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Center Lift Trailer Support Structure

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Abstract

Center lift pontoon trailers have a high center of gravity, therefore have the potential for the pontoon and trailer to tip while cornering. There is need for a component to stabilize the pontoon on the trailer while towing. Different concepts of a support system were sketched to determine the best option. The characteristics that were deemed important for the support system were narrowed down to determine the best concept and the connections of the best design were sketched in greater detail. The forces that the support system would have to withstand to stabilize the pontoon on the trailer were calculated. Then the size of the components of the design were selected and 3D modeled. Manufacturing of the support system was optimized by using like materials to lower costs. This design is a low cost option to increase the safety of center lift pontoon trailers while cornering.

Acknowledgement

Special thanks to our project advisor, Dr. Donald Quinn and our readers, Dr. Chang Ye and Dr. Josh Wong. Also, thank you to Terry and Janice Hayes for support during field visits for this project.

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Introduction

Background

The inspiration of this project comes from an entrepreneur, Terry Hayes, who is a center lift pontoon trailer owner. Currently on the market there are two separate types of pontoon trailers, the center lift and the bunk trailer. A center lift trailer is a style of pontoon trailer where the pontoon is supported by the trailer at the mid-section or deck of the pontoon as opposed to a wider trailer with a bunk support under each toon on the pontoon. The toon is the part of the pontoon that is an aluminum cylinder used for flotation. The center lift trailer has the advantage of being able to adjust the height at which the pontoon is suspended above the ground. This is beneficial for clients who launch their pontoon off a ramp in shallower water. Another benefit is that the center lift trailer is more convenient for storage because it allows the pontoon to fit within a garage. Due to these reasons, center lift trailers are very popular, increasing the demand for the design of a support system.

Without a support system, the issue that arises for many center lift owners is the raising of the center of gravity for the trailer and pontoon, reducing the stability of the trailer when hauling the pontoon. Another contributing factor to instability is the small frame of a center lift trailer. The image in figure 1 shows how small the support frame of a center lift trailer is compared to the pontoon.

Figure 1. Center lift trailer with a small frame compared to the large pontoon boat.

Due to the small support frame and high center of gravity, many accidents have occurred like the one shown here in figure 2, where a vehicle was making a turn too abruptly and the trailer tipped over.

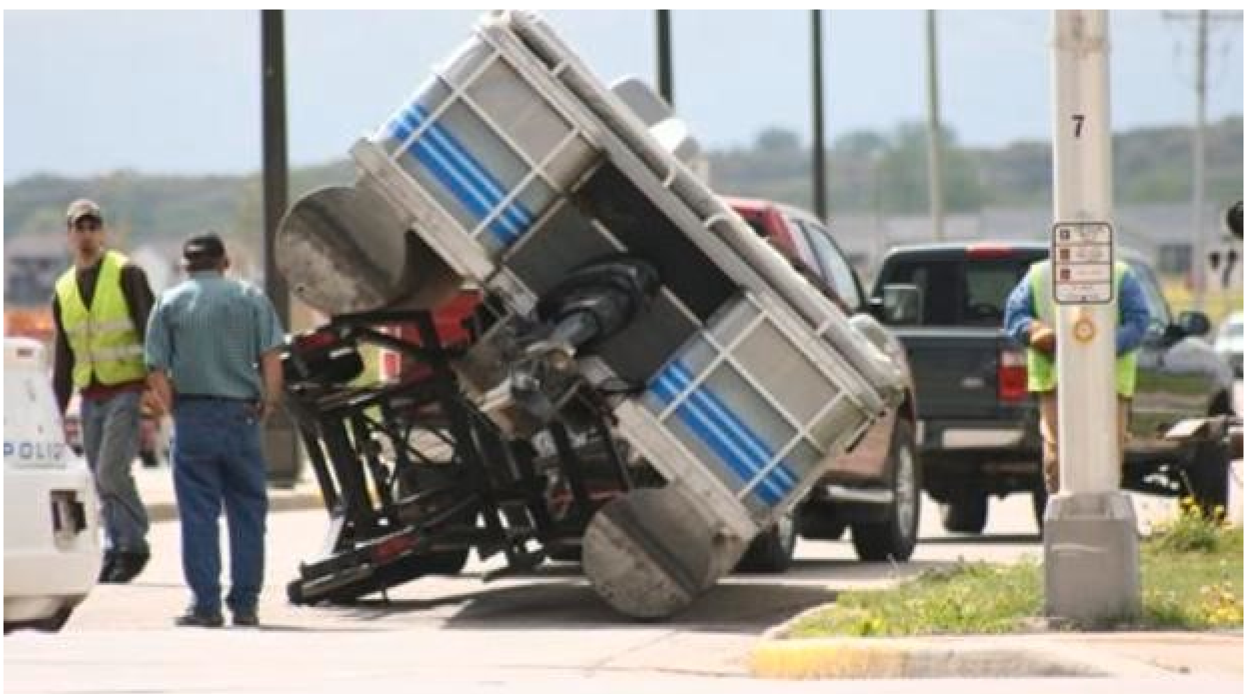


Figure 2. Accidents resulting from sharp turns with a center lift trailer.

The goal of this project was to design a prototype for a removable apparatus that can support the pontoon. The support system would prevent the pontoon from sliding on the trailer. When the pontoon slides on the trailer, the center of gravity of the pontoon shifts from the center of the trailer and the system is susceptible to tipping. This is most apparent during cornering, when the centripetal acceleration of a turn acts on the pontoon and the frictional forces between the pontoon and the trailer are overcome. This allows the pontoon to slide on the trailer and shift the center of gravity. Once designed, this part could be produced and sold as an aftermarket addition for all center lift pontoon owners.

Providing this design to Terry Hayes encompasses the core values of engineering, improvement, and safety. With a support system installed on a center lift pontoon trailer, it will be much safer to travel with and much more efficient to load and unload. These type of improvements are what engineers strive for.

Product Definition

There is need for a designed support structure for a center lift pontoon trailer to stabilize the pontoon on the trailer during cornering. The device will be in contact with the pontoon to support and stabilize. Center-lift pontoons are typically 32 feet in length and 50 inches wide. The height of the bottom of the center lift trailer beams are typically 15 inches from the ground. The device should be lightweight and easily assembled.

Conceptual Design

The premise of this design was to fulfill a need for having an aftermarket, removable attachment for a center lift pontoon trailer to help with stabilization of the pontoon while transporting. The attachment is connected to the base frame of the trailer and in contact with the toon to help stabilize the pontoon on the trailer while in transit. The designed support system is placed in four locations, two on each side of the trailer with a pair in the front and a pair in the back.

Function Structures

The first step in the conceptual design was to determine the functions of the product by creating a black box. Figure 3 shows the black box diagram used to describe the inputs and the outputs of the product.

Figure 3. Black box diagram for trailer support system.

The pontoon support system begins with the pontoon being loaded on the trailer. Next, human energy is needed in the form of putting the support system into position. Once the system is configured properly, the pontoon is able to be towed. The external forces are applied to the pontoon and trailer during transit. This can be from centripetal acceleration and other external forces during towing. The main objective is to keep the pontoon stabilized on the trailer. After towing, the pontoon is still on the trailer and accidents have been averted.

The next step is to create a more detailed black box diagram. Figure 4 shows a more detailed black box, a function structure.

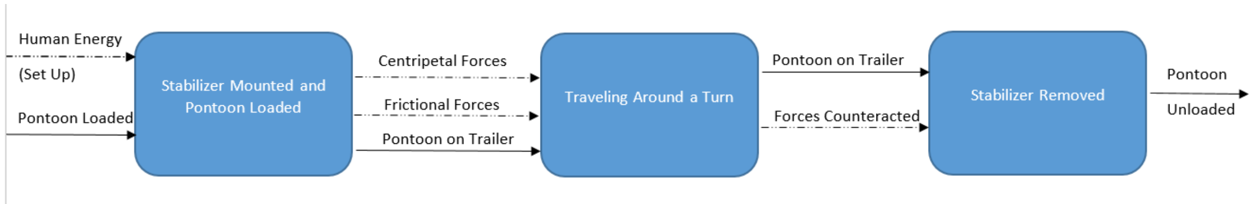


Figure 4. Detailed function structure diagram of pontoon support system.

The function structure adds more details into the entire process of the pontoon design support system. First, the pontoon is loaded onto the trailer. Next, human energy is needed to setup the pontoon stabilizer. Once the system is connected, it is ready for the trailer to be towed. During transit, the pontoon experiences frictional forces between it and the trailer. When cornering, the pontoon experiences centripetal acceleration. The forces that are not countered by the frictional forces are compensated by the designed support system. After the turn, the pontoon is still on the trailer in its intended position. Once towing is complete, the support system needs to be removed for the pontoon to be unloaded from the trailer.

<u>Morphological Chart</u>

The next step in the design process was to brainstorm possible components that could be utilized as a potential solution. In figure 5, a morphological chart displays possible designs.

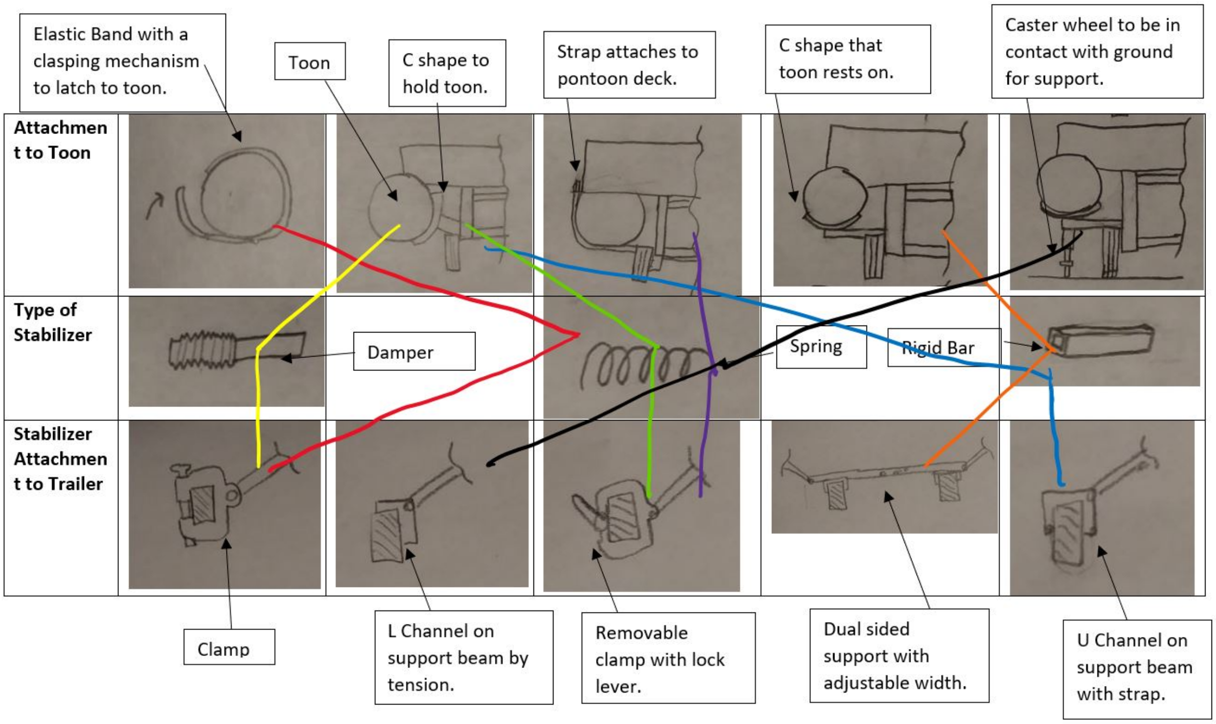


Figure 5. Morphological chart containing possible design solutions.

Included in Design A, the red line, is the elastic band method of attachment to the toon, a spring for stabilizing, and a clamp to attach to the trailer. For Design B, the yellow line, a C-shape side support is to be the form of connection to the toon, a damper for stabilizing, and a clamp to connect to the trailer. The method for Design C, the green line, consists of a C-shape side support attached to the toon, a spring for stabilizing, and a removable clamp with lock lever for attachment to the trailer. Design D, the blue line, is made up of a C-shape side support connected to the toon, a rigid bar for stabilizing, and a U-channel for attaching to the trailer. Incorporated in Design E, the purple line, is a strap for attaching to the toon, a spring for stabilizing, and a removable clamp for connecting to the trailer. Design F, the orange line, is a C-shape bottom support to attach to the toon, a rigid bar for stabilizing, and a dual sided support for connecting to the trailer. The components for Design G, the black line, include a C-shape bottom support to attach to the toon, a spring for stabilizing, and an L-channel with a caster wheel ground support for the trailer connection. Taking each of the prior designs, concept sketches were then drafted.

<u>Concept Sketches</u>

The concept sketches are simplified representations of the different combinations of the components. It was a good start to visual how each component will interact with the pontoon and the trailer. Below are the concept sketches of the different methods. Figure 6 shows a concept sketch of a support system.

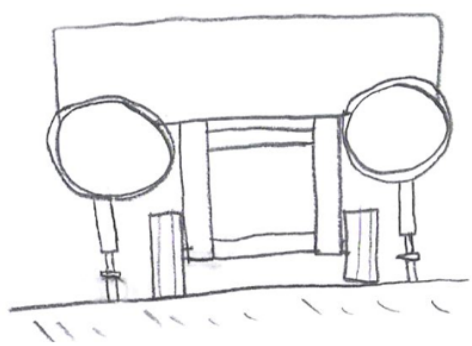


Figure 6. First concept sketch of support system on pontoon.

Above, the use of a spring was used to help gradually stabilize the pontoon as it rotates up and down on the trailer. The caster wheel is in contact with the pontoon and the ground at all times. There is no contact with the pontoon trailer.

The following sketches will only show one half of the pontoon for simplicity. Figure 7 displays another concept sketch of the support system.

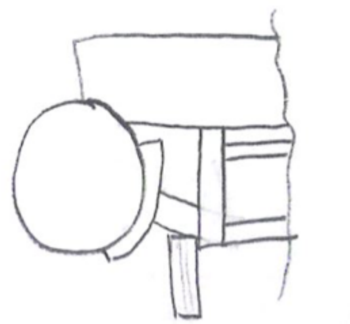


Figure 7. Second concept sketch of support system on pontoon.

This sketch displays a connection to the pontoon trailer with a rigid bar connected to a c-shaped section. The c-shaped section is in contact with the toon to support it. This concept has no contact with the ground, only the pontoon and trailer.

The next sketch is much different from the last sketch. Figure 8 shows another concept sketch of a support system.

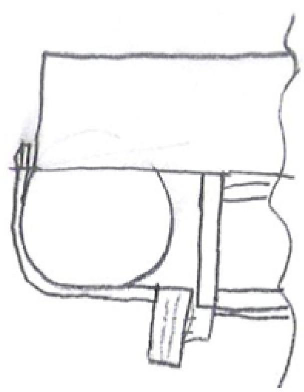


Figure 8. Third concept sketch of support system on pontoon.

