cdot \left(b_0 + a_0\right)$

$$J\_u\_1 := \frac{\left(b_0 + a_0\right)^3}{6}$$

$J\_t\_1 := .707 \cdot h\, J\_u\_1$

$$\tau'\_t\_1 := \frac{V\_t\_1}{A\_t\_1}$$

$$\tau''\_t\_1 := \frac{M\_t\_1 \cdot r\_w_0}{J\_t\_1}$$

$$\tau\_t\_1 := \sqrt{\tau'\_t\_1^2 + \tau''\_t\_1^2}$$

$\tau\_t\_max\_1 := \max(\tau\_t\_1)$

$$\boxed{SF\_t\_1 := \frac{S\_sy}{\tau\_t\_max\_1} = 4.299}$$

Case 1 Bending Stress

$V\_b\_1 := R\_x$

$M\_b\_1 := V\_b\_1 \cdot d\_b_1$

$A\_b\_1 := 1.414 \cdot h \cdot \left(b_1 + a_1\right)$

$$I\_u\_1 := \frac{\left(a_1\right)^2 \left(3 \cdot b_1 + a_1\right)}{6}$$

$I\_b\_1 := .707h \cdot I\_u\_1$

$$\tau'\_b\_1 := \frac{V\_b\_1}{A\_b\_1}$$

$$\tau''\_b\_1 := \frac{M\_b\_1 \cdot r\_w_1}{I\_b\_1}$$

$$\tau\_b\_1 := \sqrt{\tau'\_b\_1^2 + \tau''\_b\_1^2}$$

$\tau\_b\_1\_max := max(\tau\_b\_1)$

$$SF\_b\_1 := \frac{S\_sy}{\tau\_b\_1\_max} = 12.982$$

Case 6 Bending

$V\_b\_6 := F\_cyl \cdot sin(theta\_c)$

$M\_b\_6 := V\_b\_6 \cdot d\_b_2$

$A\_b\_6 := 1.414 \cdot h \cdot \left(b_2 + a_2\right)$

$$I\_u\_6 := \frac{\left(a_2\right)^2 \left(3 \cdot b_2 + a_2\right)}{6}$$

$I\_b\_6 := .707h \cdot I\_u\_6$

$$\tau'\_b\_6 := \frac{V\_b\_6}{A\_b\_6}$$

$$\tau''\_b\_6 := \frac{M\_b\_6 \cdot r\_w_2}{I\_b\_6}$$

$$\tau\_b\_6 := \sqrt{\tau'\_b\_6^2 + \tau''\_b\_6^2}$$

$\tau\_b\_max\_6 := max(\tau\_b\_6)$

$$SF\_6 := \frac{S\_sy}{\tau\_b\_max\_6} = 22.517$$

Truss Attachment Weld Analysis

Given Variables

$W\_l := 1500 \cdot lbf$

$d\_l := 120 \cdot in$

$r\_w := 1.7 \cdot in$

$h := .375in$

Strength of Weld Material

$S\_y := 57000 \cdot psi$ Electrode Yield Strength (E70xx)

$S\_sy := .577 \cdot S\_y = 3.289 \times 10^{4} psi$

Case 1 Torsional Stress

$$V\_t := \frac{W\_l}{4}$$

$M\_t := V\_t \cdot d\_l$

$A\_t := 1.414 \cdot h \cdot \pi\, r\_w$

$J\_u := 2 \cdot \pi \cdot r\_w^{3}$

$J\_t := .707 \cdot h\, J\_u$

$$\tau'\_t := \frac{V\_t}{A\_t}$$

$$\tau''\_t := \frac{M\_t \cdot r\_w}{J\_t}$$

$$\tau\_t := \sqrt{\tau'\_t^{2} + \tau''\_t^{2}}$$

$\tau\_t\_max := \max(\tau\_t)$

$$\boxed{SF\_t := \frac{S\_sy}{\tau\_t\_max} = 3.518}$$

Case 1 Bending Stress

$$V\_b\_1 := R\_y$$

$$M\_b\_1 := V\_b\_1 \cdot d\_b_1$$

$$A\_b\_1 := 1.414 \cdot h \cdot \left(b_1 + a_1\right)$$

$$I\_u\_1 := \frac{\left(a_1\right)^2 \left(3 \cdot b_1 + a_1\right)}{6}$$

$$I\_b\_1 := .707h \cdot I\_u\_1$$

$$\tau'\_b\_1 := \frac{V\_b\_1}{A\_b\_1}$$

$$\tau''\_b\_1 := \frac{M\_b\_1 \cdot r\_w_1}{I\_b\_1}$$

$$\tau\_b\_1 := \sqrt{\tau'\_b\_1^2 + \tau''\_b\_1^2}$$

$\tau\_b\_1\_max := \max(\tau\_b\_1)$

$$SF\_b\_1 := \frac{S\_sy}{\tau\_b\_1\_max} = \blacksquare$$

Case 6 Bending

$V\_b\_6 := F\_cyl \cdot \sin(theta\_c)$

$M\_b\_6 := V\_b\_6 \cdot d\_b_2$

$A\_b\_6 := 1.414 \cdot h \cdot \left(b_2 + a_2\right)$

$$I\_u\_6 := \frac{\left(a_2\right)^2 \left(3 \cdot b_2 + a_2\right)}{6}$$

$I\_b\_6 := .707h \cdot I\_u\_6$

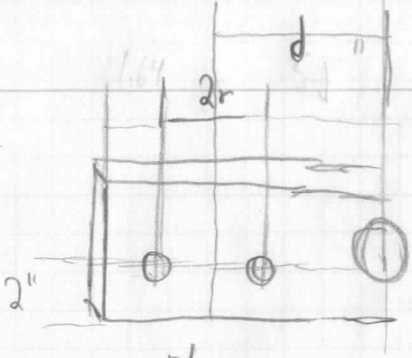
$$\tau'\_b\_6 := \frac{V\_b\_6}{A\_b\_6}$$

$$\tau''\_b\_6 := \frac{M\_b\_6 \cdot r\_w_2}{I\_b\_6}$$

$$\tau\_b\_6 := \sqrt{\tau'\_b\_6^2 + \tau''\_b\_6^2}$$

$\tau\_b\_max\_6 := \max(\tau\_b\_6)$

$$SF\_6 := \frac{S\_sy}{\tau\_b\_max\_6} = \blacksquare$$

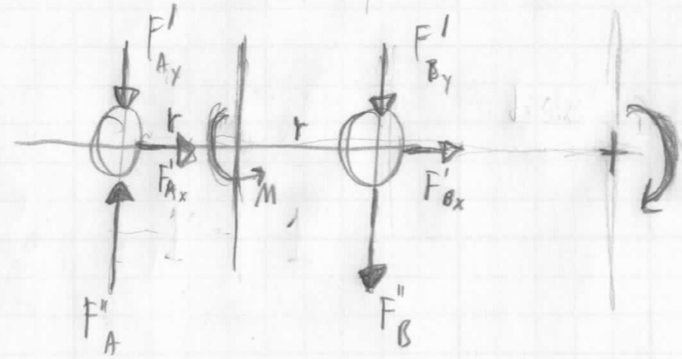


$S_{sy} = .577 S_y$ $\quad A_s = \frac{\pi \varnothing^2}{4}$

$V = F_y$ $\quad F'_y = \frac{V}{n} = \frac{F_y}{2}$

$r = 1''$

$n = \#$ of bolts $= 2$



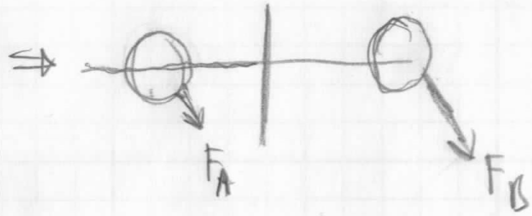
$F'_{A_y} = F'_{B_y} = F'_y$

$M = F_y(d)$

$F''_B = F''_A = F'' = \frac{M}{2r}$

$F_B = \sqrt{(F'' + F'_y)^2 + F'^2_{Bx}}$

$F_A = \sqrt{\left(|F'' - F'_y|\right)^2 + F'^2_{A_x}}$



$\tau_{max} = \frac{F_{max}}{A_s}$ $\quad n = \frac{S_{sy}}{\tau_{max}}$ Bolt shear

0 Degree Configuration Bolt Analysis

*for all direction and sign conventions refer to static analysis sheets

Retrieve Static Analysis Data

data :=
Forces and Moments 0 Degree Position.xlsx

Parse Data into Variables

$R\_y := data^{\langle 0 \rangle} \cdot lbf$ Force in y-direction

$R\_x := data^{\langle 1 \rangle} \cdot lbf$ Force in x-direction

Calculate Forces on Bolts

$S\_y := 130000psi$ Yield Strength (Grade 8 bolt)

$S\_sy := .577 \cdot S\_y$

$n := 2$

$d\_bolt := 1.00in$

$$A\_bolt := \frac{\pi \cdot d\_bolt^2}{4}$$

$d := 6.25in$

$r\_a := 2.25in$

$r\_b := r\_a$

$$F'\_x := \frac{R\_x}{n}$$

$$F'\_a := \frac{R\_y}{n}$$

$M := R\_y \cdot d$

$$F''\_a := \frac{(M \cdot r\_a)}{r\_a^2 + r\_b^2}$$

$$F\_a := \sqrt{(F'\_a + F''\_a)^2 + F'\_x^2}$$

$$\tau\_a := \frac{max(F\_a)}{A\_bolt}$$

$$\boxed{SF\_bolt := \frac{S\_sy}{\tau\_a} = 7.696}$$

45 Degree Configuration Bolt Analysis

*for all direction and sign conventions refer to static analysis sheets

Retrieve Static Analysis Data

data :=

Forces and Moments - 45 Degree Position.xlsx

Parse Data into Variables

$R\_y := data^{\langle 0 \rangle} \cdot lbf$ Force in y-direction

$R\_x := data^{\langle 1 \rangle} \cdot lbf$ Force in x-direction

Calculate Forces on Bolts

$S\_y := 130000psi$ Yield Strength (Grade 8 bolt)

$S\_sy := .577 \cdot S\_y$

$n := 2$

$d\_bolt := 1.00in$

$$A\_bolt := \frac{\pi \cdot d\_bolt^2}{4}$$

$d := 6.25in$

$r\_a := 2.25in$

$$F'\_x := \frac{R\_x \cdot \cos\left(\frac{\pi}{4}\right) + R\_y \cdot \sin\left(\frac{\pi}{4}\right)}{n}$$

$$F'\_a := \frac{\left(R\_y \cdot \cos\left(\frac{\pi}{4}\right) - R\_x \cdot \cos\left(\frac{\pi}{4}\right)\right)}{n}$$

$$M := \left(R\_y \cdot \cos\left(\frac{\pi}{4}\right) - R\_x \cdot \cos\left(\frac{\pi}{4}\right)\right) \cdot d$$

$$F''\_a := \frac{(M \cdot r\_a)}{r\_a^2 + r\_b^2}$$

$$F\_a := \sqrt{(F'\_a + F''\_a)^2 + F'\_x^2}$$

$$\tau\_a := \frac{\max(F\_a)}{A\_bolt}$$

$$\boxed{SF\_bolt := \frac{S\_sy}{\tau\_a} = 2.78}$$

90 Degree Configuration Bolt Analysis

*for all direction and sign conventions refer to static analysis sheets

Retrieve Static Analysis Data

data := Forces and Moments - 90 Degree Position.xlsx

Parse Data into Variables

$R\_y := data^{\langle 0 \rangle} \cdot lbf$ Force in y-direction

$R\_x := data^{\langle 1 \rangle} \cdot lbf$ Force in x-direction

Calculate Forces on Bolts

$S\_y := 130000psi$ Yield Strength (Grade 8 bolt)

$S\_sy := .577 \cdot S\_y$

$n := 2$

$d\_bolt := 1.00in$

$$A\_bolt := \frac{\pi \cdot d\_bolt^2}{4}$$

$d := 6.25in$

$r\_a := 2.25in$

$r\_b := r\_a$

$$F'\_x := \frac{R\_y}{n}$$

$$F'\_a := \frac{R\_x}{n}$$

$M := R\_x \cdot d$

$$F''\_a := \frac{(M \cdot r\_a)}{r\_a^2 + r\_b^2}$$

$$F\_a := \sqrt{(F'\_a + F''\_a)^2 + F'\_x^2}$$

$$\tau\_a := \frac{\max(F\_a)}{A\_bolt}$$

$$\boxed{SF\_bolt := \frac{S\_sy}{\tau\_a} = 9.061}$$

FASTENERS (Tensile Loading)

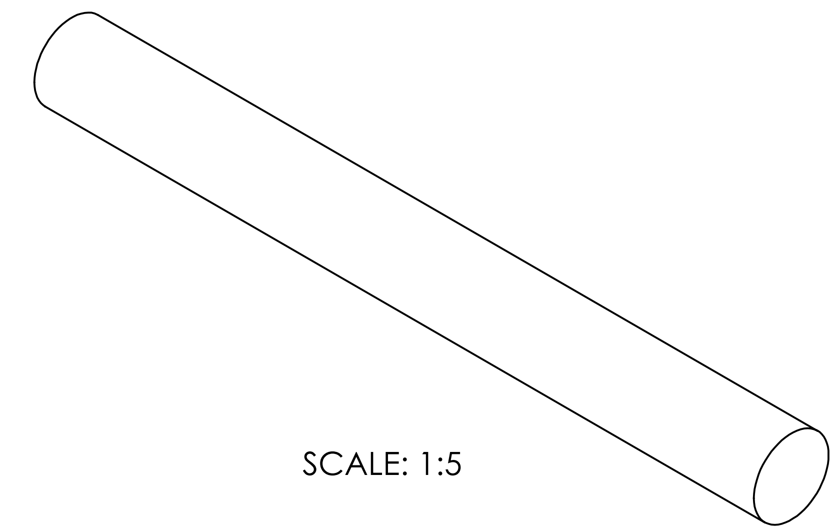
Description	Variable (Units)	Value	Notes/Comment
Number of fasteners	N	8	Fixed
Fastener diameter	d (in)	3/4	
Washer thickness	t (in)	0.148	used W (Wide) from Table A-32, pg 1064
Grip length	l (in)	1 1/2	measure to check and update
Nut thickness	H (in)	41/64	Table A-31, regular hexagonal, pg 1063
Head Width	W (in), D_m (in)	1 1/8	Table A-31, pg 1063
Fastener Length	L (in)	2	
Threaded Length	L_T (in)	3/4	
Length of unthreaded portion in grip	l_d (in)	1 1/4	
Length of threaded portion in grip	l_t (in)	1/4	Table 8-7, figure a, pg 418
Area of unthreaded portion	A_d (in^2)	0.44	
Area of threaded portion (coarse)	A_t (in^2)	0.302	Table 8-2, pg 405 coarse, **coarse currently used in equations**
Area of threaded portion (fine)	A_t (in^2)	0.351	Table 8-2, pg 405 fine
Young's Modulus	E (lbf/in^2)	30000000	medium carbon steel
Fastener stiffness	k_b (lbf/in)	8202916.575	eqn 8-17, pg 419
Half apex angle	α (deg)	30	used in Shigley's
Diameter of washer used for eqns	d_w (in)	1 1/8	
Member Stiffness (washer)	k_mw (lbf/in)	24974478.43	for washers, eqn 8-21, pg 421
member Length	t_m (in)	0.5	
Member Stiffness (frame)	k_mf (lbf/in)	61497911.65	for members, thickness of 1/2" , eqn 8-20, pg 421
Total Member Stiffness	k_m (lbf/in)	8880743.707	treated lock washer as a washer, assumed both members had same thickness
Proof strength	S_p (lbf/in^2)	120000	Table 8-9,Grade 8 pg 425
Proof load	F_p (lbf)	36240	eqn 8-32, 434
Preload	F_i (lbf)	27180	eqn 8-31,non-permanet connections
External tensile load	P_load (lbf)	22500	
External tensile load per bolt	P (lbf)	5625	(a) , pg 428
Portion of P taken by bolt	P_b (lbf)	10803.63	(d), pg 428
Portion of P taken by members	P_m (lbf)	2924.09	portion c, pg 428
Fraction of external load P carried by bolt	C	0.48	(f), pg 428
Resultant bolt load	F_b (lbf)	37983.63	eqn 8-24, pg 428
Resultant load on the connected memebers	F_m (lbf)	-24255.91	has to be negative (compression), eqn 8-25, pg 428
Yeilding factor of safety guarding against the static stress exceeding the proff strength	n_p	1.212814524	pg 432 (b)
Factor of safety against joint separation	n_o	2.32	eqn 8-30, pg 433
Alternating stress experienced by the bolt	σ_a (lbf/in^2)	4471.70	eqn 8-39, pg 438
Midrange stress experienced by the bolt	σ_m (lbf/in^2)	94471.70	eqn 8-40, pg 438
Preload Stress	σ_i (lbf/in^2)	90000.00	Assuming preload is constant, eqn 8-41, pg 438
Minimum Tensile Strength (used as ultimate tensile strength)	S_ut (lbf/in^2)	150000	Table 8-9,Grade 8, pg 425
Endurance Strength	S_e (lbf/in^2)	23200	Table 8-17, Grade 8, pg 437
Alternating Strength?	S_a (lbf/in^2)	9279.94	bottom of pg 437
Fatigue factor of safety	n_f	2.08	eqn 8-37, pg 438
Fatigue factor of safety, Goodman	n_f	1.80	Goodman, eqn 8-45, pg 439
Fatigue factor of safety, Gerber	n_f	2.77	Gerber, eqn 8-46, pg 439
Fatigue factor of safety, ASME-elliptic	n_f	2.72	ASME-elliptic, eqn 8-47, pg 439
Tensile stress in the bolt	σ_b (lbf/in^2)	98943.40611	pg 432 (a)

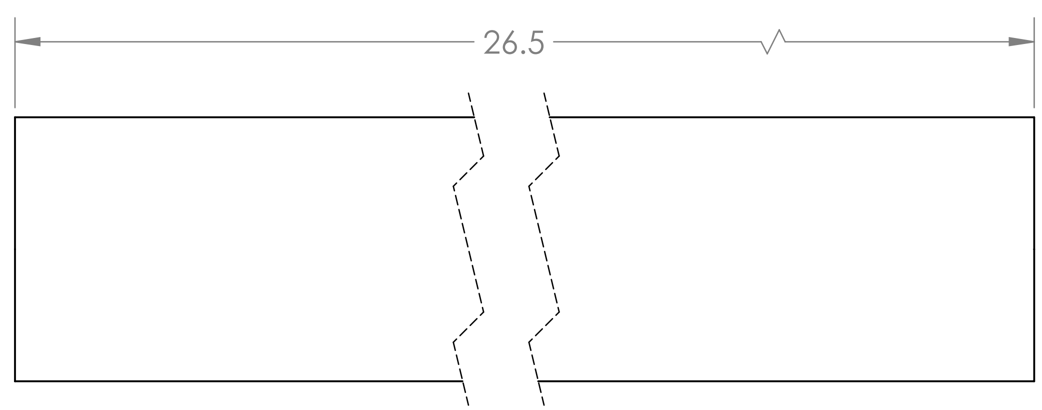
Gantt Chart

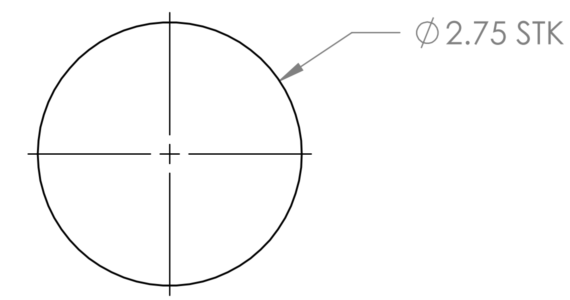
For the complete Gantt chart, please see attached file on Polylearn.

