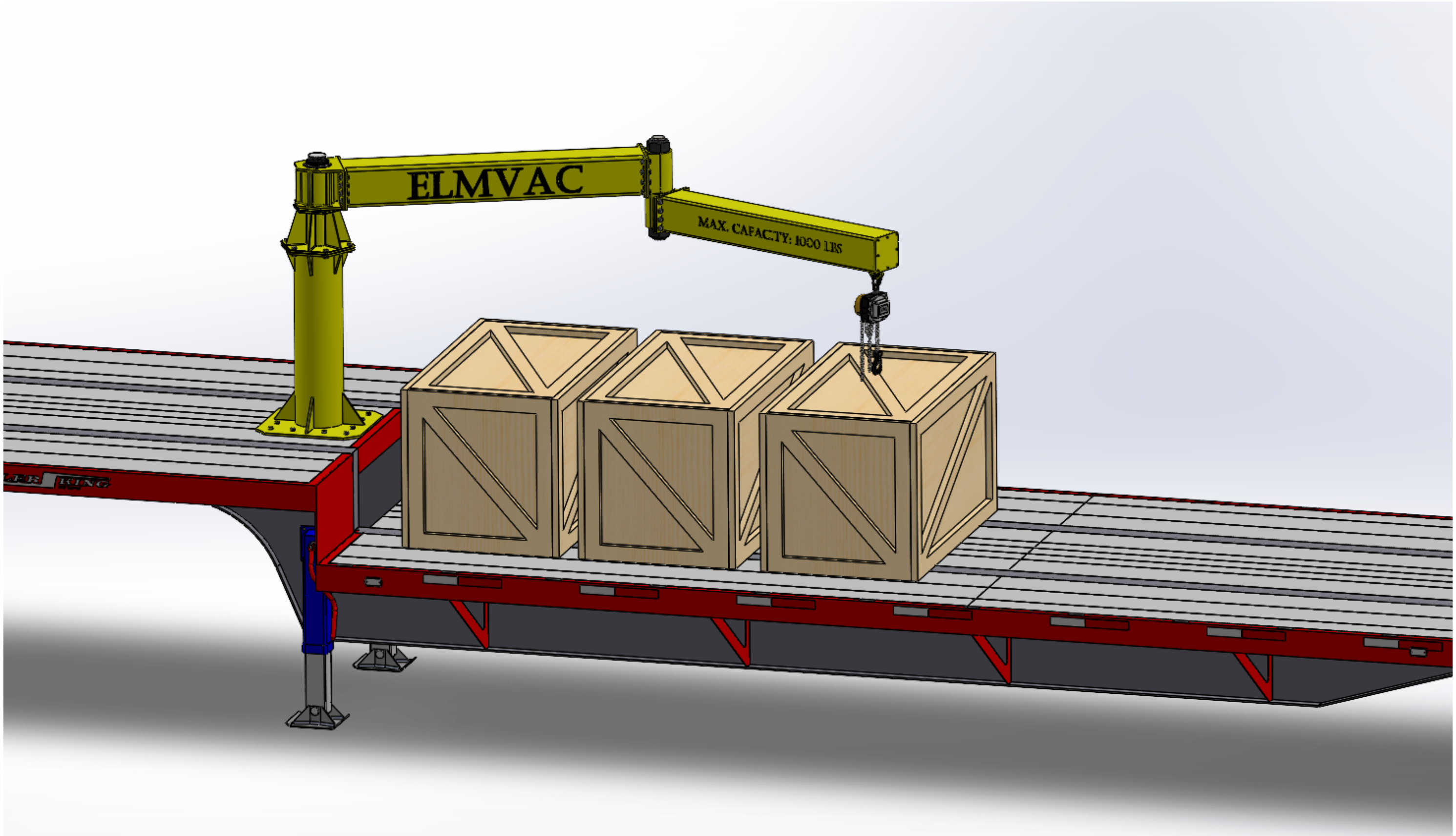

Articulated Jib Crane for Cargo Unloading

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Introduction

Elmvac, a small company that manufactures Ultra High Vacuum chambers (UHVs) needs mechanical device to assist in the unloading of the chambers to customers who do not have the convenience of a fork truck or other equipment. Specific design requirements were as follows:

1. Machine must weigh less than 1500 lbs
2. Machine must be mounted to the upper deck of a standard drop-deck flatbed semi trailer
3. Machine must be capable of lifting a minimum of 500 lbs
4. Machine must be able to deposit the load safely a minimum of 2 ft from either the right, left, or rear of the trailer
5. The machine must be either human powered, or electrically powered with a maximum power draw of 2400 watts (120VAC 20A or 12VDC 200A)

This task was assigned to a group of engineering interns working at the company.

In order to design the device more efficiently, the team first laid out the priorities for device. Practicality, both in terms of versatility, ease of operation and ease of manufacture, was deemed a priority, as well as cost effectiveness. These guiding principles were used to aid in decision-making throughout the project.

Due to the expedited nature of the project, it was decided to design various components of the crane concurrently. This method has the disadvantage of requiring more iterations of each element, as many design parameters must be first chosen by “gut feel” and then revised through analysis. Such an iterative process is in contrast to a sequential process, where all possible variables are defined before moving to another component. Regardless, drawing from prior experience of team members allowed the crane to be efficiently and concurrently designed.

The design chosen was an articulated jib crane, shown in Figure 1. The crane is human powered, as the hoisting of cargo is handled through a manual chain fall and once hoisted the cargo can be freely moved by hand to the desired position.

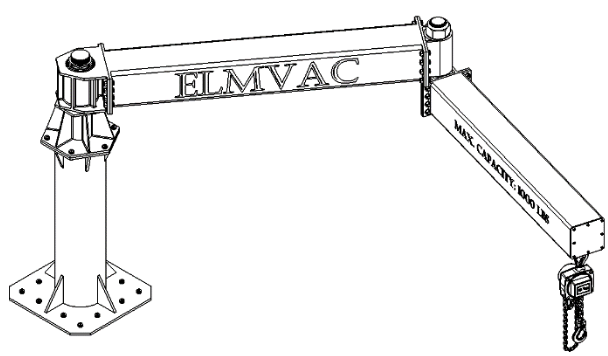


Figure 1: Elmvac crane isometric view

Design

Design Requirements

The typical loading conditions for the crane, as defined by Elmvac, is as follows:

1. Maximum load weight of 500 lbs
2. Maximum load dimensions of 5 ft x 3 ft x 3 ft (W x H x D)

Additional requirements defined by the team were to maximize reach along the trailer as well as to be able to drop the load on either side of the trailer. Finally, the team settled on a factor of safety of 5 as a goal for the crane. A value of 5 is commonly found in engineering of permanent structures, and also provides security in the event that the crane is misused and overloaded.

The articulated jib crane design achieves this goal by maximizing range of motion with a simple design. The team first researched existing designs in order to get a sense of industry design standards. From there, team members were assigned various elements of the design which will be described in following sections.

Design Parameters & Operation

The articulated jib crane design achieves all of the design goals set out for this project.

1. Maximum load weight of 500lbs and dimensions of 5 ft x 3 ft x 3 ft (W x H x D)

The designed crane has a maximum rated capacity of 1000lbs and a maximum reach of 11.2 ft. A minimum of 4 crates (at maximum dimensions) can be unloaded from the trailer without the need to shuffle cargo around the deck.

2. Must deposit the load a minimum of 2 ft from the side(s) or back of the trailer

The articulated design of the crane allows for significant range of motion, allowing for cargo to be deposited over 6 ft off either side of the trailer, as shown in Figure 2& Figure 3.

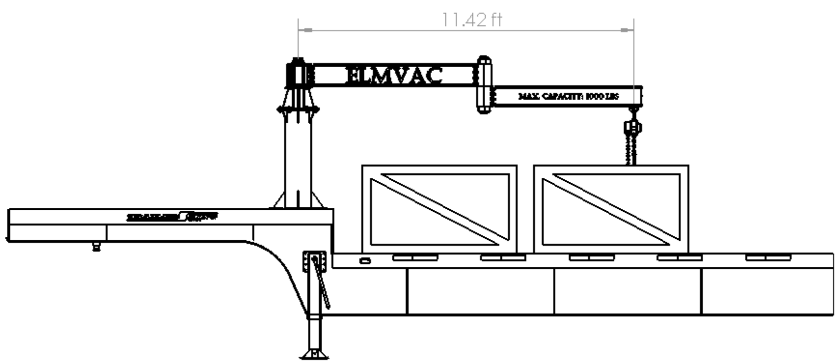


Figure 2: Elmvac crane mounted & in fully extended loading position
Note: Crates arranged 2 deep on trailer (4 in total for config. shown)

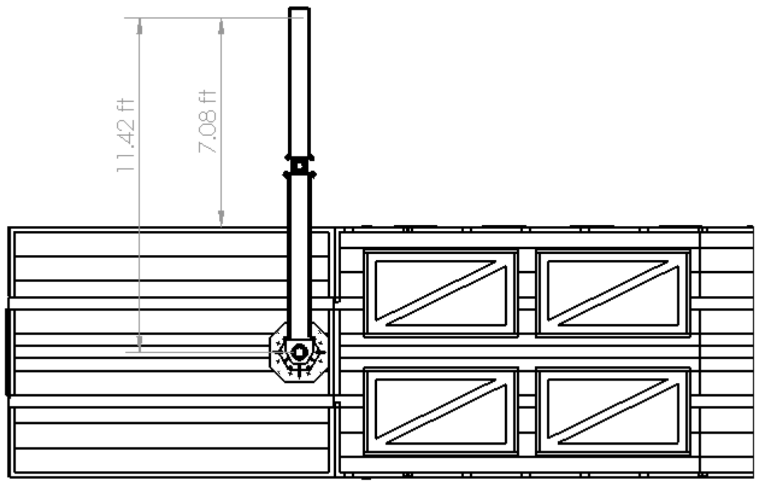


Figure 3: Elmvac crane mounted & in fully extended unloading position

3. Machine must weigh less than 1500 lbf

The articulated jib crane design outlined in this report weighs in at 1347.49.

4. The crane must be a) human powered, or b) require less than 2400 watts

The crane design is easily operated by a single person, if necessary, although having at least 2 workers present is recommended for safety reasons.

Tower Design

The tower for the crane began as cylindrical tower. A cylinder was selected for its uniform bending cross section in all directions. A constant cylindrical profile, in accordance with the design priorities laid out, was selected as opposed to a tapered design for ease of manufacture, as large diameter steel pipes are more widely available off the shelf.

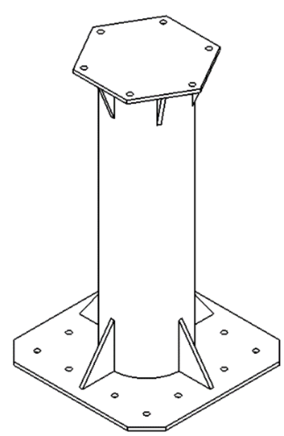


Figure 4: Crane tower isometric view

Continuing the theme of practical engineering selections, the material was chosen to be ASTM A106 steel, as this is a common off-the-shelf material for large diameter steel pipe. A local steel supplier was contacted by the team and provided insight into commonly available products that heavily influenced design.

The base of the tower is designed to be bolted to the trailer deck. A bolted connection was selected to add versatility to the crane, allowing the crane to be mounted on any trailer, regardless of deck material. It is made out of a ½" sheet of plain carbon steel. The base is welded to the tower and further supported by 4 fins welded around the base of the tower. The fins help to redirect the bending of the tower from the welds on the base plate into the trailer deck.

The top of the tower is capped with a hexagonal plate of ½" carbon steel to mate with the support structure for the crane's primary pivot point. A 6-bolt pattern provides more even stress distribution in bending than a 4-bolt pattern. The holes are sized to accommodate ¾" bolts, similarly to the holes used to mount the tower to the trailer. These bolts will be discussed later. The team felt that additional bolts (i.e., an 8 or 10 bolt pattern) would be superfluous. To support the hexagonal mounting plate, 6 fins, made of ½" steel, are welded to oppose deformation of the plate under bending loads.

The bolts to hold the tower to the trailer were sized the same as all other bolts used to fasten load-bearing segments of the crane together to reduce the number of tools needed for assembly. The length of the base-plate bolts was not specified, however, as this would be trailer dependent to reach through the deck. Originally, an 8-bolt pattern was implemented on the tower base. Analysis showed it was necessary to bolster this aspect of tower design. One extra bolt was added per base quadrant, totaling 12 bolts in the final design.

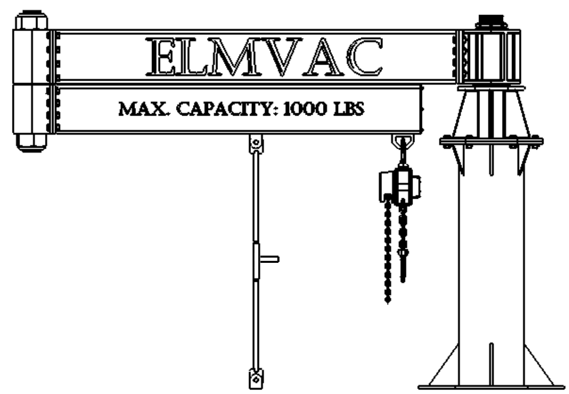


Figure 5: Crane in transport configuration

Boom Design

To maintain uniformity in the crane's appearance, the upper and lower boom were designed to be the same cross-sectional shape: hollow square tubing (Figure 6). The team spent significant time considering this decision, with an I (or H) beam cross section coming in second. Ultimately it was the box channel that won out due to its ability to withstand both bending and torsion. Due to the nature of an articulating jib, some crane positions place high torsional loading on the upper boom that must be accounted for.

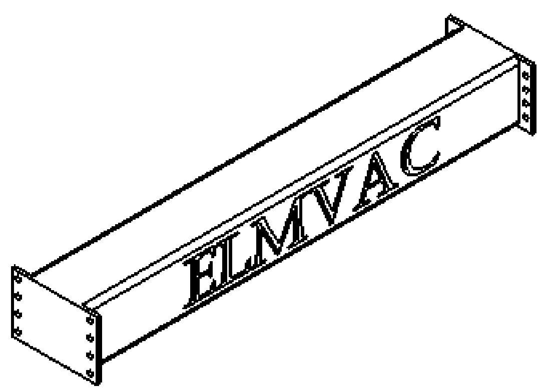


Figure 6: Upper boom isometric drawing

In addition to beam cross section, beam length parameters were chosen to be 5.5ft and 5ft for the upper and lower boom respectively. This combination allows for plenty of reach while also allowing for clearance for the lower boom to tuck under the upper boom for transport (Figure 5).

Design of the lower boom was done using SMath, a computer program that allows for mathematical worksheets to be created and calculations easily repeated. A simple method of trial and error, whereby common beam dimensions were input into the worksheet and the mass and stresses in the beam were calculated based some input load. For this load, 2500 lbf was used to ensure no factor of safety issues later on. Final parameters of the lower & upper boom are shown in Table 1.

Table 1: Relevant beam dimensions

Crane Boom - Relevent Dimension			
Primary Boom		Secondary Boom	
Height	9in	Height	8in
Thickness	0.375in	Thickness	0.25in
Length	5.5ft	Length	5ft

Onto the outer end of the lower boom a flange was added in order to cap the end of the beam. This cap (Figure 7) serves two purposes. Firstly, it protects the inside of the boom from the elements, which will help to extend the life of the crane. Secondly, its removability allows for easy access inside for inspection of the boom.

Onto the end of the boom which mounts to the center articulation point is another flange. This flange is thicker than the outer flange and is welded on 2 sides to the channel using fillet welds. This flange is critical as it transmits all of weight of the boom and load into the articulation point, and in turn the upper boom. It is also made of ½" plate steel to resist high stresses. The welds on this flange were sized according to industry standard procedure and sizes were limited by the thickness of the box channel walls. The fillet welds are 1/8".