The concept sketch above shows an elastic material in connection with the pontoon and the trailer. The elastic material supports the toon as a spring to position it back to its original position when it slides. The elastic material is not a traditional spring, but it can be equated as one to simplify the morphological chart.

Figure 9 shows another concept sketch of a support system on the pontoon.

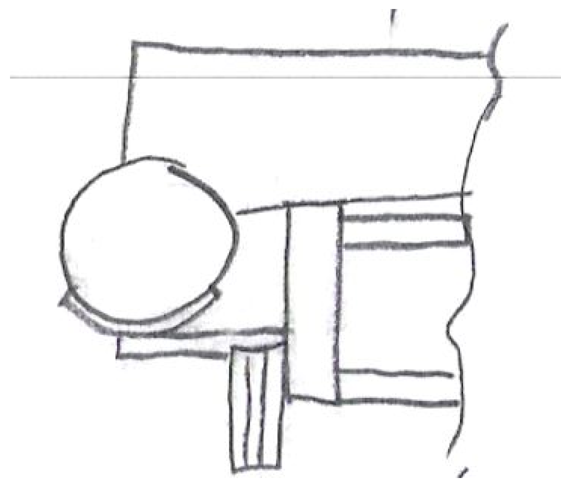


Figure 9. Fourth concept sketch of support system on pontoon.

This concept sketch contains the c-shaped section to cup the toon. It is connected to a rigid bar which is mounted to the trailer. Once again there is no contact with ground. The c-shaped section is in constant contact with the pontoon to deter any movement while towing.

Figure 10 displays another concept sketch of a support system on a pontoon.

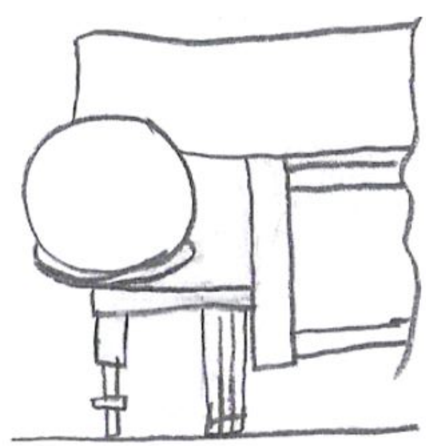


Figure 10. Fifth concept sketch of stabilizer on pontoon.

The final concept sketch contains the c-shaped section and the caster wheel. These are both connected to a rigid bar. The rigid bar is mounted to the trailer. This concept does not allow much movement of the pontoon, but contains many components.

The next step was to determine what characteristics are important and decide which concept to use.

Weighted Decision Matrix

After completing all the potential concept designs, the important characteristics were evaluated that needed to be the focus of the design. To organize the ideas, an objective tree was created, containing all the brainstormed characteristics. Figure 11 shows the objective tree.

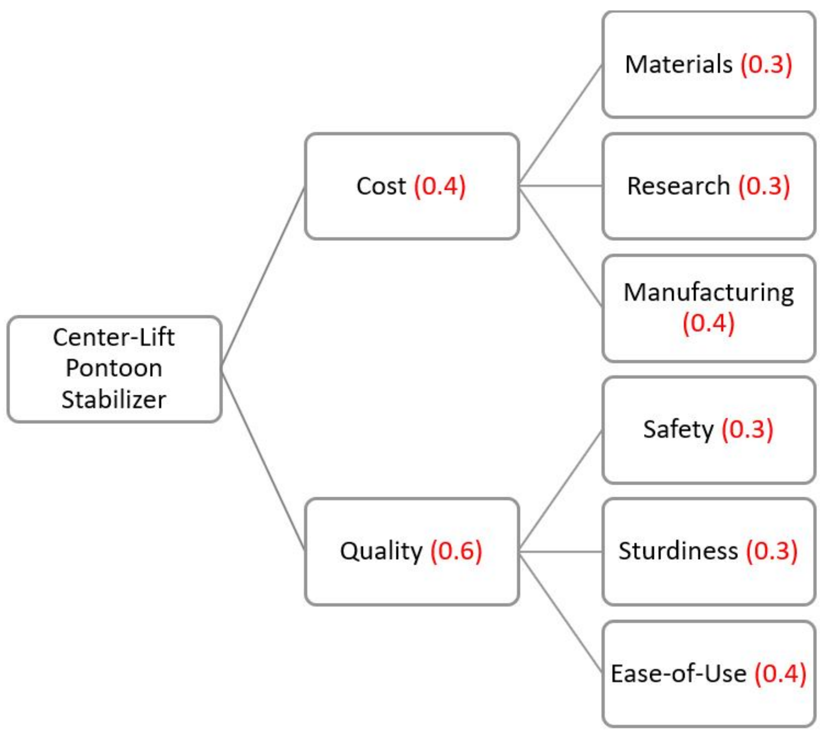


Figure 11. Objective tree containing level of importance values.

Next, the weighted decision matrix was created to compare all the conceptual designs to determine the best option to follow through with as our design.

Table 1

Weighted decision matrix for selecting the best design

Category	Weight Factor	Design A		Design B		Design C		Design D		Design E		Design F		Design G	
		Score	Weight	Score	Weight	Score	Weight	Score	Weight	Score	Weight	Score	Weight	Score	Weight
Materials	0.12	7	0.84	5	0.6	6	0.72	7	0.84	6	0.72	8	0.96	4	0.48
Research	0.12	7	0.84	6	0.72	7	0.84	8	0.96	7	0.84	7	0.84	5	0.6
Manufacturing	0.16	7	1.12	8	1.28	8	1.28	8	1.28	7	1.12	8	1.28	6	0.96
Safety	0.18	6	1.08	8	1.44	7	1.26	7	1.26	6	1.08	5	0.9	3	0.54
Sturdiness	0.18	5	0.9	7	1.26	7	1.26	8	1.44	5	0.9	8	1.44	5	0.9
Ease-of-Use	0.24	8	1.92	6	1.44	6	1.44	7	1.68	8	1.92	8	1.92	7	1.68
Total			6.70		6.74		6.80		7.46		6.58		7.34		5.16

Based on the results of the weighted decision matrix, Design D was the prefered design. Design D is made up of a C-shape side support connected to the toon, a rigid bar for stabilizing, and a U-channel for attaching to the trailer.

Embodiment Design

<u>Product Architecture</u>

The first step in the embodiment design was to determine the functions of the product by using a function structure diagram. This diagram broke down each component of the design into their main functions and what is transmitted between them. Figure 12 shows the function structure diagram of the designed component.

Figure 12. Function structure diagram of support system.

The input force from the turn is transmitted between the contact point, support bar, and mounting point. This meant that each of these separate components and their connections must be designed to withstand the force exerted while turning. The connections between the main components will also have a mechanical purpose such as a length adjuster or a stow away mechanism.

Configuration Design

The best configurations for the design were evaluated using embodiment design principles: division of tasks, simplicity, self help, and stability. With the intention of this product being available to the public, simplicity was at the forefront of most of the decisions. Simplicity would also lower the cost needed to manufacture the product. The division of tasks were completed by having two designed support systems on either side of the trailer to add stability. Figure 13 shows the general layout of the configurations.

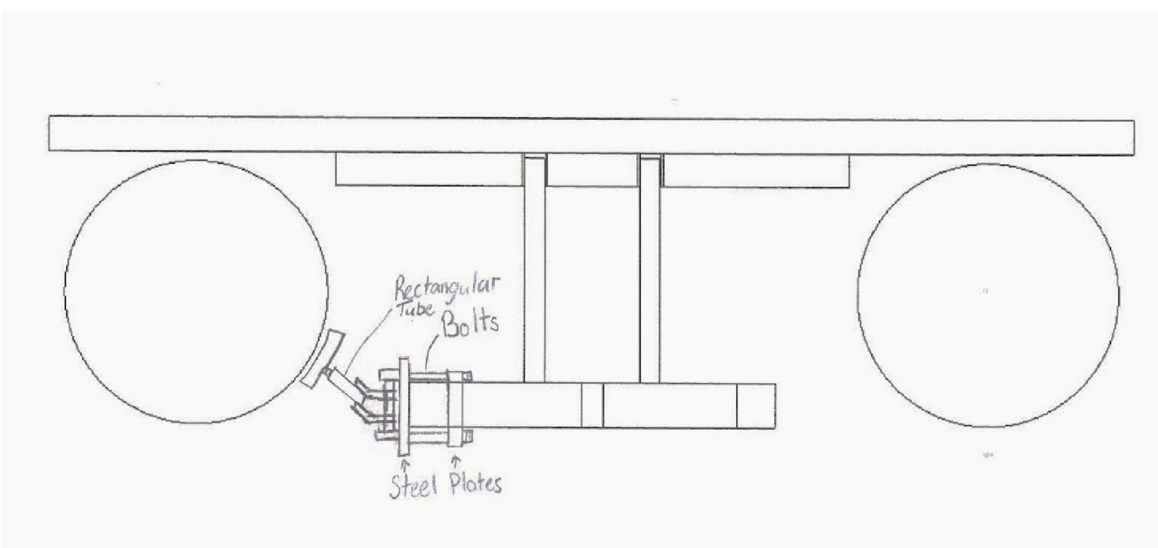


Figure 13. General configuration layout of the pontoon support system.

The general layout gives a simplified idea of how each component interacts with each other. The next step was to further clarify the connections.Figure 14 shows a more detailed connection layout.

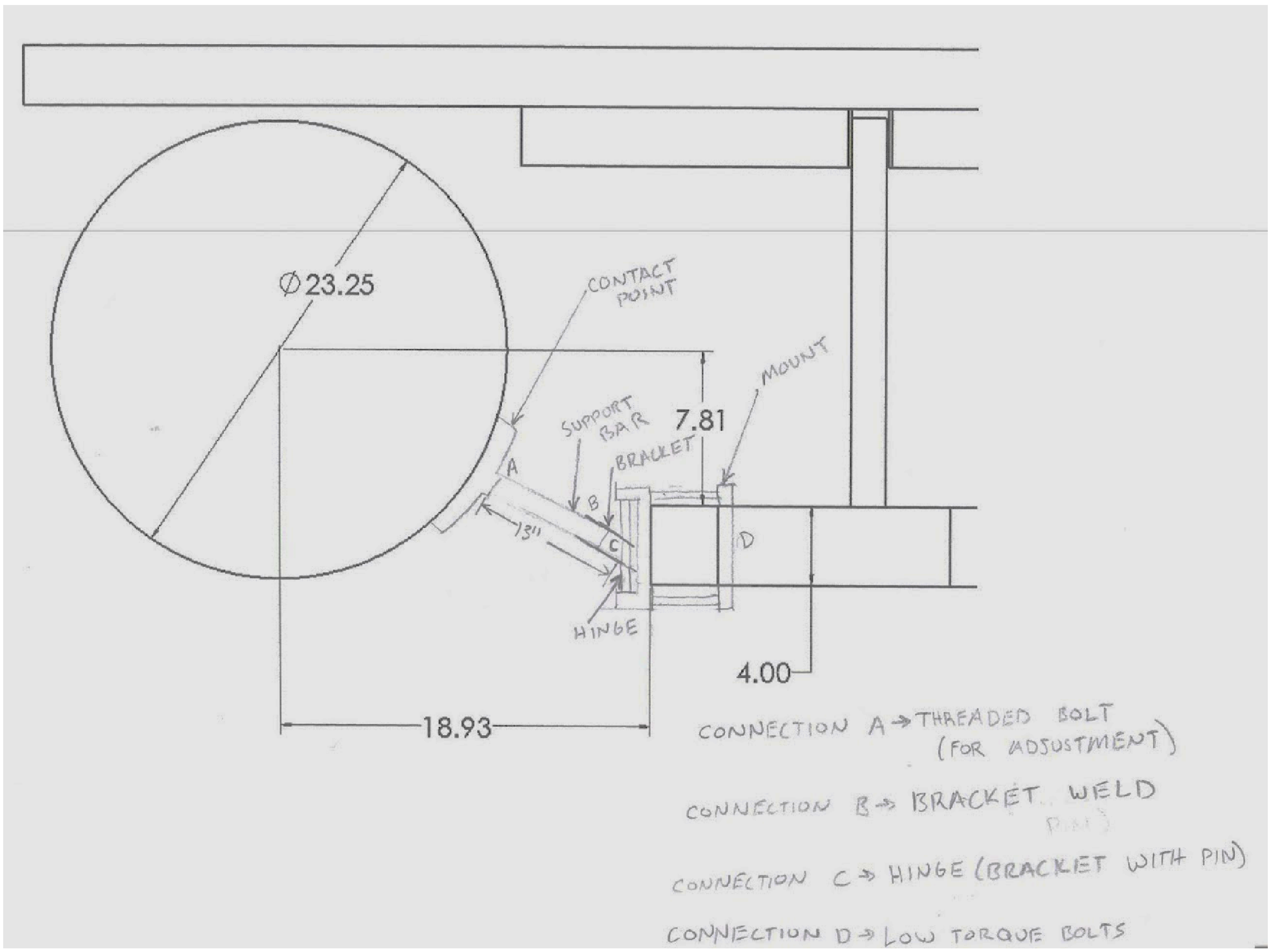


Figure 14. Detailed connection layout diagram of pontoon support system.

The first connection, Connection D, is the mounting plate to the trailer. This was determined to be bolted to the trailer. This was done for aftermarket applications and easy installations. The next configuration, Connection C, is the connection between the pontoon mounting plate and the bracket. This was determined to be a pin welded to the mounting. The bracket would be able to swivel, allowing the stabilizer to be mounted on the trailer while the pontoon is loaded. Connection B, between the bracket and the rigid bar, is welded for a permanent connection. The next connection, Connection A, is between the rigid bar and the contact point for the toon. That was decided to be an adjustable bolt that would move the contact point in and out to maintain contact with the toon, regardless of the toon's position.

<u>Parametric Design</u>

The parametric design focuses on detailing the connection designs and identifying the major forces involved in the overall design.

Connection Designs

The general connections were already determined above in the configuration design. Connection A was determined to be a bolted connection between a plastic contact point and the support bar. Figure 14 shows the bolted connection. The end of the support rod will be drilled and tapped.

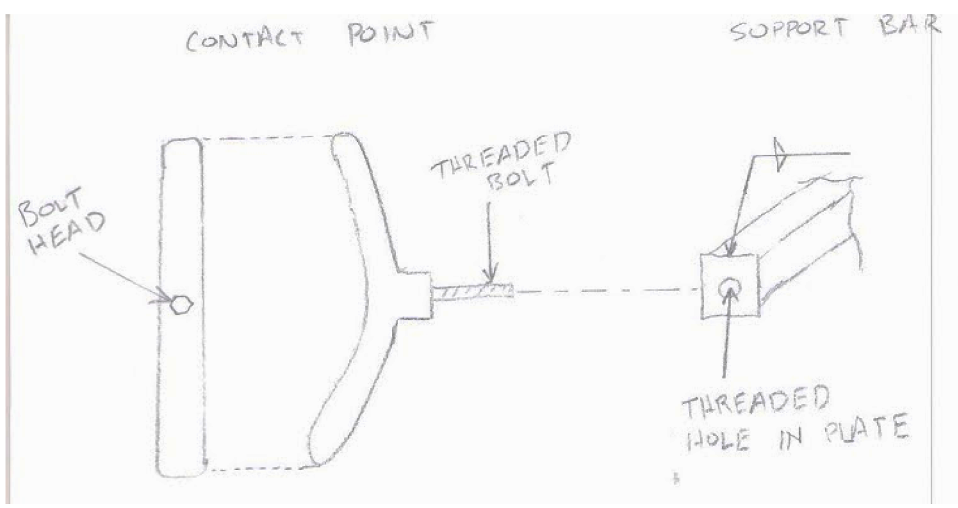


Figure 15. Connection A, threaded bolt and hole.