NOTES:

1. MATERIAL: AISI 1045 CD STEEL.







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NEXT ASSY	USED ON
101	SHAFT ASSY
APPLICATION	

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 1/8
ANGULAR: MACH ± 2°
ONE PLACE DECIMAL ± .1
TWO PLACE DECIMAL ± .05
THREE PLACE DECIMAL ± .01

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL: SEE NOTE 1

FINISH: N/A

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	B. TRAN	10/4/17
CHECKED	A. HARAKE	10/4/17
ENG APPR.		
MFG APPR.		
Q.A.		

HIGHRISE

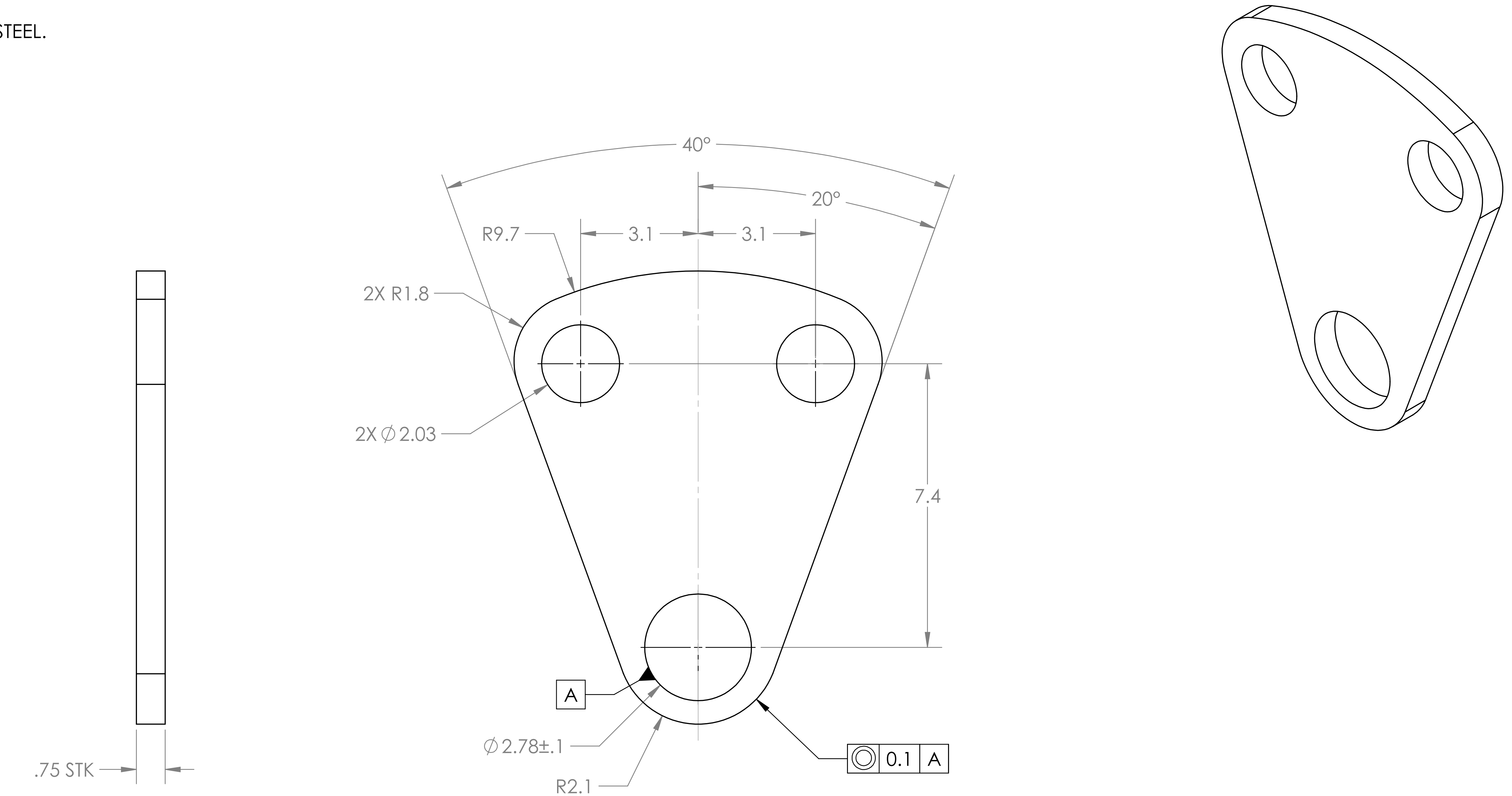
TITLE: MAIN SHAFT

SIZE	DWG. NO.	REV
B	001	02

SCALE: 1:2 | WEIGHT: 44.63 LB | SHEET 1 OF 1

NOTES:

1. MATERIAL: ASTM A36 STEEL.



		UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ±1/8 ANGULAR: MACH ±2° ONE PLACE DECIMAL ±.1 TWO PLACE DECIMAL ±.05 THREE PLACE DECIMAL ±.01	
		INTERPRET GEOMETRIC TOLERANCING PER:	
101	SHAFT ASSY	MATERIAL SEE NOTE 1	
NEXT ASSY	USED ON	FINISH N/A	
APPLICATION		DO NOT SCALE DRAWING	

	NAME	DATE
DRAWN	B. TRAN	10/5/17
CHECKED	A. HARAKE	10/5/17
ENG APPR.		
MFG APPR.		
Q.A.		

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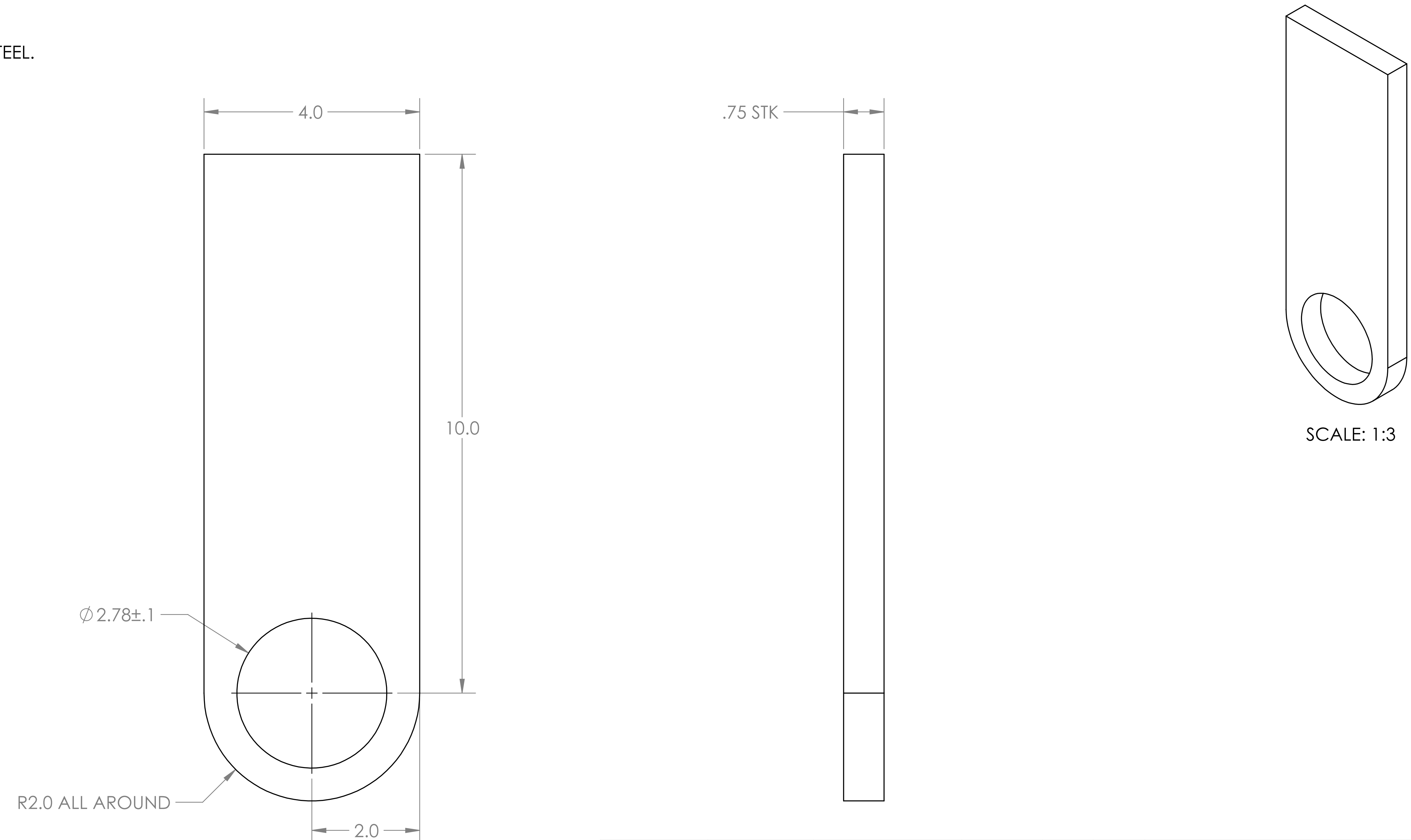
HIGHRISE

TITLE: CYLINDER LEVER ARM

SIZE	DWG. NO.	REV
B	002	02
SCALE: 1:3	WEIGHT: 13.85 LB	SHEET 1 OF 1

NOTES:

1. MATERIAL: ASTM A36 STEEL.



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101	SHAFT ASSY
NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 1/8
ANGULAR: MACH ± 2°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.05
THREE PLACE DECIMAL ±.01

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL SEE NOTE 1

FINISH N/A

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	B. TRAN	10/5/17
CHECKED	A. HARAKE	10/5/17
ENG APPR.		
MFG APPR.		
Q.A.		

HIGHRISE

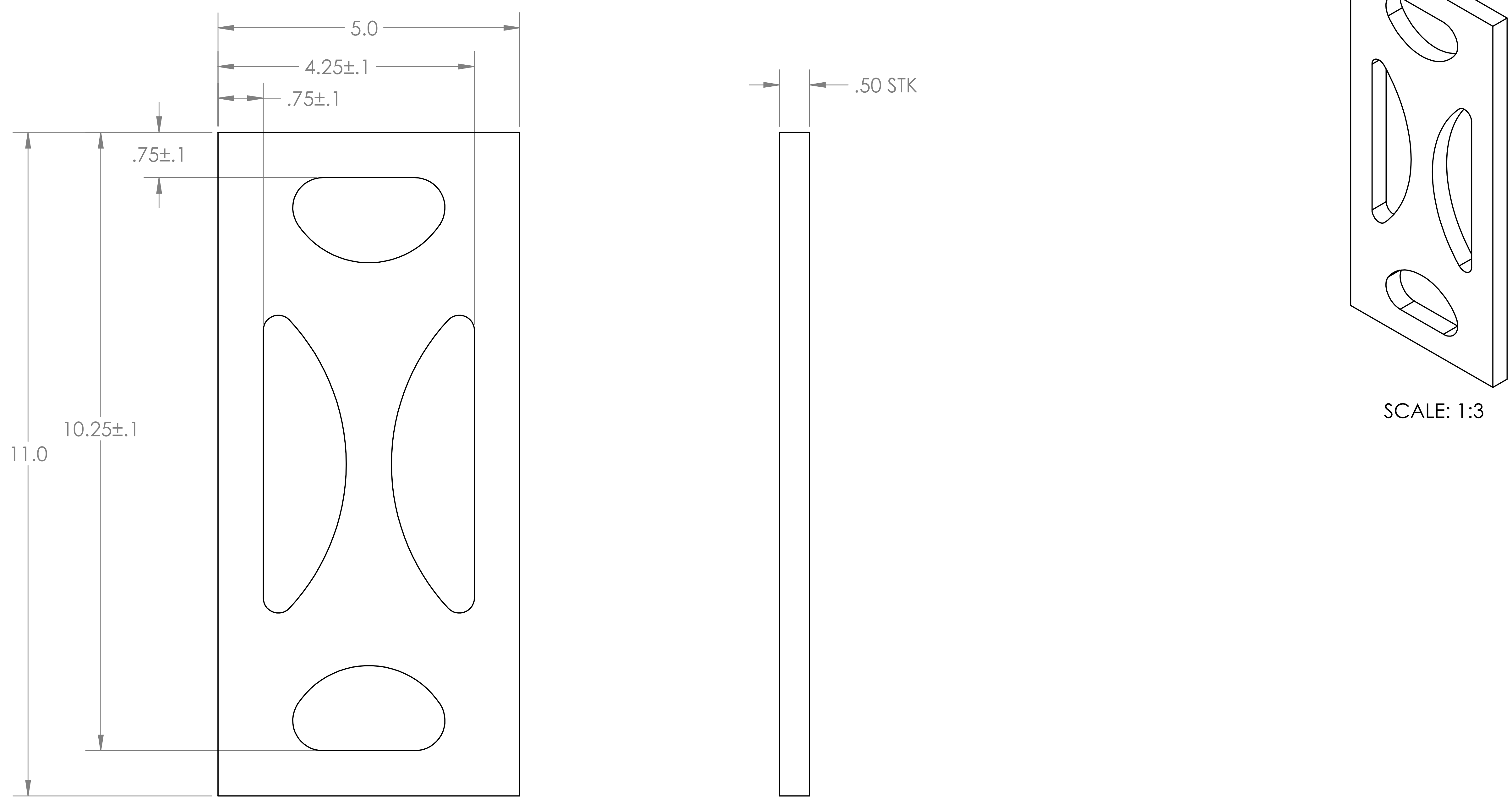
TITLE: TRUSS MOUNT PLATE

SIZE	DWG. NO.	REV
B	003	02

SCALE: 1:2 | WEIGHT: 8.55 LB | SHEET 1 OF 1

NOTES:

1. MATERIAL: ASTM A36 STEEL.



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NEXT ASSY	USED ON
101	SHAFT ASSY

APPLICATION

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 1/8
ANGULAR: MACH ± 2°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.05
THREE PLACE DECIMAL ±.01

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL: SEE NOTE 1

FINISH: N/A

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	B. TRAN	10/5/17
CHECKED	A. HARAKE	10/5/17
ENG APPR.		
MFG APPR.		
Q.A.		

HIGHRISE

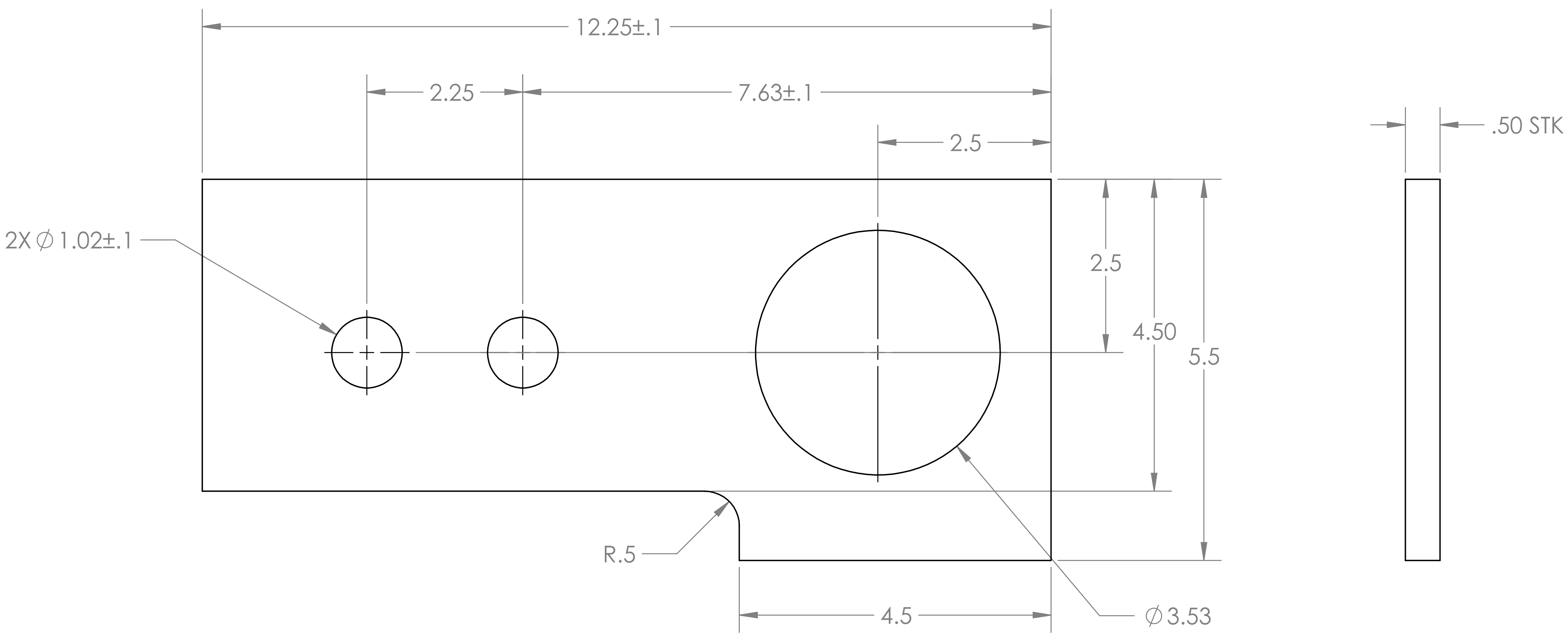
TITLE: TRUSS BRACE

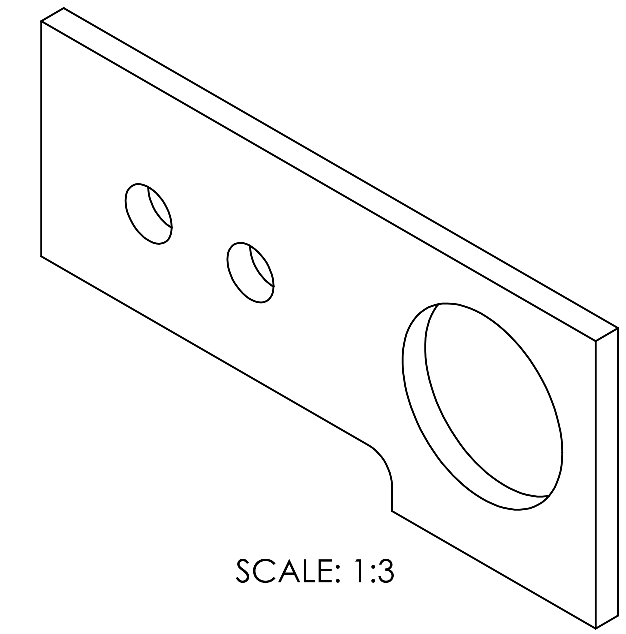
SIZE	DWG. NO.	REV
B	004	01

SCALE: 1:2 | WEIGHT: 5.51 LB | SHEET 1 OF 1

NOTES:

1. MATERIAL: ASTM A36 STEEL.





SCALE: 1:3

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101	SHAFT ASSY
NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 1/8
ANGULAR: MACH ± 2°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.05
THREE PLACE DECIMAL ±.01

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL: SEE NOTE 1

FINISH: N/A

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	B. TRAN	10/5/17
CHECKED	A. HARAKE	10/5/17
ENG APPR.		
MFG APPR.		
Q.A.		