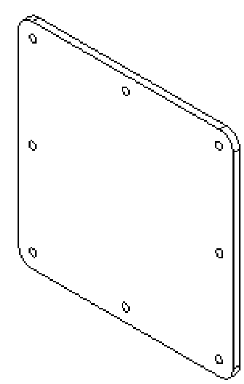
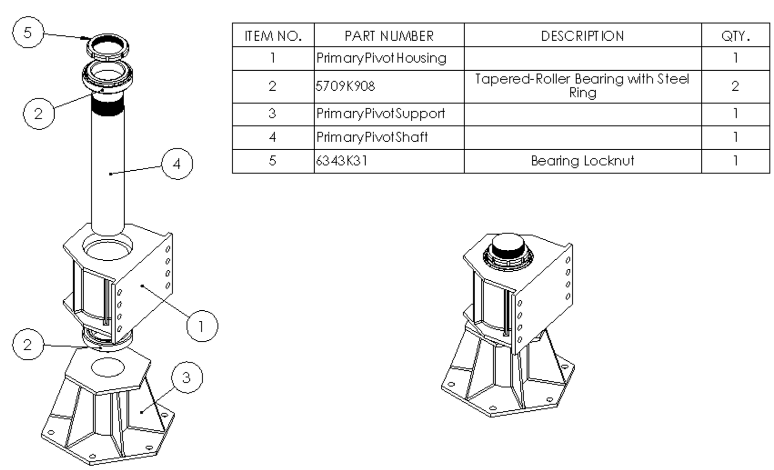


Figure 7: Lower boom end cap

Primary Pivot Design

Pivot Point Overview

The primary pivot point was designed to connect the crane arms to the tower. It needs to accept high bending and axial loads as well as allow for 360 degrees of rotation. In order to accomplish this, a high strength steel cylinder is pressed into a lower housing. An upper housing rides on this shaft via 2 tapered roller bearings to provide radial and axial support while maintaining rotational freedom. A diagram of the assembly is shown in Figure 8.



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	PrimaryPivotHousing		1
2	5709K908	Tapered-Roller Bearing with Steel Ring	2
3	PrimaryPivotSupport		1
4	PrimaryPivotShaft		1
5	6343K31	Bearing Locknut	1

Figure 8: Primary pivot point - exploded and isometric view

In addition to rotational freedom, the pivot must be able to be locked for transport so that the crane does not swing freely. This is accomplished by a simple flange protruding from the back of both the upper and lower housing. A pin is inserted through this flange when the operator wishes to lock crane rotation.

Upper Housing

The primary pivot point of the crane is the one of the highest points of stress for the machine. It is here that all bending moments and lifting forces are transferred from the boom sections to the tower.

In order to accept these forces while maintaining 360 degrees of rotational freedom, the point was designed with two large off-the-shelf tapered roller bearings that can accept both high axial and radial loading. The capacity of this bearing far exceeds the necessary loading, but the sizing of the bearing was dictated by the necessary size of the cylindrical member on which it rides rather than by its loading.

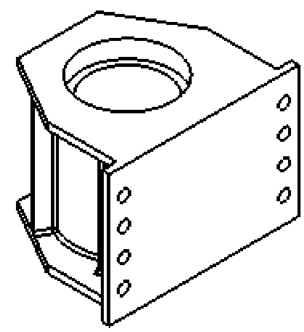


Figure 9: Primary pivot upper housing - isometric view

In order to withstand the high bending loads created by the boom at maximum extension, the cylindrical member is 4 inches in diameter. Due to the nature of the primary pivot assembly, the axial loads on the cylindrical member should be insignificant, as vertical load is taken up by the lower roller bearing. Still, the bending moment in the cylindrical member is significant. The material for this member is 4140 steel. The steel is to be quenched and tempered once machined in order to increase its yield strength to 238 kpsi.

The upper portion of the cylindrical member is threaded to 3.918" - 12. This thread allows for a bearing retaining nut to be threaded on. While the axial load in the shaft is negligible, this nut helps to maintain joint stiffness in the event of rough transport or a rapid unloading of the crane.

Lower Support Housing

The upper pivot housing sets atop a tapered steel housing that serves to transmit the forces from the large cylindrical member described above into the tower down to the trailer. This housing is shown in Figure 10.

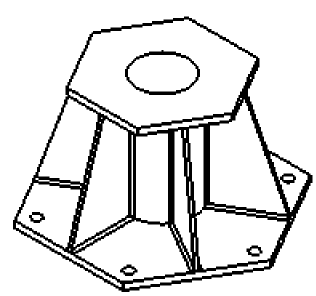


Figure 10: Primary pivot lower support housing - isometric view

The center of the housing is made from a hollow steel cylinder with a ½" wall thickness. The inside is sized to accept the primary pivot pin in a tight press fit. The wall thickness of this central cylinder both ensures rigidity as well as providing support for the vertical load applied to the crane. The lower thrust bearing of the upper support section rests atop the inner steel cylinder such that the cylinder will take the brunt of the vertical load. The fins arranged around the outside of the hexagonal profile serve to take up the high bending loads applied to the pivot pin. All plates and fins are made from the same ½" steel used in the upper housing. The bolt pattern on the bottom hexagonal plate serves to connect the housing to the top of the tower. The holes are sized for ¾" bolts.

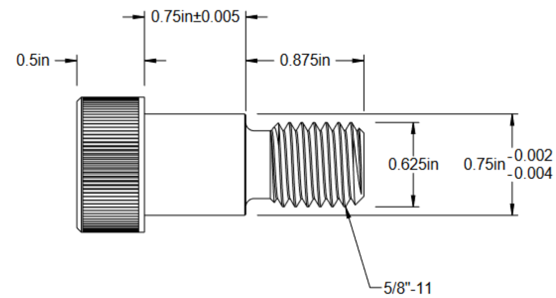


Figure 11: 5/8" - 11 bolt used for all major structural bolted connections (Drawing provided by McMaster Carr)

The bolts selected for the task are 5/8" - 11 shoulder bolts with a ¾" diameter shank, as shown in Figure 11. Analysis shows that in some places, these bolts are significantly oversized. The team felt that to offset the inconvenience of the unusual head of the bolt (allen key vs a hex head), it would be helpful to size all structural bolts the same. Since the cost of hardware is relatively insignificant compared to the overall machine cost, this trade off was deemed adequate.

Articulation Pivot Design

The articulation point pivot (Figure 13)was designed similarly to the primary pivot point. The only major difference is the center pin size, which is down to 3 inches at this point. This was done to maintain off-the-shelf compatibility. Load capacity was unaffected.

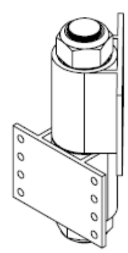


Figure 13: Articulation point

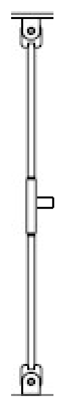


Figure 12: Transportation support

Transportation Locking Mechanism Design

The transportation locking support was deemed essential, ad during the transportation of the crane, uneven roads could create high impact loading on the primary and secondary pivot joints. The transport mechanism consists of two members, pinned at both ends, and a turnbuckle for pretensioning (Figure 12).

Analysis

Beam Analysis

The two boom sections, primary and secondary, were considered independently to better account for their respective mass effects. Two conditions were considered:

1. Beams in a parallel configuration for peak bending moment.
2. Beams in a perpendicular configuration for peak torsion on the primary boom.

Secondary Boom

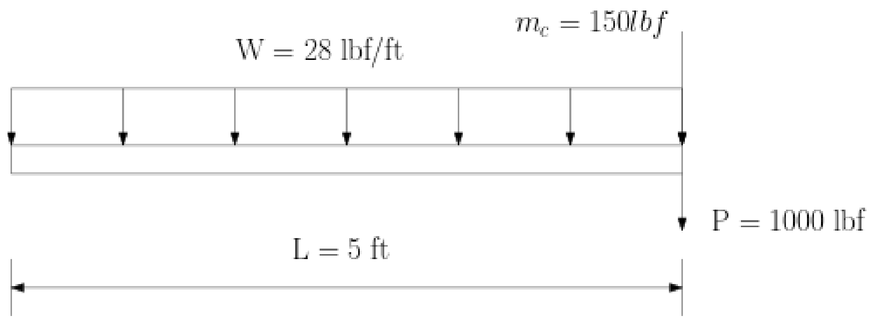


Figure 14: Secondary boom free body diagram

The secondary boom will be analyzed first, as its results will be necessary to analyze the primary boom. The first step is to find the applied bending moment and associated stress.

For both conditions, the bending moment on the secondary boom will be identical. The bending moment is computed using equation 1:

$$M_0 = P * L + m_b * \frac{L}{2} \quad (1)$$

$$M_0 = (1150\ lbf) * 5ft + 200\ lbf * 2.5\ ft = 6200\ ft\ lb$$

Where

M_o is the bending moment

P is an applied force at the end of the boom consisting of both the desired load (1000 lbs) and the weight of the chain fall (150 lbs)

L -is the length of the boom

m_b is the mass of the boom

To find the peak bending stress in the beam, the area moment of inertia of its cross-section (Figure 13) must be determined. Equation 2 is used to perform this calculation.

$$I = \frac{h^4 - (h - 2 * t)^4}{12} = 0.0037\, ft^4 \qquad 2$$

Equation 2 calculates the area moment of inertia about a horizontal axis through the center of the cross-section.

The value derived from equation 2 is next used to find the peak bending stress in the beam. In general, the peak bending stress in any member is given by equation 3:

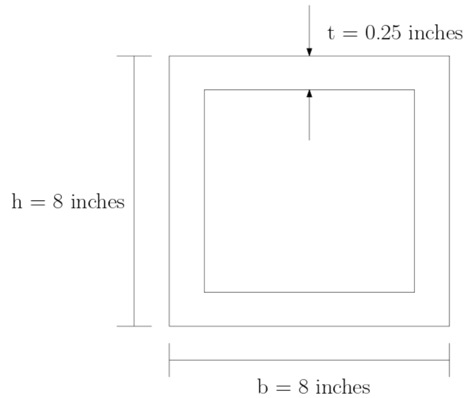


Figure 15: Secondary beam cross-section

$$\sigma_b = \frac{My}{I} \qquad 3$$

where y is the distance from the center of the beam to the outer most edge of the member. In the case of the square channel used for the secondary boom, equation 3 evaluates to:

$$\sigma_b = \frac{M_0 * \frac{h}{2}}{I} = \frac{6200\, lbf - ft\ *\ 0.33\, ft}{0.0037\, ft^4} = 3.9\, kpsi$$

Evaluating equation 3 for the secondary boom yields a bending stress of 3.9 kpsi. Considering that the yielding strength of steel is 30 kpsi, this result points to a significant factor of safety. In order to calculate the factor of safety, equation 4 is used:

$$n = \frac{S_y}{\sigma_b} = \frac{30\, kpsi}{3.9\, kpsi} = 7.69 \qquad 4$$

As laid out in the design section, the factor of safety target for the crane is 5. Consequently, the calculation in equation 4 confirms that the design of the beam can meet this requirement in bending.

In addition to bending stress, shear stress is also a point of concern in the crane booms. From the free body diagram in Figure 12 a shear force diagram can be derived, shown here in Figure 14.

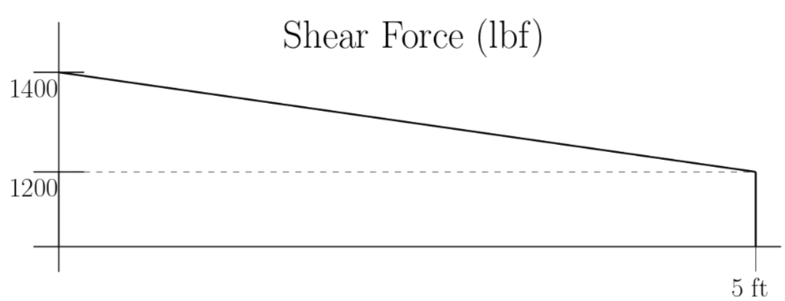


Figure 16: Secondary boom shear force diagram

The peak shear force present in the beam occurs at the connection point with the central pivot, as this is the only reaction point for the beam. The value of 1400 lbf comes from the summation of all vertical forces acting on the beam, namely: the weight of the beam, the applied load (1000 lbf), and the approximate weight of the chain fall (150 lbf).

The following calculation for shear stress is simply the peak internal shear in the beam divided by the cross-sectional area, demonstrated in equation 5:

$$\tau = \frac{V}{A} = \frac{1400\ lbf}{0.66\ ft^2} = 0.17\ kpsi \qquad 5$$

To calculate the factor of safety, the conservative *Maximum Shear Stress Theory* is used to calculate the yield strength in shear of the beam:

$$S_{ys} = 0.5 * S_y = 15\ kpsi \qquad 6$$

Considering the vast difference here, it is safe to say that failure in shear will not occur, but for the sake of thoroughness the factor of safety was found using equation 4 above, substituting the new yield strength in shear:

$$n = \frac{S_{ys}}{\tau} = \frac{15kpsi}{0.17kpsi} = 88$$

While not required for safety reasons, the static deflection under load was computed to check that it came to a reasonable value. From beam tables in Shigley's, equations for the peak deflection for a cantilevered beam under a uniform load and a point load at the end were taken. These were summed together to provide an expression for the deflection of the secondary beam from this analysis and evaluated,

$$\delta = \frac{m_{unit} * L^4}{8EI} + \frac{PL^3}{3EI} = 0.032\ in \qquad 7$$

In equation 7, the mass per unit length (m_{unit}) comes from the product of cross-sectional area of the secondary beam and the density of steel. The value is shown in the free body diagram of the secondary beam (Figure 12). The applied point load, P, is the summation of the two point-loads, also shown in the free body diagram. The computed value of 0.032 inches is more than acceptable for this application and demonstrates a rigid beam suitable for the application.

Primary Boom

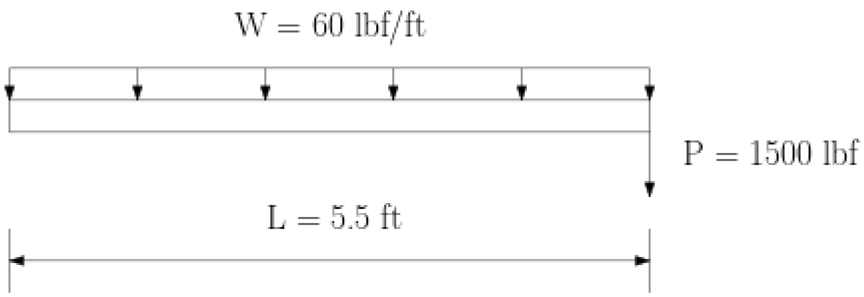


Figure 17: Primary beam free body diagram

The analysis for the primary boom follows the same path and equations as the secondary boom, with minor changes. The first major change is in the area moment of inertia. The primary boom has a larger cross section (Figure 16), and therefore a new area moment was calculated using equation 2 from above:

$$I = \frac{h^4 - (h - 2 * t)^4}{12} = 0.0078\ ft^4$$

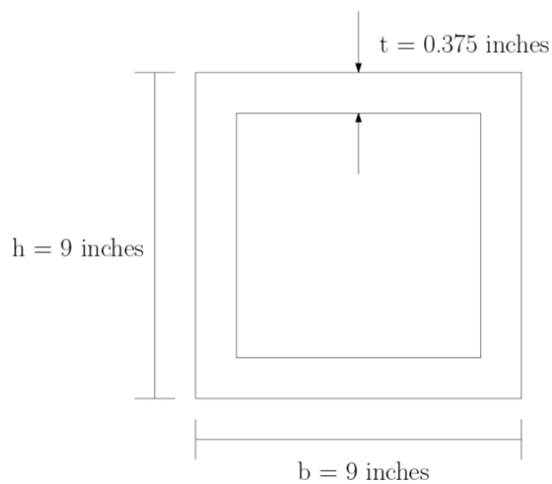


Figure 18: Primary beam cross-section

Once again, using the area moment of inertia and the general bending stress equation (equation 3 above), the bending stress of the primary boom is calculated. For this calculation, the applied loading is changed, but the method of solving for the bending stress is unchanged.

Firstly, the bending moment due to the applied vertical forces is found through equation 8:

$$M_{op} = M_0 + P * L + m_b * \frac{L}{2} = 1.6 * 10^4\ lbf\ ft \qquad 8$$

where

M_{op} is the overall bending moment

M_o is the bending moment from the secondary beam

m_b is the mass of the primary beam

L is the length of the primary beam

P is the total mass of the secondary beam, central pivot connection, chain fall, and applied load.

It is necessary in this section to note that the weight of the bolts holding the beams together as well as the smaller bolts connecting the end cap of the secondary beam has been left out of the moment analysis. While these components do contribute to the moment on the beam, their contribution is insignificant in proportion to the applied loads and mass of the beams. Therefore, they were neglected for simplicity.

Once again, using equation 3, the peak bending stress in the beam is found. Substituting values for the primary beam yields:

$$\sigma_b = \frac{M_{op}\left(\frac{h_p}{2}\right)}{\mathrm{I_p}} = 5.2\ kpsi$$

Applying this value to equation 4 to find the factor of safety yields:

$$n = \frac{S_y}{\sigma_b} = \frac{30kpsi}{5.2kpsi} = 5.77$$

The result shown here confirms that the primary beam design meets the criteria set forth for the project.