Connection A was determined to be threaded due to the embodiment design concept of division of task. This connection bears the forces of support but also serves as a way to make fine adjustments to the length of the support. Turning the bolt clockwise would shorten the support and turning the bolt counterclockwise would lengthen the support. This was thought of when considering the fact that trailering a pontoon is not an exact science and the boat may have shifted off center slightly when trailered. These adjustments will guarantee a secure support system even when the boat is not perfectly centered.

Connection B was then considered for design. This connection was determined to be permanent, therefore could be welded. Figure 16 shows the welded joint. Two brackets would be welded to the support bar with equal parallel welds on each bracket.

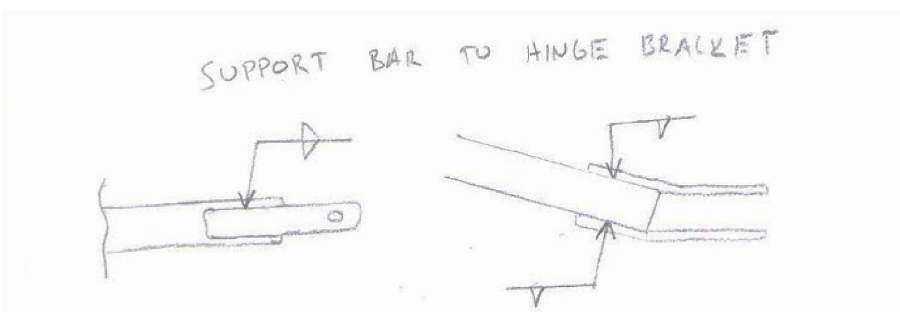


Figure 16. Connection B, brackets welded to support bar.

A welded connection was chosen over a bolted connection for two main reasons. With a bolted connection, wear and tear may be more prevalent if the bolt can shift in the hole in any way or if any rust is present to weaken the bolt. The second reason for not choosing a bolted connection is the extra steps that would be required in manufacturing. Holes would have to be drilled in both the support bar and the hinges, which requires more time and possibilities for error. This will be a relatively small weld, requiring small amounts of time and effort.

Connection C is the the hinge connection between the mount and the support bar. A hinge was needed to allow rotation of the support to be out of the way when launching the pontoon. Figure 17 shows the hinge design..

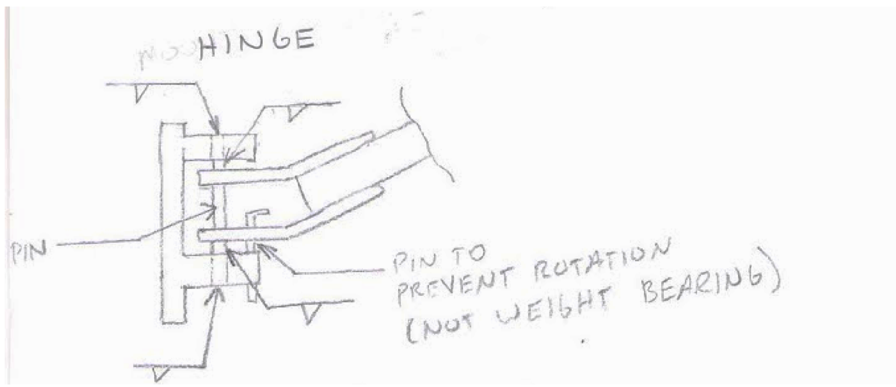


Figure 17. Connection C, hinge connection between support bar and mount.

Many options were considered for this connection due to the fact that it had to be able to move the support out of the way when launching the boat. A side to side rotation was decided on because it would allow the support to be adjusted to length and rotated into position tightly up against the toon. Once in position, the design of a locking pin was added to prevent any rotation out of place due to vibrations when traveling.

Connection D is the mount to the trailer. Four bolts are used to clamp the mount plate onto the bottom bar of the trailer. Figure 18 shows this mount. It will clamp on the inside and outside of the bottom bar of the trailer.

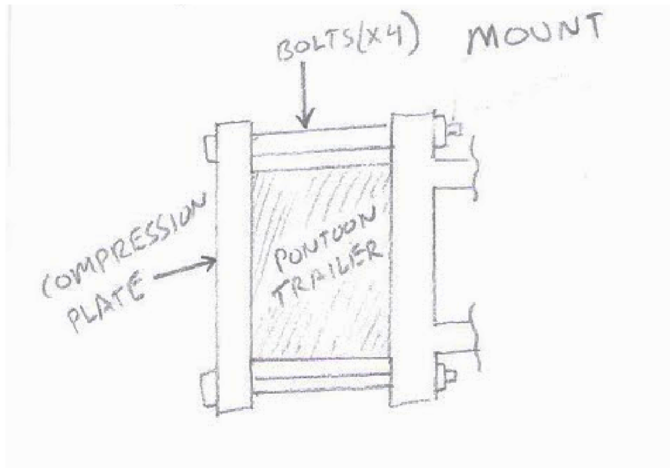


Figure 18. Connection D, bolted trailer mount.

This connection design uses the embodiment principle of self-help by putting all of the force on the already existing trailer beam. By mounting the plated on the inside and outside, the largest force being transmitted will be pushing against the beam instead of an extra part added on. The four bolts are simply there to hold the weight of the support, therefore they will not have to be torqued down too tightly and can be installed by a customer.

Major Forces

The next part of the parametric design was to determine the major forces that would affect the design of the components. The product definition states that the components will support the pontoon while towing. The forces that will affect the pontoon when towing are from centripetal acceleration encountered when turning.

To calculate these forces, a free body diagram was set up with a top down view of the pontoon and trailer going around a turn. This can be seen in figure 19. The forces included are the friction force between the pontoon and the trailer (F_T) and the effects from centripetal acceleration (F_C). It was assumed that no slip occurred between the tires and the road, so no force was considered.

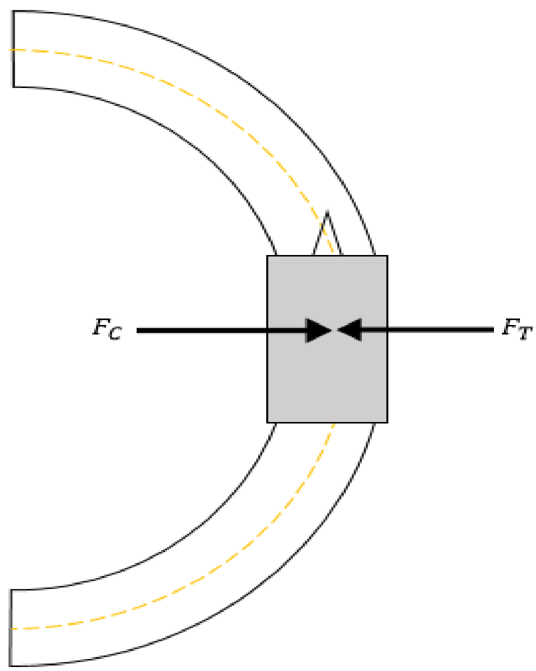


Figure 19. Free body diagram of pontoon traveling around a curve.

The force to be supported by the designed system, F_S , was then added to the free body diagram as seen in Figure 20. The force being supported by the designed system is opposite the direction of the centripetal force.

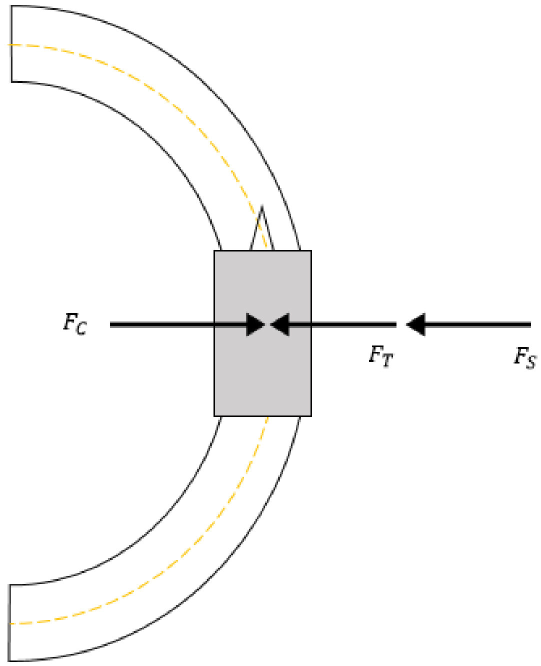


Figure 20. Free body diagram of pontoon traveling around a curve with support force.

To find the support force needed, the free body diagram in figure 20 was solved for static equilibrium,

$$F_S = F_C - F_T \qquad \text{(Equation 1)}$$

F_C was found using the equation for centripetal force,

$$F_C = \frac{mv^2}{r} \qquad \text{(Equation 2)}$$

where m is the mass of the pontoon, v is the tangential velocity, and r is the radius of the turn.

F_T was found using the equation for static friction,

$$F_T = \mu N \qquad \text{(Equation 3)}$$

where μ is the coefficient of friction between the pontoon and the trailer and N is the normal force acting on the trailer from the pontoon.

The final equation used to determine the support force needed to stop the pontoon from slipping on the trailer is found by combining equations 1,2, and 3,

$$F_S = \frac{mv^2}{r} - \mu N \qquad \text{(Equation 4)}$$

In equation 4, there are two constants: mass of the pontoon (*m*) and normal force (*N*). The weight of the pontoon was given as 2,200 lbf. This means *N*=2,200 lbf. Dividing the normal force by 32.2 ft/s/s gives the mass of the pontoon: *m*=68.32 lbm. The remaining three variables in equation 4 were varied for different towing velocities (*v*), different turn radii (r), and different friction factors between the boat and trailer (μ). Friction factor was varied between wet galvanized steel (0.3) and dry galvanized steel (0.1). Table 2 displays the different iterations used to find the largest support force needed to stop the pontoon from sliding.

Table 2

Iterations of different forces exerted on pontoon from centripetal acceleration

Weight(lbf)	2200	2200	2200	2200	2200	2200	2200
Friction Factor	0.3	0.1	0.1	0.1	0.1	0.1	0.1
Speed(mph)	15	15	25	10	45	60	60
Turn radius(ft)	50	50	200	50	200	300	250
Friction Force(lbf)	660	220	220	220	220	220	220
Centripital Force (lbf)	661.3664596	661.3665	459.2823	293.9406	1488.075	1763.644	2116.373
Support Force Needed	1.366459627	441.3665	239.2823	73.94065	1268.075	1543.644	1896.373

From table 2, it was found that the largest support force needed was at a speed of 60 mph on a turn with radius 250 ft, and a friction factor of 0.1. Substituting these values into equation 3 gave

$$F_S = \frac{\frac{2200\ lbf}{32.2\ ft/s/s}\left[(60\frac{mi}{hr})(5280\ \frac{ft}{mi})(\frac{1hr}{60\ min})(\frac{1\ min}{60\ sec})\right]^2}{250\ ft} - (.1)(2200\ lbf) = 1,896.373\ lbf$$

Therefore the horizontal force required to stop the pontoon from sliding is 1,896.373 lbf.

Failure Modes and Effect Analysis

Safety was the main focus of this design. The design support system adds extra security for the user to avoid the pontoon tipping while driving. If the design fails, then the design was for nought. Failure Modes and Effect Analysis (FMEA) was conducted to determine any potential failures of the support system. Only two potential modes were considered for failure. Figure 21 shows the results of the FMEA.

Process Step/Input	Potential Failure Mode	Potential Failure Effects	SEVERITY (1 - 10)	Potential Causes	OCCURRENCE (1 - 10)	Current Controls	DETECTION (1 - 10)	RPN
What is the process step, change or feature under investigation?	In what ways could the step, change or feature go wrong?	What is the impact on the customer if this failure is not prevented or corrected?		What causes the step, change or feature to go wrong? (how could it occur?)		What controls exist that either prevent or detect the failure?		
Putting the stabilizer in postion	*Pin does not lock into position*	*Stabilizer will not be effective*	*10*	*User not putting pin in position*	*1*	*Visual Inspection*	*4*	*40*
Stabilizer is not in contact with toon	*Pontoon is able to slide on the trailer*	*Pontoon could tip*	*10*	*User not putting stabilizer in position*	*2*	*Visual Inspection*	*4*	*80*

Figure 21. FMEA of support system of two potential modes.

The risk priority number (RPN) was rather low for the two cases, the maximum RPN being 300. The scores were well below that and even below the threshold of 100, meaning that the design is not likely to fail. It is worth noting that the severity if there is failure is an extremely high chance of severity. That is mainly due to the two necessary functions of the support system. If those are not achieved, the support system has no way to be effective. The user is responsible for positioning the support system and ensuring it is in position for their specific trailer-pontoon combination. The likelihood of the user not utilizing the support system correctly is unlikely because of the simplicity of the process. It should not take very long to effectively position the support system in its proper location. From above, the chances of failure for these modes are unlikely for the support system.