HIGHRISE

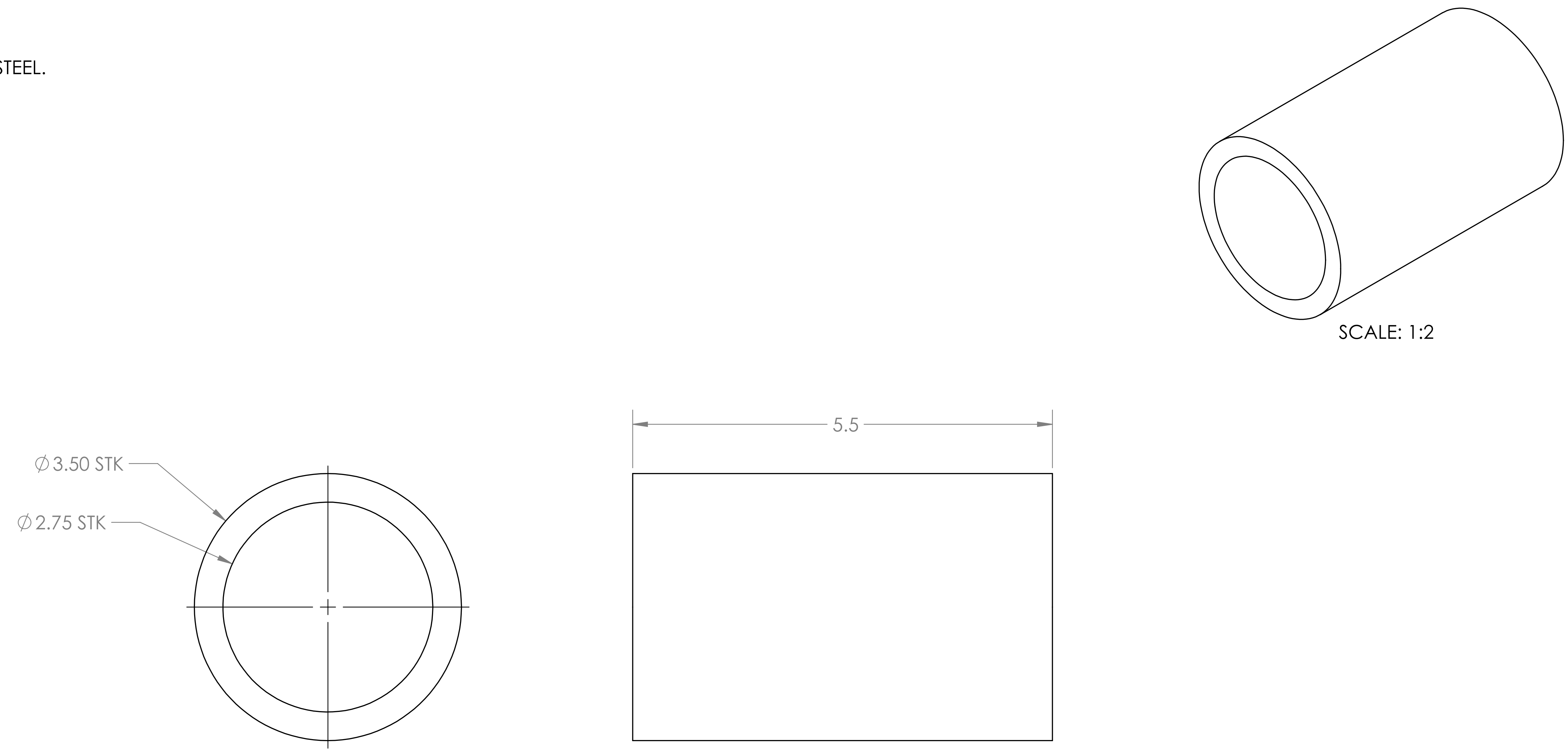
TITLE: SIDE BEARING PLATE

SIZE: B | DWG. NO. 005 | REV 02

SCALE: 1:2 | WEIGHT: 6.84 LB | SHEET 1 OF 1

NOTES:

1. MATERIAL: A513 STEEL.

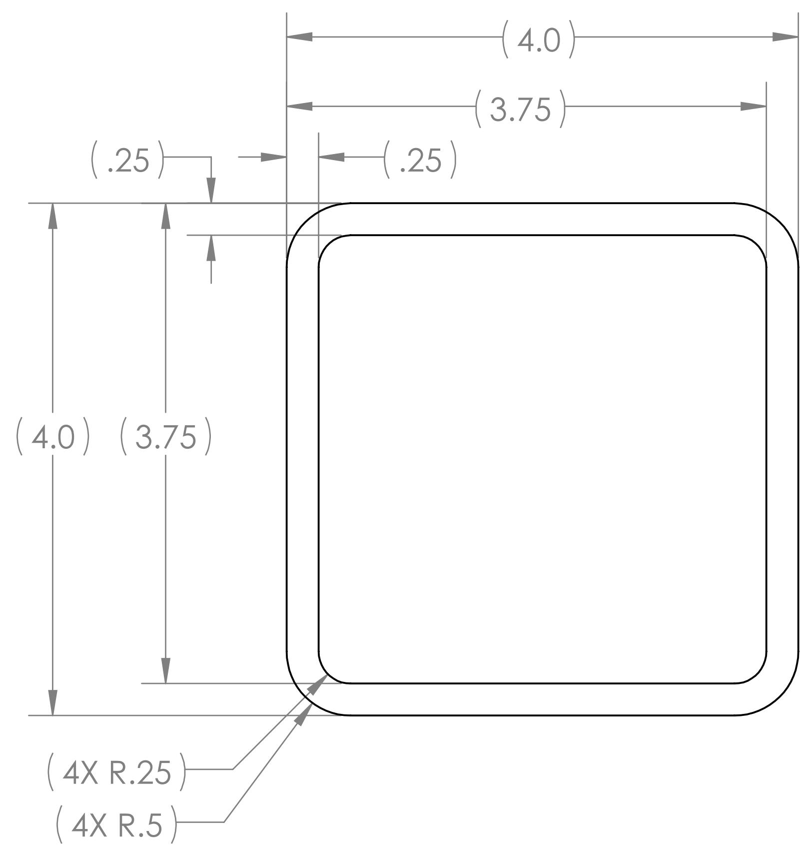


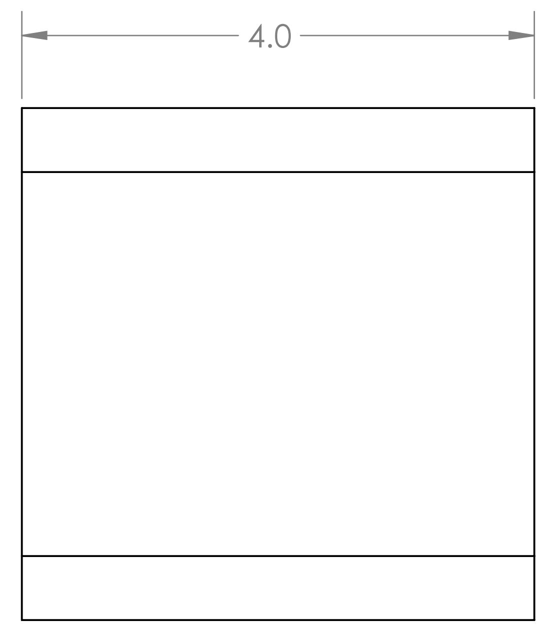
UNLESS OTHERWISE SPECIFIED:		NAME	DATE	TITLE:	HIGHRISE
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± 1/8 ANGULAR: MACH ± 2° ONE PLACE DECIMAL ± .1 TWO PLACE DECIMAL ± .05 THREE PLACE DECIMAL ± .01	DRAWN	B. TRAN	10/5/17		PIPE SLEEVE
	CHECKED	A. HARAKE	10/5/17		
	ENG APPR.				
	MFG APPR.				
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.				
MATERIAL: SEE NOTE 1				SIZE: B	DWG. NO. 006 REV 02
FINISH: N/A				SCALE: 1:1.5 WEIGHT: 5.74 LB	SHEET 1 OF 1
DO NOT SCALE DRAWING					

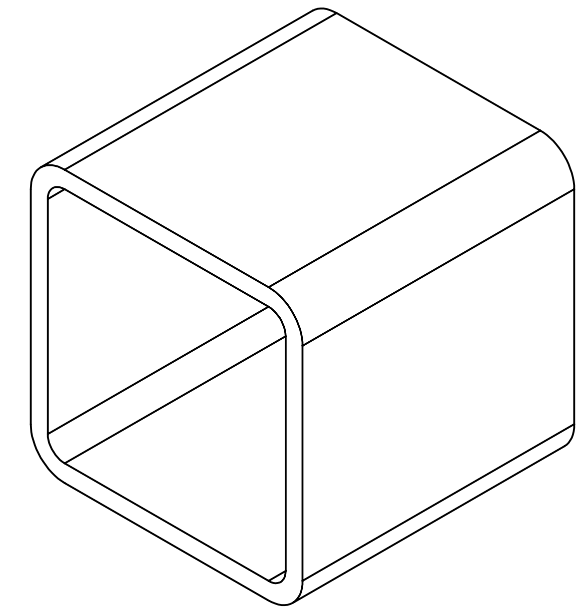
101	SHAFT ASSY
NEXT ASSY	USED ON
APPLICATION	

NOTES:

1. MATERIAL: ASTM A36 STEEL.







SCALE: 1:2

		UNLESS OTHERWISE SPECIFIED:		NAME	DATE	HIGHRISE
		DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± 1/8 ANGULAR: MACH ± 2° ONE PLACE DECIMAL ± .1 TWO PLACE DECIMAL ± .05 THREE PLACE DECIMAL ± .01	DRAWN	B. TRAN	10/5/17	TITLE: SQUARE PIPE SLEEVE
			CHECKED	A. HARAKE	10/5/17	
			ENG APPR.			
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
101	SHAFT ASSY	MATERIAL SEE NOTE 1				SIZE B · DWG. NO. 007 · REV 01
NEXT ASSY	USED ON	FINISH N/A				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 1:1.5 · WEIGHT: 4.07 LB · SHEET 1 OF 1

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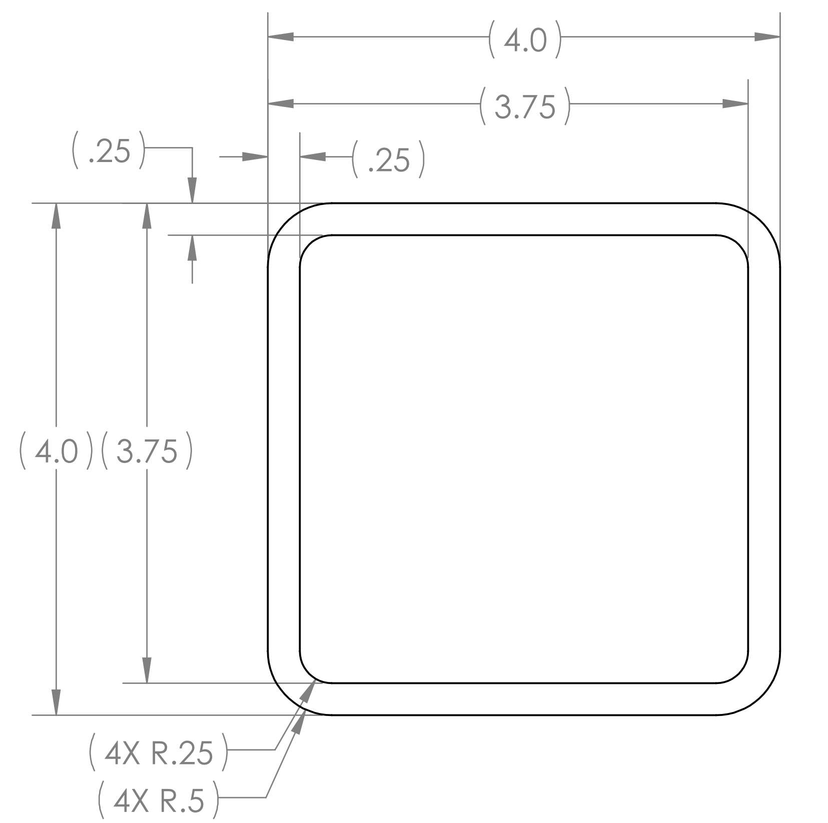
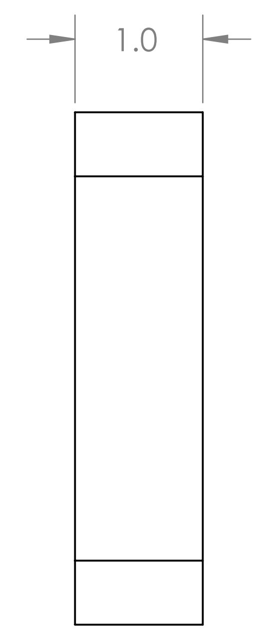
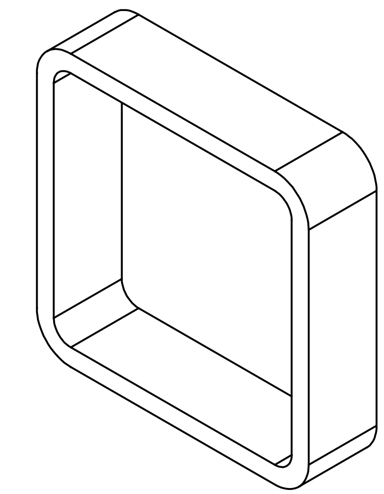
NOTES:

1. MATERIAL: ASTM A36 STEEL.







SCALE: 1:2

101	SHAFT ASSY
NEXT ASSY	USED ON
APPLICATION	

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UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 1/8
ANGULAR: MACH ± 2°
ONE PLACE DECIMAL ± .1
TWO PLACE DECIMAL ± .05
THREE PLACE DECIMAL ± .01

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL: SEE NOTE 1

FINISH: N/A

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	B. TRAN	10/5/17
CHECKED	A. HARAKE	10/5/17
ENG APPR.		
MFG APPR.		
Q.A.		

HIGHRISE

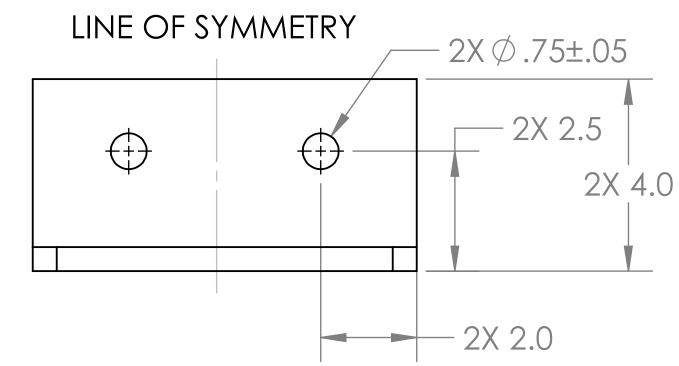
TITLE: SQUARE PIPE SPACER

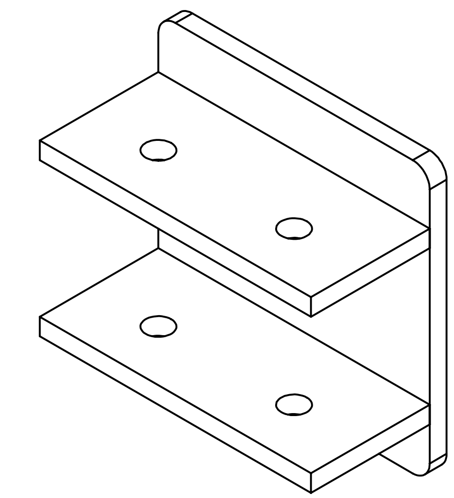
SIZE	DWG. NO.	REV
B	008	01

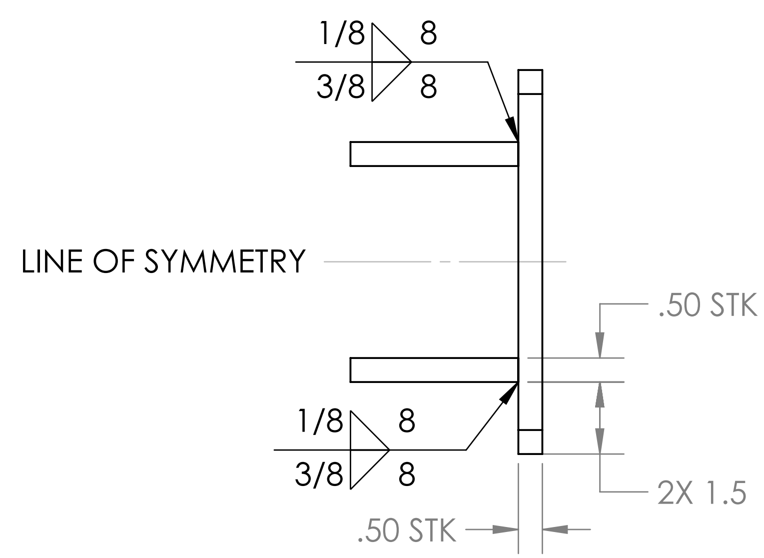
SCALE: 1:1.5 | WEIGHT: 1.02 LB | SHEET 1 OF 1

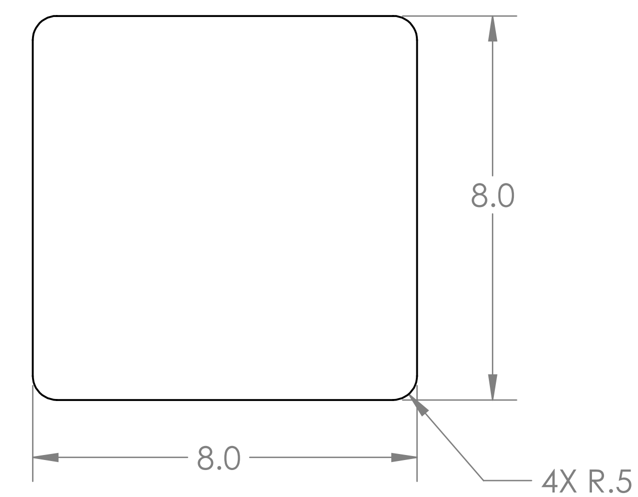
NOTES:

1. MATERIAL: ASTM A36 STEEL.
2. 7018 ROD REQUIRED FOR WELDING









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102	CYLINDER ASSY
NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 1/8
ANGULAR: MACH ± 2°
ONE PLACE DECIMAL ± .1
TWO PLACE DECIMAL ± .05
THREE PLACE DECIMAL ± .01

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL: SEE NOTE 1

FINISH: N/A

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	B. TRAN	10/5/17
CHECKED	A. HARAKE	10/5/17
ENG APPR.		
MFG APPR.		
Q.A.		