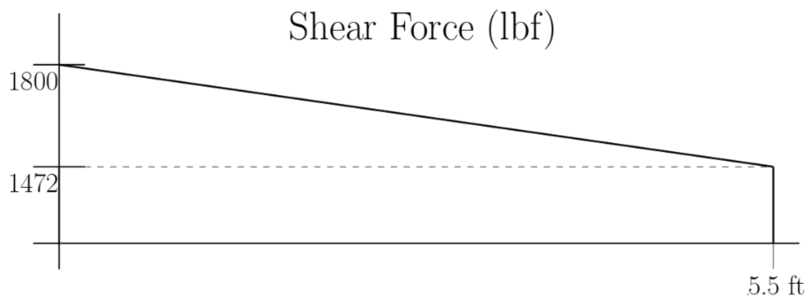


Figure 19: Primary beam shear force diagram

Calculating the peak shear stress is done as for the secondary beam. The peak shear load is found by constructing a shear force diagram (Figure 17) from the free body diagram (Figure 15)of the beam. Again, using the process used on the secondary beam, the shear stress and shear factor of safety is found using the *Maximum Shear Stress Theory*.

$$\tau = \frac{V_{max}}{A} = \frac{1800lbf}{14in^2} = 0.14kpsi$$

$$n = \frac{S_{ys}}{\tau} = \frac{15kpsi}{0.14kpsi} = 107$$

Similarly, to the secondary beam, shear failure is far from an issue.

For completeness, the static deflection under load of the primary beam is calculated and added to that calculated for the secondary beam to get an overall deflection using equation 9:

$$\delta_p = 0.055in \qquad 9$$

In addition to bending, the primary beam will undergo substantial torsion when the beams are oriented 90 degrees to one another. To simplify the calculations, the magnitude of the torque applied to the primary beam is equivalent to the peak bending moment found in the analysis of the secondary beam.

To calculate torsional stresses the polar moment of inertia takes the place of the area moment of inertia. The polar moment is calculated using equation 10:

$$J_p = \frac{2t_p {h_p}^4}{2h_p}\left(1 - \left(\frac{t_p}{h_p}\right)\right)^4 = 230in^4 \qquad 10$$

The relevant dimensions, minus their subscripts (p for primary beam), are shown in Figure 16 above.

With the polar moment of inertia found, the torsional stress and angle of twist are given by equations 11 and 12 respectively

$$\gamma = \frac{M_0\left(\frac{h_p}{2}\right)}{J_p} = 1.5kpsi \qquad 11$$

$$\theta = \frac{M_0 L_p}{J_p G} = 0.11deg \qquad 12$$

Before the analysis can be completed, the *Von Mises Effective* stress must be calculated for the combined loading of the beam. The peak bending stress found above as well as the torsional stress are entered into equation 13 as stress in the x direction and shear stress respectively:

$$\sigma' = \sqrt{\sigma_a^2 + 3\tau_{xy}^2} = 5.8kpsi \qquad 13$$

Reperforming the factor of safety calculation yields a factor of safety that is still over 5, so the beam is adequately sized:

$$n = \frac{30kpsi}{5.8kpsi} = 5.16$$

Tower Analysis

Due to the complex nature of the tower geometry, a solidworks simulation was performed to obtain stress and displacement results. The results of that study are shown below in Figure 19. Loading was applied to the tower as shown in the simplified free body diagram (Figure 18). The magnitude of the bending moment comes from the analysis

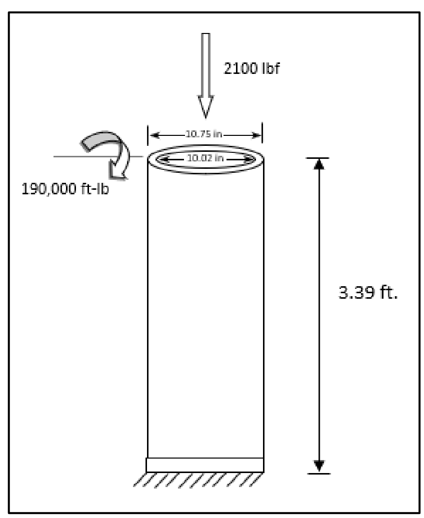


Figure 20: Simplified tower free body diagram

of the primary beam, while the magnitude of the vertical force comes from a weight analysis of the entire upper assembly of the crane.

The results show that the peak Von Mises stress is lower than the yield strength of the material. Performing a factor of safety calculation yields:

$$n = \frac{35kpsi}{13.7kpsi} = 2.55$$

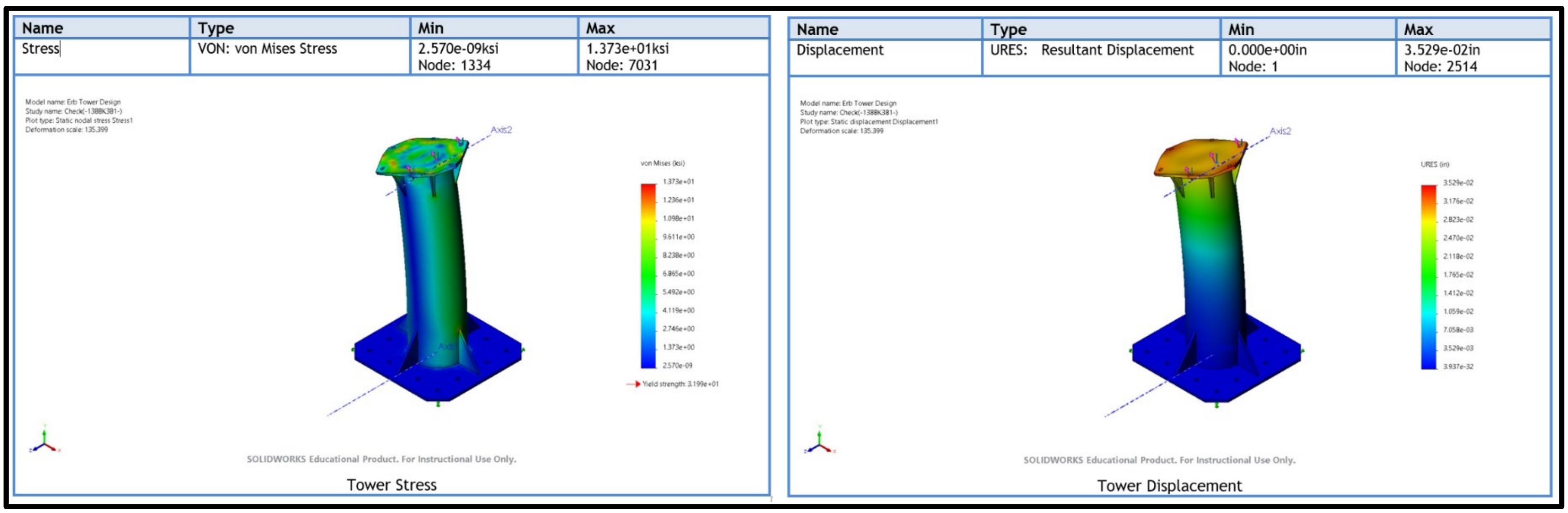


Figure 21: Solidworks stress and displacement results for crane tower; Von Mises stress (left) & Displacement (right)

Central Pivot Point Analysis

Pin Analysis

The pin for the central pivot sees two big stresses: axial and bending. The axial load on the pin is calculated simply using equation 14:

$$\sigma = \frac{F}{A} \tag{14}$$

$$\sigma = \frac{1500lbf}{7.1in^2} = 0.21kpsi$$

The bending stress in the pin is calculate using equation 3 above:

$$\sigma_b = \frac{1.8 * 10^4 lbfft * 0.125ft}{4in^4} = 81kpsi$$

The area moment of inertia was calculated using equation 15:

$$I_{pin} = \frac{\pi}{64} * \left(d_{pin}^4\right) = 4in^4 \tag{15}$$

The pin also sees significant shear stress at the point where it rests on the bearings. This shear stress is calculated by first dividing the overall bending moment over the distance between the two points to find the moment couple shear forces using equation 16:

$$V = \frac{M_0}{L_{couple}} = 27kips \tag{16}$$

With this value, a shear stress can be calculated using equation 5:

$$\tau = \frac{V}{A} = \frac{27kips}{7.1in^2} = 3.8kpsi$$

Due to the nature of the forces on the pin, two factors of safety must be checked. At the location where the shear force is at a maximum, the bending stress will be zero, and the opposite is true where the bending stress peaks. Therefore, both the factor of safety in shear and bending are determined. Equations 4 and 6, as well as the *Maximum Shear Stress Theory* for the shear yield strength, are used:

$$S_{ys} = 0.5 * S_y = 0.5 * 238kpsi = 119kpsi$$

$$n_{shear} = \frac{S_{ys}}{\tau} = \frac{119kpsi}{3.8kpsi} = 31$$

$$n_{bending} = \frac{S_y}{\sigma} = \frac{238kpsi}{82.21kpsi} = 2.9$$

Housing Analysis

Due to the complex nature of the housing used to contain the bearings and pin, a solidworks simulation was performed to determine peak stresses and deflections.

The lower housing was analyzed first. The applied load is a combined load consisting of an applied moment and vertical force, applied at the face of the bolted connection. The magnitudes of these loadings was determined in the analysis of the secondary beam, and will be reused here. The applied loadings used in the simulation are shown in Figure 20.

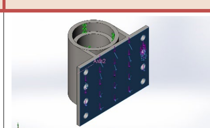
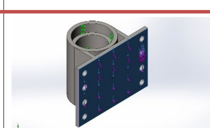
Load name	Load Image	Load Details
Torque-1		Entities: 1 face(s) Reference: Axis2 Type: Apply torque Value: -75,000 lbf.in
Force-1		Entities: 1 face(s) Reference: Edge< 1 > Type: Apply force Values: ---, ---, 1,400 lbf

Figure 22: Applied loading for lower articulation housing Solidworks simulation.

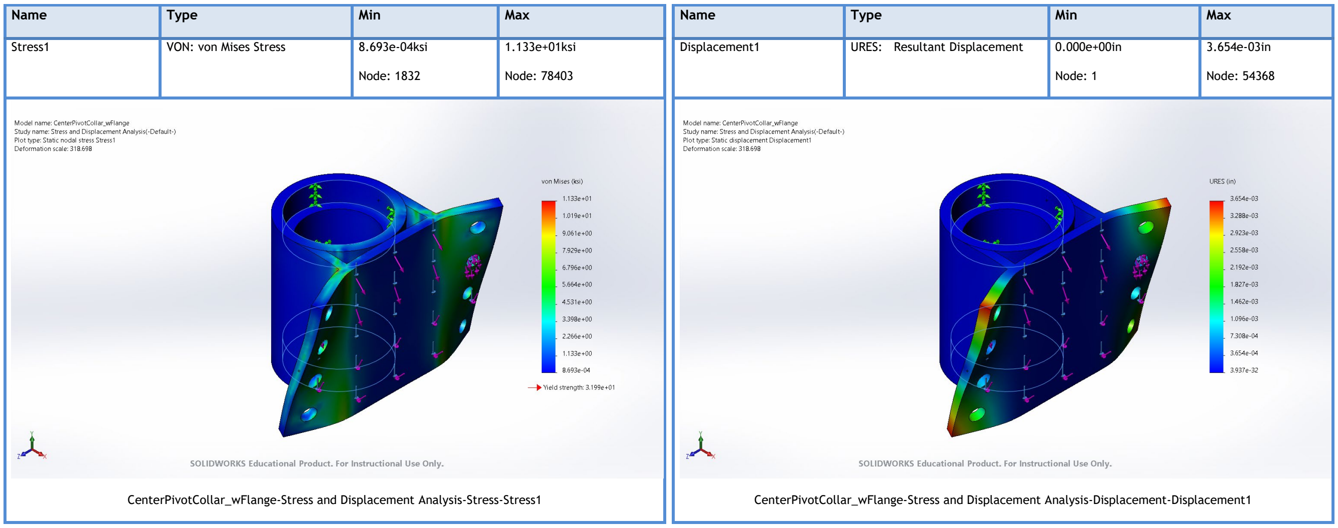


Figure 21: Solidworks simulation results for lower articulation point housing; Von Mises stress (left) & Displacement (right)

The results of the simulation, shown below in Figure 21 were positive. The peak von mises resultant stress came out to $\sigma_{max} = 11.33\ kpsi$ which yields a factor of safety from equation 4 as follows:

$$n = \frac{S_y}{\sigma_{max}} = \frac{31.99kpsi}{11.33kpsi} = 2.82$$

In addition to stress, the peak displacement was computed for reference. While no particular target for this value was set, ensuring that it remained significantly small was important.

The upper housing was analyzed in a similar manner, using the same applied bending moment, and adding the weight of the articulation assembly to the vertical loading component. The results of that simulation are shown below in Figure 22. The peak Von

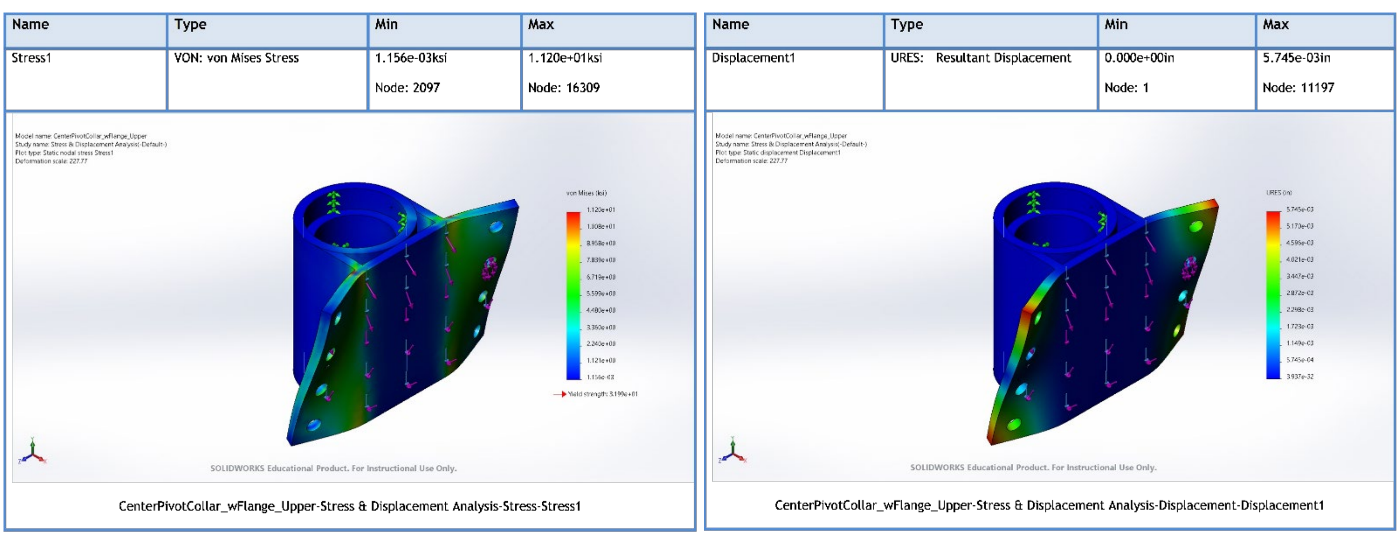


Figure 22: Solidworks simulation results for upper articulation point housing; Von Mises (left) & Displacement (right)

Mises stress evaluated to $\sigma_{max} = 11.2kpsi$ while the peak displacement came to a more than suitable value of $\delta_{max} = 0.00575in$. Computing the factor of safety as before yields:

$$n = \frac{31.99kpsi}{11.2kpsi} = 2.85$$

Primary Pivot Point Analysis

Pin Analysis

The pin used for the primary pivot point was analyzed in the same was as the pin for the articulation pivot, with only a couple of key differences. Firstly, the outer diameter of this pin is 4 inches, rather than 3. Secondly, there is no axial load on this pin. Rather, it is always in bending.