Detail Design

Load Paths

The main load on the designed component is the horizontal force needed to prevent sliding of the pontoon. This was found to be 1,896.373 lbf. This was divided between two stabilizers so $F_s = 948.19\ lbf$ or $F_s \approx 950\ lbf$. Figure 22 shows how this force, F_s, interacts with the designed component. Force F_s was broken into an x-component and a y-component, using the coordinate directions shown in the figure. This coordinate system was chosen so that the x-direction would be the longitudinal compressive forces on the system and the y-direction would be the direction considered when finding the moment acting on the system. The force in the x-direction ($F_{s,x}$) acts at 23.5° from the horizontal, therefore can be found using trigonometry,

$$F_{s,x} = F_s cos(\theta) \qquad \text{(Equation 5)}$$

Substituting values for F_s and θ gives

$$F_{s,x} = (950\ lbf)cos(23.5°) = 871\ lbf$$

This force acts on the contact point of the designed component, shown in the figure as a red arrow. The transmission of this force is also

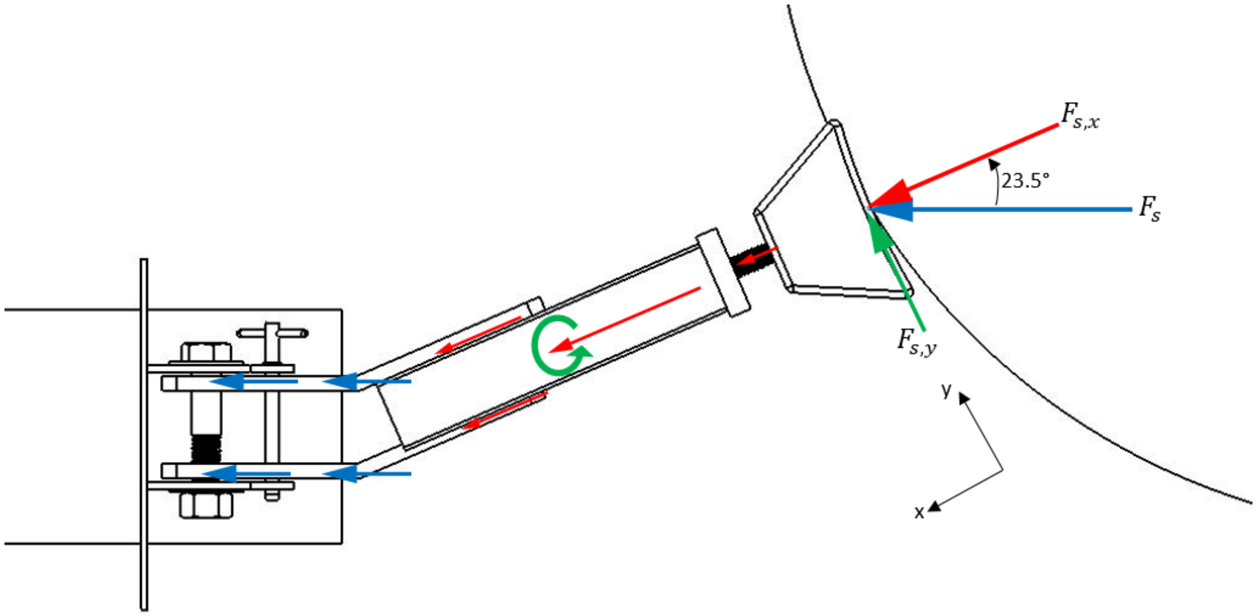


Figure 22. Load paths of forces exerted on support system.

shown in red arrows in the figure. This force is transferred longitudinally along the threaded rod. The tapped plate transfers this force to the rectangular tube, where it acts longitudinally along it. This force is distributed along the four welds connecting the brackets to the rectangular tube. The next force considered was the y-direction ($F_{s,y}$) acting on the contact point, shown as a green arrow. The magnitude of $F_{s,y}$ is also found using trigonometry,

$$F_{s,y} = F_s sin(\theta) \qquad \text{(Equation 6)}$$

Substituting values for F_s and θ gave

$$F_{s,y} = (950\ lbf)sin(23.5°) = 378\ lbf$$

This force creates a moment on the component, shown as a green moment arrow. The last transfer of force to consider is the force along the brackets and to the hinge in the horizontal direction. This was the entire support force (F_s) divided by two due to having two brackets.

This force and its transmission is shown in blue. The divided compression force is then transferred to the pin as a shear force at both points of contact with the hinge.

Calculation of Cross-Sectional Dimensions

Adjustable Bolt Compression

The first component considered was the adjustable bolt. A threaded rod's tensile strength can be found by comparing it to an unthreaded rod with area called tensile-area (A_t). The suitable tensile area for the designed component was found using

$$A_t = \frac{F}{\sigma} \qquad \text{(Equation 7)}$$

where F is the compressive force and σ is the allowable stress of the material. For this component, the material selected was stainless steel, to prevent rusting due to frequent water contact. The allowable stress of stainless steel is 70 kpsi (McMaster-Carr) and compressive force is $F_{s,x} = 871\ lbf$. Substituting these values into equation 7 gave

$$A_t = \frac{871\ lbf}{70000\ psi} = 0.0124\ in^2$$

The diameter (d) of the bolt required can be found using

$$d = \sqrt{\frac{4A_t}{\pi}} \qquad \text{(Equation 8)}$$

Substituting the value found for A_t into equation 8 gave

$$d = \sqrt{\frac{4(0.0124\ in^2)}{\pi}} = 0.126\ in$$

Therefore, to withstand the compressive force, $F_{s,x}$, the bolt must have a diameter of at least 0.126 in.

Square Tube Compression

The next component considered was the square tube. The equation for stress in a hollow rectangular tube is

$$\sigma = \frac{F}{A} + \frac{Mc}{I} \qquad \text{(Equation 9)}$$

where F is the compressive force, A is the cross sectional area, M is the moment acting on the tube, c is the distance to the centroid, and I is the moment of inertia of the tube. The compressive force is known, $F_{s,x}$ = 871 lbf. The moment of inertia of a hollow rectangular tube is found with

$$I = \frac{b_o h_o^3}{12} - \frac{b_i h_i^3}{12} \qquad \text{(Equation 10)}$$

where b_o is the outer base dimension, h_o is the outer height dimension, b_i is the inner base dimension, and h_i is the inner height dimension. In this case, because the tube was square, $b_o = h_o$ and $b_i = h_i$ and equation 10 can be written as

$$I = \frac{x_o^4}{12} - \frac{x_i^4}{12} \qquad \text{(Equation 11)}$$

The moment acting on the square tube is found with

$$M = F_m d \qquad \text{(Equation 12)}$$

where F_m is the force causing the moment and d is the moment arm length. The force causing the moment is $F_{s,y} = 378$ lbf and the moment arm length is 5.89 in. Substituting these values into equation 12 gives

$$M = (378\ lbf)(5.89\ in) = 2,226\ \ lbf \bullet in$$

Because the tube is square, the distance to the centroid is

$$c = \frac{x_o}{2} \qquad \text{(Equation 13)}$$

Substituting the value for compressive force, equation 11, the value for moment, and equation 13 into equation 9 gave

$$\sigma = \frac{871\ lbf}{A} + \frac{(1{,}113\ lbf \bullet in)x_o}{\frac{x_o^4}{12} - \frac{x_i^4}{12}} \qquad \text{(Equation 14)}$$

The materials selection section of this report explores the options for square tubes using equation 14.

Hinge Bracket

The bracket connecting the square tube to the pin was addressed next. The area of the bracket where there is the hole for the pin would have the least strength, so it was considered for calculations. A plate with a hole in it has max stress of

$$\sigma_o = \sigma_n k_t \qquad \text{(Equation 15)}$$

where σ_o is the max stress, σ_n is the nominal stress (without the hole), and k_t is the stress concentration factor. This factor is found using figure 23, which shows the chart of theoretical stress concentration factors.

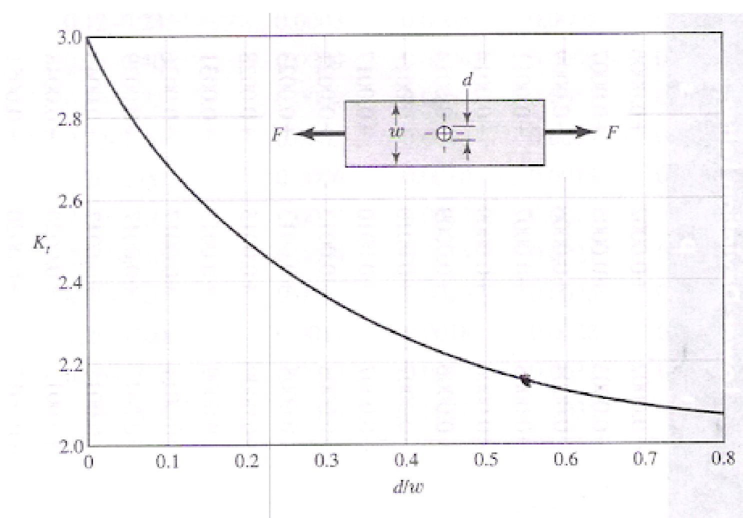


Figure 23. Bar in tension or simple compression with a transverse hole.

The width of the plate (w) and the diameter of the hole (d) are determined once the square tube dimensions and the hinge pin dimensions are finalized in the materials selection section of this report. The nominal stress was found using

$$\sigma_n = \frac{F}{A} \qquad \text{(Equation 16)}$$

Where F is the compressive force and A is the cross sectional area at the location of the hole. The compressive force is half of the support force, F_s =950 lbf because there are two hinge brackets. Therefore, the compressive force is $F=475$ lbf. The cross sectional area was found using

$$A = (w - d)t \qquad \text{(Equation 18)}$$

where t is the thickness of the plate. Substituting equation 18 into equation 16 gave

$$\sigma_n = \frac{F}{(w-d)t} \qquad \text{(Equation 19)}$$

Substituting equation 19 and the value for the compressive force into equation 15 gave

$$\sigma_o = \frac{F k_t}{(w-d)t} = \frac{(475\ lbf)k_t}{(w-d)t} \qquad \text{(Equation 20)}$$

Solving for t in equation 20 and substituting in the strength of steel from McMaster-Carr gave

$$t = \frac{(475\ lbf)k_t}{(36{,}000\ psi)(w-d)} \qquad \text{(Equation 21)}$$

Once w, d, and k_t are determined in the material selection section, the thickness required for the bracket can be found using equation 21.

Hinge Pin Shear

The next component considered was the pin for the hinge. This pin is in single shear at both contact points with the brackets. Because the pin is in single shear, ultimate shear stress is needed which is found with

$$\tau_u = 0.82(\sigma_u) \qquad \text{(Equation 22)}$$

where σ_u is the ultimate strength. The equation for shear stress for a bolt in single shear is,

$$\tau_u = \frac{F}{A} \qquad \text{(Equation 23)}$$

Where F is the force applied and A is the cross sectional area of the bolt. Substituting equation 22 into equation 23 and solving for A gave

$$A = \frac{F}{0.82(\sigma_u)} \qquad \text{(Equation 24)}$$

The force applied was 950 lbf because it is parallel with the direction of the support force. A stainless steel pin was considered to prevent rusting due to frequent exposure to water. The ultimate strength of stainless steel is 70 kpsi (McMaster-Carr). Substituting these values into equation 24 gave

$$A = \frac{950\ lbf}{0.82(70000\ psi)} = 0.01655\ in^2$$

Equation 8 was used to find the diameter required

$$d = \sqrt{\frac{4A}{\pi}} = \sqrt{\frac{4(0.01655)}{\pi}} = 0.145\ in$$

<u>Materials Selection</u>

The materials selected for this project were found on the McMaster-Carr website. Decisions were made based on a factor of safety of 2 or above using the equations and numbers found in the cross section calculation section of this report.

Adjustable Bolt

The adjustable bolt was chosen to be stainless steel to prevent rusting due to frequent water exposure and outdoor use. It was important that it did not rust because it was designed to be easily turned frequently for length adjustments. From equation 8, this bolt required a minimum diameter of 0.126 in. With a factor of safety of 2, this required diameter was 0.252 in. A ⅜"-16 x 3" stainless steel bolt was chosen from McMaster-Carr to satisfy this requirement.

Square Tube

The square tube was determined to be carbon steel for ease of welding. Using equation 14 to find the stress in the selected tube left three unknowns: the cross sectional area (*A*), the inner width (x_i), and the outer width (x_o). Because there are three unknowns, an iterative process was used to select the best option for carbon steel tubing from McMaster-Carr. A square tube with side width of 1 ¼" and wall thickness 1/16" was selected for the last iteration. Using equation 14, the stress in the selected tube was

$$\sigma = \frac{950\ lbf}{0.285\ in^2} + \frac{(1{,}092.5\ lbf\text{•}in)(1.25\ in)}{\frac{(1.25\ in)^4}{12} - \frac{(1.125\ in)^4}{12}} = 13{,}437.44\ psi$$

The steel in these tubes had a max allowable stress of 32 kpsi (McMaster-Carr). The selected tube gives a factor of safety of 2.38, which is above the desired factor of safety of 2. The next thing to consider for this design is buckling of a thin-walled tube. The critical stress where this tube will buckle was found using

$$\sigma_{cr} = S_y - \left(\frac{S_y l}{2\pi k}\right)^2 \frac{1}{CE} \quad \text{(Equation 25)}$$

where S_y is the yield strength of the material, *l* is the length of the tube, *k* is the radius of gyration, *C* is the end-condition constant, and *E* is the modulus of elasticity. The yield strength is given as 32 kpsi,the modulus of elasticity is given as 29.7 Mpsi (Engineering Toolbox), and the length of the tube is 6 in. The end-condition constant was determined from figure 24, which gives constant values for different end conditions of a column.

Column End Conditions	End-Condition Constant C Theoretical Value	Conservative Value	Recommended Value*
Fixed-free	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$
Rounded-rounded	1	1	1
Fixed-rounded	2	1	1.2
Fixed-fixed	4	1	1.2

*To be used only with liberal factors of safety when the column load is accurately known.

Figure 24. End condition constant, C, for buckling equation.

For this design scenario, the column is fixed at one end and free at the other, therefore the value for the end-condition constant is ¼. The radius of gyration was found using,

$$k = \sqrt{\frac{I}{A}} \qquad \text{(Equation 26)}$$

where *I* is the moment of inertia and *A* is the cross sectional area. *I* can be found using equation 11,

$$I = \frac{(1.25\ in)^4}{12} - \frac{(1.125\ in)^4}{12} = 0.06997\ in^4$$

Substituting this value for moment of inertia and the value for cross sectional area into equation 26 gives

$$k = \sqrt{\frac{I}{A}} = \sqrt{\frac{0.06997\ in^4}{0.285\ in^2}} = 0.4955\ in$$

Substituting all known and calculated values into equation 25 gives,

$$\sigma_{cr} = 32\ kpsi - \left(\frac{(32\ kpsi)(6\ in)}{2\pi(0.4955\ in)}\right)^2 \frac{1}{(0.25)(29.7\ Mpsi)} = 31.5\ kpsi$$

Therefore, buckling would occur at a stress of 31.5 kpsi. This leaves a factor of safety of 2.3. The tube with side width 1 ¼" and wall thickness 1/16" will work for this design.

Hinge Pin

The hinge pin was originally going to be a carbon steel pin inserted and tack welded in place to act as the pivot point for the hinge brackets. It was later determined that a stainless steel bolt was the better choice. This would protect against rust on a moving part and also would provide easier assembly and disassembly. The same equations apply for the bolt in shear as did for the pin in shear. From equation 8, the required diameter of the bolt was 0.145”. A ½”-13 x 2 ¾” stainless steel bolt was selected for a factor of safety of 3.44.

Hinge Bracket

Once the square tube and the hinge pin were selected, the hinge bracket could be selected. The rectangular tube is 1.25” wide. To leave space for welding on each side of the hinge bracket to the rectangular tube, the hinge bracket selected was to be 1” wide. Also, a ½” bolt was selected for the hinge pin, so a 0.55” diameter hole was determined to be drilled in the hinge bracket. With values for *w* and *d* determined, k_t could be determined using figure 23. The $\frac{d}{w}$ ratio is 0.55, giving a k_t value of approximately 2.15. Now the thickness needed for the hinge bracket can be found using equation 21,

$$t = \frac{(475\ lbf)k_t}{(36{,}000\ psi)(w-d)} = \frac{(475\ lbf)(2.15)}{(36{,}000\ psi)(1\ in - .55\ in)} = 0.063\ in$$

The thinnest option from McMaster-Carr that still offers a factor of safety of at least 2 is a plate with 3/16” thickness. This is a factor of safety of 2.97. The hinges can be cut to size from this plate.