HIGHRISE

TITLE:

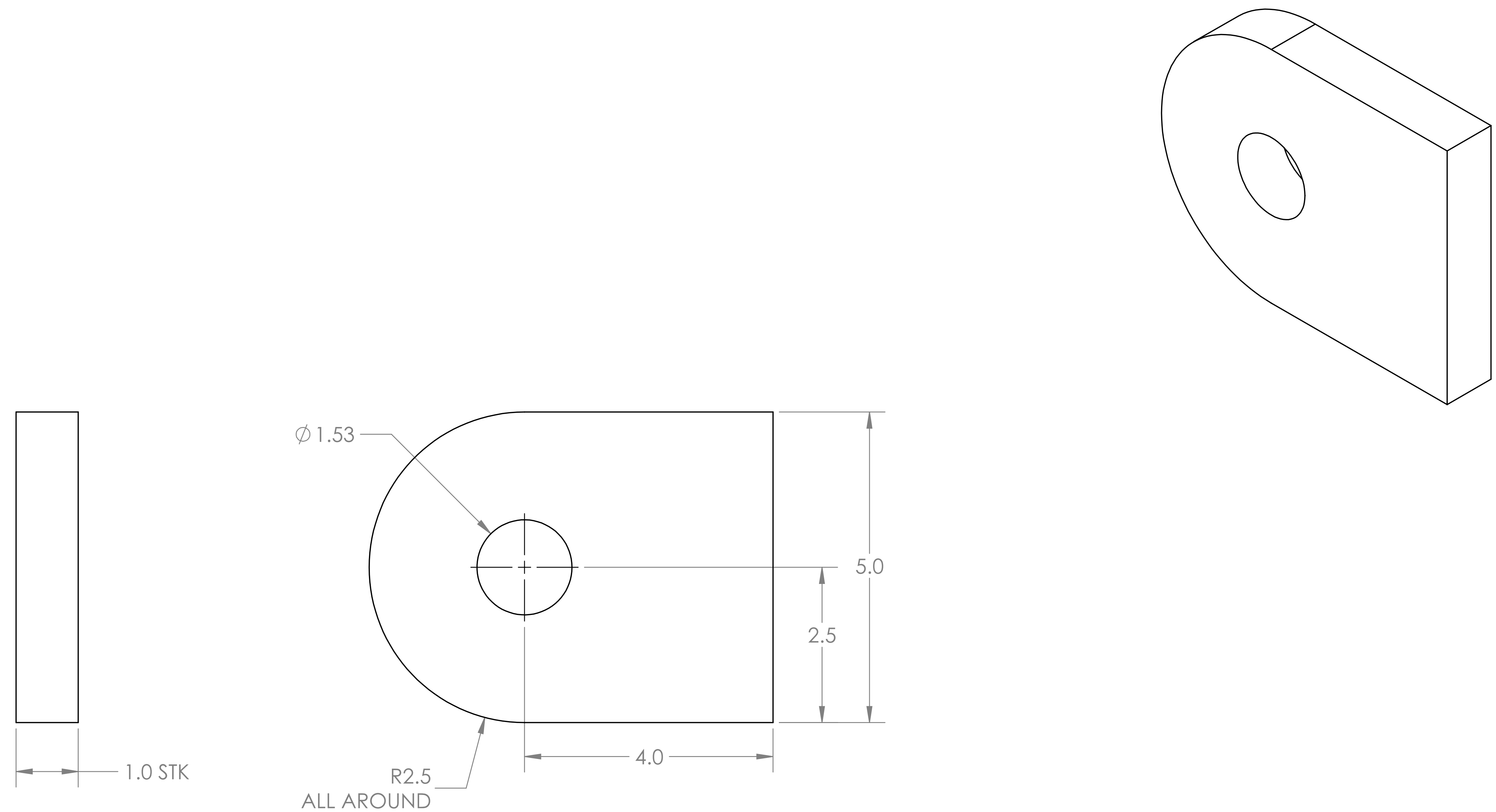
CYLINDER BACK MOUNTING PLATE

SIZE	DWG. NO.	REV
B	009	02

SCALE: 1:4 | WEIGHT: 16.73 LB | SHEET 1 OF 1

NOTES:

1. MATERIAL: ASTM A36 STEEL.



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NEXT ASSY	USED ON
102	CYLINDER ASSY

APPLICATION

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±1/8
ANGULAR: MACH ±2°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.05
THREE PLACE DECIMAL ±.01

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL: SEE NOTE 1

FINISH: N/A

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	B. TRAN	10/5/17
CHECKED	A. HARAKE	10/5/17
ENG APPR.		
MFG APPR.		
Q.A.		

HIGHRISE

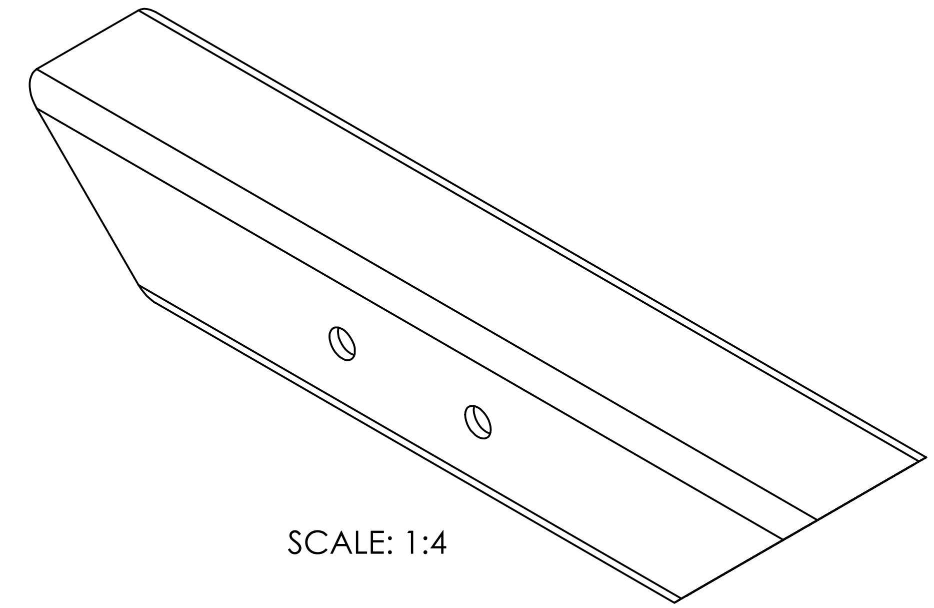
TITLE: FIXED END PIN MOUNT

SIZE	DWG. NO.	REV
B	010	02

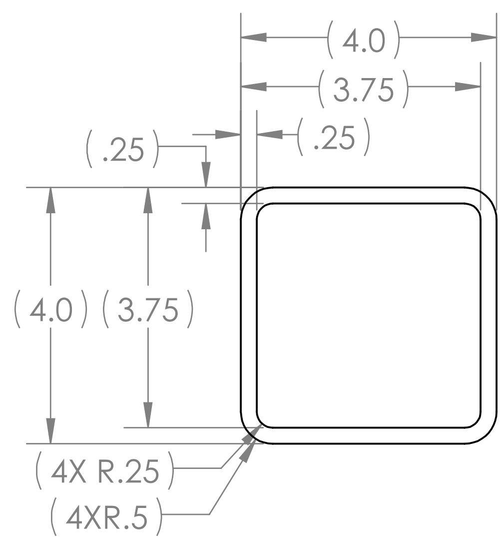
SCALE: 1:2 WEIGHT: 7.93 LB SHEET 1 OF 1

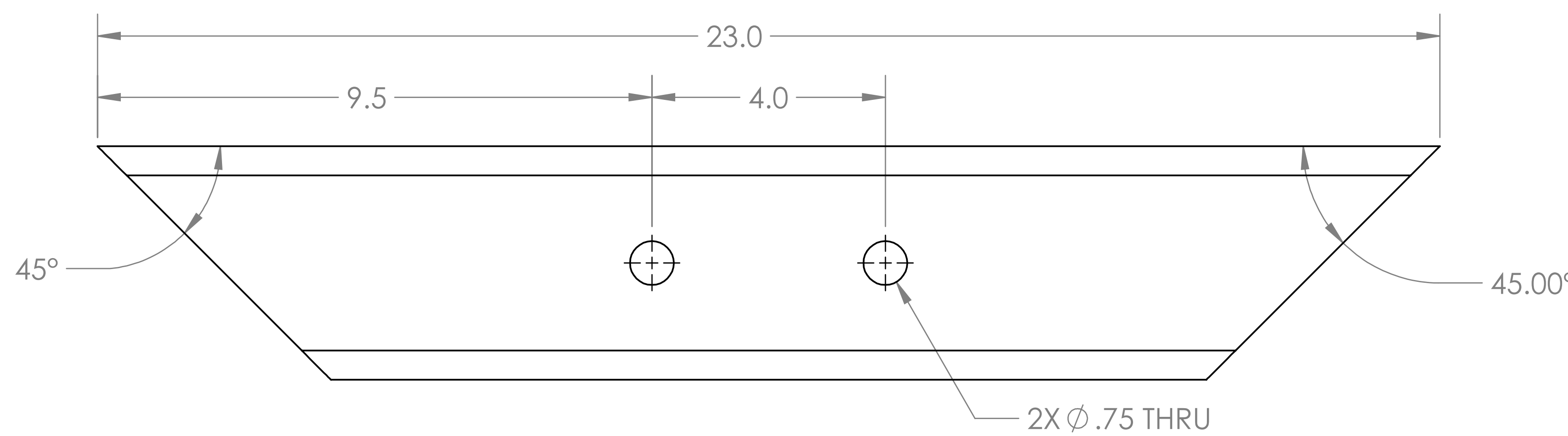
NOTES:

1. MATERIAL: ASTM A36 STEEL.



SCALE: 1:4





PROPRIETARY AND CONFIDENTIAL

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103	TUBE FRAME ASSY
NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 1/8
ANGULAR: MACH ± 2°
ONE PLACE DECIMAL ± .1
TWO PLACE DECIMAL ± .05
THREE PLACE DECIMAL ± .01

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL: SEE NOTE 1

FINISH: N/A

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	B. TRAN	10/5/17
CHECKED	A. HARAKE	10/5/17
ENG APPR.		
MFG APPR.		
Q.A.		

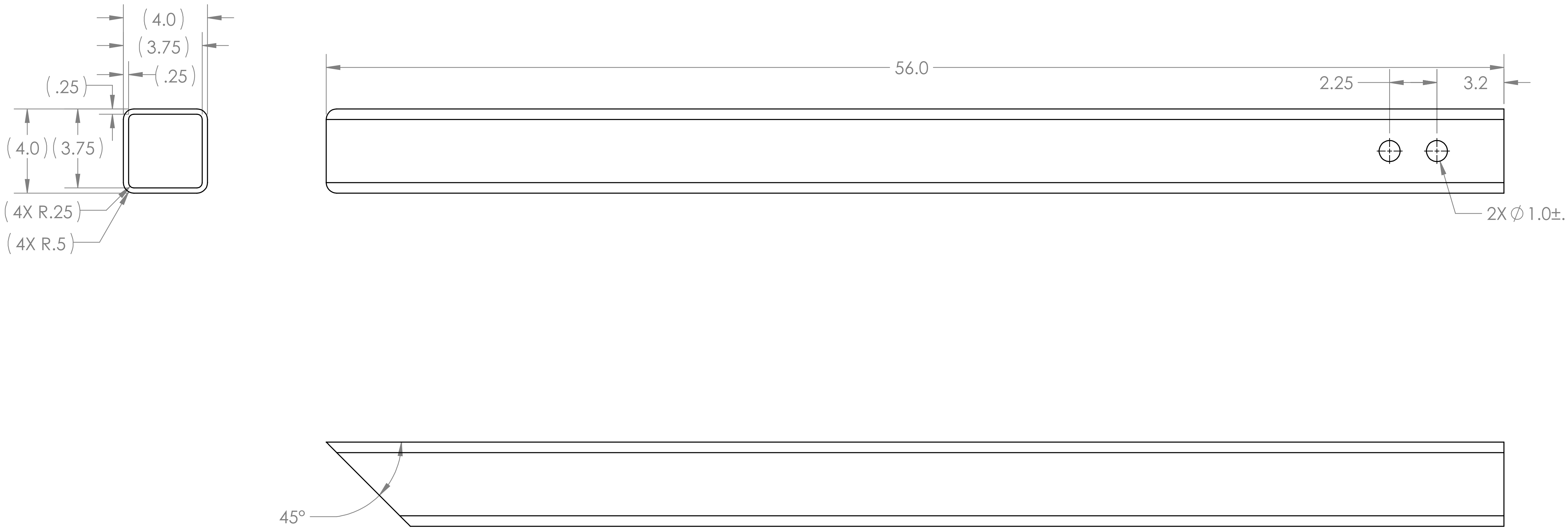
HIGHRISE

TITLE: TUBE FRAME BACK

SIZE	DWG. NO.	REV
B	011	01
SCALE: 1:3	WEIGHT: 19.21 LB	SHEET 1 OF 1

NOTES:

1. MATERIAL: ASTM A36 STEEL.



PROPRIETARY AND CONFIDENTIAL

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NEXT ASSY	USED ON
103	TUBE FRAME ASSY

APPLICATION

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 1/8
ANGULAR: MACH ± 2°
ONE PLACE DECIMAL ± .1
TWO PLACE DECIMAL ± .05
THREE PLACE DECIMAL ± .01

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL: SEE NOTE 1

FINISH: N/A

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	B. TRAN	10/5/17
CHECKED	A. HARAKE	10/5/17
ENG APPR.		
MFG APPR.		
Q.A.		

HIGHRISE

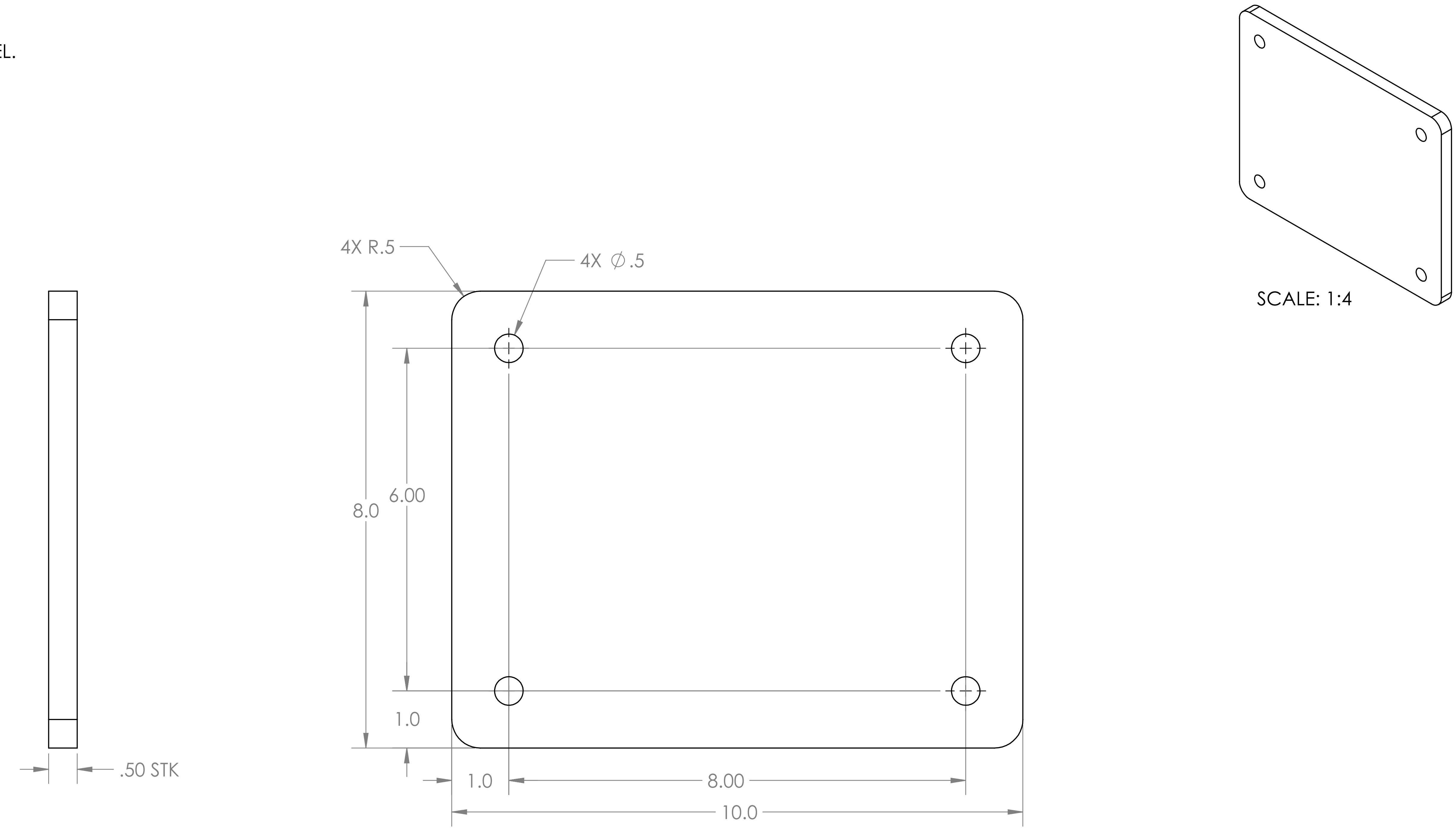
TITLE: TUBE FRAME SIDES

SIZE	DWG. NO.	REV
B	012	01

SCALE: 1:5 | WEIGHT: 54.74 LB | SHEET 1 OF 1

NOTES:

1. MATERIAL: ASTM A36 STEEL.



PROPRIETARY AND CONFIDENTIAL

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NEXT ASSY	USED ON
101	SHAFT ASSY

APPLICATION

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 1/8
ANGULAR: MACH ± 2°
ONE PLACE DECIMAL ± .1
TWO PLACE DECIMAL ± .05
THREE PLACE DECIMAL ± .01

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL: SEE NOTE 1

FINISH: N/A

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	B. TRAN	10/5/17
CHECKED	A. HARAKE	10/5/17
ENG APPR.		
MFG APPR.		
Q.A.		