First, using equation 13, a new moment of inertia is found:

$$I = \frac{\pi d^4}{64} = 13in^4$$

Next, the bending stress is found using equation 3 the bending stress is found:

$$\sigma_b = \frac{M_0 \frac{d_{shaft}}{2}}{I} = 30kpsi$$

Since there is no torsion or axial loads on the pin, this stress can be used for a factor of safety calculation:

$$n = \frac{S_y}{\sigma_b} = \frac{238kpsi}{30kpsi} = 7.93$$

Similarly to the articulation pin, the shear stress in the pin must also be found. By dividing the bending moment over the distance between the two thrust bearings, the shear force for the moment couple is found:

$$V = \frac{\sigma_b}{8in} = 24kips$$

From there, the shear stress is found:

$$\tau = \frac{V}{A} = 1.9kpsi$$

Using the *Maximum Shear Stress Theory,* the yielding stress is 119 kpsi, and the factor of safety is as follows:

$$n = \frac{119kpsi}{1.9}kpsi$$

Housing Analysis

The housing that accepts the primary pivot bearings, and onto which the primary boom attaches was analyzed under two conditions. Firstly, with both booms fully extended, the peak bending and vertical load was applied to the connection face. Figure 23 shows these loads and their respective directions. The applied bending moment is the same as the peak bending moment from the primary boom analysis, and acts about an axis through the centerline of the housing, parallel to the bolt face. The vertical load is 1800 lbf from the weight of the boom assembly. The results of this study show that the peak Von Mises stress is 17.1 kpsi and the peak displacement is 0.00197 in. Performing a factor of safety calculation using equation 4 yields:

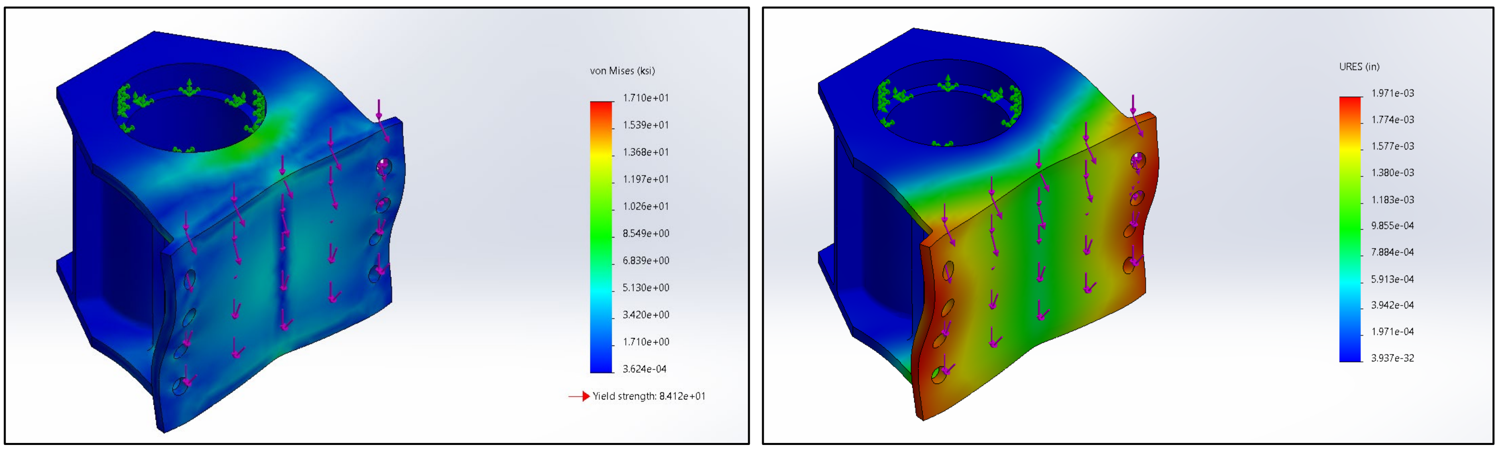


Figure 23: Bending stress (left) and displacement (right) of primary pivot housing in full extension position

$$n = \frac{84.12kpsi}{17.1kpsi} = 4.92$$

A second study was then performed on the housing, this time in bending and torsion with a vertical load. This represent the crane when the lower arm is arranged perpendicular to the primary arm. The vertical load remained the same, at 1800 lbf. The torsional load is equivalent to the peak bending moment of the lower boom and acts about an axis through the center of the housing perpendicular to the bolt face. The applied loadings as well as the results are shown in Figure 24.

The results of the stress analysis yield a peak Von Mises stress of 15.89 kpsi and a peak displacement of 0.00232 in. Performing the factor of safety calculation yields:

$$n = \frac{84.12kpsi}{15.89kpsi} = 5.29$$

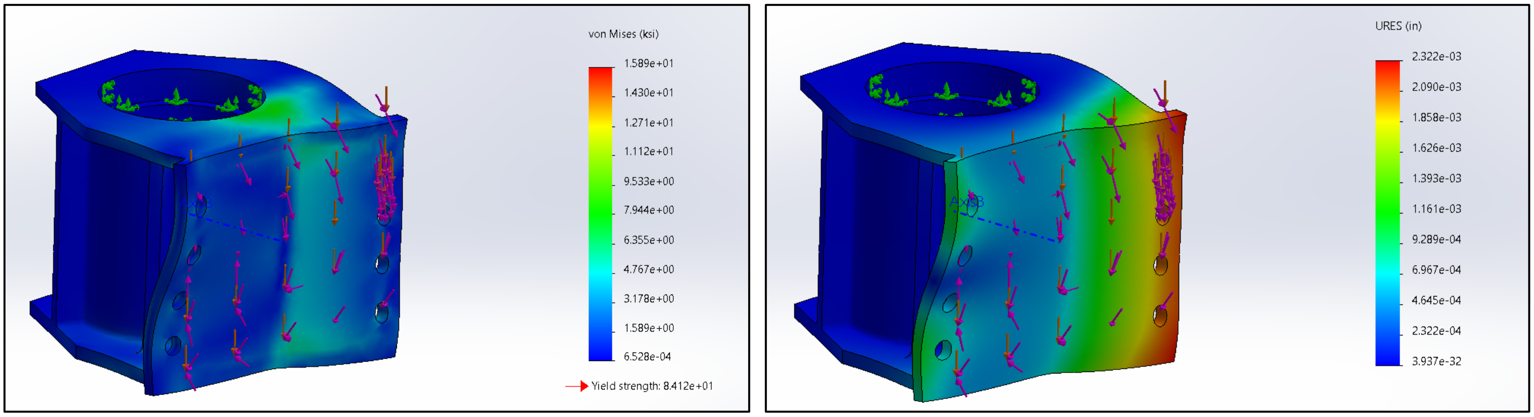


Figure 24: Primary pivot point von mises stress (left) and displacement (right) from combined bending, torsion, and vertical loading

The final result of the analysis of the primary pivot housing yields a minimum factor of safety of 4.92 and more than adequately small deformation.

Transportation Locking Mechanism Analysis

The transportation support is not expected to take the full load of the crane most of the time. Rather, it is expected to prevent impact loading from rough roads to be transmitted through the pivot supports in the form of a bending impact.

Vertical Support Member

Since the nature of the loading on this member is expected to be cyclic, a basic endurance limit calculation is performed before the member could be sized.

Firstly, the uncorrected endurance limit for the material is found by equation 17:

$$S_e' = 0.5 * S_{ut} = 27.5kpsi \qquad 17$$

Next, the corrected endurance limit is found through equation 16:

$$S_e = K_a K_c K_d K_e S_e' \qquad 18$$

The various K values represent correction factors as follows:

- $K_a = Surface\ correction\ factor\ = 0.811$
- $K_c = Axial\ load\ correction\ factor = 0.85$
- $K_d = Temperature\ correction\ factor = 1$
- $K_e = Reliability\ correction\ factor = 0.814(for 99\% reliability)$

Applying the selected values to equation 15 yields a corrected endurance limit as follows:

$$S_e = 15.4kpsi$$

In order to calculate the factor of safety for the member under a fluctuating load, the Modified Goodman approach was used, as described in equation 19:

$$\frac{1}{n} = \frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{ut}} \tag{19}$$

Since the member is not expected to take significant load statically, the mean stress (σ_m) goes to zero. Equation 19 was then rearranged to solve for the maximum alternating stress:

$$\sigma_a = \frac{S_e}{n} = \frac{15.4}{5} = 3.08 kpsi$$

With the peak alternating stress found, the proper area for the circular member could be found by rearranging equation 12 to solve for area:

$$A = \frac{F}{\sigma_a} = \frac{1100 lbf}{3.08 kpsi} = 0.357 in^2$$

Solving for the rod diameter yields $d = 0.674\ in$. For convenience, this value was rounded to 0.75 as this is a more common size. Recalculating the actual factor of safety with this diameter yields:

$$n = \frac{S_e}{\sigma_a} = \frac{15.4 kpsi}{2.49 kpsi} = 6.18$$

Finally, the member must be checked for buckling due to its long slender nature. First, using equation 20, the minimum moment of inertia to prevent buckling is found:

$$I = \frac{PL^2}{\pi^2 E} = 0.00615 in^4 \tag{20}$$

Next, using equation 13, the moment of inertia of the long slender members and the turnbuckle are found:

$$I_{member} = 0.0155 in^4$$

$$I_{turnbuckle} = 0.0336 in^4$$

Since both of these values are sufficiently high, the support is deemed sufficient.

End Connection Bracket

On either end of the long member is a plate through which a pin sits and takes the load. This plate was analyzed for tear out stress. Using the weight of the entire boom assembly as a reference force, the bearing stress in the pin and tear out stress in the plate were calculated using equations 21 and 22 respectively:

$$\sigma_{bearing} = \frac{P}{t * d} = 2.93 kpsi \tag{21}$$

$$\sigma_{tearout} = \frac{p}{2 * t * b} = 1.46kpsi \quad (20)$$

In the equations above, t represents the plate thickness, d represents the diameter of the pin, and b represents the distance from the hole in the plate to the edge of the plate. These dimensions are shown in detail in the drawing package included in the appendix.

Weld Analysis

The plates on either end of the support are connected to the rods via welds. The length of these welds will be twice the diameter of the rod, as they are double fillets. A quick check is performed to ensure that they will hold up.

Using the 3/16" sizing for the welds, the necessary length is calculated using equation 21:

$$l = \frac{P}{4h(0.707)(0.3S_{ut})} = 0.126in \quad (21)$$

Since the actual length of the welds is 1.5 inches, they will be more than sufficient, and factor of safety was not calculated.

Fastener Analysis

In order to confirm that the selected fasteners will meet the requirements of the design, they must be analyzed in several ways. Firstly, the bolt connection between the secondary beam and pivot housing must be checked both for separation and shear. The bolt connection between the upper pivot housing and the primary beam must be analyzed in separation and shear as well in addition to an analysis in torsion. Finally, the pretension of the bolts must be determined and checked against the compressive force in the members. If the pretension is too low, then the compressive force on the lower bolts resulting from bending stress under load will surpass the pretension and cause the bolts to come loose, which would result in failure of the connection.

Since all load bearing fasteners have been sized the same, the allowable stress in these bolts can be found once and then compared to the actual stress in each fastener to confirm minimum factor of safety.

The proof strength of the chose fasteners is 86 kpsi. For a factor of safety of 5, the maximum allowable axial stress in the bolts is as follows:

$$\sigma_{allow} = \frac{86kpsi}{5} = 17.2kpsi$$

From this an allowable proof force can be found using the bolt root diameter:

$$F_p = \sigma_{allow} * \frac{\pi d_{root}^2}{4} = 6.192kips \quad 22$$

Articulation Point Upper & Lower Connection

Since the crane arm is rigidly attached to a solid steel plate, it is assumed that the bending moment of the arm is transferred uniformly into that plate. Therefore, the area moment of inertia of the plate, given below in equation 23, is used for calculations.

$$I_{plate} = \frac{b_{plate} h_{plate}^3}{12} = 400in^4 \quad 23$$

To find the stress in the bolts due to the bending moment applied by the crane arm, the compressive stress at the distance of each bolt is found using equation 3, with bending moment taken from the secondary beam analysis:

$$\sigma_1 = 0.19kpsi$$

$$\sigma_2 = 0.56kpsi$$

The first stress is the member compressive stress present at the bolts nearest to the midplane, and the second at the further bolts. It follows that the peak stress due to the applied load in the bolt will have a magnitude equal to σ_2 on the upper most bolts. To find the force in each bolt, this stress is divided by the total bolt area, which is two bolts:

$$F_{applied} = 0.56kpsi * \frac{\pi(d_{root})^2}{4} = 100lbf$$

Next, the pretension in the bolt is calculated using equation 24:

$$F_i = 0.75F_p = 4.644\ kips \quad 24$$

Finally, the total force in the bolt is found using equation 25:

$$F_{total} = F_i + F_{applied} = 4.744kips \quad 25$$

Since the total force in the bolt is lower than the amount allowed for the factor of safety, then this bolt is adequately sized.

The bolts at this connection also see a vertical shear load. This shear stress is found simply by dividing the vertical load, taken from the weight of the secondary beam and all attached components, and dividing it by the total bolt shank area:

$$\tau = \frac{1400lbf}{8 * A_{shank}} = 0.396kpsi$$

In the case of this connection, since the shear stress is an order of magnitude lower than the axial load on the fasteners, it was determined that it will not reduce the machine's overall factor of safety.

In order to achieve the preload specified here, the torque spec for the fastener is determined by equation 24:

$$T_i = 380 ftlb \qquad 24$$

Primary Pivot Beam Connection

The analysis of the bolt connection between the primary boom and primary pivot connection in bending is done similarly to the articulation connection, with the only differences being in the bolt distances. Recalculation of the stresses, using the same equations and new geometry yields similar stresses, and an identical torque spec.

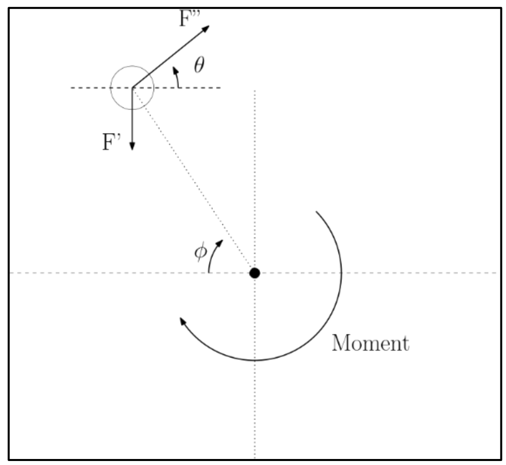


Figure 25: Trigonometric diagram for bolt calculations

For this connection, it is necessary to check the bolts in shear when the primary boom is in torsion. Using the values for torsion found in the analysis of the primary boom as well as the associated bending moment and shear forces, the shear in the bolts can be found.

The calculation was performed using a MATLAB code that looped through each bolt as the bolt pattern is not uniform. In this section the process for just 1 bolt will be described. Figure 25 shows the relevant angles and force vectors for an arbitrary bolt.

From Figure 25, the angle from the centroid of the bolt pattern can be determined using the horizontal and vertical distances of the bolt, as equation 25 shows:

$$\phi = tan^{-1}\left(\frac{y_{bolt}}{x_{bolt}}\right) \qquad 25$$

Next, the reference angle for the resultant force, F'', can be derived using trigonometry as equation 26 shows:

$$\theta = 90\, deg - \phi \qquad 26$$

These angles provide all the trigonometry needed to resolve the applied forces. Next, the uniformly applied shear force, F', from the vertical load on the joint, can be found with equation 27:

$$F' = \frac{P}{N} \tag{26}$$

To find the resultant force as a result of the moment is more complex. Equation 27 gives the magnitude of the shear force applied to each bolt as a reaction to the applied moment:

$$F'' = \frac{Mr_i}{\sum_{i=1}^{n} r_i^2} \tag{27}$$

From there the force can be broken into horizontal and vertical components, using equations 28 & 29:

$$F_x'' = F''cos(\theta) \tag{28}$$

$$F_y'' = F''sin(\theta) \tag{29}$$

Once the forces have been resolved into their components, the vertical components are summed with the uniformly applied shear force, F', from above, and then recombined into a final resultant magnitude using equation 30:

$$F = \sqrt{F_x''^2 + \left(F_y'' + F'\right)^2} \tag{30}$$

This process was coded in MATLAB and iterated over each bolt, taking care to account for the sign of each force component. The maximum shear stress was then found by dividing the peak shear force over the shank area:

$$\tau_{max} = 10.28kpsi$$

Before factor of safety can be calculated, the axial stress in the bolts must also be found in two capacities. Firstly, the peak axial stress in the fasteners must be checked to ensure safety. This is done at the threads. Secondly, the axial stress at the point of shear must be found. This will be lower than the peak axial stress as it occurs on the shank of the bolt, where the diameter is larger.

The process to find the axial loading of the fasteners follows the same path as the fasteners for the articulation point. The moment of inertia for the fastener plate is found using equation 21:

$$I = \frac{bh^3}{12} = 400in^2$$

Using equation 3, the bending stresses at the bolt locations are found:

$$\sigma_1 = 0.43kpsi$$

$$\sigma_2 = 1.3kpsi$$

For the upper bolts, this is a tensile force to separate. It follows that the peak applied stress in the bolts would therefore be 1.3 kpsi. Treating the joint as a non-permanent connection means the preload in the bolt is the same as before, 4.644 kips. The applied load can then be converted to a force by multiplying by the root area of the bolt:

$$F_{applied} = 120lbf$$

Summing this with the preload force calculated earlier yields a total axial force at the threads:

$$F = 4.76\ kips$$

Since this is lower than the maximum allowable found earlier (6.129 kips), the bolt is sufficient in this condition.