Other Selections

Other selections to be made include the materials to be used for the mounting plate, the contact point, and the end plate for the rectangular tube. The mounting plate will be constructed out of the same plate as the hinges. Because these plates have a much bigger cross sectional area than the brackets, it is known they will have sufficient strength to support the same loads. The bolts for the mounting plate were determined to be ⅜” with length enough to span the cross beam of the trailer. The only load these bolts bear is the weight of the designed component, making ⅜” a sufficient size bolt. The contact point was determined to be made of nylon to avoid any scratching to the toons from a harder material. Nylon has a compressive strength of 12,500 psi (Laird Plastics), making it apt to support the compressive forces of this design. The end cap of the rectangular tube will be cut from a ⅜” thick steel plate to ensure enough threads are engaged with the adjustable bolt.

<u>Manufacturing Processes</u>

The processes to manufacture the stabilizer include machining and welding. A drill press, bandsaw, and bend brake are needed to manufacture many of the parts. The majority of parts all come from the same sheet. Therefore, the first operation is to cut away any parts from the sheet to be able to complete other processes. Welding is needed to assemble the main components of the support system. The final process that all of the plain carbon steel go through is the application of a rust inhibiting coat. With frequent exposure to water, the low carbon steel will rust very quickly, but with the application of a rust inhibiting coat, the steel parts will last longer and not rust. Many of the parts require little processing, resulting in fast manufacturing.

Contact Point

The contact point between the designed support system and the toon is connected to the rectangular tube by a bolt. This bolt needs to not contact the toon in order to prevent damaging the toon. This is done by counterboring the contact point. The bolt head sits in the counterbore to not contact the toon and still clamps the contact point to the support system with a nut and lock washer to hold in position.

Tube End Plate

This feature involves two manufacturing processes. First, the individual part is to be cut from the sheet. Next the plate is to be drilled and tapped for the adjustable bolt. For the final assembly, it is spot welded on the tube. The spot weld was determined to be 1/16" leg size. It is limited by the thickness of the square tube. The spot weld will be on all four sides of the end plate.

Square Tube

The square tube only needs to be sawed from the long length of tube ordered. Two other components need to be welded to the tube, but no other operation is needed on the tube.

Bracket

The bracket is to be cut from the sheet. After it is cut out, it is to be bent to the specified angle. Next, holes are drilled. The final operation is the bracket being welded to the square tube. The leg size is limited to 1/16" because the thickness of the square tube. Going beyond that thickness is a waste of material.

It was then checked that this size weld could withstand the forces being applied to the designed support system. The longitudinal force, $F_{s,x}$ =871 lbf is directed straight through the tube. This force was divided by four because it will be welded on the top and bottom of the tube on either side of the bracket. The resulting force per weld was 218 lbf. The allowable shear stress on the fillet weld is found from Shigley's Mechanical Engineering Design, table 9-6, shown in figure 25.

Schedule A: Allowable Load for Various Sizes of Fillet Welds

	Strength Level of Weld Metal (EXX)						
	60*	70*	80	90*	100	110*	120
	Allowable shear stress on throat, MPa of fillet weld or partial penetration groove weld						
$\tau =$	124	145	165	186	207	228	248
	Allowable Unit Force on Fillet Weld, N/mm						
$^{\dagger}f =$	$87.67h$	$102.52h$	$116.66h$	$131.5h$	$146.35h$	$161.2h$	$175.34h$
Leg Size h, mm	Allowable Unit Force for Various Sizes of Fillet Welds N/mm						
25	2192	2563	2916	3288	3659	4030	4383
22	1929	2255	2566	2893	3220	3546	3857
20	1753	2050	2333	2630	2927	3224	3506
16	1403	1640	1866	2104	2342	2579	2805
12	1052	1230	1400	1578	1756	1934	2104
11	964	1127	1283	1447	1610	1773	1927
10	877	1025	1167	1315	1463	1612	1753
8	701	820	933	1052	1171	1290	1403
6	526	615	700	789	878	967	1052
5	438	513	583	658	732	806	877
3	263	308	350	395	439	484	526
2	175	205	233	263	293	322	351

*Fillet welds actually tested by the joint AISC-AWS Task Committee.
$^{\dagger}f = 0.707h\,\tau_{all}$.

Figure 25. Allowable load for various sizes of fillet welds.

With a strength level of 60, the allowable shear stress is given to be 17,840 psi. The length of the weld required is calculated with

$$l = \frac{1.414F}{h\tau} \qquad \text{(Equation 27)}$$

where F is the force applied, h is the leg size, and τ is the allowable shear stress. Substituting values in for these variables gives

$$l = \frac{1.414F}{h\tau} = \frac{1.414(218\ lbf)}{(1/16\ in)(17840\ psi)} = 0.276\ in$$

This minimum length is to be welded on both sides of the top and bottom bracket. This component requires every manufacturing process making it the most time-consuming component to manufacture.

Stop

This part only needs to be cut from the sheet and welded to Mounting Plate 1. It is a safety measure to help the consumer not overextend the support system when placing it in position.

Mounting Plate 1

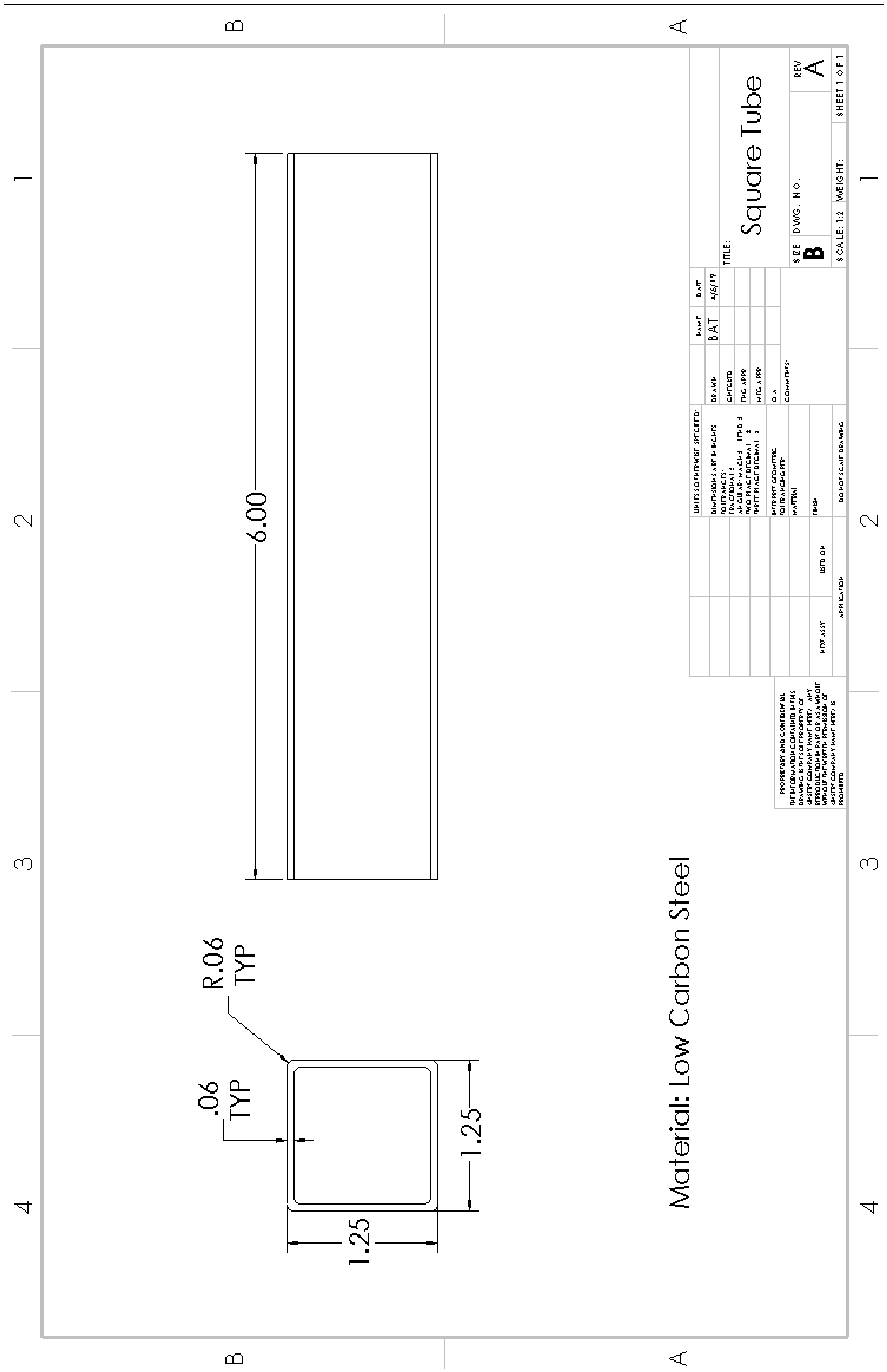
These two mounting plates need to be cut from the sheet. Once that is complete, then the holes may be drilled. This mounting plate holds the hinge bolt and locking pin. It is then welded to one plate from the Mounting Plate 2 pair, which contacts the trailer. The weld leg size is ⅛”. This ensures safety for the mounting to never have a failure.

Mounting Plate 2

These two mounting plates are the contact between the trailer and the designed support system. They are cut from the sheet, then drilled. Then one of the plates is welded to the Mounting Plate 1 pair. The two plates are connected to the trailer by compressing them with bolts.

<u>Part and Assembly Drawings</u>

Below are the assembly and part drawings of the designed support system.The part drawings will be shown first. Then the the assembly drawing and exploded view.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4/6/19
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REV
A
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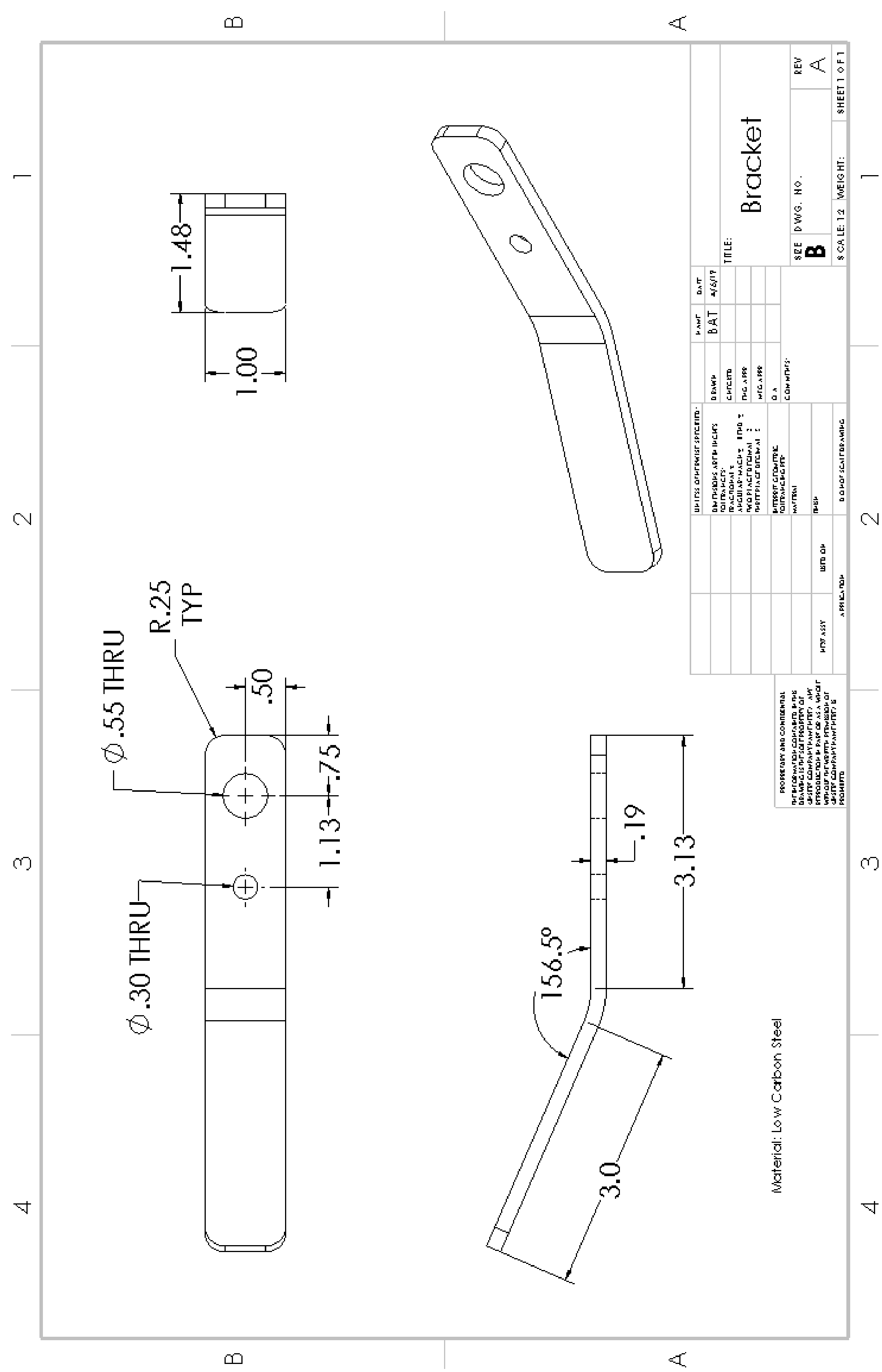
WEIGHT:
SHEET 1 OF 1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TITLE:
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REV
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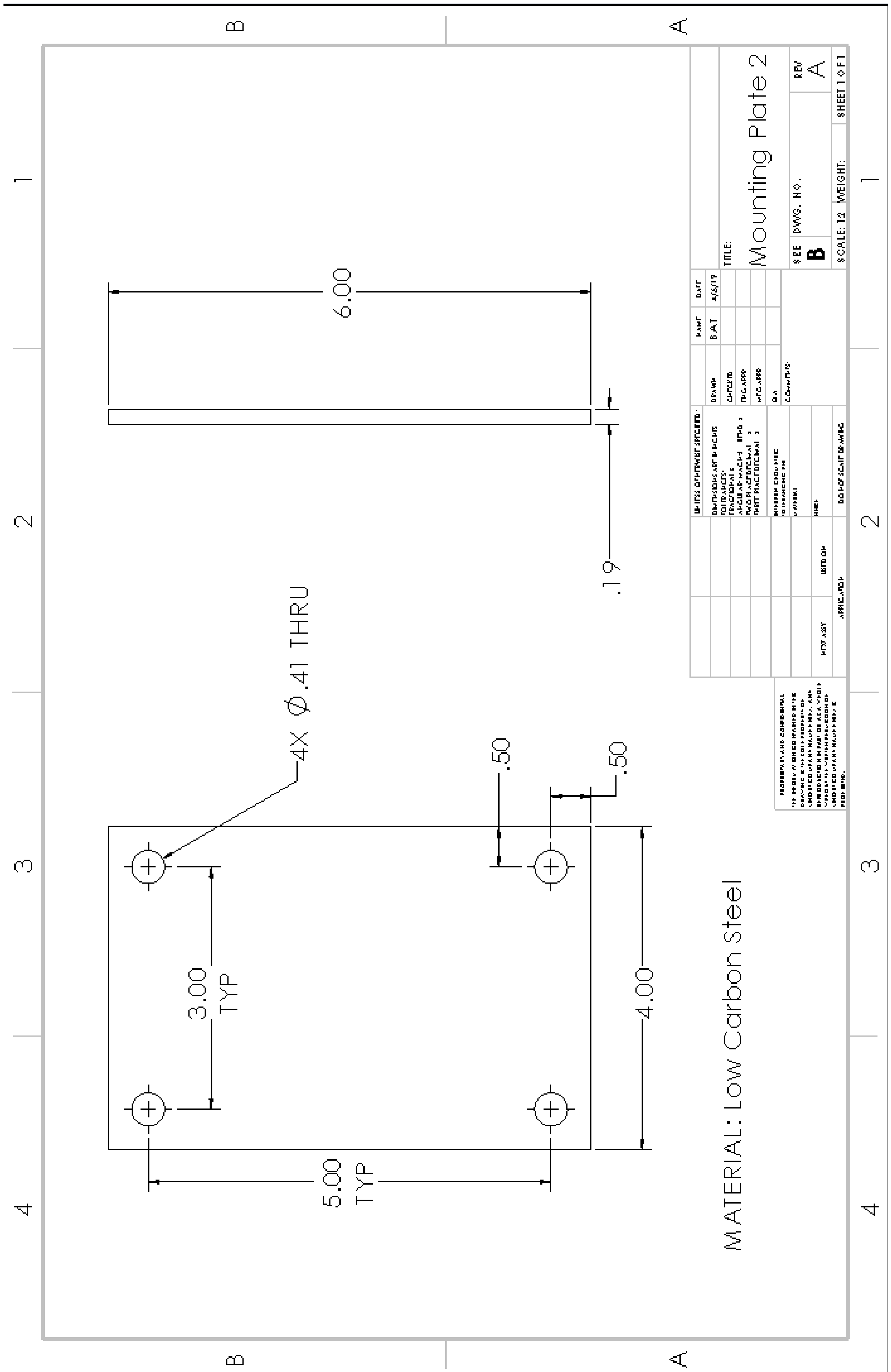
SHEET 1 OF 1
DRAWN
BAT
4/6/17