HIGHRISE

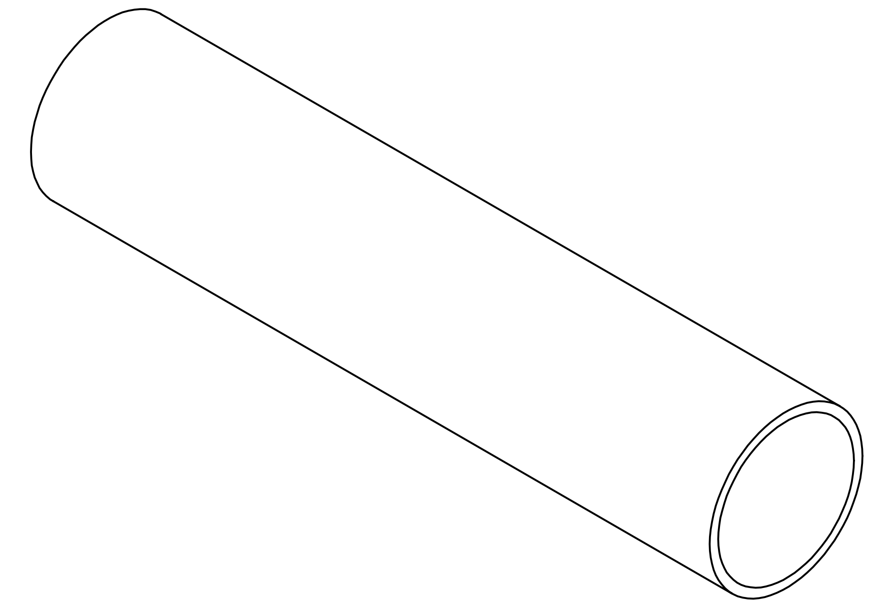
TITLE: MAIN MATCH PLATE

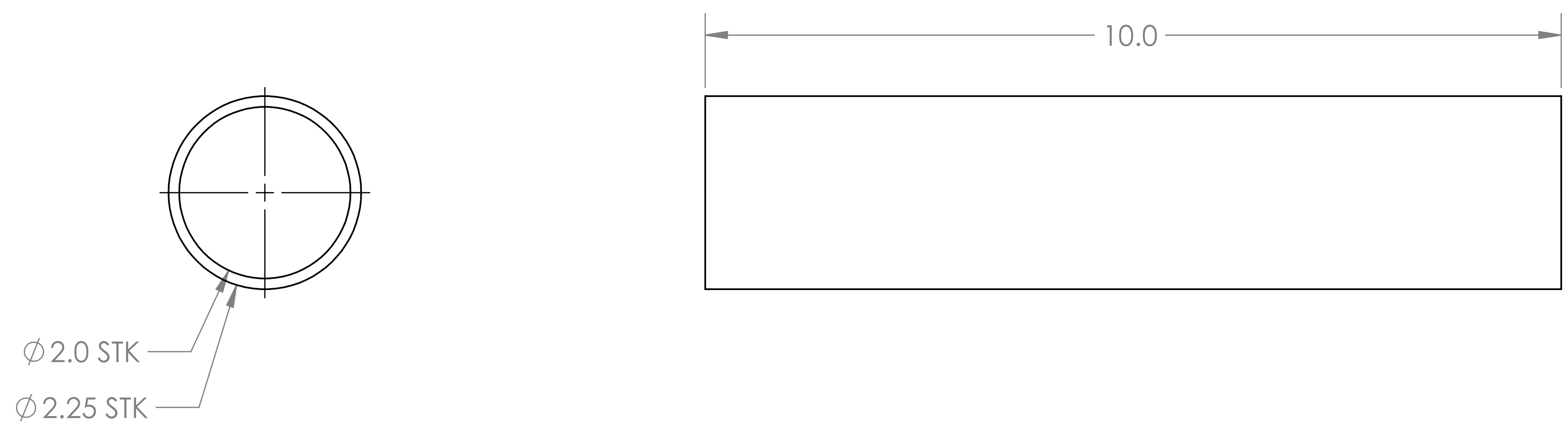
SIZE	DWG. NO.	REV
B	013	01

SCALE: 1:2 | WEIGHT: 14.56 LB | SHEET 1 OF 1

NOTES:

1. MATERIAL: ASTM A36 STEEL.





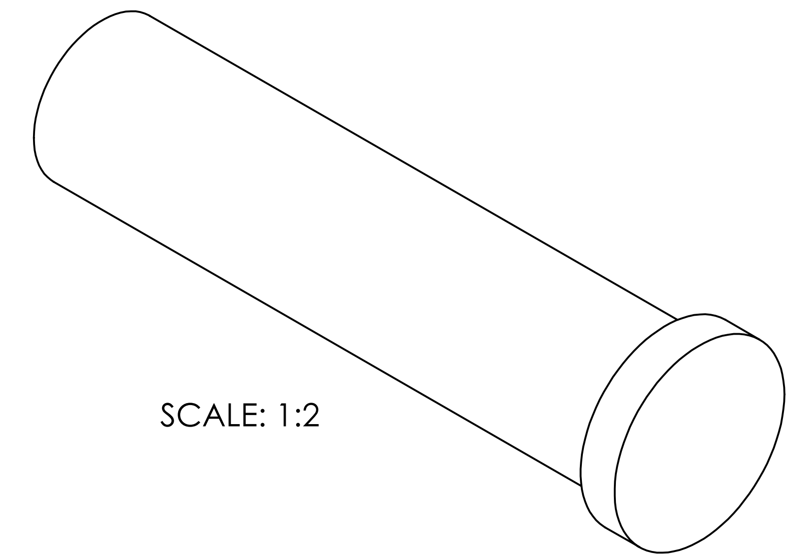
UNLESS OTHERWISE SPECIFIED:		NAME	DATE		HIGHRISE
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± 1/8 ANGULAR: MACH ± 2° ONE PLACE DECIMAL ±.1 TWO PLACE DECIMAL ±.05 THREE PLACE DECIMAL ±.01	DRAWN	B. TRAN	10/5/17	TITLE:	CYLINDER COLLAR
	CHECKED	A. HARAKE	10/5/17		
	ENG APPR.				
	MFG APPR.				
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.				
MATERIAL SEE NOTE 1				SIZE B	DWG. NO. 014 REV 01
FINISH N/A					
DO NOT SCALE DRAWING				SCALE: 1:2 WEIGHT: 2.37 LB	SHEET 1 OF 1

105	MAIN ASSY
NEXT ASSY	USED ON
APPLICATION	

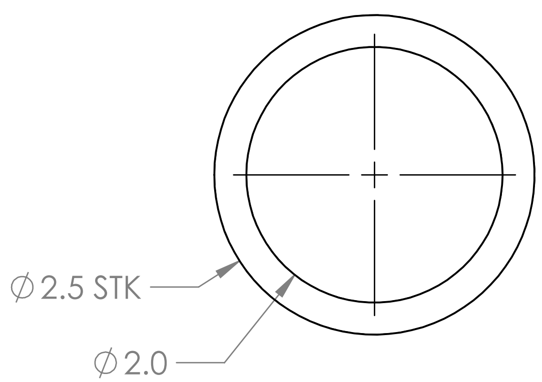
PROPRIETARY AND CONFIDENTIAL

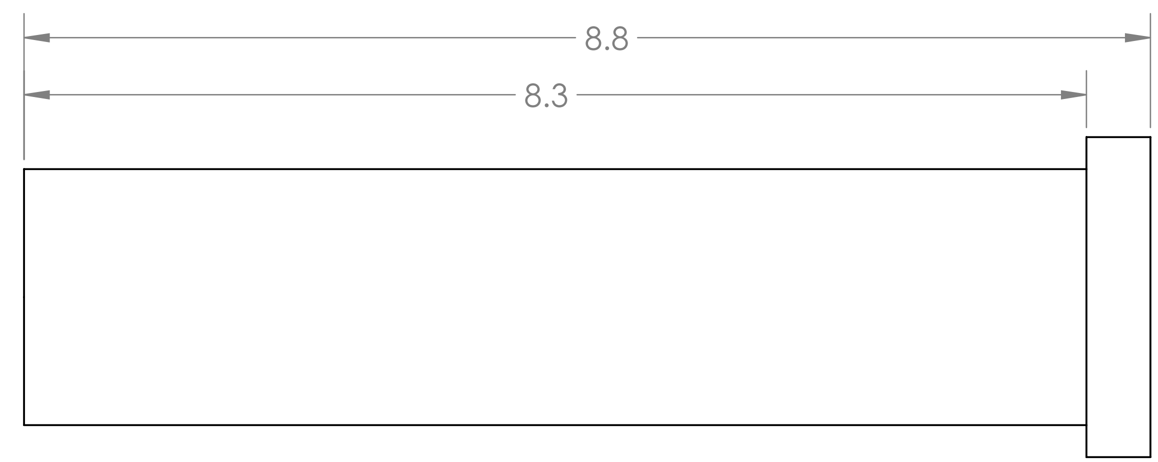
NOTES:

1. MATERIAL: AISI 1045 CD STEEL.



SCALE: 1:2





PROPRIETARY AND CONFIDENTIAL

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105	MAIN ASSY
NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 1/8
ANGULAR: MACH ± 2°
ONE PLACE DECIMAL ± .1
TWO PLACE DECIMAL ± .05
THREE PLACE DECIMAL ± .01

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL: SEE NOTE 1

FINISH: N/A

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	B. TRAN	10/5/17
CHECKED	A. HARAKE	10/5/17
ENG APPR.		
MFG APPR.		
Q.A.		

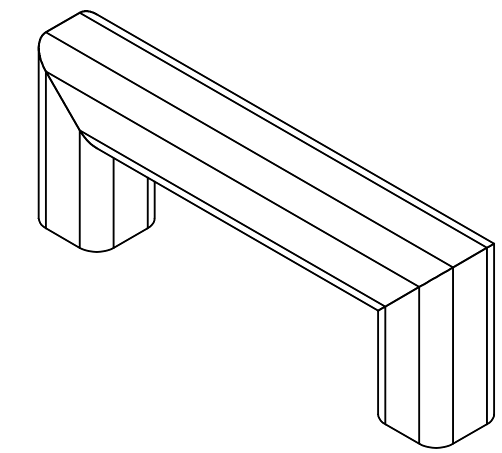
HIGHRISE

TITLE: CYLINDER PIN

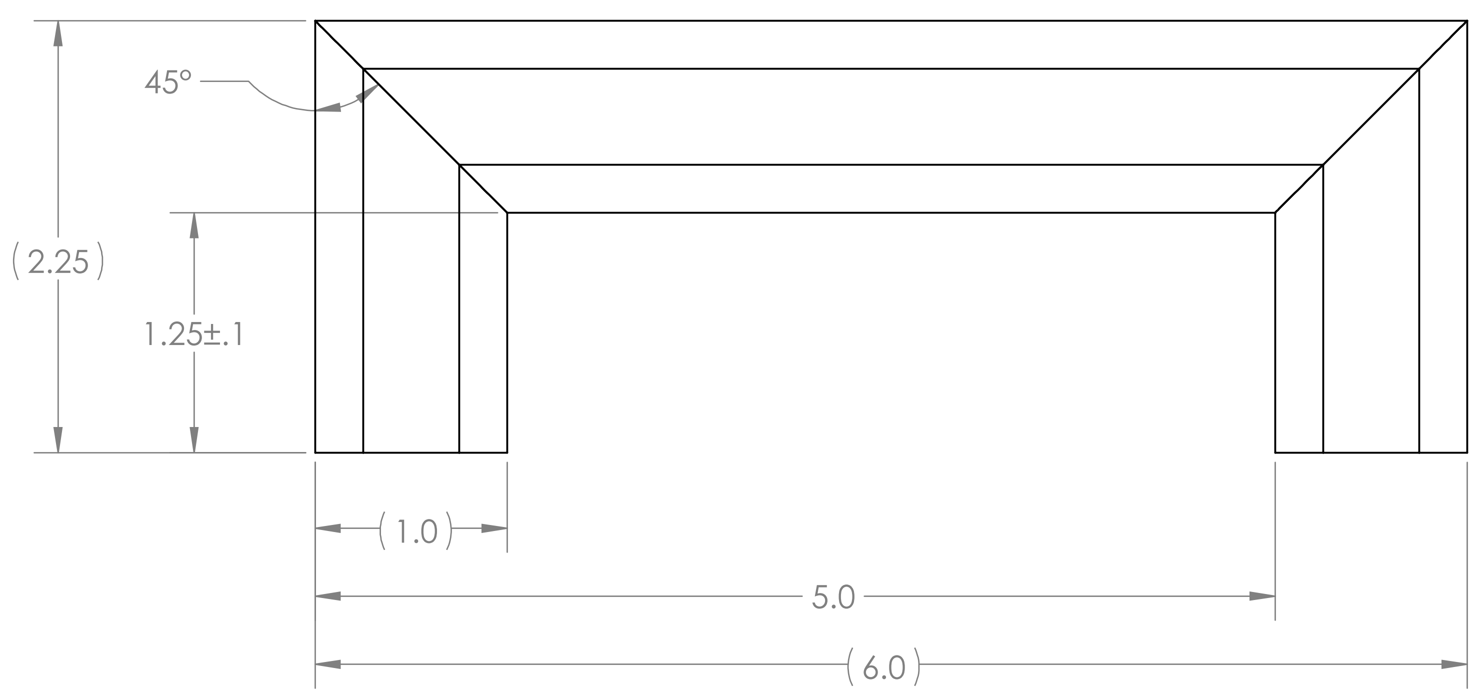
SIZE	DWG. NO.	REV
B	015	02
SCALE: 1:1.5	WEIGHT: 8.09 LB	SHEET 1 OF 1

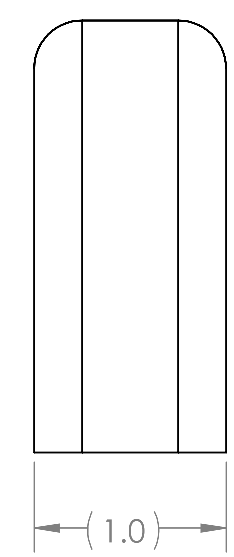
NOTES:

1. MATERIAL: ASTM A36 STEEL.



SCALEI: 1:2





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NEXT ASSY	USED ON
105	MAIN ASSY

APPLICATION

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 1/8
ANGULAR: MACH ± 2°
ONE PLACE DECIMAL ± .1
TWO PLACE DECIMAL ± .05
THREE PLACE DECIMAL ± .01

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL: SEE NOTE 1

FINISH: N/A

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	B. TRAN	10/20/17
CHECKED	A. HARAKE	10/21/17
ENG APPR.		
MFG APPR.		
Q.A.		

HIGHRISE

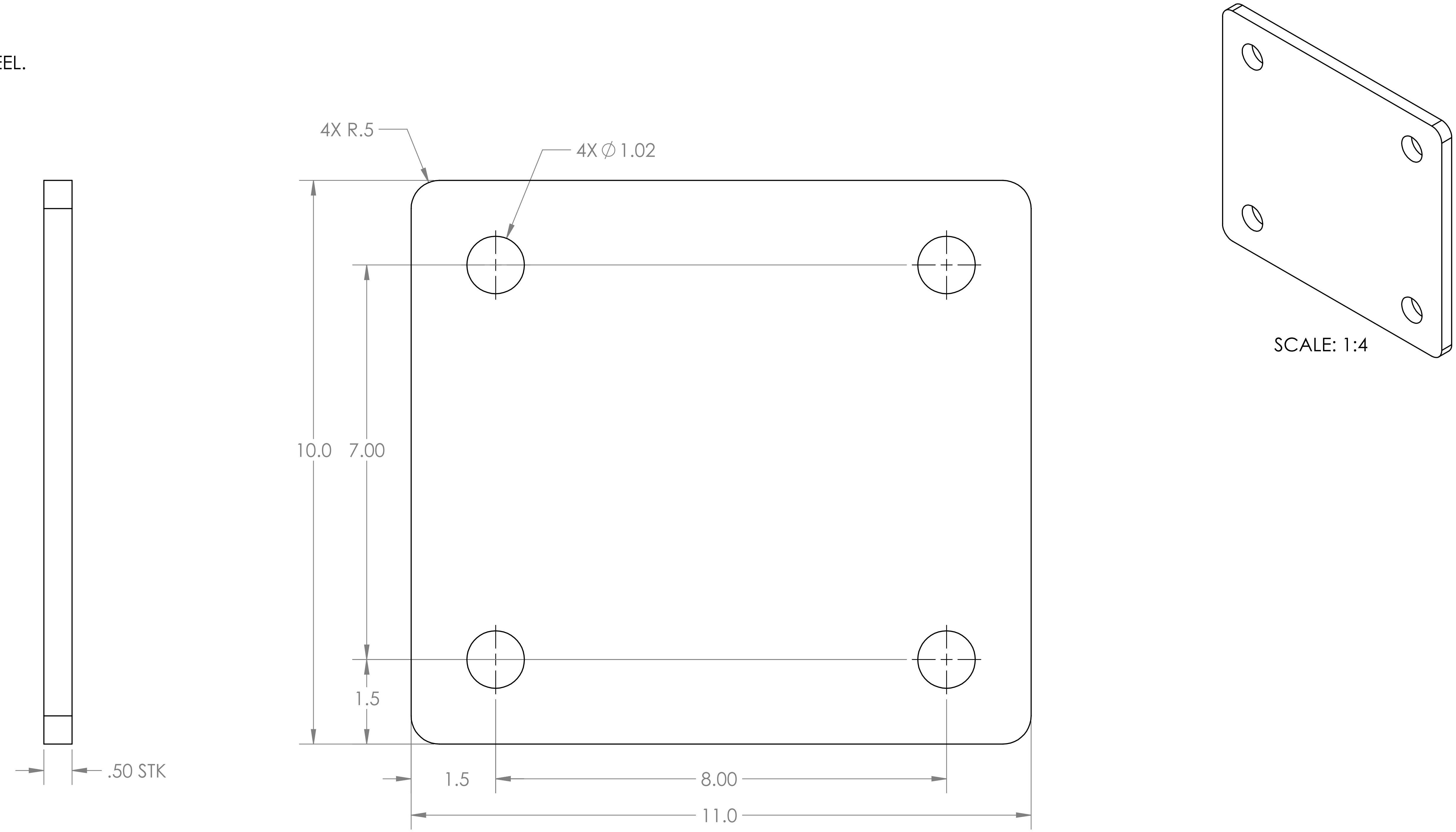
TITLE: SAFETY HANDLE

SIZE	DWG. NO.	REV
B	016	01

SCALE: 1:1 | WEIGHT: 0.41 LB | SHEET 1 OF 1

NOTES:

1. MATERIAL: ASTM A36 STEEL.



101	SHAFT ASSY	
NEXT ASSY	USED ON	
APPLICATION		

PROPRIETARY AND CONFIDENTIAL

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UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 1/8
ANGULAR: MACH ± 2°
ONE PLACE DECIMAL ± .1
TWO PLACE DECIMAL ± .05
THREE PLACE DECIMAL ± .01

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL: SEE NOTE 1

FINISH: N/A

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	B. TRAN	3/16/18
CHECKED	A. HARAKE	3/16/18
ENG APPR.		
MFG APPR.		
Q.A.		

HIGHRISE

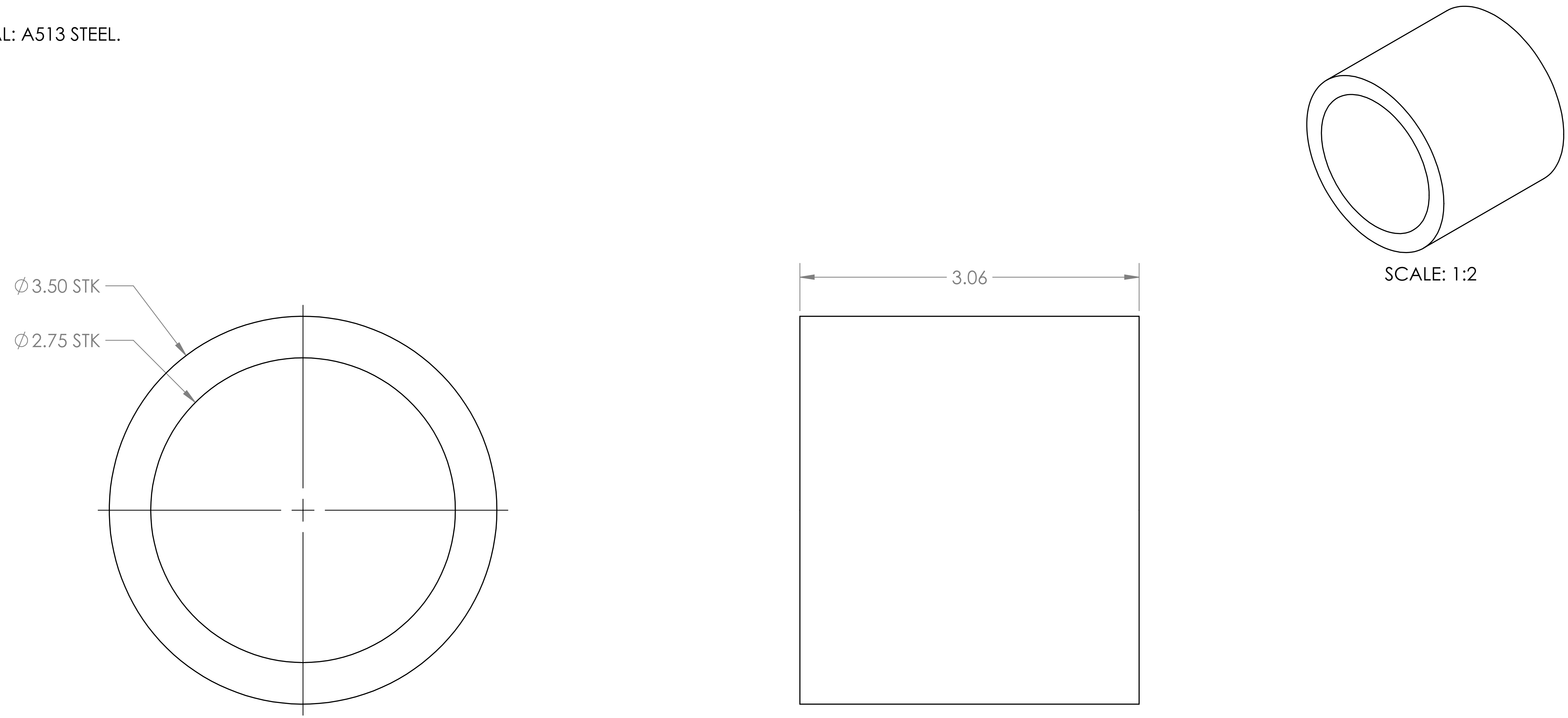
TITLE: TRUSS MATCH PLATE

SIZE	DWG. NO.	REV
B	017	01

SCALE: 1:2 | WEIGHT: 15.10 LB | SHEET 1 OF 1

NOTES:

1. MATERIAL: A513 STEEL.



		UNLESS OTHERWISE SPECIFIED:		NAME	DATE
		DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± 1/8 ANGULAR: MACH ± 2° ONE PLACE DECIMAL ±.1 TWO PLACE DECIMAL ±.05 THREE PLACE DECIMAL ±.01	DRAWN	B. TRAN	3/16/18
			CHECKED	A. HARAKE	3/16/18
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
101	SHAFT ASSY	MATERIAL SEE NOTE 1			
NEXT ASSY	USED ON	FINISH N/A			
APPLICATION		DO NOT SCALE DRAWING			

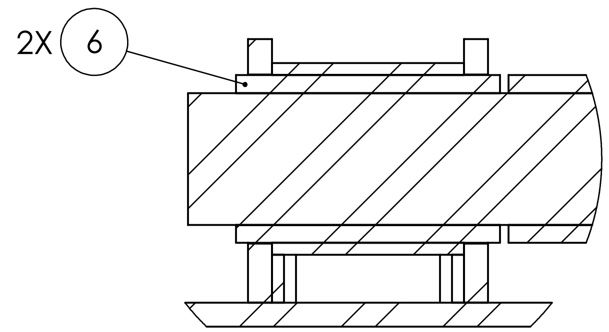
HIGHRISE

TITLE: PIPE SPACER

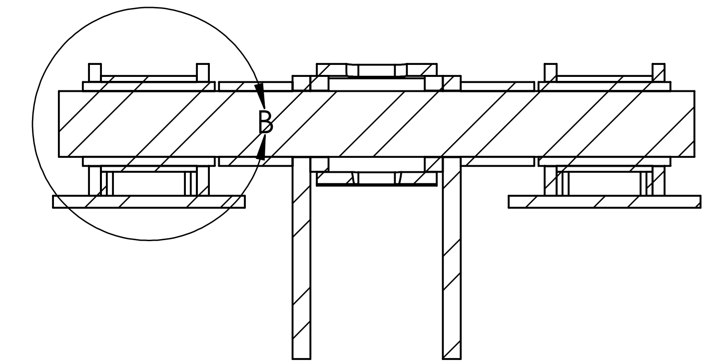
SIZE	DWG. NO.	REV
B	018	01
SCALE: 1:1	WEIGHT: 3.20 LB	SHEET 1 OF 1

NOTES:

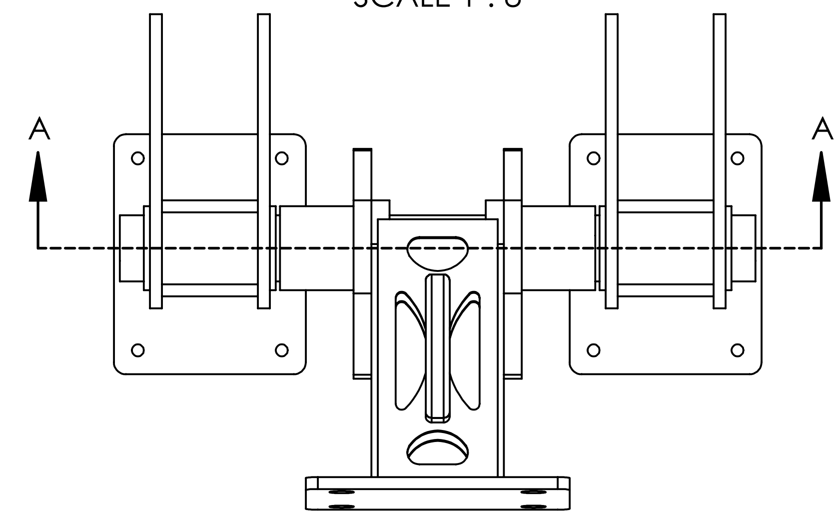
1. 7018 ROD REQUIRED FOR WELDING

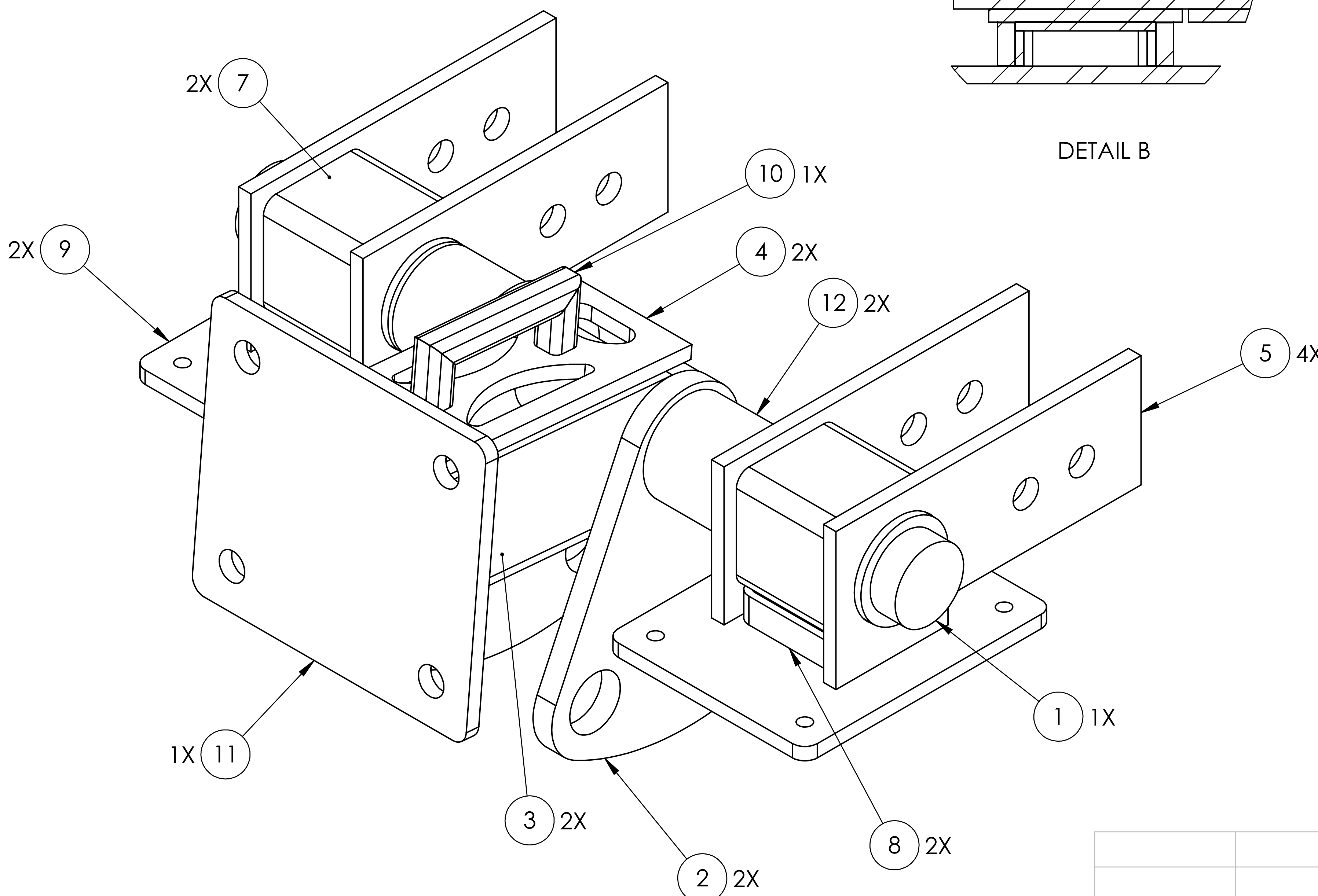


DETAIL B



SECTION A-A
SCALE 1 : 8





ITEM NO.	PART NUMBER	DESCRIPTION	Default/ QTY.
1	001	MAIN SHAFT	1
2	002	CYLINDER LEVER ARM	2
3	003	TRUSS MOUNT PLATE	2
4	004	TRUSS BRACE	2
5	005	SIDE BEARING PLATE	4
6	006	PIPE SLEEVE	2
7	007	4X4 PIPE SLEEVE	2
8	008	4X4 PIPE SPACER	2
9	013	MAIN MATCH PLATE	2
10	016	SAFETY HANDLE	1
11	017	TRUSS MATCH PLATE	1
12	018	PIPE SPACER	2

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 1/8
ANGULAR: MACH ± 2°
ONE PLACE DECIMAL ± .1
TWO PLACE DECIMAL ± .05
THREE PLACE DECIMAL ± .01

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL N/A

FINISH N/A

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	B. TRAN	10/5/17
CHECKED	A. HARAKE	10/5/17
ENG APPR.		
MFG APPR.		
Q.A.		

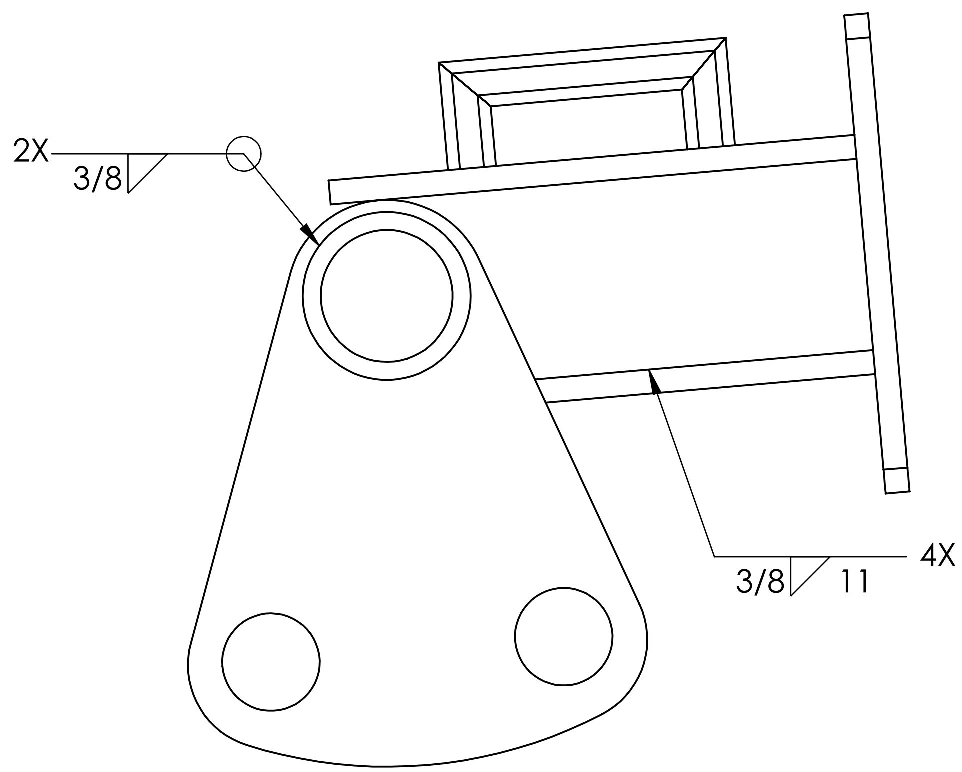
105	MAIN ASSY
NEXT ASSY	USED ON
APPLICATION	

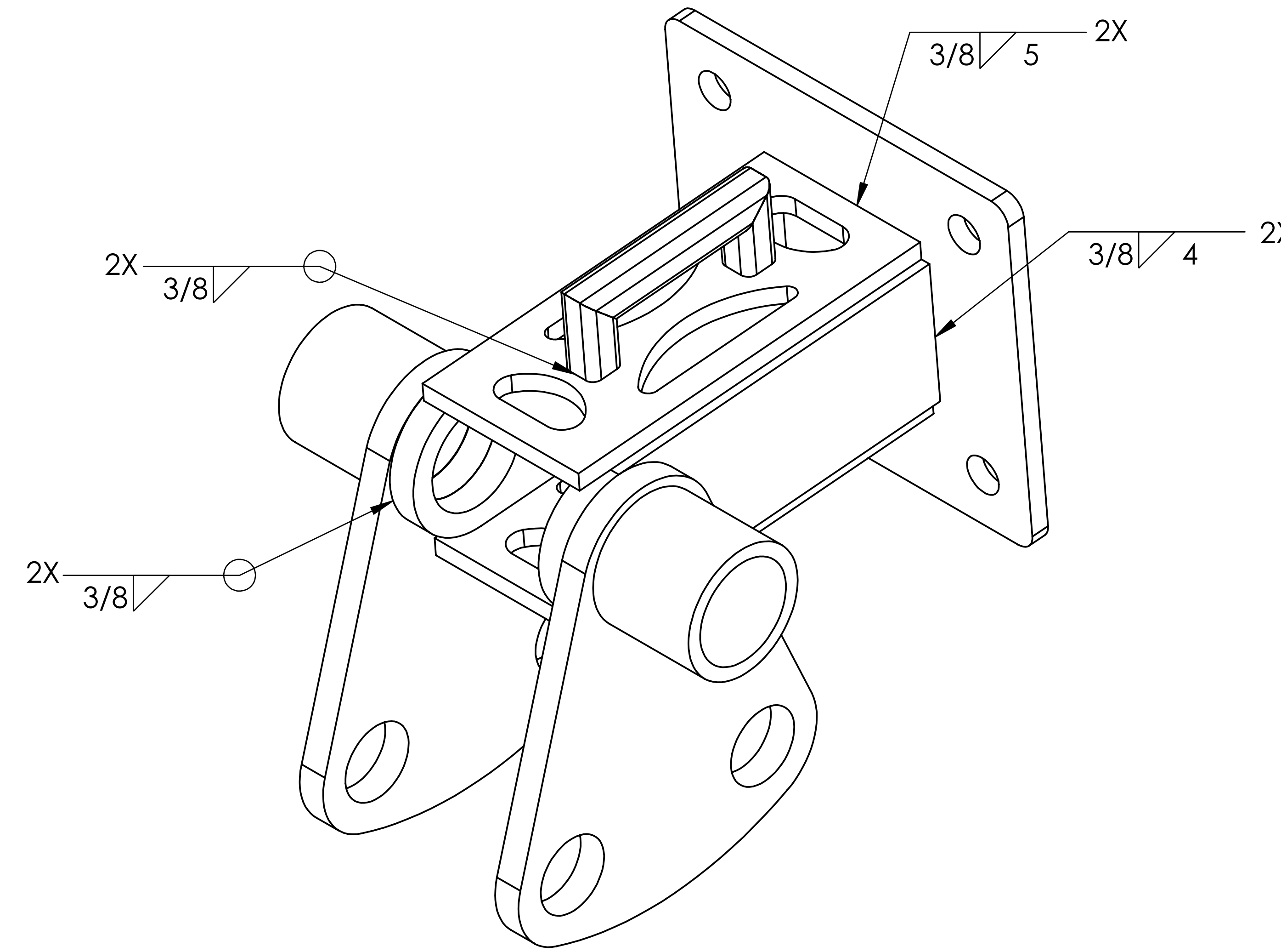
HIGHRISE

TITLE:
SHAFT ASSEMBLY

SIZE B | DWG. NO. 101 | REV 02

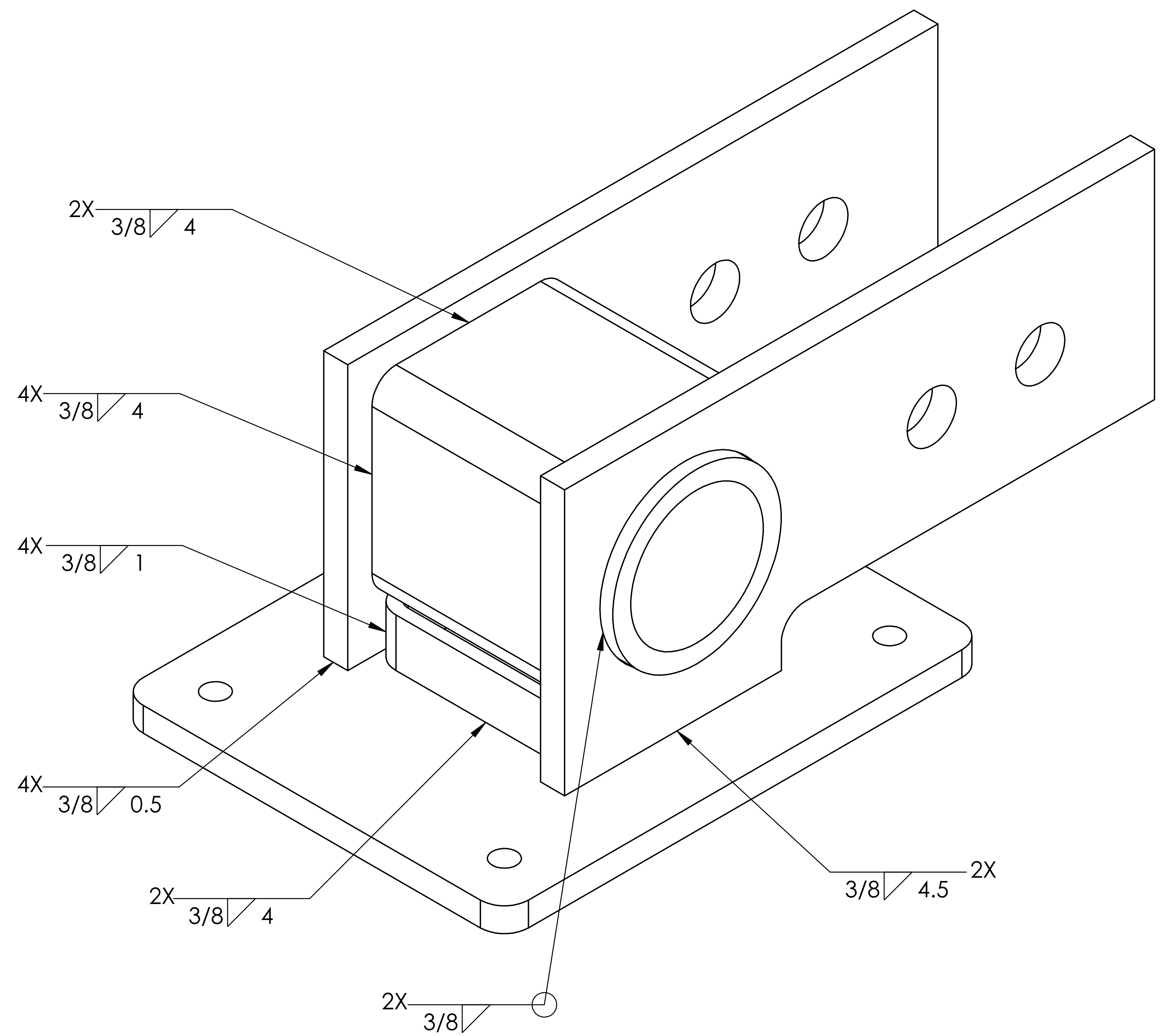
SCALE: 1:4 | WEIGHT: 194.36 LB | SHEET 1 OF 3