To check the bolt for failure at the point of shear, the axial stress at the shear point must be found. This is done using the shank diameter as that is where the shear occurs. Dividing the axial force by the shank area yields an axial stress of:

$$\sigma_a = \frac{4.76}{0.442in^2} = 10.8kpsi$$

With the shear and axial stress found, they can be combined using the *Von Mises Effective* stress, shown in equation 32:

$$\sigma' = \sqrt{\sigma_a^2 + 3\tau_{xy}^2} = 20.82kpsi$$

Finding the factor of safety is then done using the minimum yield strength provided by the manufacturer:

$$n = \frac{84kpsi}{20.82kpsi} = 4.03$$

Primary Pivot Support Tower Connection

Securing the ½" carbon steel hexagonal plate at the top of the tower to the primary pivot support was achieved using 6 x ¾" alloy steel shoulder screws. Axial stress (tensile) exerted on these bolts at full boom extension was calculated by taking a moment about an imaginary center plane perpendicular to the hexagonal plate's face. This setup assumes a moment of zero exerted on the bolts split by the plane. Figure 26 shows the two bolts farthest from the boom arm on the hexagonal plate are assumed to be in tension and the two bolts nearest the boom arm in compression. Using Equation 31, the moments experienced by the bolts in tension are considered.

$$\Sigma M = 0 = F * d \tag{31}$$

In this equation, force is represented by *F* and the distance on which the force acts from the moment plane is represents by *d*. Within the scope of the two bolts in tension, the full boom arm exerts a moment of 190,000in-lb at full extension when considered from the half-hexagon moment plane shown in Figure 26. A moment relation may also be drawn using the distance from the bolts in tension to the moment plane (6.3 inches). The force, *P*, exerted on each bolt (accounted for by dividing the total boom moment by 2 and multiplying the bolt to plane moment by 2) may be derived accordingly.

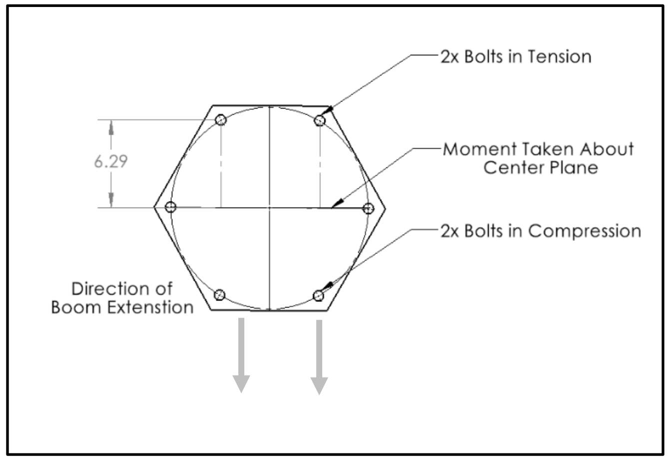


Figure 26: Tower upper bolt pattern drawing

$$\Sigma M = 0 = \left(\frac{190000in - lb}{2}\right) - \left(2 * (P * 6.3in)\right)$$

Rearranging, the force *P* exerted on each bolt in tension may be calculated.

$$P = \frac{190000in - lb}{4 * 6.3in} = 7539.68\ lb$$

Using a variation of Equation 14 which includes a shoulder stress concentration value *k = 1.5*, the maximum stress exerted on each bolt is calculated. The minimum bolt cross-sectional area, *A*, is .198 in^2

$$\sigma_{axial} = \frac{P}{A} * k$$

$$\sigma_{axial} = \frac{7539.68\ lb}{.198in^2} * 1.5 = 57118.79\ \frac{lb}{in^2}$$

This particular bolt has a tensile strength of 140ksi. Using this value and the maximum axial stress calculated above, the factor of safety is calculated using a modified version of Equation 4.

$$n = \frac{S_{tensile}}{\sigma_{axial}}$$

$$n = \frac{140000psi}{57118.79} \frac{lb}{in^2} = 2.45$$

Tower Base Connection

The tower base connection is characterized by the use of 12 x ¾" alloy steel shoulder screws. Analysis of this connection demanded using a similar process to that of the primary pivot support tower connection. Using this approach, the axial stress experienced by the 4 bolts in tension (Figure 27) is calculated using the same formulas from above. A distance of 8.66 inches is measured between each of the 4 bolts in tension and the moment plane. The force on each bolt individually is accounted for by dividing the total boom moment by 4 and multiplying the bolt to plane moment by 4.

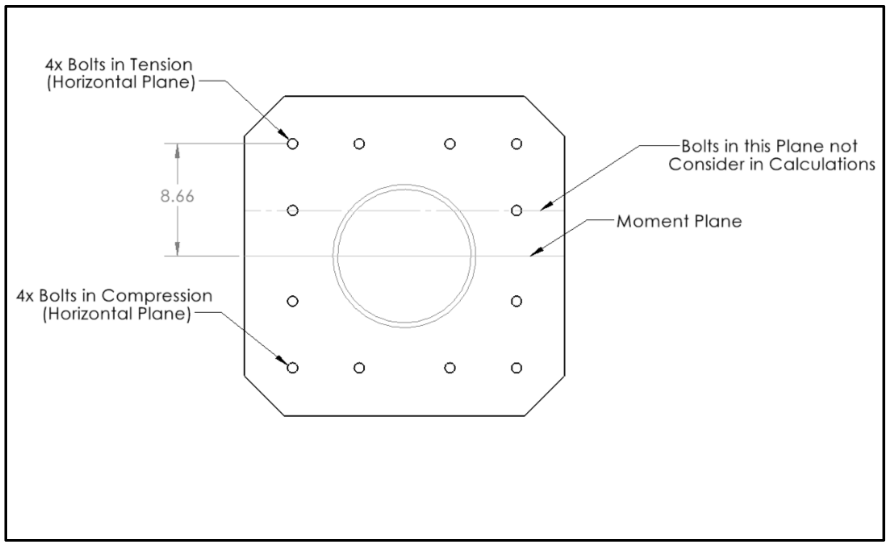


Figure 27: Tower base bolt diagram

$$\Sigma M = 0 = \left(\frac{190000in - lb}{4}\right) - \left(4 * \left(P * (8.66in)\right)\right)$$

Rearranging, the force *P* exerted on each bolt in tension is calculated.

$$P = \frac{190000in - lb}{16 * 8.33in} = 1425.75\ lb$$

Including a shoulder stress concentration value *k* = *1.5*, the maximum stress exerted on each bolt is calculated. The minimum bolt cross-sectional area, A, is .198 in^2

$$\sigma_{axial} = \frac{1425.75\ lb}{.198 in^2} * 1.5 = 7200.76\ \frac{lb}{in^2}$$

Using the maximum axial stress calculated above and the tensile strength of the bolt, the factor of safety is calculated.

$$n = \frac{140000 psi}{7200.76}\frac{lb}{in^2} = 19.44$$

In an effort to streamline these calculations, the two bolts falling on the horizontal plane between the 4 in-line bolts at the top of Figure 27 are not considered. It is understood that accounting for these bolts in more complicated calculations will only increase the factor of safety.

Results & Conclusions

Reviewing this report in totality reveals some holes in the analysis of this design. This report is non-inclusive and does not analyze all machine componentry, fixtures, and connections. One obvious example of this may is recognized in the absence of weld analysis for the fins connected to the tower base and upper tower hexagon. This report is intended to be a preliminary design, and further design review would be required to move ahead with the project.

Accounting for all machine componentry, the total weight of this design comes to 1,350.38 lb. In analyzing stresses present in this assembly, a design factor of safety of 5 was initially set for all individual part calculations. This value was chosen for is prevalence in permanent engineering structures and the large safety buffer it provides in the event of machine overload.

For a bending moment of 6,200 ft-lb, the secondary boom will experience a maximum bending stress of 3.9 kpsi. This represents a loading condition of 1000 lb load. Using a conservative yield strength value of 30 kpsi (low-carbon steel) will result in a factor of safety over 7. This value was deemed satisfactory under the preestablished factor of safety parameters.

A peak bending stress of 5.2 kpsi is calculated from the analysis of the primary boom. With a yield strength of 30 kpsi, a 5.77 factor of safety is calculated. Using

Maximum Shear Stress Theory, a .14 kpsi shear force is calculated. A yield strength in shear value of 15 kpsi results in a factor of safety over 100.

Considering the mass of the primary and secondary jib arms and the 1000 lb. applied load at the end of the secondary arm yields a total moment of 190,000 in-lb. on the tower structure. Accounting for additional stresses resulting from vertical loading on the tower (+1,096 lb. from the pivot and beam masses), a von Mises stress 10,400 $\frac{lbf}{in^2}$ is realized. From a 35,000 kpsi yield strength for Grade B Schedule 40 steel pipe (ASTM A106), a 2.55 factor of safety is calculated.

The articulating pivot (central pivot) mechanism may be broken into two distinct components, the pin, and the housing. The pivot's pin experiences an axial stress of 0.21 kpsi and a bending stress of 81 kpsi. In shear, this results in a factor of safety of 31 and a bending factor of safety of 2.9. The housing comprises two components in and of itself, the upper and lower housing. For the lower housing, SolidWorks loading simulations showed a von Mises stress of 11.33 kpsi, resulting in a factor of safety of 2.82. For the upper housing, a von Mises stress of 11.2 kpsi is determined, resulting in a 2.85 factor of safety.

Calculations for the primary pivot point required insight from articulating pivot and secondary beam calculations. Following a similar analysis process to that of the articulating pivot, the primary pivot pin and housing were analyzed separately. The primary pivot pin experiences a bending stress of 30 kpsi and a shear stress of 1.9 kpsi. This results in factors of safety values of 7.93 and 62.63 from bending and shear stress, respectively. Unlike the articulating pivot, there exists only one housing element in the primary pivot. The housing was analyzed under full boom extension conditions and separately, when the lower crane arm is perpendicular to the primary arm (in bending and torsion with a vertical load). Under the first conditions, SolidWorks loading simulations showed a von Mises stress of 17.1 kpsi, resulting in a factor of safety of 4.92. Under the second conditions, a von Mises stress of 15.89 kpsi is determined, resulting in a 5.29 factor of safety.

The diameter required of the locking mechanism boom support factor of safety was calculated using a Modified Goodman approach. Accounting for stress concentration factors, an endurance limit of 15.4 kpsi is calculated. Using a desired factor of safety of 5, a maximum bending stress of 3.08 kpsi is calculated. Understanding this component will experience a force of 1,100 lb. from the weight of the two booms, a required rod area of 0.357 in^2 is calculated. From this, a required rod diameter of 0.674 in is calculated. A 0.75 in is chosen. Importantly, a required moment of inertia of 0.00615 in^4 is calculated to prevent buckling of this boom support using a rod length of 40.7 in and a force of 1,000 lb.

Under proper loading conditions, all safety consideration of this design prove adequate. Outside of these calculations, there does exist a concern as to how a 'swinging' load will affect the integrity of the primary and secondary articulating arms and their accompanying pivot points. A swinging load, resulting from improper rigging of the load or from impulse transit, may exert dangerous torsional forces to the structure. It is therefore critical that precautions are taken by the jib operator to ensure all loads are secured in a fashion which prevents this motion.

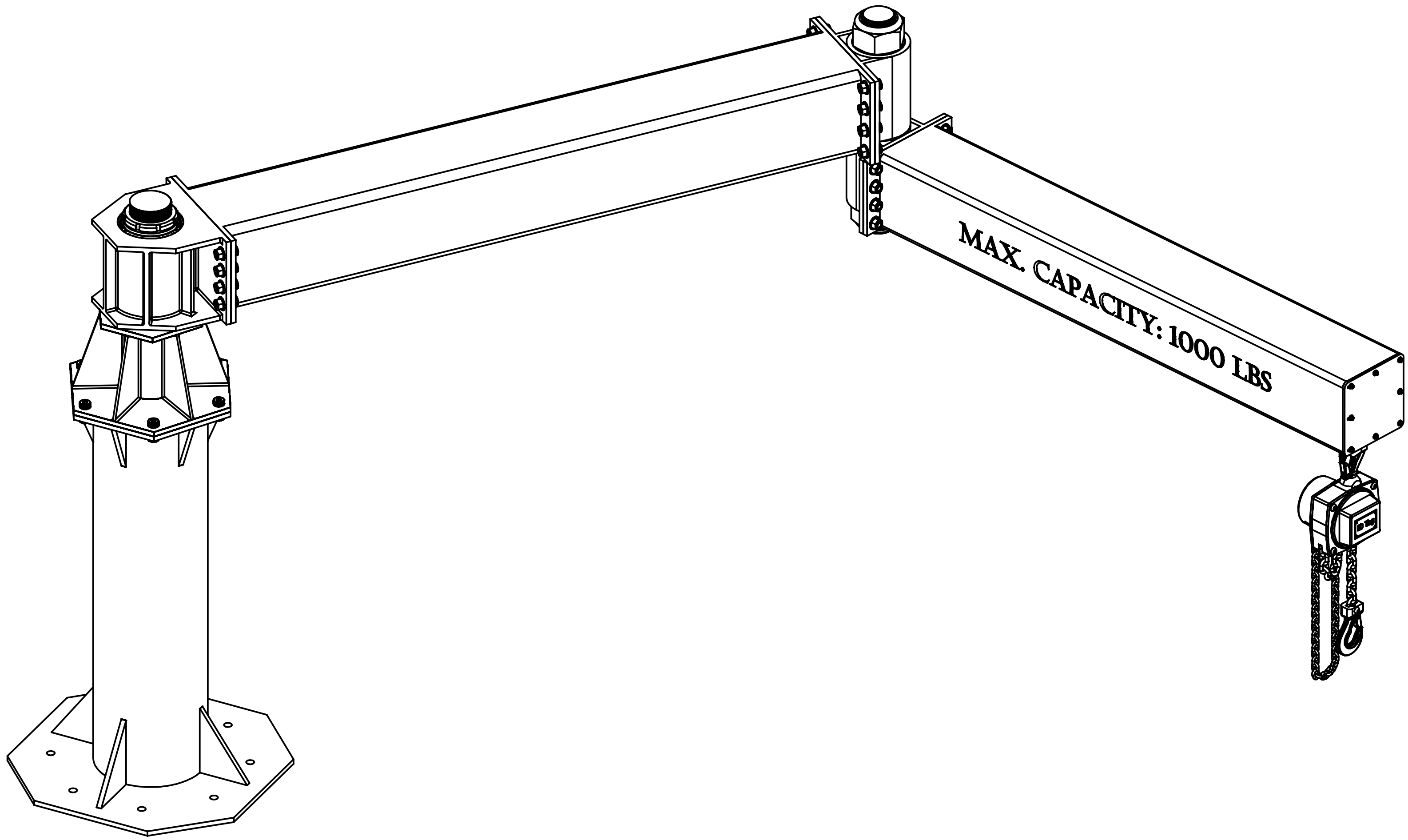
Some of the major suspected limitations of this design may not be recognized in the factor of safety calculations provided. A potential concern of this design is its ease of operation. Under 1000lb maximum loading, it is unknown how difficult it will be for the jib operator to articulate the crane's arms in effort to move the load off the trailer bed. A significant factor dictating the efficiency in which the jib articulates is the quality of the pivot bearings. In practice, a high-quality bearing must be selected to provide low-friction articulation.

In reviewing the calculations made in this analysis, it becomes evident that the desired factor of safety of 5 was not achieved in every calculation. This point should be taken with a grain of salt. As the design process progressed, priority was taken to increase the loading capacity of the articulating jib from 500lb. to 1,000lb. Consequentially, increasing the loading capacity of the machine decreased the overall factor of safety of its components. It was determined that a 1,000lb. load at an average factor of safety of 3 was preferred to that of a 500lb load closer to 5. The final minimum factor of safety for the design comes to 2.45

This design prioritizes simplicity by incorporating only two pivot points and no electronic componentry. Further, this articulating jib design offers significant onloading and offloading flexibility. With a maximum reach of 11 ft, over 180° of freedom at full extension, and 360° of motion at partial extension, this design enables delivery pickup and drop-off in virtually any location accessible by the jib's carrier vehicle. The impressive reach of this design allows for up to four 5' x 3' x 3' 500 lb. crates to be offloaded from the trailer bed without the use of a pallet jack or load rearrangement.