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.50
3.00 TYP
5.00 TYP
4.00
6.00
.19
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REV
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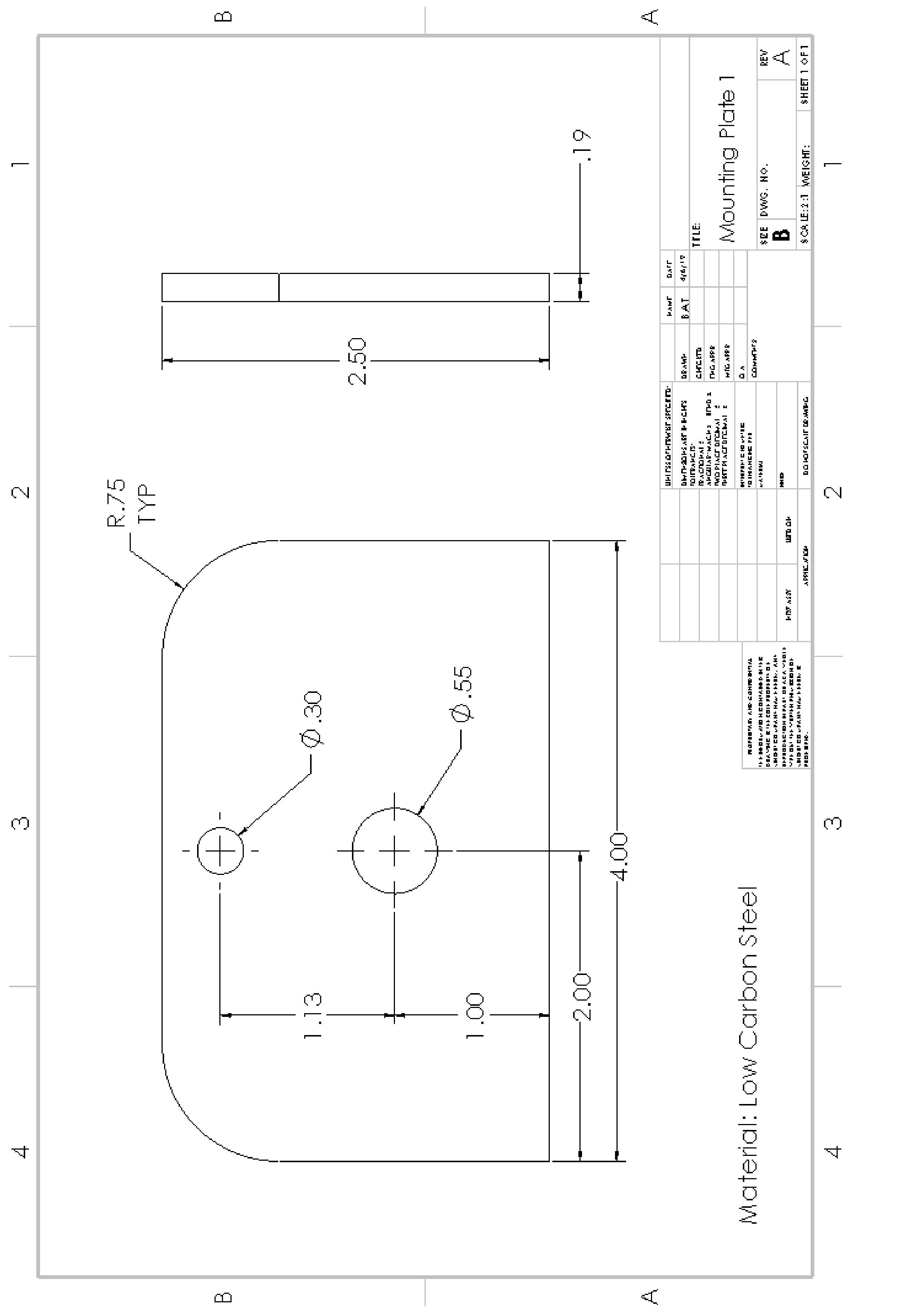
SHEET 1 OF 1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REV
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WEIGHT:
SHEET 1 OF 1
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BAT
CHECKED
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MFG APPR.
Q.A.
COMMENTS:
NAME
DATE
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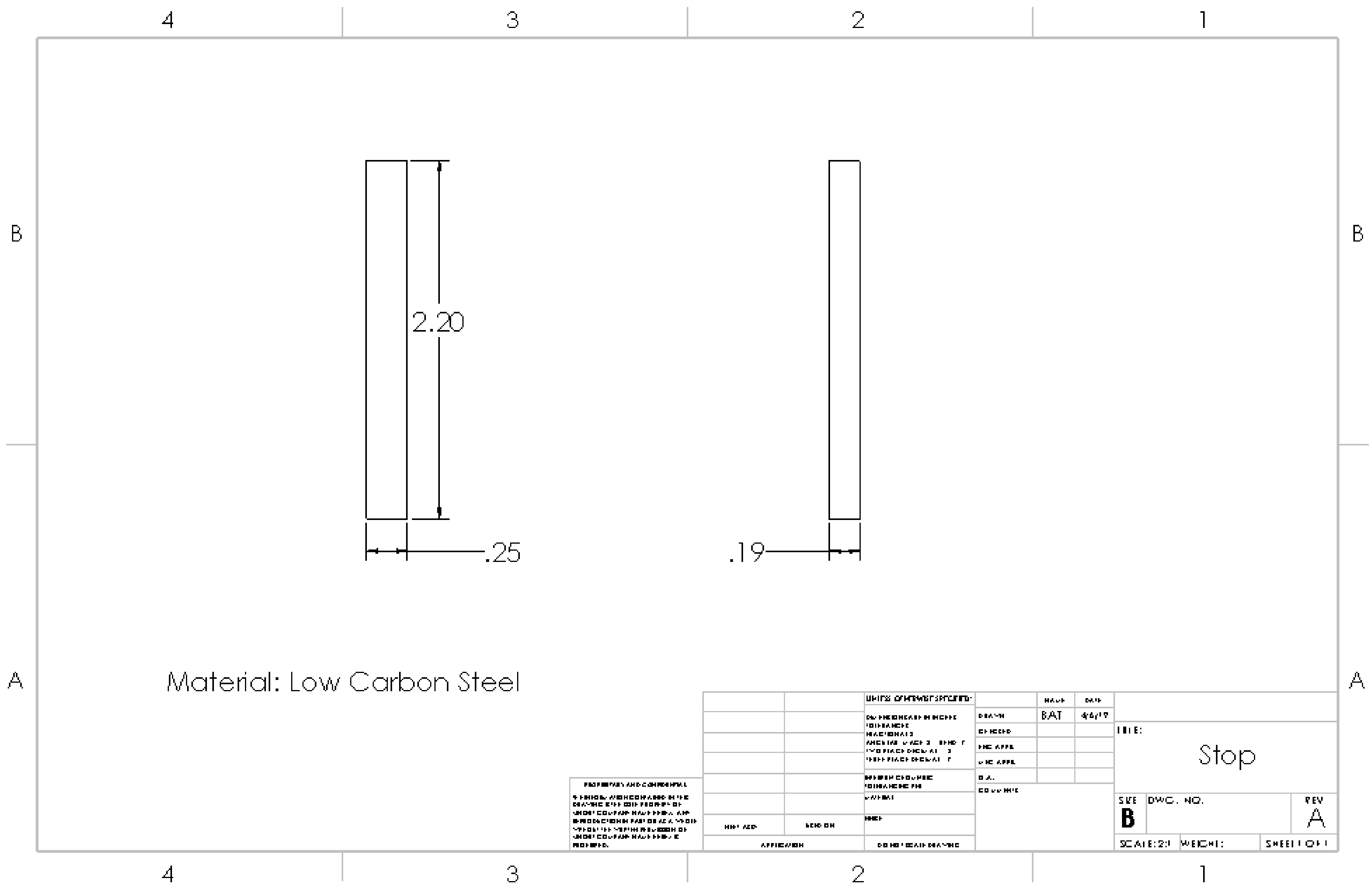
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REV A
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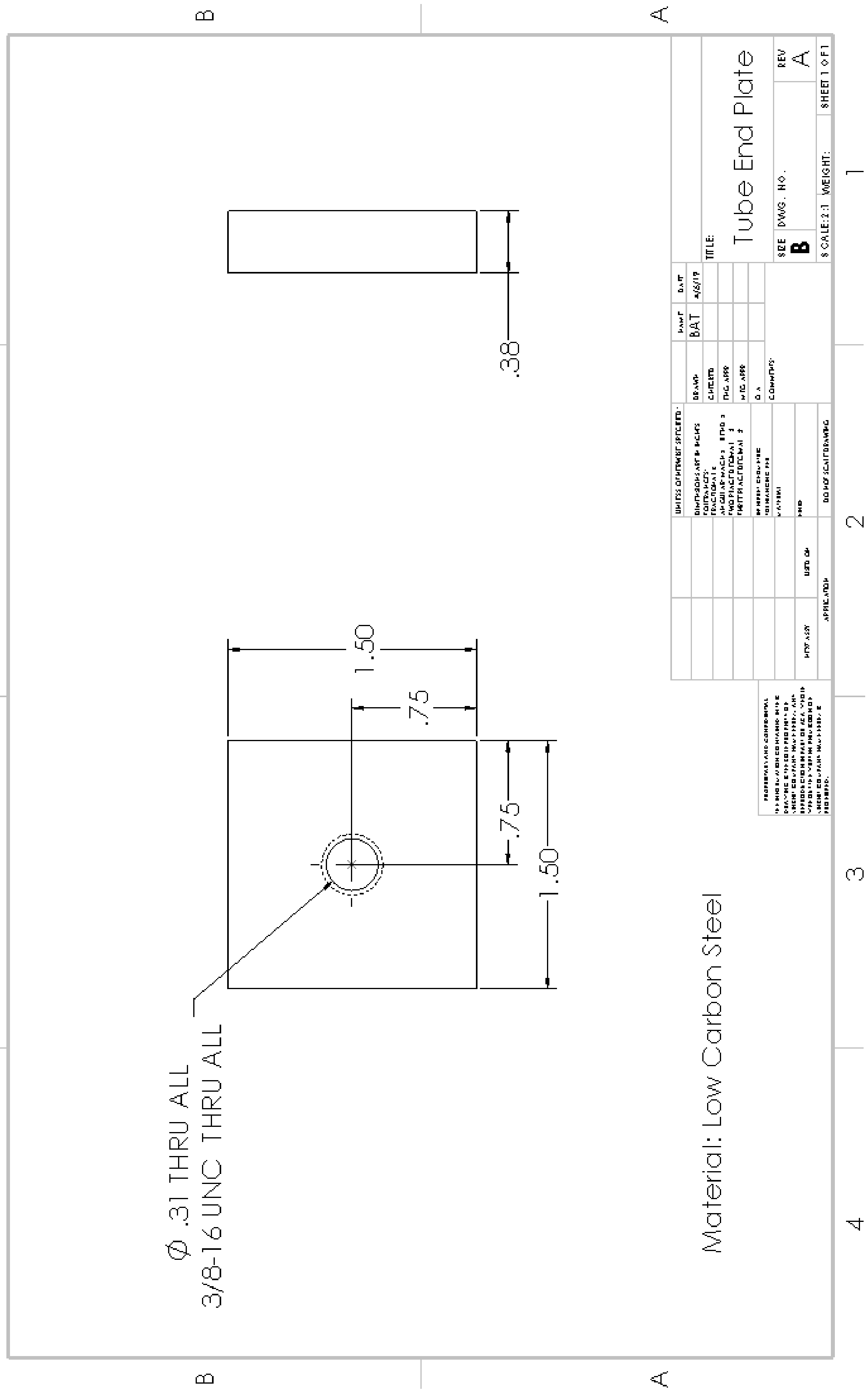
SHEET 1 OF 1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1.50
.38
Tube End Plate
TITLE:
DWG. NO.
SIZE
B
REV
A
SCALE:2:1
WEIGHT:
SHEET 1 OF 1
DRAWN
BAT
CHECKED
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COMMENTS:
UNLESS OTHERWISE SPECIFIED:
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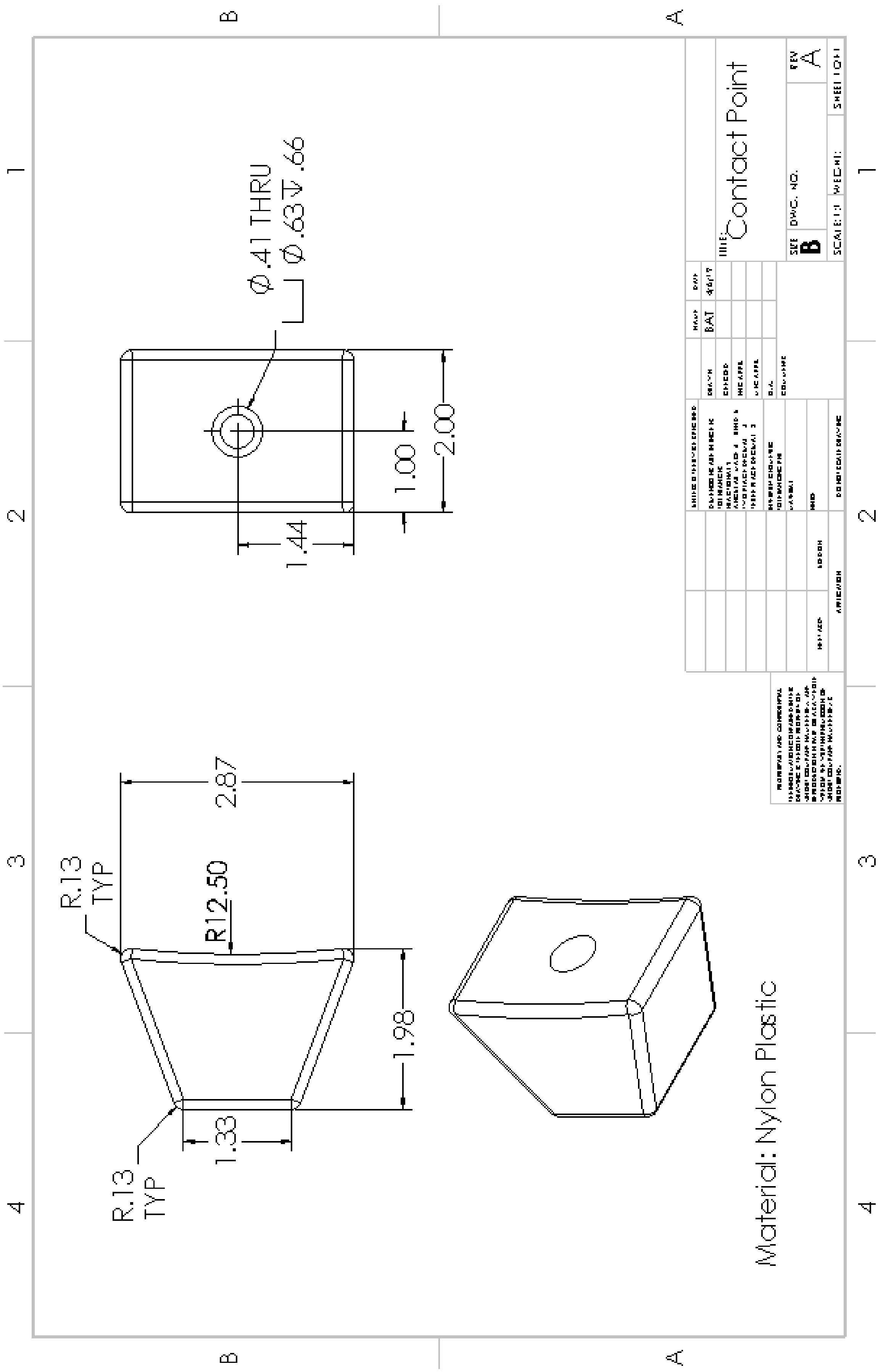
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R.13
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REV
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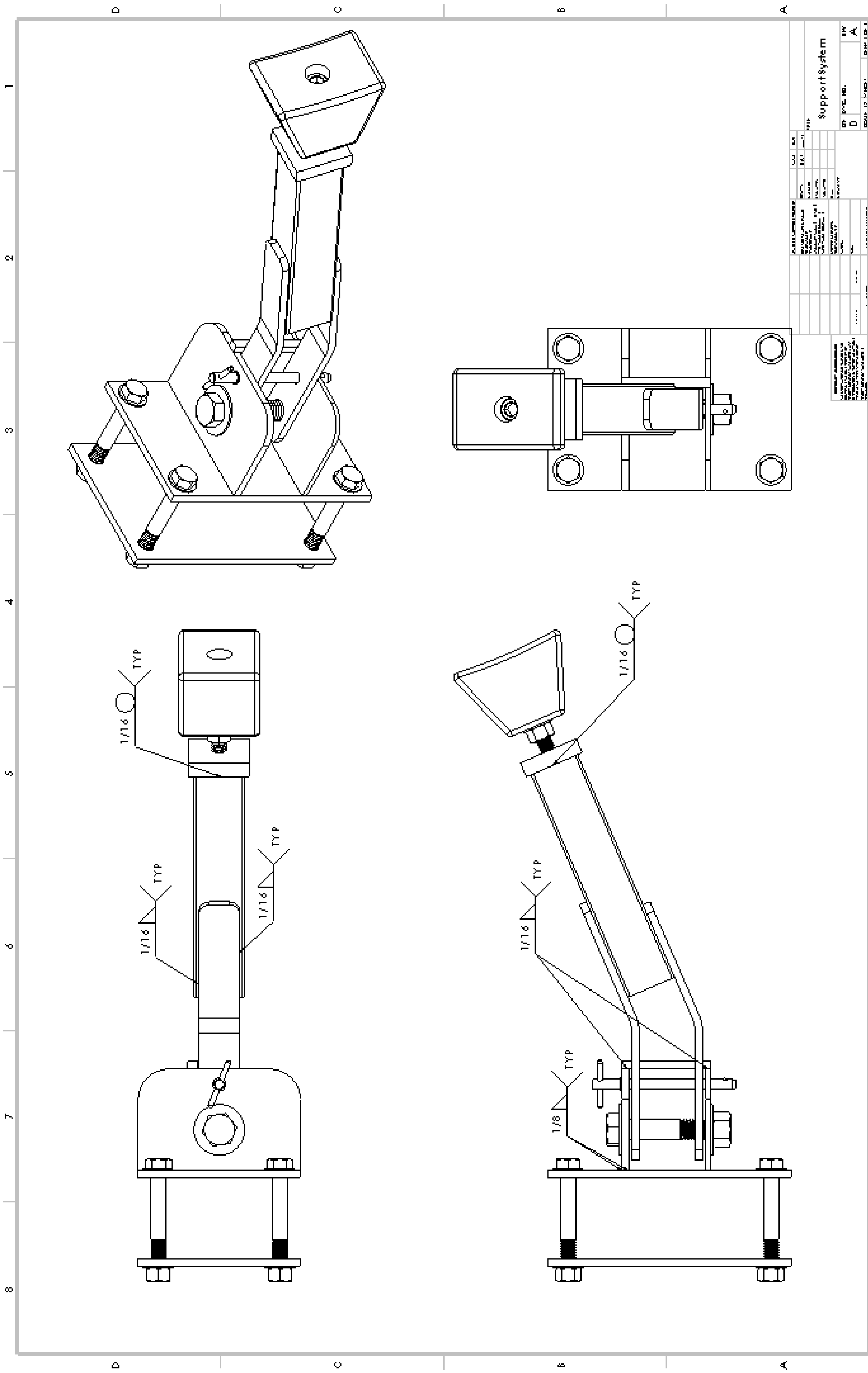
SCALE: 1:1
WEIGHT:
SHEET 1 OF 1
DRAWN
BAT