CENTER SUBASSEMBLY

SHEET 2 OF 3

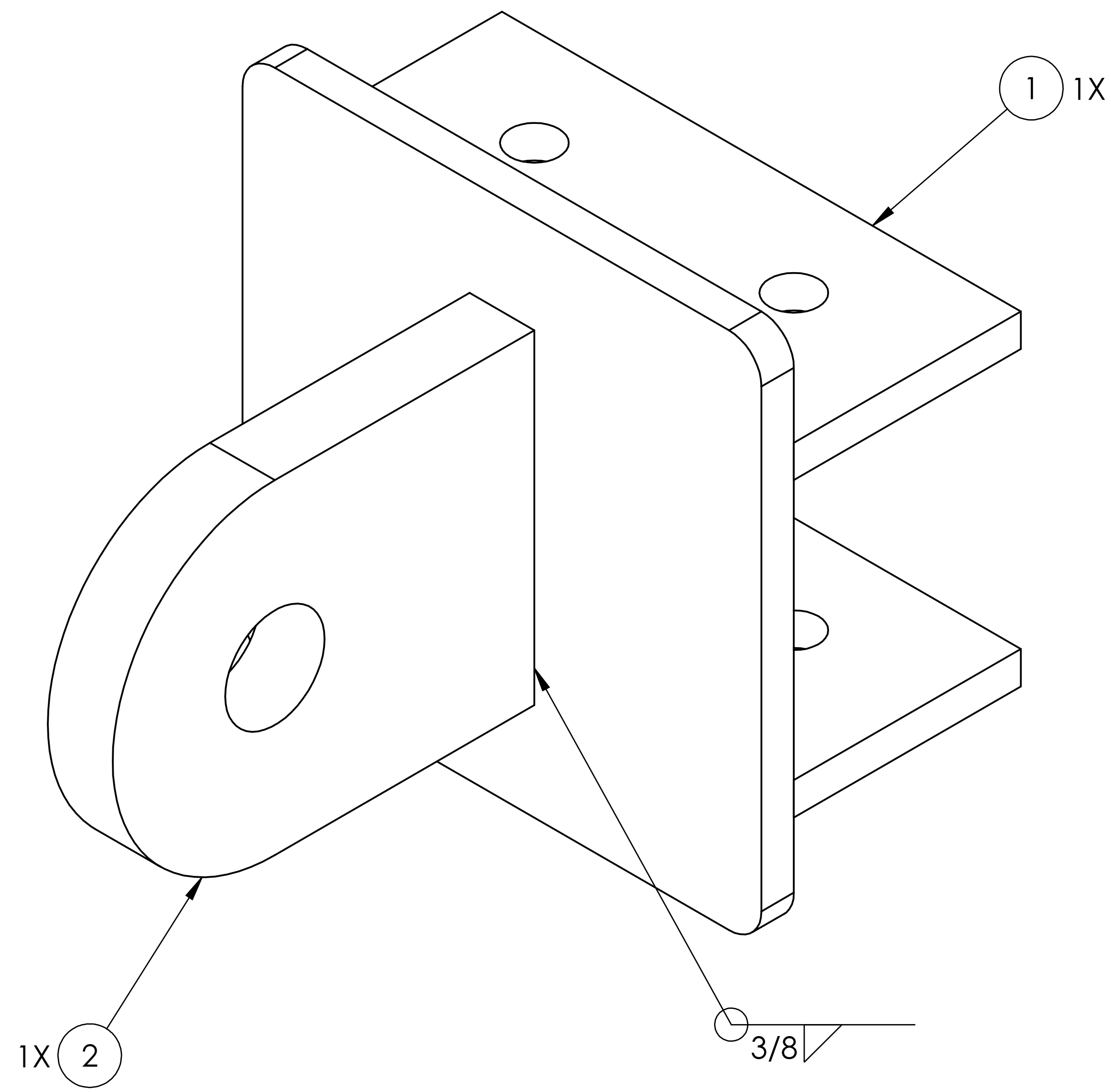


SIDE SUBASSEMBLY

SHEET 3 OF 3

NOTES:

1. 7018 ROD REQUIRED FOR WELDING



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	009	CYLINDER BACK MOUNTING PLATE	1
2	010	FIXED END PIN MOUNT	1

PROPRIETARY AND CONFIDENTIAL

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105	MAIN ASSY
NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±1/8
ANGULAR: MACH ±2°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.05
THREE PLACE DECIMAL ±.01

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL N/A

FINISH N/A

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	B. TRAN	10/5/17
CHECKED	A. HARAKE	10/5/17
ENG APPR.		
MFG APPR.		
Q.A.		

HIGHRISE

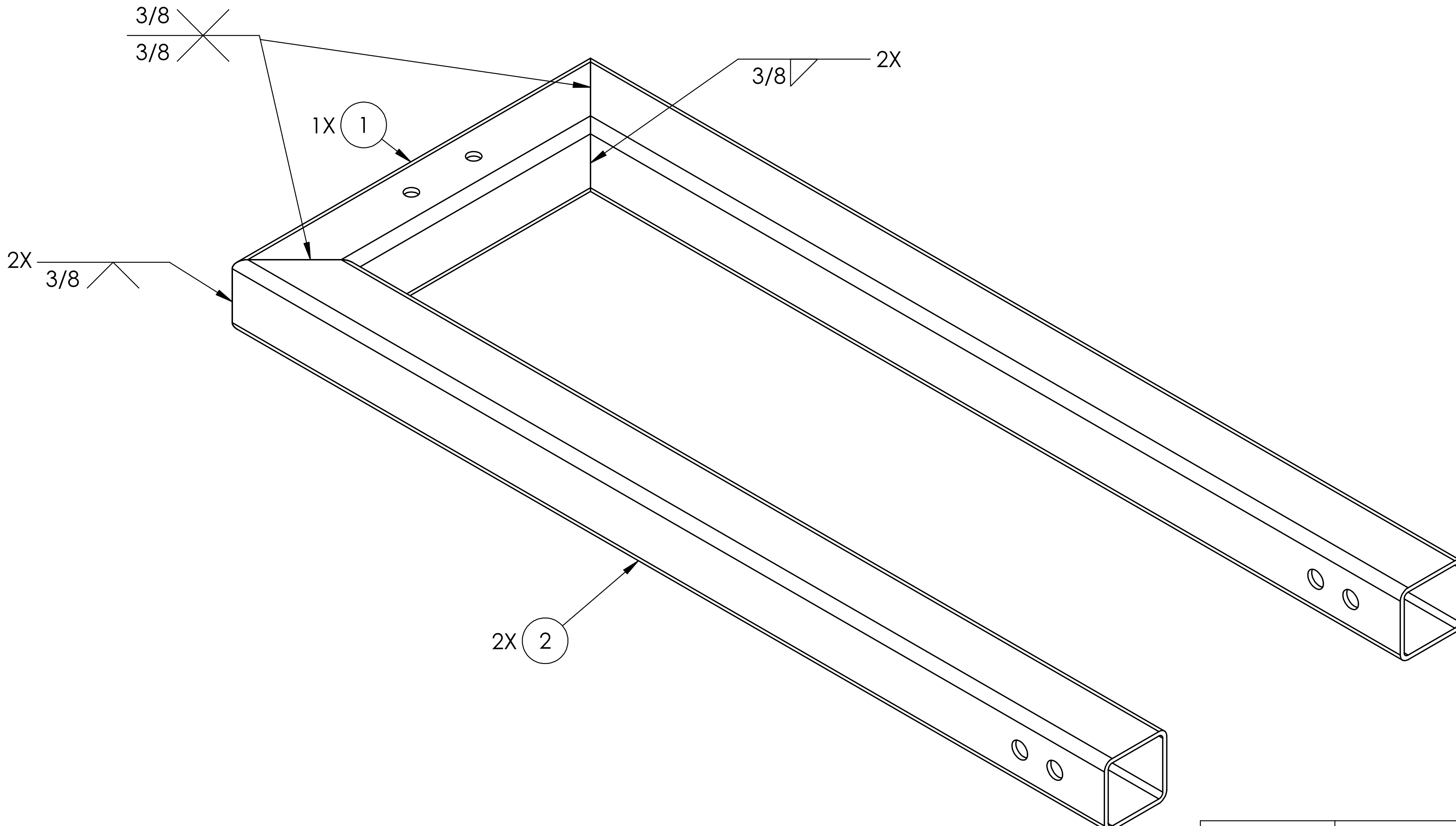
TITLE:
CYLINDER MOUNT ASSEMBLY

SIZE B | DWG. NO. 102 | REV 02

SCALE: 1:2 | WEIGHT: 24.67 LB | SHEET 1 OF 1

NOTES:

1. 7018 ROD REQUIRED FOR WELDING



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	011	TUBE FRAME BACK	1
2	012	TUBE FRAME SIDES	2

PROPRIETARY AND CONFIDENTIAL

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NEXT ASSY	USED ON
104	FRAME ASSY

APPLICATION

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 1/8
ANGULAR: MACH ± 2°
ONE PLACE DECIMAL ± .1
TWO PLACE DECIMAL ± .05
THREE PLACE DECIMAL ± .01

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL N/A

FINISH N/A

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	B. TRAN	10/5/17
CHECKED	A. HARAKE	10/5/17
ENG APPR.		
MFG APPR.		
Q.A.		

HIGHRISE

TITLE: TUBE FRAME ASSEMBLY

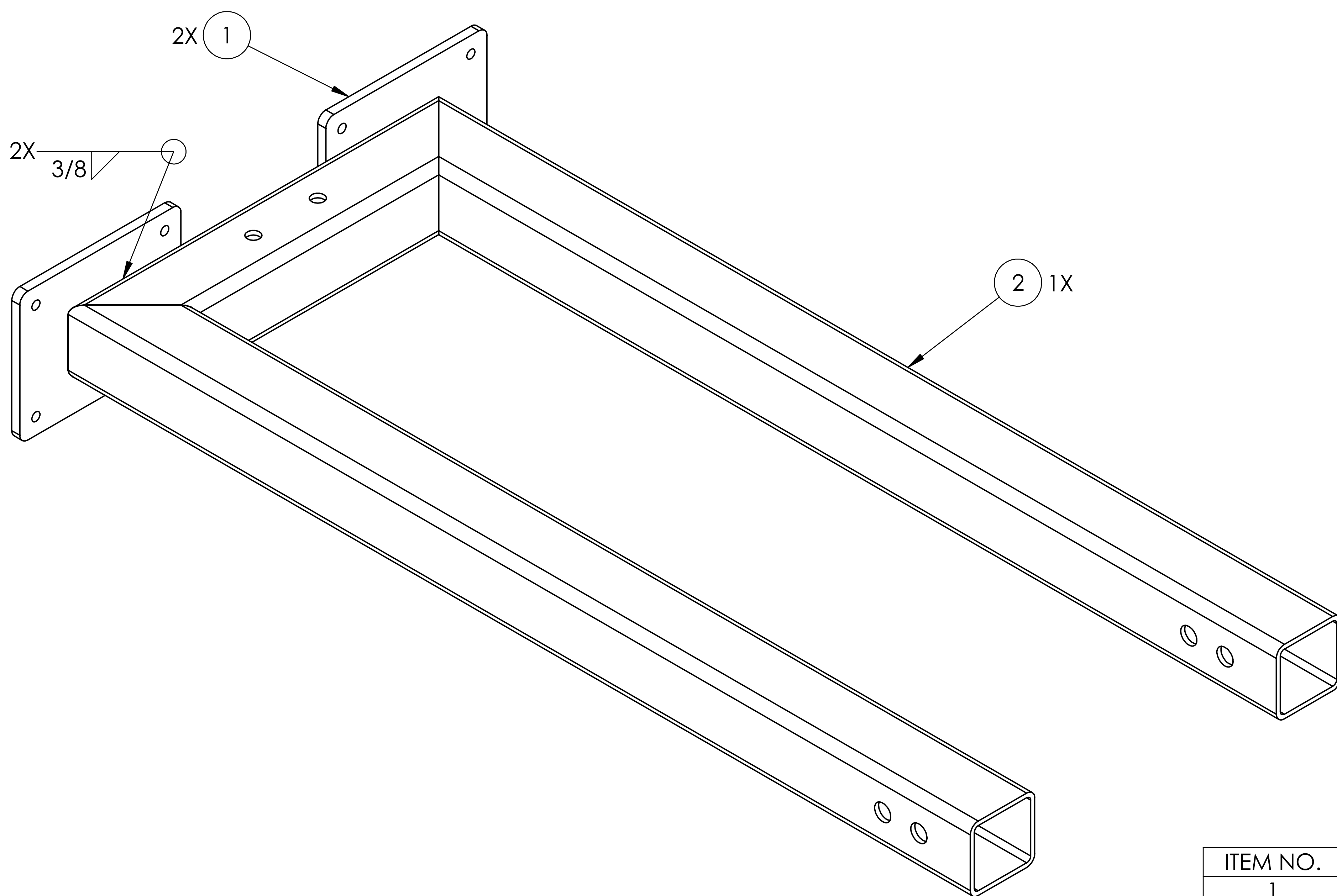
SIZE B | DWG. NO. 103 | REV 02

SCALE: 1:6 | WEIGHT: 128.70 LB | SHEET 1 OF 1

4 3 2 1

NOTES:

1. 7018 ROD REQUIRED FOR WELDING



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	013	MAIN MATCH PLATE	2
2	103	TUBE FRAME ASSEMBLY	1

PROPRIETARY AND CONFIDENTIAL

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105	MAIN ASSY
NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 1/8
ANGULAR: MACH ± 2°
ONE PLACE DECIMAL ± .1
TWO PLACE DECIMAL ± .05
THREE PLACE DECIMAL ± .01

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL N/A

FINISH N/A

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	B. TRAN	10/5/17
CHECKED	A. HARAKE	10/5/17
ENG APPR.		
MFG APPR.		
Q.A.		

HIGHRISE

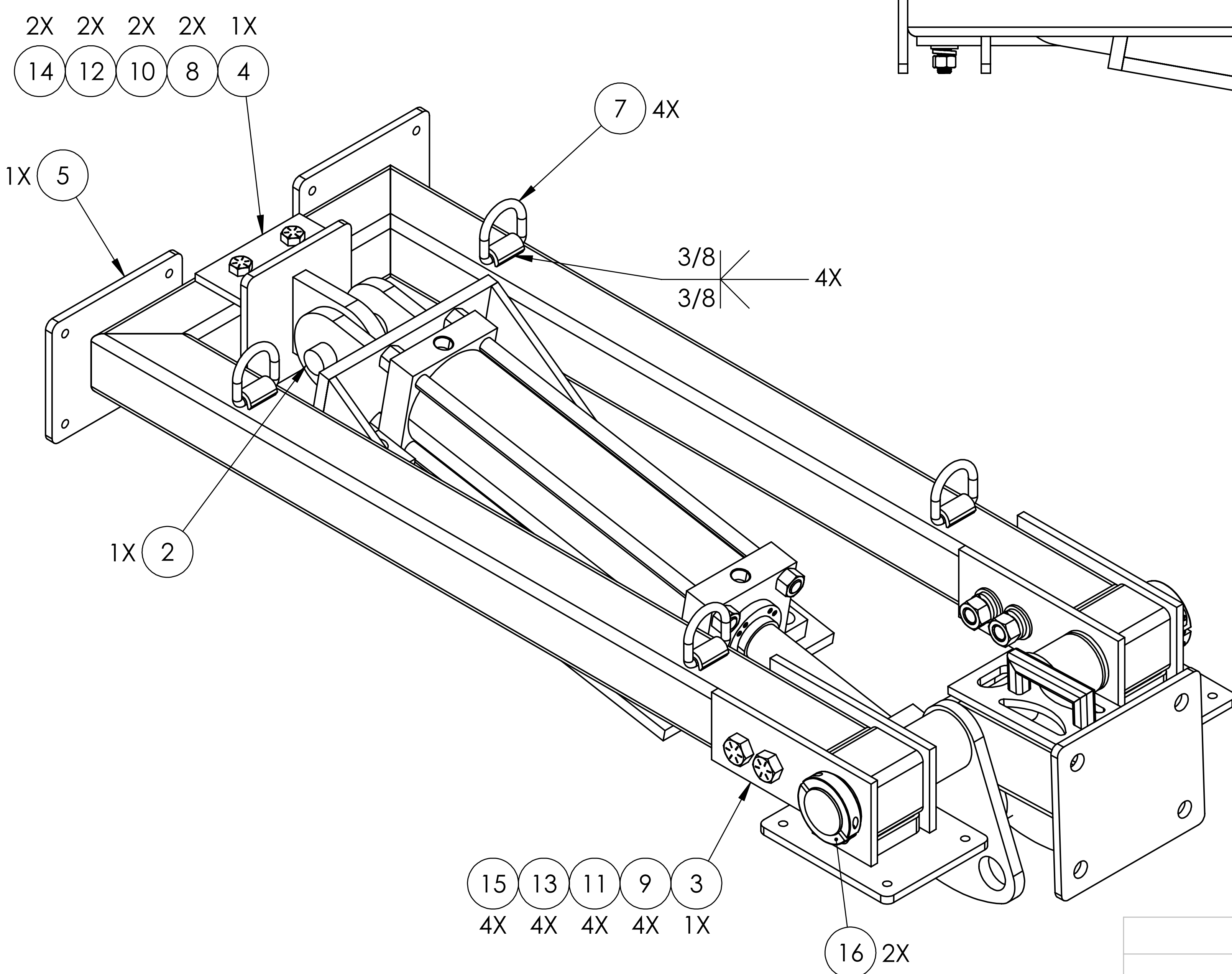
TITLE: FRAME ASSEMBLY

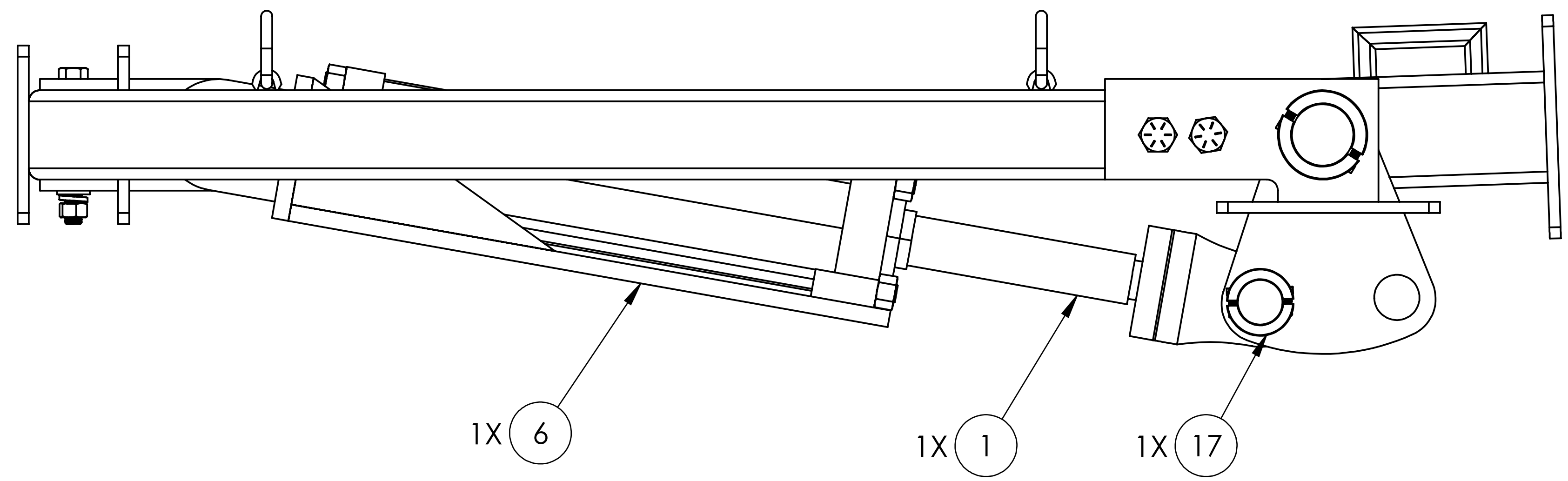
SIZE B | DWG. NO. 104 | REV 02

SCALE: 1:6 | WEIGHT: 151.10 LB | SHEET 1 OF 1

NOTES:

1. 7018 ROD REQUIRED FOR WELDING





ITEM NO.	PART NUMBER	DESCRIPTION	Default/ QTY.
1	014	CYLINDER COLLAR	1
2	015	CYLINDER PIN	1
3	101	SHAFT ASSEMBLY	1
4	102	CYLINDER MOUNT ASSEMBLY	1
5	104	FRAME ASSEMBLY	1
6	N/A	CYLINDER ASSEMBLY	1
7	V3028T32	D-RING	4
8	V91257A860	SCREW, HEX, 3/4"-10 X 6-1/2"	2
9	V91257A970	SCREW, HEX, 1"-8 X 6-1/2"	4
10	V98023A036	WASHER, FLAT, 3/4"	2
11	V98023A038	WASHER, FLAT, 1"	4
12	V91104A047	WASHER, SPLIT LOCK, 3/4"	2
13	V91104A049	WASHER, SPLIT LOCK, 1"	4
14	V94895A036	NUT, HEX, 3/4"-10, GRADE 8	2
15	V94895A038	NUT, HEX, 1"-8, GRADE 8	4
16	V6436K85	CLAMPING SHAFT COLLAR 2.75" DIA.	2
17	V6436K27	CLAMPING SHAFT COLLAR 2" DIA.	1

NEXT ASSY | USED ON

APPLICATION

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±1/8
ANGULAR: MACH ±2°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.05
THREE PLACE DECIMAL ±.01

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL N/A

FINISH N/A

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	B. TRAN	10/5/17
CHECKED	A. HARAKE	10/5/17
ENG APPR.		
MFG APPR.		
Q.A.		