2 | 1

B | B

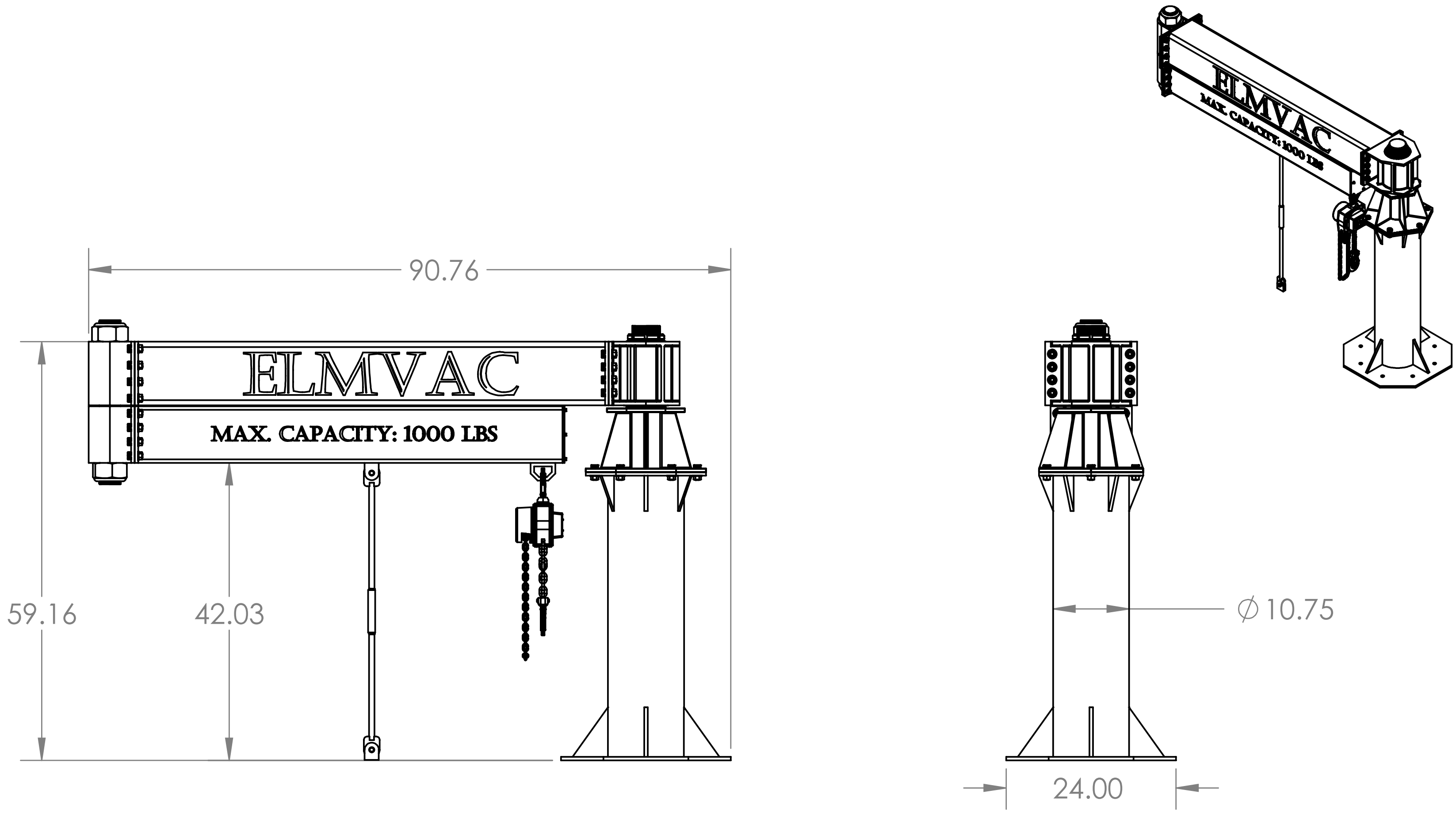


A | A

Dpt. of Research & Development ELMVAC Inc. 101 Thornton Dr., Suite 3 Burlington VT, 05401	**Unless Otherwise Specified** Dimensions in Inches Tolerances Two Places: +/- 0.01 Three Places: +/- 0.003 Angular: +/- 1 Deg	Drawn By: Ian Ammerman	Title: Crane - Isometric	
		Date: 4/21/2022	File Name: FullCrane_ISO	
		Material:	Size: **A**	Scale: 1:16	Rev.

2 | 1

B B



A A

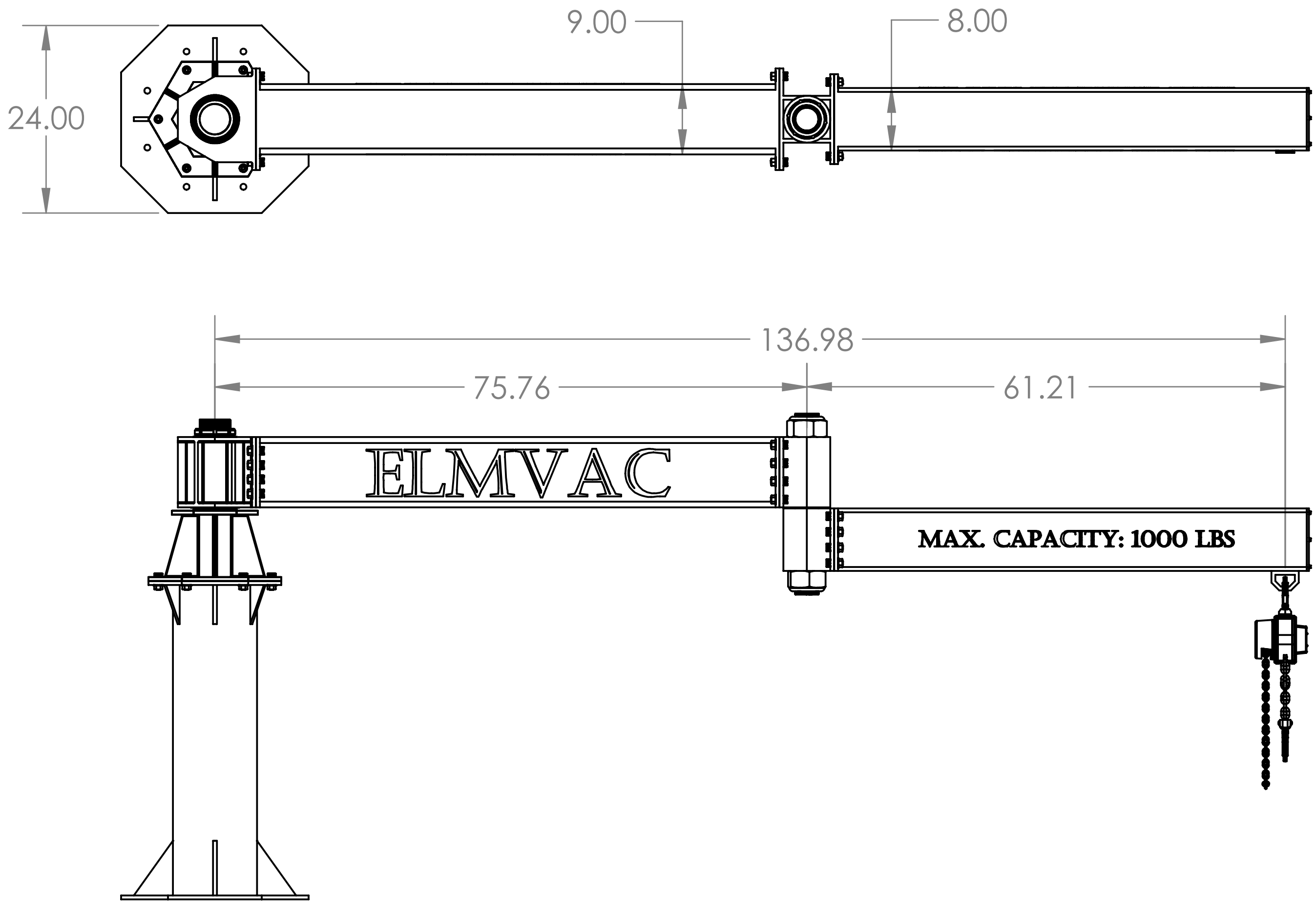
Dpt. of Research & Development
ELMVAC Inc.
101 Thornton Dr., Suite 3
Burlington VT, 05401

Unless Otherwise Specified
Dimensions in Inches

Tolerances

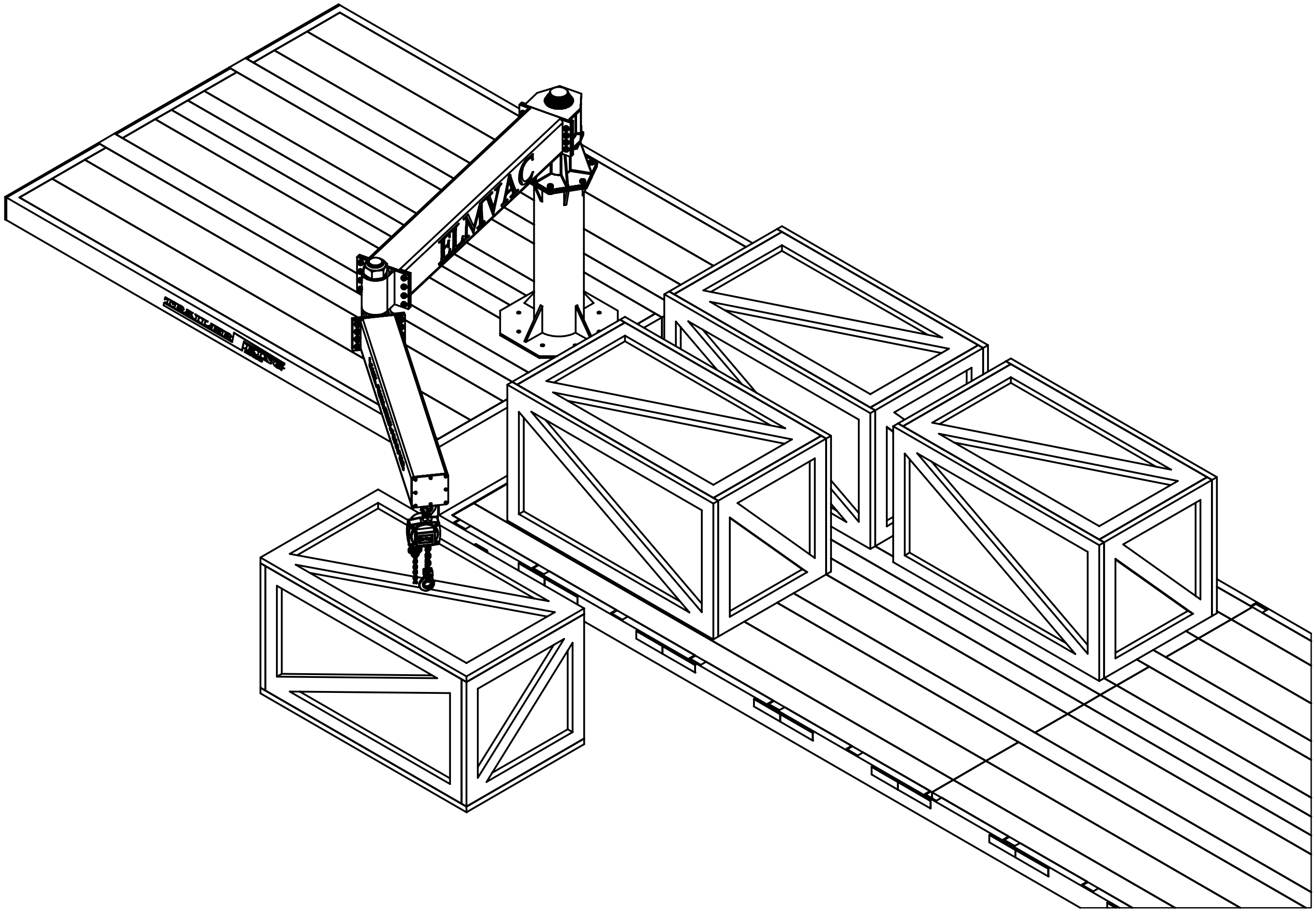
Two Places: +/- 0.01
Three Places: +/- 0.003
Angular: +/- 1 Deg

Drawn By: Ian Ammerman	Title: Transport Position		
Date: 4/21/2022	File Name: FullCrane_Transport		
Material:	Size: **A**	Scale: 1:24	Rev.



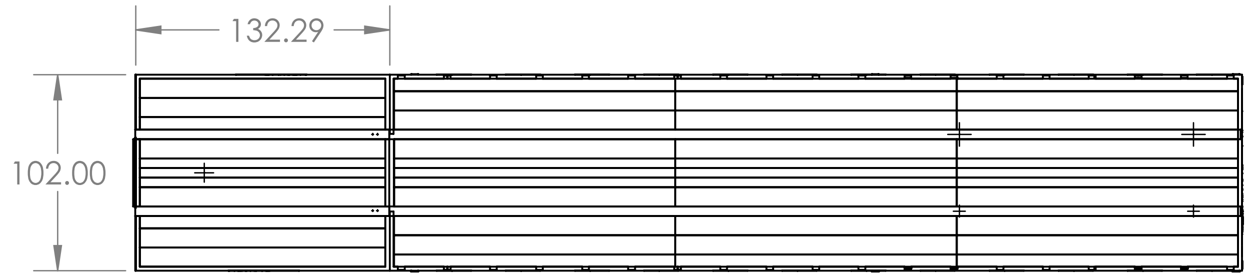
Dpt. of Research & Development ELMVAC Inc. 101 Thornton Dr., Suite 3 Burlington VT, 05401	**Unless Otherwise Specified** Dimensions in Inches Tolerances Two Places: +/- 0.01 Three Places: +/- 0.003 Angular: +/- 1 Deg	Drawn By: Ian Ammerman	Title: Loading Position	
		Date: 4/21/2022	File Name: CraneAssm_Loading	
		Material:	Size: A	Scale: 1:24 / Rev.

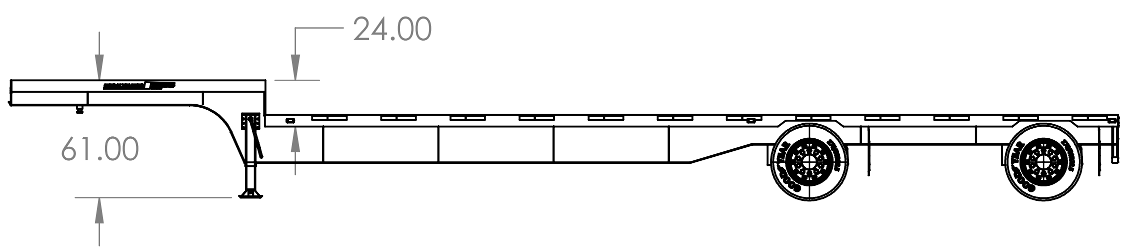
B B



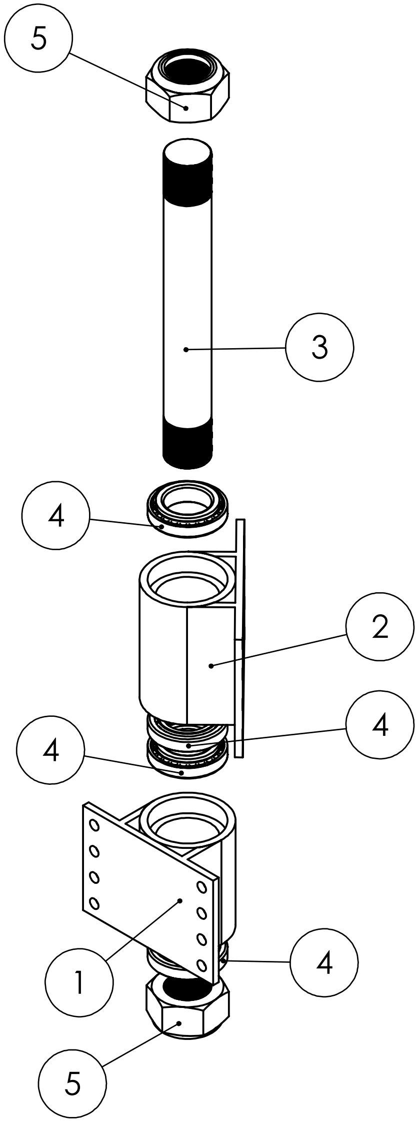
A A

Dpt. of Research & Development ELMVAC Inc. 101 Thornton Dr., Suite 3 Burlington VT, 05401	**Unless Otherwise Specified** Dimensions in Inches <u>Tolerances</u> Two Places: +/- 0.01 Three Places: +/- 0.003 Angular: +/- 1 Deg	Drawn By: Ian Ammerman	Title: Unloading	
		Date: 4/21/2022	File Name: Full_ISO_Unloading	
		Material:	Size: **A**	Scale: 1:40 / Rev.

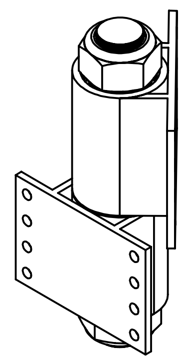




Dpt. of Research & Development ELMVAC Inc. 101 Thornton Dr., Suite 3 Burlington VT, 05401	**Unless Otherwise Specified** Dimensions in Inches Tolerances Two Places: +/- 0.01 Three Places: +/- 0.003 Angular: +/- 1 Deg	Drawn By: Ian Ammerman	Title: Dropdeck Trailer
		Date: 4/21/2022	File Name: Dropdeck
		Material:	Size: **A** Scale: 1:100 Rev.