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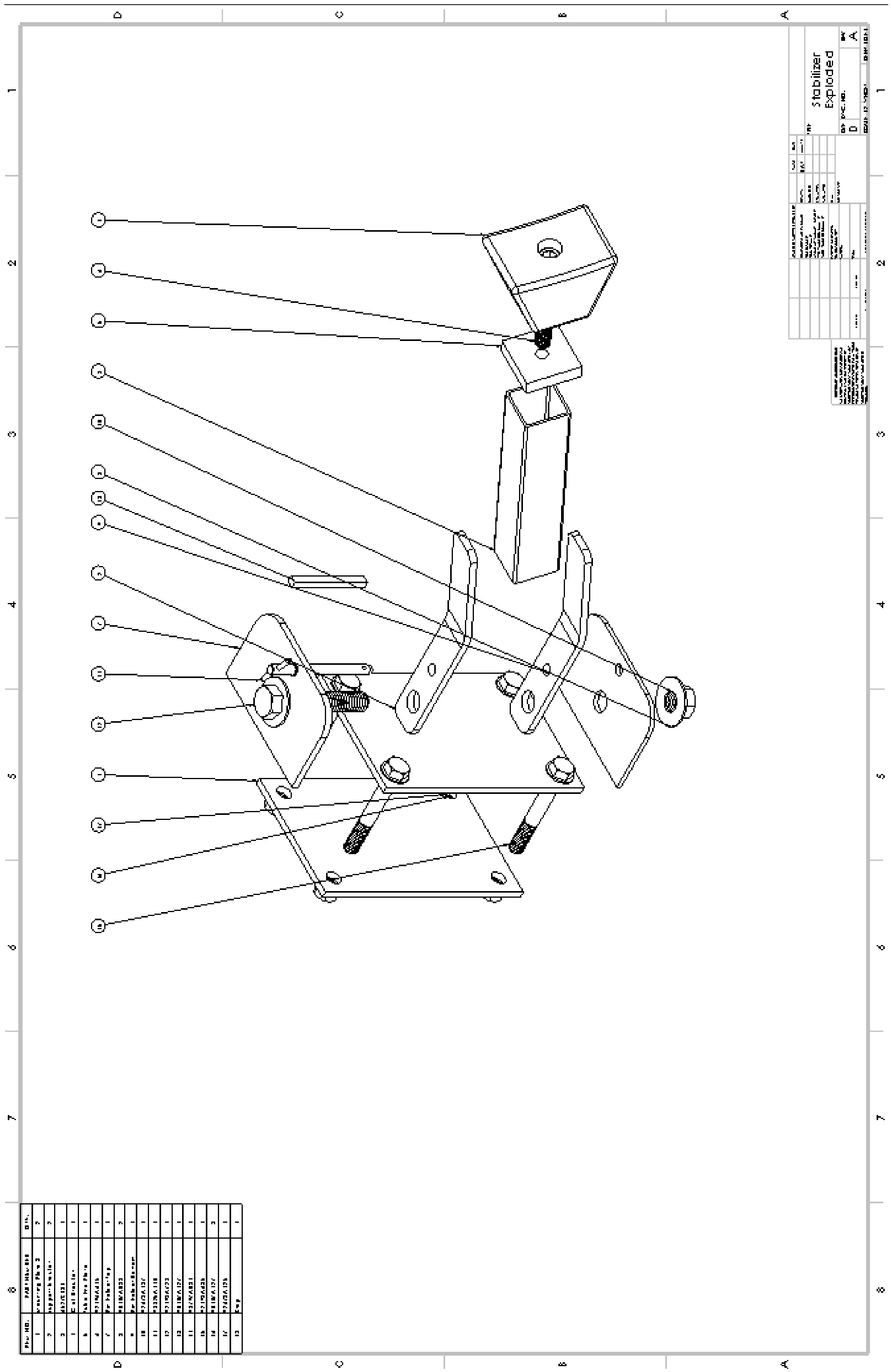

Stabilizer
Exploded

Bill of Materials and Estimated Cost of Materials

The required parts were outlined in the sections above. All parts and materials were found through mcmaster.com. The part number in the table below displays the part number from McMaster-Carr. The mounting plates and brackets were ordered from the same low carbon steel sheet to optimize the cost of the sheet. The sheet size of 18”x18” provides 324 in^2 of material. The designed support system requires approximately 300 in^2 of material, which allows for errors. The nylon plastic contact point was done in the same manner of having the sections cut from the same sheet. Optimizing the hardware was not as feasible. For many parts, only a few components were needed, but the amount per package far exceeded the necessary required amount. This will further lower cost in the long run, if more support systems are produced. The bill of materials shown in table 3 includes all of the parts and materials to produce four support systems which is the requirement for one trailer.

Table 3

Bill of Materials for designed support system with estimated cost, details in Appendix

Item No.	Part Number	Qty.	Total Qty	Qty per package	Packages Needed	Price per package	Total Price
1	Mounting Plate 1 (1388K755)	2	8	1	1	$ -	$ -
2	Stop (1388K755)	1	4	1	1	$ -	$ -
3	Mounting Plate 2 (1388K755)	2	8	1	1	$ -	$ -
4	6527K181	1	4	2	2	$ 7.99	$ 15.98
5	C-shape (8540K328)	1	4	4	1	$ 50.33	$ 50.33
6	Tube End Plate (6544K28)	1	4	4	1	$ 35.37	$ 35.37
7	Bracket (1388K755)	2	8	1	1	$ 142.49	$ 142.49
8	92196A615	1	4	5	1	$ 9.56	$ 9.56
9	90107A033	2	8	25	1	$ 9.48	$ 9.48
10	92673A137	1	4	10	1	$ 2.89	$ 2.89
11	98325A145	1	4	1	4	$ 2.06	$ 8.24
12	92198A723	1	4	5	1	$ 5.71	$ 5.71
13	92147A031	1	4	50	1	$ 6.37	$ 6.37
14	98797A031	1	4	5	1	$ 5.66	$ 5.66
15	92198A635	4	16	10	2	$ 5.76	$ 11.52
16	90107A127	8	32	25	2	$ 8.80	$ 17.60
17	92673A125	4	16	25	2	$ 2.35	$ 4.70
18	1370K34	1	1	1	1	$ 21.91	$ 21.91
						Total	**$ 347.81**

The raw material cost is a feasible option to include on a trailer for added safety. A manufacturing cost and profit margin will need to be added to the number above if it is sold to the public. The manufacturing cost should be low because of the small amount of machining and welding that is needed for the product. The large number of materials for this product is primarily the different sizes of nuts, bolts, and washers. All of the other items are grouped together to be able to be made from the same sheet. This optimization of the items lowered the cost of the product to financially be a viable option for a center lift trailer owner.

Discussion

Working on this project showed us what it would be like to be a real engineer designing a full product for field use. We had a chance to go through all the critical steps in the process of designing a system. Many lessons were learned along the way and improvements were made. Many of the classes we took throughout our college career were useful in making decisions during this project.

Observing the process of design learned in “Concepts of Design” class really helped us organize and constantly improve our design. Initially it was thought that a spring/damper mechanism would be best for this design. After working through the conceptual design phase, it was realized that this made things more expensive and brought unnecessary complications. We learned that sometimes the simpler solution is more efficient and safer. As engineers, we had to choose the best option, and in this case, that was the simplest option.

As we moved on to later phases of design, we realized that our “simple” design concept was anything but. As we analyzed every part individually, new problems arose and new opportunities for improvement presented themselves. An example of this was with the hinge bolt. Throughout the design and calculations it was considered to be a pin. It was later realized a bolt was the better choice and the proper adjustments were made. Part of being an engineer is seeing improvements that need made and pursuing them to complete a job correctly.

Throughout the calculations of this design, we had an opportunity to use many of the skills learned throughout our classes. We used concepts such as ultimate stress, shear stress, welding processes, free body diagram development, material properties, and much more. The classes these concepts were learned in include “Statics”, “Dynamics”, “Mechanical Metallurgy”, “Design of Mechanical Components”, and many more.

Another skill we had the opportunity to practice was our customer relations skills. Interacting with Terry Hayes on a regular basis over the phone and through field visits gave us a chance to understand what he really wanted and bring him the design that suited his needs. Through our co-ops, we learned that communication is a strong tool that all engineers use every day and this project was no exception. The economics of designing new components was also learned through interactions with the customer on this project. Mr. Hayes was not ready to finance this project without further analysis of our suggested design and techniques of manufacturing. Therefore, a prototype was not constructed simply due to lack of funding.