HIGHRISE

TITLE: MAIN ASSEMBLY

SIZE B | DWG. NO. 105 | REV 02

SCALE: 1:8 | WEIGHT: 684.2 LB | SHEET 1 OF 1

FMEA Form v3.0

FMEA Form

Process/Product Name: Overheight Mechanism

Prepared By: Sergio Gutierrez

Process Step/Input	Potential Failure Mode	Potential Failure Effects	SEVERITY (1 - 10)	Potential Causes	OCCURRENCE (1 - 10)	Action Recommended	DETECTION (1 - 10)
What is the process step or feature under investigation?	In what ways could the step or feature go wrong?	What is the impact on the customer if this failure is not prevented or corrected?		What causes the step or feature to go wrong? (how could it occur?)		What are the recommended actions for reducing the occurrence of the cause or improving detection?	
Hydraulic Cylinder	Seal breaking	Cylinder will not work properly, needs repair or replacement	5	Dirt, mad, etc getting in the lines	3	Inspect the lines, improve storage and treatment of lines, clean lines	5
	Rod fracture	Cylinder needs to be replaced, system design needs changing	8	Incorrect loading on rod	2	Proper set up of cylinder	7
	Insufficient Pressure	Cylinder will not push/pull, cylinder sizing needs change since pressure cannot change	4	Force required exceeds the capabilty of the pressure we can supply	3	Proper sizing of cylinder so the pressure supplied will produce the proper force	5
Shaft-in-Pipe Bearing	Grease stiffing up	Rotation about the pivot will be more diffilcult or not occur at all	6	Improper grounding when welding	2	Don't ground connecting material	5
Mounts	Breaking	Cylinder will be out of place, fall if broken	6	Improper welding or too much stress on welds	3	Proper welding, inspecting the welds	8
Main Shaft	Buckling	System failure, will not work. Damage will extend to other cmponents	9	Selected sizing or material of shaft is incapable of handling the loading	2	Design mechanism with a shaft that is capable of handling the loading	4
Fasteners	Shear	System damage, misalignment. Undesired & uncontrolled motion	8	Too much shear force on the fasteners from the weight	3	Using high grade fasteners. Calculate the safety factor of the bolts in shear	2
Welds	Welds break	Depends on location, can increase the force that other components feel which can cause further failures	7	Incorrect welding procedure	3	Proper welding procedure and calculate safety factor on the welds	5

ME428/429/430 Senior Design Project 2015-2016

SENIOR PROJECT CONCEPT DESIGN HAZARD IDENTIFICATION CHECKLIST

Team: HighRise Advisor: Fabijanic

Y	N	
☒	☐	Will any part of the design create hazardous revolving, reciprocating, running, shearing, punching, pressing, squeezing, drawing, cutting, rolling, mixing or similar action, including pinch points and sheer points?
☒	☐	Can any part of the design undergo high accelerations/decelerations?
☒	☐	Will the system have any large moving masses or large forces?
☐	☒	Will the system produce a projectile?
☒	☐	Would it be possible for the system to fall under gravity creating injury?
☒	☐	Will a user be exposed to overhanging weights as part of the design?
☐	☒	Will the system have any sharp edges?
☐	☒	Will any part of the electrical systems not be grounded?
☐	☒	Will there be any large batteries or electrical voltage in the system above 40 V either AC or DC?
☒	☐	Will there be any stored energy in the system such as batteries, flywheels, hanging weights or pressurized fluids?
☐	☒	Will there be any explosive or flammable liquids, gases, or dust fuel as part of the system?
☐	☒	Will the user of the design be required to exert any abnormal effort or physical posture during the use of the design?
☒	☐	Will there be any materials known to be hazardous to humans involved in either the design or the manufacturing of the design?
☐	☒	Can the system generate high levels of noise?
☒	☐	Will the device/system be exposed to extreme environmental conditions such as fog, humidity, cold, high temperatures, etc?
☒	☐	Is it possible for the system to be used in an unsafe manner?
☐	☒	Will there be any other potential hazards not listed above? If yes, please explain on reverse.

For any "Y" responses, add a complete description, list of corrective actions to be taken, and dates to be completed on the reverse side.

Description of Hazard	Corrective Actions to Be Taken	Planned Completion Date	Actual Completion Date
The retracting cylinder presents a pinch point.	A pinch point warning sticker will be placed in a visible space near the hazard.	2/15/17	5/1/18
If broken, the overheight structure will quickly accelerate downwards due to gravity.	one method to prevent the structure from falling quickly would be to have a chain that wraps through a handle to hold the structure up.	2/13/17	5/1/18
The overheight mechanism will be lifting a massive section of the float weighing between 1000 lbs to 1500 lbs.	The safety factor for all components will be at least 1.5 for a maximum capacity of 1500 lbs.	10/4/17	10/4/17
when unpressurized, the structure will fall under gravity.	Again, the chain and handle would address this problem.	2/13/17	5/1/18
When the overheight mechanism is in the up position, a weight of up to 1500 lbs is being held in the air.	When people are going to be around the mechanism, the chain will be hooked on. The crew compartment will not be under the mechanism.	2/15/17	5/1/18
Because Cal Poly Rose Float utilizes hydraulic systems, the fluid is highly pressurized to create enough force to support the structure.	we will attach a sticker indicating that there is a pressurized fluid hazard.	2/15/17	5/1/18

Description of Hazard	Corrective Actions to Be Taken	Planned Completion Date	Actual Completion Date
The mechanism design requires hydraulic fluid, which presents a hazard. During manufacturing, the welding fumes are a potential hazard.	In the instructions manual, we will state the precautions that should be taken with the hydraulic fluid. We will also take precautions when welding, such as getting fresh air.	1/22/17	3/22/18
Since the Rose Parade is on New Year's, the mechanism must be able to withstand temperatures as low as 30-40°F.	The mechanism will be designed to withstand cold temperatures, and it will also be covered by the float during operation.	9/27/12	9/27/17
Since the mechanism is designed to lift 1000 to 1500 lbs, it can be used as a forklift, which creates a very unsafe scenario.	In the instructions manual, we will emphasize that our mechanism is for use as an overheight mechanism on floats only.	2/15/17	3/22/18

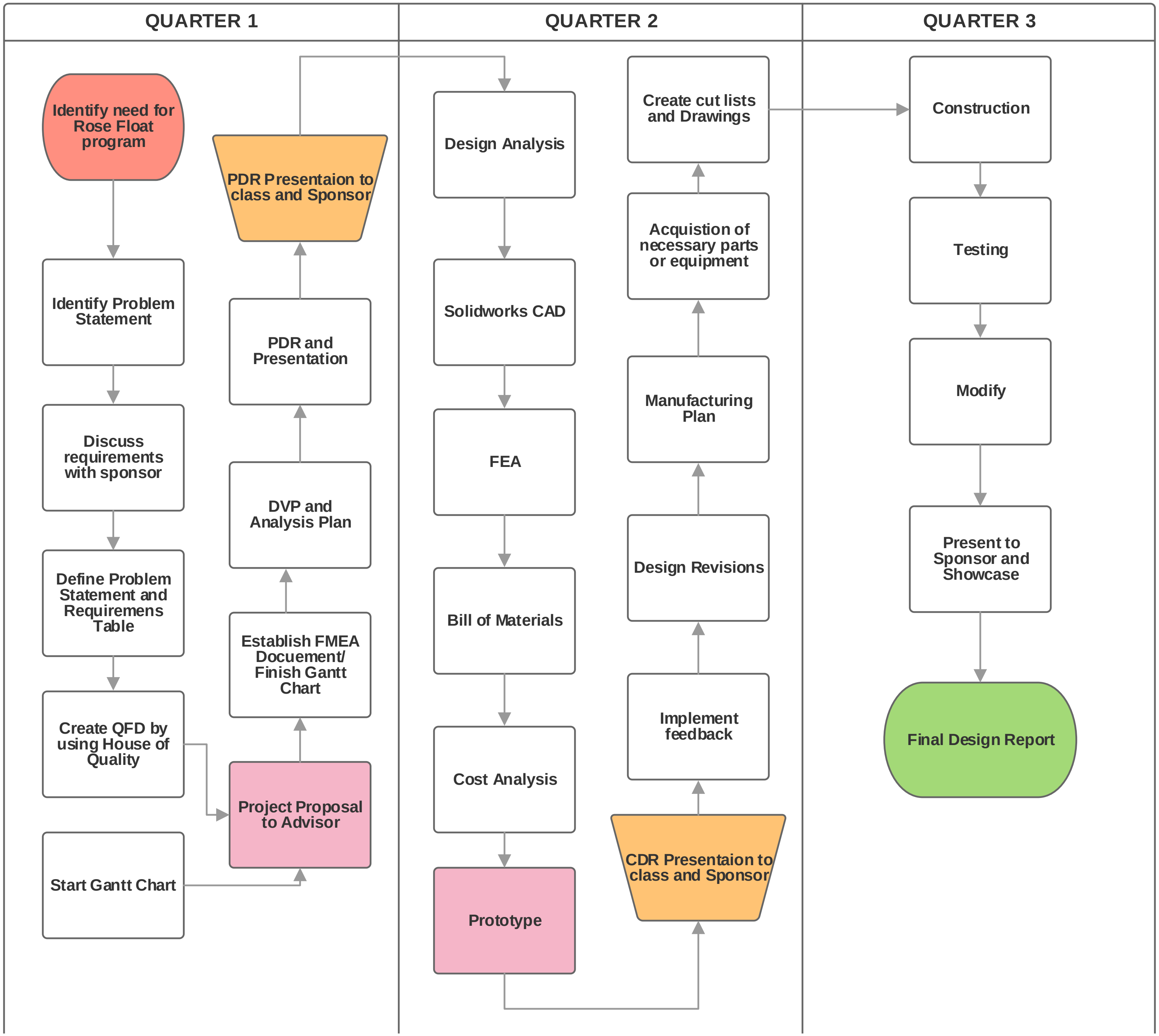
ME428/ME481 DVP&R Format

Report Date		2/19/2018	Sponsor	Josh D'Acquisto	Component/ Assembly		Overheight Mechanism	REPORTING ENGINEER:	Breanna Tran

TEST PLAN										TEST REPORT			
Item No	Specification or Clause Reference	Test Description	Acceptance Criteria	Test Responsibility	Test Stage	SAMPLES TESTED		TIMING		TEST RESULTS			NOTES
						Quantity	Type	Start date	Finish date	Test Result	Quantity Pass	Quantity Fail	
1	Lift Capacity	Overheight mechanism will be loaded with a weight of 1500 pounds to ensure it meets our maximum requirement.	1250 ± 250 lbs	AH	DV	9	B	3/11/2018	3/11/2018	1. Pass 2. Pass 3. Pass 4. Pass 5. Pass 6. Pass 7. Pass 8. Pass 9. Pass	9	0	Operated as expected
2	Distance of Load from Pivot Point	The length of the load lever arm will be measured using a tape measure.	8 ± 2 ft	BT	DV	2	B	3/11/2018	3/11/2018	1. 10 ft 2. 10 ft	2	0	Derived from 3D model
3	System Pressure	A pressure gage will be used to measure the pressure of the hydraulic fluid.	1400 psi MAX	MM	DV	2	B	3/11/2018	3/11/2018	1. 1370 psi 2. 1200 psi	2	0	Value 1 is at initiation, 2 is constant the rest of the lifting time
4	Gravity Drop Time	The overheight mechanism will be released from the up position and the time it takes for the arm to fall down due to gravity will be recorded with a stopwatch.	60 s MAX	SG	DV	3	B	3/11/2018	3/11/2018	1. 50.58 s 2. 29.52 s 3. 21.24 s	3	0	1st run was slow due to caution
5	Height Difference	The height of the end of the load lever arm will be recorded when in the up and down position to calculate the difference.	9 ± 5 ft	AH	DV	2	B	3/11/2018	3/11/2018	1. 10'9" 2. 10'9"	2	0	
6	Operators Required	Each member will attempt to operate the mechanism individually, and if all members can, then the mechanism	1 Operator MAX	MM	DV	4	B	3/11/2018	3/11/2018	1. Sergio 2. Morgan 3. Breanna 4. Ali	4	0	One operator with a director

		satisfies the requirement.											
7	Locking System	The mechanism will be placed in the up position and the hydraulic system turned off. If the lever arm stays relatively fixed when the hydraulic system is shut off, then the requirement is satisfied.	1 Lock MIN	SG	DV	3	B	3/17/2018	3/17/2018	1. Up 2. Up 3. Up	3	0	Operated as expected
8	Unpressurized Position	When the hydraulic system is shut off, the load lever arm should be in the down position or should fall to the down position if the locking system is not activated.	Down Position	AH	DV	3	B	3/11/2018	3/11/2018	1. Down 2. Down 3. Down	3	0	Operated as expected
9	Floats can be used on	Significantly different float designs from the previous 20 years will be analyzed to determine whether our device could be implemented on those floats.	5 MIN	MM	DV	10	B	3/17/2018	3/18/2018	1. 2018-Y 2. 2017-Y 3. 2016-N 4. 2015-Y 5. 2014-Y 6. 2013-Y 7. 2012-Y 8. 2011-N 9. 2010-N 10. 2008-Y	7	3	"Y" denotes the mech would've worked; "N" denotes it would not have worked
10	Volume	The necessary dimensions will be measured with a tape measure, and the volume will be calculated.	15 ft^3 MAX	SG	DV	1	B	3/11/2018	3/11/2018	1. 14.5 ft^3	1	0	

Appendix V


QUARTER 1
QUARTER 2
QUARTER 3
Identify need for Rose Float program
Identify Problem Statement
Discuss requirements with sponsor
Define Problem Statement and Requiremens Table
Create QFD by using House of Quality
Start Gantt Chart
Project Proposal to Advisor
Establish FMEA Docuement/ Finish Gantt Chart
DVP and Analysis Plan
PDR and Presentation
PDR Presentaion to class and Sponsor
Design Analysis
Solidworks CAD
FEA
Bill of Materials
Cost Analysis
Prototype
CDR Presentaion to class and Sponsor
Implement feedback
Design Revisions
Manufacturing Plan
Acquistion of necessary parts or equipment
Create cut lists and Drawings
Construction
Testing
Modify
Present to Sponsor and Showcase
Final Design Report

Modular Overheight Operator's Manual

Where to find on drive:

https://drive.google.com/drive/folders/0B0BrBtXUDIoybG4xZG1kMjBUaUE?usp=sharing

Safety Hazards

- Exercise caution around mechanism, when system is upright there is potential energy, make sure to de-energize system before working on hoses/cylinder.
- Structure is heavy (>685 lbs), utilize mechanized lift to maneuver structure into place.
- Pinch points are plentiful, do not place hands or tools near the system when in use.
- **Max Lift Capacity: 1500 lbs placed at 10 feet from pivot point**
- **Do Not Exceed Max Lift Cap. hydraulic cylinder will physically not be able to lift it.**
- **Max System Pressure: 1400 psi**

<u>Features Overview</u>

- **<u>Removability</u>**
  - Back cylinder mount uses ¾ x 6.5” hex bolts, and front bearing mounts are attached with 1 x 6.5” hex bolts. System is bolted together for ease of removal and so that the cylinder can be removed.
- **<u>Narrow Design</u>**
  - **Volume:** 14.5 ft^3
  - **Length:** 69”
  - **Width:** 24”
  - **Height:** 15.2”
- **<u>Multiple Configurations</u>**
  - **3 setups:** Horizontal, Vertical, and 45**°** mounting options described below in configuration setup
- **<u>Ease of Movement/Mounting</u>**
  - D-Rings attached at 4 locations on top of structure
- **<u>Safety</u>**
  - Safety Chain can be used when working underneath the system to protect from accidental lowering of mechanism or extreme failure
- **<u>Maintenance</u>**
  - Zerk Fittings on pipe sleeves for lubrication and upkeep of shaft

<u>Initial Setup</u>

1. Determine which orientation will work best for the current year’s overheight requirements. See below for 3 different orientation setups
2. Consult with team on if this device will fit in space.
3. Build mounting supports that will matchplate to the mechanism.
4. Build support for overhanging structure using special 1 in bolt match plates (different than the standard match plates).
5. Avoid welding to the mechanism to preserve life.
6. Perform initial maintenance.
7. Mount system and plumb into PO’s hydraulic system with accumulators (if necessary).

<u>Different Configurations Setups</u>

<u>Horizontal/Vertical:</u>

1. To use the mechanism in either the vertical or horizontal orientation you must change the pin location from 2 to 1 on the clevis mount (2 plates on the pivot point. Refer to image below.

Pin Placement for Horizontal/Vertical Setup

2. Make sure to build supports for all 4 match plates. See below.

Positions of 4 match plates

3. Overview of setups as seen through CAD.

Vertical Setup (Left) and Horizontal Setup (Right)

45° Setup:

1. To use the overheight mechanism in the 45° orientation first find/attach the 45° angled match plates seen below.

45° Angled Match plates

2. Then you must change the pin location from 1 to 2 on the clevis mount (2 plates on the pivot point. Refer to image below.

Pin Placement for 45° setup

3. Overview of setup as seen through CAD.

Maintenance

- Grease zerk fittings every 6-8 months, using **NLGI 2 Grease**.
- Make sure to keep cylinder in dry area with fluid inside to avoid drying out internal seals and creating leakage.
  - Make sure no warpage or cracking has occurred in the structure before loading.

Grease Fitting location

Additional Notes/Pictures

Testing at 1500 lbs with 10 ft truss.

Back side of mechanism, showing back mounting location and two rear match plate locations

Rear Cylinder mount, must be removed completely to remove cylinder