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	CenterPivotCollar_wFlange		1
2	CenterPivotCollar_wFlange_Upper		1
3	5947K34_Rotary Shaft		1
4	5709K46	Tapered-Roller Bearing with Steel Ring	4
5	93126A210	Extra-Wide Thin Nylon-Insert Locknut	2



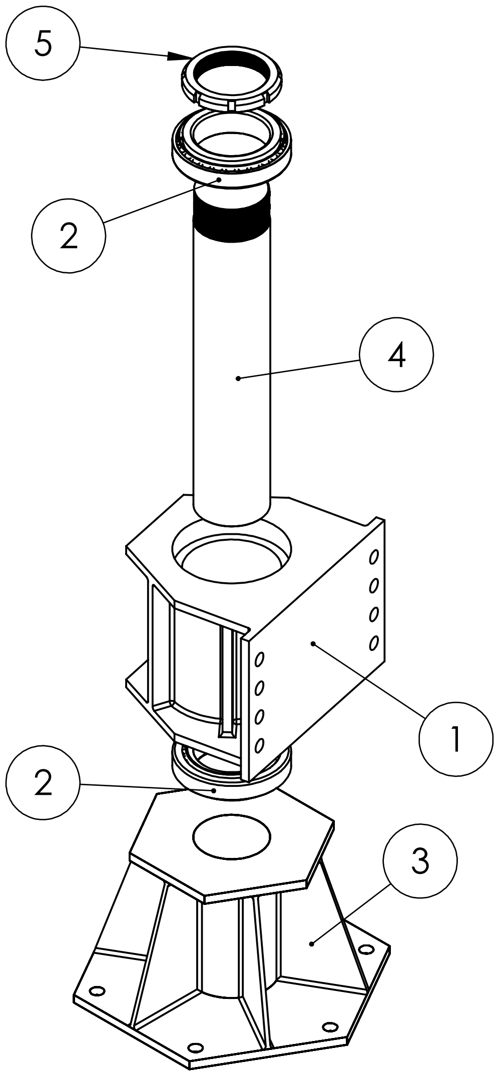
Dpt. of Research & Development
ELMVAC Inc.
101 Thornton Dr., Suite 3
Burlington VT, 05401

Unless Otherwise Specified
Dimensions in Inches

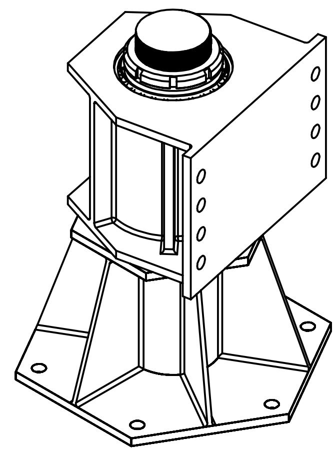
Tolerances

Two Places: +/- 0.01
Three Places: +/- 0.003
Angular: +/- 1 Deg

Drawn By: Ian Ammerman	Title: Articulation Sub-Assembly	
Date: 4/21/2022	File Name: ArtSub_Ex_ISO	
Material:	Size: **A**	Scale: 1:12	Rev.



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	PrimaryPivotHousing		1
2	5709K908	Tapered-Roller Bearing with Steel Ring	2
3	PrimaryPivotSupport		1
4	PrimaryPivotShaft		1
5	6343K31	Bearing Locknut	1



Dpt. of Research & Development
ELMVAC Inc.
101 Thornton Dr., Suite 3
Burlington VT, 05401

Unless Otherwise Specified
Dimensions in Inches

Tolerances

Two Places: +/- 0.01
Three Places: +/- 0.003
Angular: +/- 1 Deg

Drawn By: Ian Ammerman

Title: Primary Pivot Assembly

Date: 4/21/2022

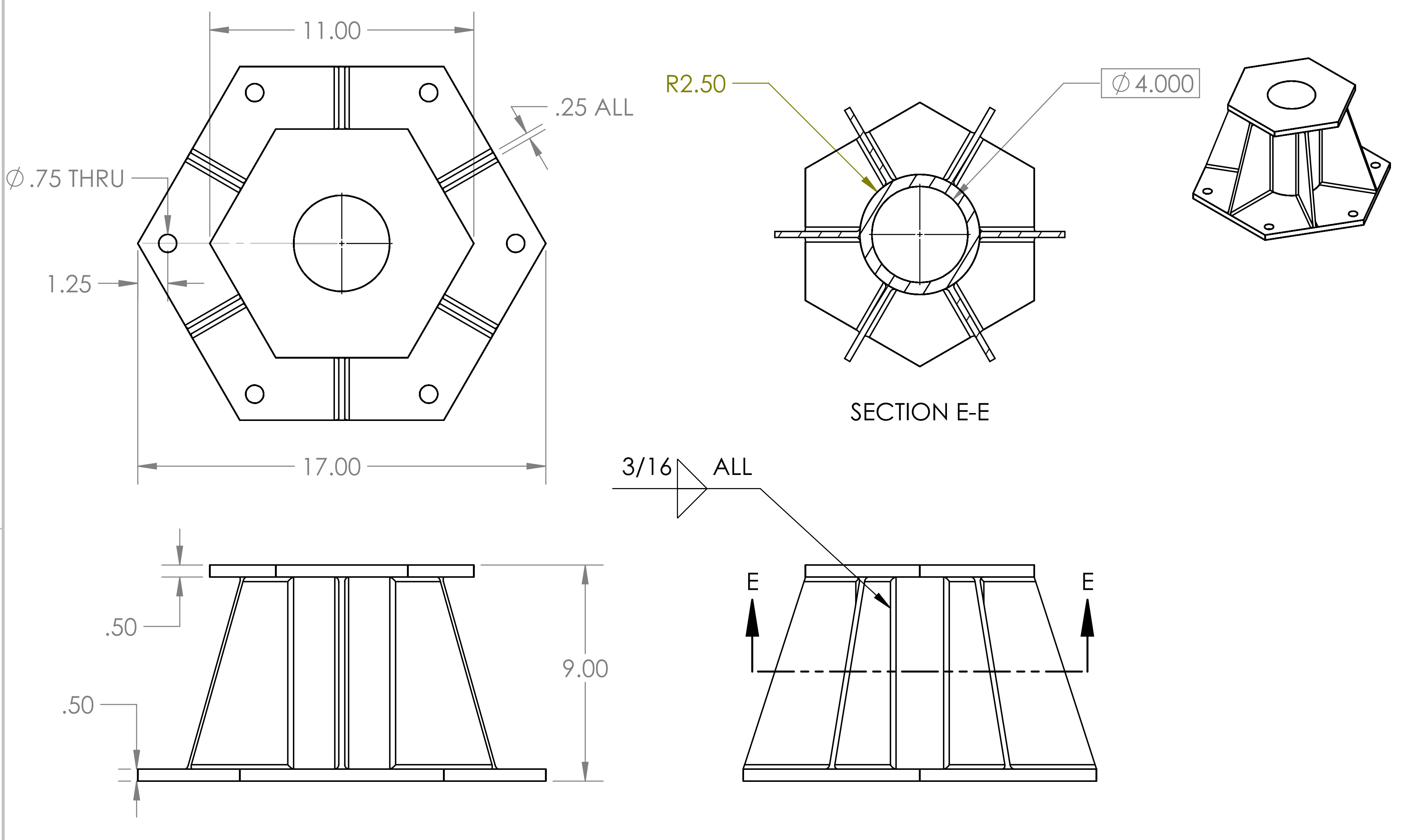
File Name: PPAssm_Exp_Iso

Material:

Size: **A**

Scale: 1:10

Rev.



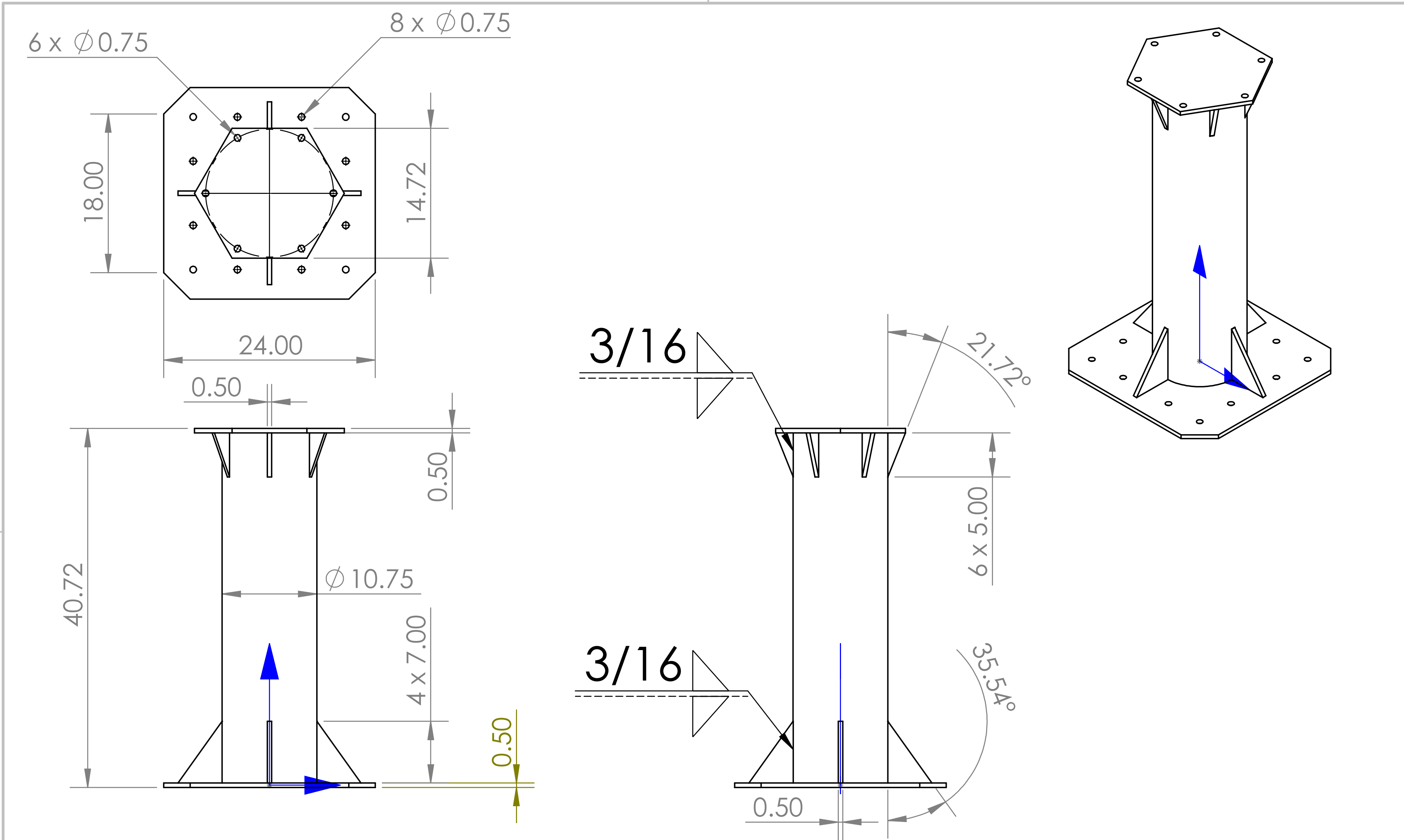
Dpt. of Research & Development
ELMVAC Inc.
101 Thornton Dr., Suite 3
Burlington VT, 05401

Unless Otherwise Specified
Dimensions in Inches

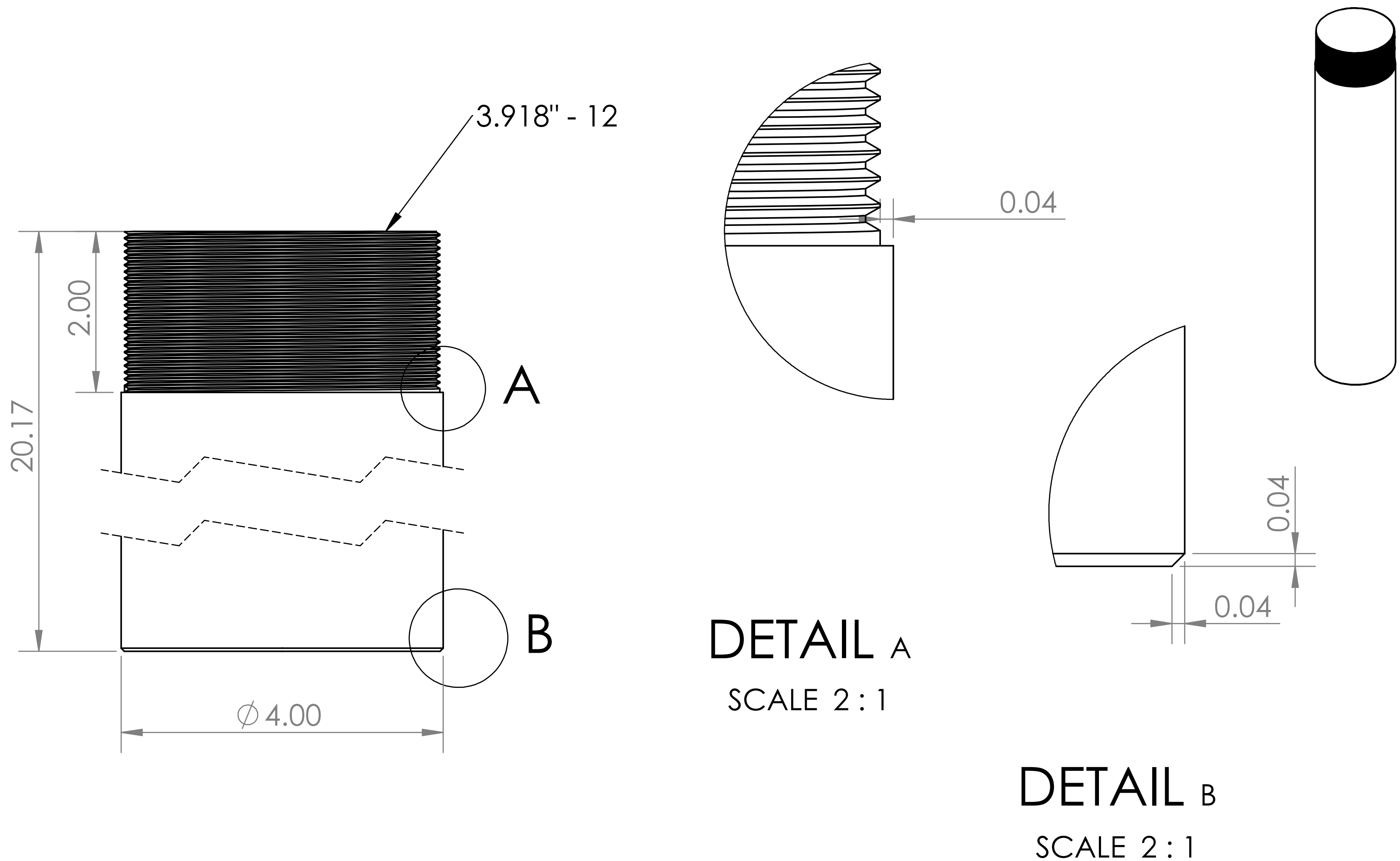
Tolerances

Two Places: +/- 0.01
Three Places: +/- 0.003
Angular: +/- 1 Deg

Drawn By: Ian Ammerman	Title: Primary Pivot Support
Date: 4/21/2022	File Name: PrimaryPivotSupport
Material: Plain Carbon Steel	Size: A / Scale: 1:6 / Rev. 1



Dpt. of Research & Development ELMVAC Inc. 101 Thornton Dr., Suite 3 Burlington VT, 05401	**Unless Otherwise Specified** Dimensions in Inches <u>Tolerances</u> Two Places: +/- 0.01 Three Places: +/- 0.003 Angular: +/- 1 Deg	Drawn By: Christopher Erb	Title: Tower		
		Date: 4/22/2022	File Name:		
		Material: ASTM A106	Size: **A**	Scale: 1:24	Rev.