This project provided us with the opportunity to learn new skills and practice ones we have learned throughout college and co-op. It encompassed many aspects of engineering including the technical sides and the non-technical sides. We were very pleased with how this project process developed and believe we learned a great amount that will be applied to our careers in mechanical engineering.

Conclusion

The support system addition to a center-lift trailer creates a safer option to add to trailers to prevent tipping while cornering. First, the product had to be defined: what the product should be and the performance of it. Next there had to be many different concepts to display the possible designs. Defining what was important in the design through the decision matrix narrowed down the design. Then the different connections were sketched to provide the layout of the design in greater detail. Next by determining the forces that the support system will have to withstand, the exact dimensions of the critical components of the design were found. By applying a factor of safety to ensure absolute safety of the support system, the components were selected. A 3D model was created to ensure proper fits and to be able to make technical drawings. Selecting components with similar thicknesses was done to lower costs and optimize the manufacturing processes. By providing a low cost safety option to the center-lift trailer market, this design is a reasonable option for trailer manufacturers and consumers to utilize.

References

Budynas, R. G., & Nisbett, J. K. (2016). *Shigley's Mechanical Engineering Design* (10th ed., pp.150-1050). Tamil Nadu, India: McGraw Hill.

Engineering ToolBox. (2003). Young's modulus - tensile and yield strength for common material*s*. Retrieved from https://www.engineeringtoolbox.com/young-modulus-d_417.html [12 April 2019]

Laird Plastics. (2019). Nylon 101. Retrieved from http://www.lairdplastics.com/product/materials/nylon/3350-nylon-101

McMaster-Carr. (2019). McMaster-Carr. Retrieved from https://www.mcmaster.com/

Appendix

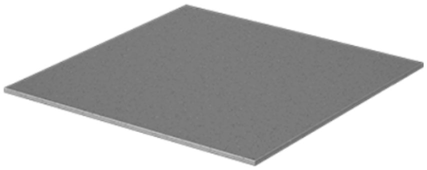
Below are the web pages used for material selection.

McMASTER-CARR

1388k755

Low-Carbon Steel Sheet

with Decarb-Free Surface, 18" x 18" x 3/16"



Each

ADD TO ORDER

In stock
$142.49 Each
1388K755

Material	Low-Carbon Steel
Cross Section Shape	Rectangle
Construction	Solid
Appearance	Plain
Thickness	3/16"
Thickness Tolerance	-0.003" to 0.003"
Tolerance Rating	Standard
Width	18"
Width Tolerance	-1/16" to 1/2"
Length	18"
Length Tolerance	-1/16" to 1/2"
Yield Strength	36,000 psi
Temper Rating	Not Rated
Hardness	Not Rated
Hardness Rating	Not Rated
Heat Treatable	Yes
Maximum Hardness after Heat Treatment	Rockwell C65
Features	Decarb-Free Surface
Mechanical Finish	Ground
Temperature Range	Not Rated

McMASTER-CARR

steel hollow bars

Low-Carbon Steel Square Tube

0.060" Wall Thickness, 1-1/4" x 1-1/4" Outside Size, Long

Length, ft.
✓ 3

Each

ADD TO ORDER

In stock
$14.53 Each
6527K184

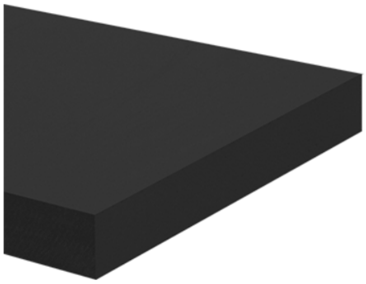
Material	Low-Carbon Steel
Cross Section Shape	Rectangle
Corner Shape	
Outside	Round
Inside	Round
Construction	Hollow
Appearance	Plain
Wall Thickness	0.06"
Wall Thickness Tolerance	-0.006" to 0.006"
Tolerance Rating	Standard
Outside	
Height	1 1/4"
Height Tolerance	-0.03" to 0.03"
Width	1 1/4"
Width Tolerance	-0.03" to 0.03"
Inside	
Height	1.13"
Width	1.13"
Yield Strength	32,000 psi
Fabrication	Hot Rolled
Temper Rating	Not Rated

McMASTER-CARR

8540k328

Black Nylon Sheet

6" x 6" x 2"



Each

ADD TO ORDER

In stock
$50.33 Each
8540K328

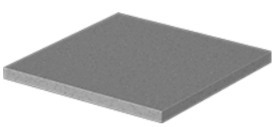
Material	Nylon Plastic
Grade	6/6
Cross Section Shape	Rectangle
Construction	Solid
Texture	Smooth
Thickness	2"
Thickness Tolerance	0" to 0.025"
Tolerance Rating	Oversized
Width	6"
Width Tolerance	-1/8" to 1/8"
Length	6"
Length Tolerance	-1/8" to 1/8"
Backing Type	Plain
Hardness	Rockwell R108-R121
Hardness Rating	Hard
For Use Outdoors	No
Temperature Range	-40° to 185° F
Impact Strength	0.6-1.4 ft.-lbs./in.
Impact Strength Rating	Poor
Tensile Strength	11,200-12,300 psi
Tensile Strength Rating	Good
Color	Black

McMASTER-CARR

6544k28

Low-Carbon Steel Sheet

6" x 6" x 3/8"



Each

ADD TO ORDER

In stock
$35.37 Each
6544K28

Material	Low-Carbon Steel
Cross Section Shape	Rectangle
Construction	Solid
Appearance	Plain
Thickness	3/8"
Thickness Tolerance	-0.013" to 0"
Tolerance Rating	Undersized
Width	6"
Width Tolerance	-1/8" to 1/8"
Length	6"
Length Tolerance	-1/8" to 1/8"
Yield Strength	60,000 psi
Fabrication	Cold Worked
Temper Rating	Hardened
Hardness	Rockwell B80 (Medium)
Maximum Hardness after Heat Treatment	Not Rated
Heat Treatable	Yes
Temperature Range	Not Rated
Flatness Tolerance	Not Rated
Density	0.28 lbs./cu. in.
Surface Resistivity	15 microhm-cm @ 32° F

McMASTER-CARR

92196a615

18-8 Stainless Steel Socket Head Screw

3/8"-16 Thread Size, 3" Long, Fully Threaded

Packs of 5

ADD TO ORDER

In stock
$9.56 per pack of 5
92196A615

Head Type	Socket
Socket Head Profile	Standard
Drive Style	Hex
System of Measurement	Inch
Thread Direction	Right Hand
Thread Size	3/8"-16
Screw Size Decimal Equivalent	0.375"
Thread Type	UNC
Thread Fit	Class 3A
Length	3"
Threading	Fully Threaded
Thread Spacing	Coarse
Head	
Diameter	9/16"
Height	3/8"
Drive Size	5/16"
Tensile Strength	70,000 psi
Hardness	Rockwell B70
Material	18-8 Stainless Steel
RoHS	Compliant

McMASTER-CARR

90107a033

316 Stainless Steel Washer

for 1/2" Screw Size, 0.531" ID, 1.25" OD

Packs of 25

ADD TO ORDER

In stock
$9.48 per pack of 25
90107A033

Material	316 Stainless Steel
For Screw Size	1/2"
ID	0.531"
OD	1.250"
Thickness	0.055"-0.069"
Washer Type	Flat
System of Measurement	Inch
Hardness	Not Rated
RoHS	Compliant

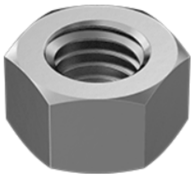
316 stainless steel washers have excellent resistance to chemicals and salt water. They may be mildly magnetic.

McMASTER-CARR

92673a137

18-8 Stainless Steel Hex Nut

1/2"-13 Thread Size, ASTM F594



Packs of 10

ADD TO ORDER

In stock
$2.89 per pack of 10
92673A137

Material	18-8 Stainless Steel
Thread Size	1/2"-13
Thread Type	UNC
Thread Spacing	Coarse
Thread Fit	Class 2B
Thread Direction	Right Hand
Width	3/4"
Height	7/16"
Specifications Met	ASTM F594
Drive Style	External Hex
Nut Type	Hex
Hex Nut Profile	Standard
System of Measurement	Inch
RoHS	Compliant

These nuts have good chemical resistance and may be mildly magnetic.

McMASTER-CARR

98325a145

Zinc-Plated Steel T-Handle Quick-Release Pin

1/4" Diameter, 2-1/2" Usable Length

Each

ADD TO ORDER

In stock
$2.06 Each
98325A145

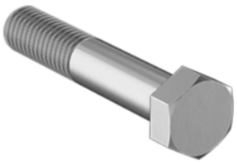
Diameter	1/4"
Usable Length	2 1/2"
Shoulder Length	3/4"
Diameter at Extended Ball Height	0.286"
Diameter Tolerance	-0.003" to 0"
Material	Zinc-Plated Steel
Min. Hardness	Rockwell B96
Breaking Strength	4,300 lbs.
Number of Retaining Balls	1
Ball Material	316 Stainless Steel
Min. Ball Hardness	Rockwell C25
Spring Material	316 Stainless Steel
Handle Material	Steel
End Style	Beveled
Handle Style	T-Grip
Locking Feature	Nonlocking
Pin Type	Quick Release
System of Measurement	Inch
RoHS	Compliant

McMASTER-CARR

92198a723

18-8 Stainless Steel Hex Head Screw

1/2"-13 Thread Size, 2-3/4" Long, Partially Threaded



Packs of 5

ADD TO ORDER

In stock
$5.71 per pack of 5
92198A723

Head Type	Hex
Drive Style	External Hex
System of Measurement	Inch
Thread Direction	Right Hand
Thread Size	1/2"-13
Screw Size Decimal Equivalent	0.5"
Thread Type	UNC
Thread Fit	Class 2A
Length	2 3/4"
Threading	Partially Threaded
Min. Thread Length	1 1/4"
Thread Spacing	Coarse
Head	
Width	3/4"
Height	5/16"
Material	18-8 Stainless Steel
Tensile Strength	70,000 psi
Hardness	Rockwell B70
Specifications Met	ASME B18.2.1
RoHS	Compliant

McMASTER-CARR

92147a031

316 Stainless Steel Split Lock Washer

for 3/8" Screw Size, 0.385" ID, 0.68" OD

Packs of 50

ADD TO ORDER

In stock
$6.37 per pack of 50
92147A031

Material	316 Stainless Steel
For Screw Size	3/8"
ID	0.385"
OD	0.680"
Thickness	0.094"
Washer Type	Split Lock
System of Measurement	Inch
Hardness	Rockwell C35
Specifications Met	ASME B18.21.1 (Dimensions Only)
RoHS	Compliant

As a screw is tightened, these washers flatten to add tension to the joint and prevent loosening from small amounts of vibration.

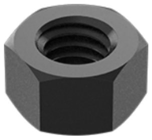
316 stainless steel washers have excellent resistance to chemicals and salt water. They may be mildly magnetic.

McMASTER-CARR

98797a031

Medium-Strength Steel Hex Nuts - Grade 5

Black Ultra-Corrosion-Resistant Coated, 3/8"-16 Thread Size



Packs of 5

ADD TO ORDER

In stock
$5.66 per pack of 5
98797A031

Material	Black Ultra-Corrosion-Resistant Coated Steel
Fastener Strength Grade/Class	Grade 5
Thread Size	3/8"-16
Thread Type	UNC
Thread Spacing	Coarse
Thread Fit	Class 2B
Thread Direction	Right Hand
Width	9/16"
Height	21/64"
Drive Style	External Hex
Nut Type	Hex
Hex Nut Profile	Standard
System of Measurement	Inch
RoHS	Not Compliant

These nuts are suitable for fastening most machinery and equipment.

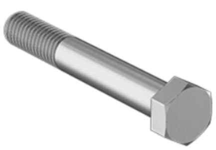
Black ultra-corrosion-resistant coated steel nuts are more corrosion resistant in wet environments than zinc-plated steel nuts. Also known as black luster nuts.

McMASTER-CARR

92198a635

18-8 Stainless Steel Hex Head Screw

3/8"-16 Thread Size, 2-3/4" Long, Partially Threaded



Packs of 10

ADD TO ORDER

In stock
$5.76 per pack of 10
92198A635

Head Type	Hex
Drive Style	External Hex
System of Measurement	Inch
Thread Direction	Right Hand
Thread Size	3/8"-16
Screw Size Decimal Equivalent	0.375"
Thread Type	UNC
Thread Fit	Class 2A
Length	2 3/4"
Threading	Partially Threaded
Min. Thread Length	1"
Thread Spacing	Coarse
Head	
Width	9/16"
Height	15/64"
Material	18-8 Stainless Steel
Tensile Strength	70,000 psi
Hardness	Rockwell B70
Specifications Met	ASME B18.2.1
RoHS	Compliant

McMASTER-CARR

90107a127

316 Stainless Steel Washer

for 3/8" Screw Size, 0.406" ID, 0.75" OD

Packs of 25

ADD TO ORDER

In stock
$8.80 per pack of 25
90107A127

Material	316 Stainless Steel
For Screw Size	3/8"
ID	0.406"
OD	0.750"
Thickness	0.040"-0.060"
Washer Type	Flat
System of Measurement	Inch
Hardness	Not Rated
RoHS	Compliant

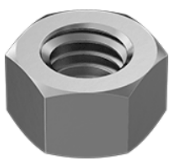
316 stainless steel washers have excellent resistance to chemicals and salt water. They may be mildly magnetic.

McMASTER-CARR

92673a125

18-8 Stainless Steel Hex Nut

3/8"-16 Thread Size, ASTM F594



Packs of 25

ADD TO ORDER

In stock
$2.35 per pack of 25
92673A125

Material	18-8 Stainless Steel
Thread Size	3/8"-16
Thread Type	UNC
Thread Spacing	Coarse
Thread Fit	Class 2B
Thread Direction	Right Hand
Width	9/16"
Height	21/64"
Specifications Met	ASTM F594
Drive Style	External Hex
Nut Type	Hex
Hex Nut Profile	Standard
System of Measurement	Inch
RoHS	Compliant

These nuts have good chemical resistance and may be mildly magnetic.

McMASTER-CARR

1370k34

LPS #3 Clean-Away Coating for Metal

11 oz. Aerosol Can

Each

ADD TO ORDER

In stock
1-11 Each $21.91
12 or more $19.63
1370K34

Type	Protective Coating
Container Size	11 oz.
Container Type	Aerosol Can
Maximum Protection Time	
Indoor	24 mo.
Outdoor	6 mo.
Mixing Required	No
Dry Time	
Touch	3.5 hrs.
Overall	24 hrs.
One-Coat Coverage	45 sq. ft. @ 3.5 mil
For Use On	Steel, Iron
Composition	Solvent Based
Material	Wax
Maximum Temperature	175° F
Manufacturer/Brand	LPS
Manufacturer/Brand Model Number	LPS #3
For Use Outdoors	Yes
Specifications Met	MIL-C-16173D, Grade 2
Color	Transparent Yellow
Additional Specifications	SDS