Dpt. of Research & Development
ELMVAC Inc.
101 Thornton Dr., Suite 3
Burlington VT, 05401

Unless Otherwise Specified
Dimensions in Inches

Tolerances

Two Places: +/- 0.01
Three Places: +/- 0.003
Angular: +/- 1 Deg

Drawn By: Ian Ammerman

Title: Primary Pivot Member

Date: 4/21/2022

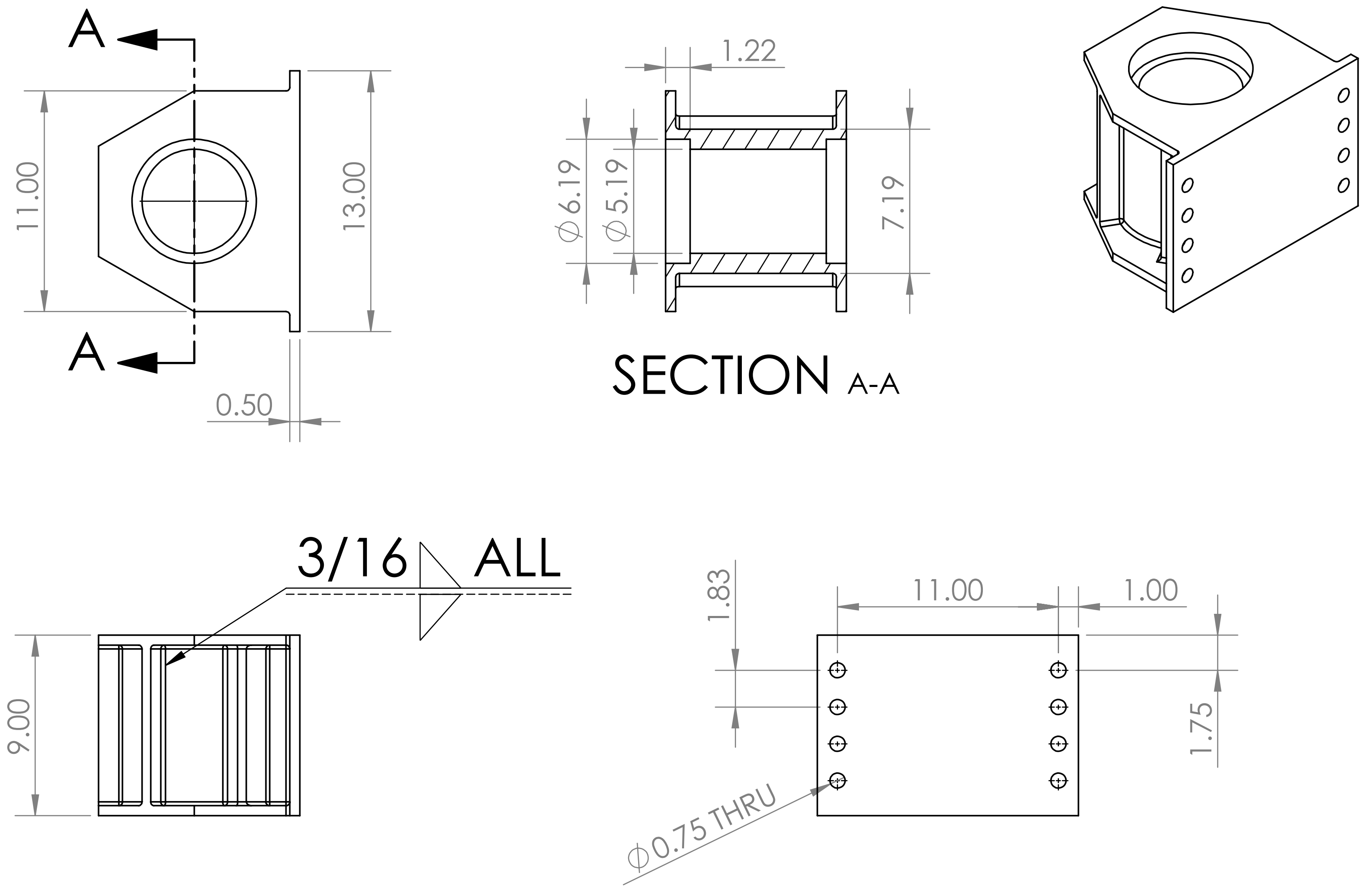
File Name: PrimaryPivotShaft

Material: Plain Carbon Steel

Size: **A**

Scale: 1:2

Rev. 2



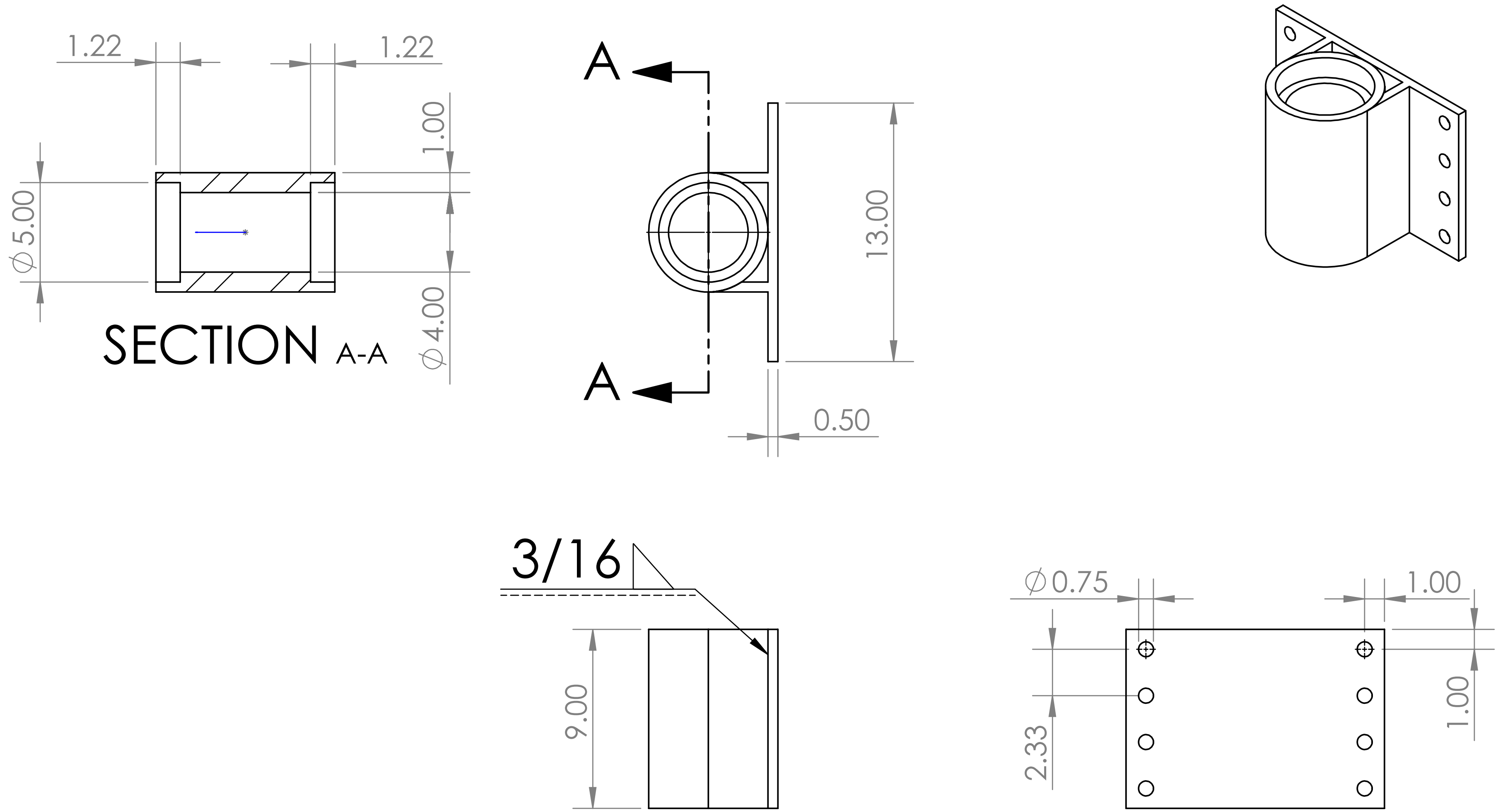
Dpt. of Research & Development
ELMVAC Inc.
101 Thornton Dr., Suite 3
Burlington VT, 05401

Unless Otherwise Specified
Dimensions in Inches

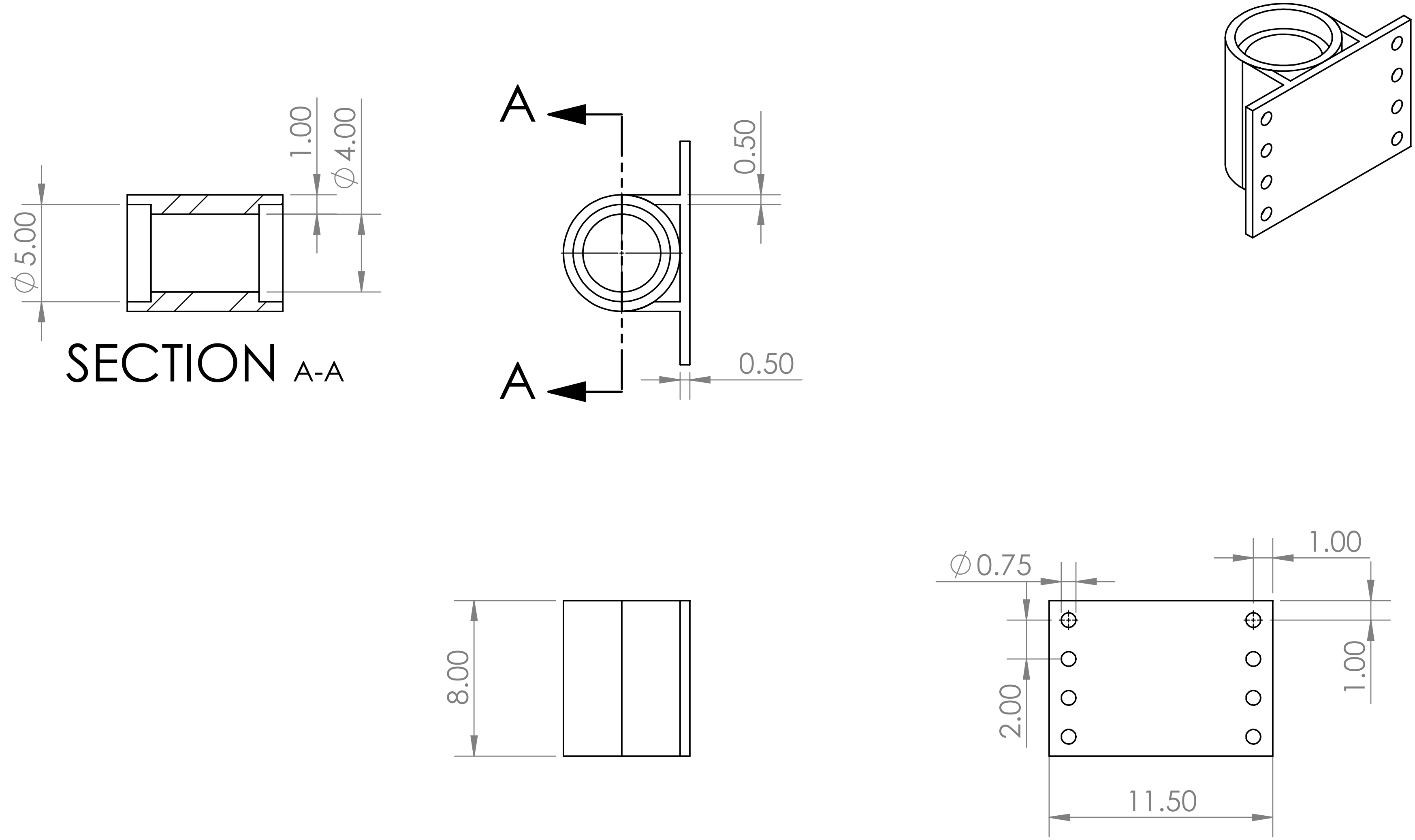
Tolerances

Two Places: +/- 0.01
Three Places: +/- 0.003
Angular: +/- 1 Deg

Drawn By: Ian Ammerman	Title: Primary Pivot Housing
Date: 4/21/2022	File Name: PrimaryPivotHousing
Material: AISI 1050 CD	Size: A / Scale: 1:8 / Rev.



Dpt. of Research & Development ELMVAC Inc. 101 Thornton Dr., Suite 3 Burlington VT, 05401	**Unless Otherwise Specified** Dimensions in Inches <u>Tolerances</u> Two Places: +/- 0.01 Three Places: +/- 0.003 Angular: +/- 1 Deg	Drawn By: Ian Ammerman	Title: Center Pivot Housing (Upper)	
		Date: 4/21/2022	File Name:	
		Material: Plain Carbon Steel	Size: **A**	Scale: 1:8	Rev.



Dpt. of Research & Development
ELMVAC Inc.
101 Thornton Dr., Suite 3
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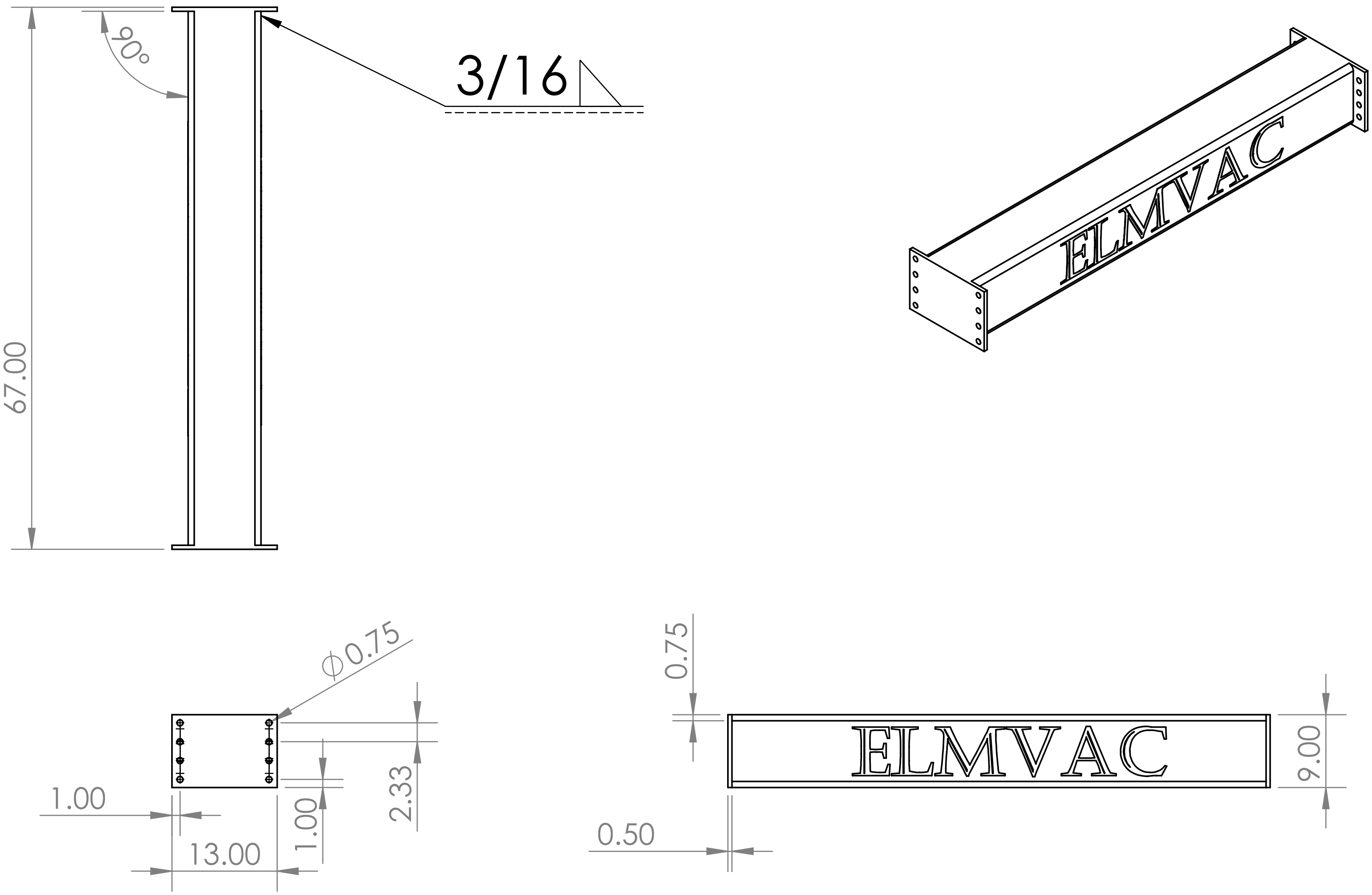
Unless Otherwise Specified
Dimensions in Inches

Tolerances
Two Places: +/- 0.01
Three Places: +/- 0.003
Angular: +/- 1 Deg

Drawn By: Ian Ammerman	Title: Center Pivot Housing (lower)
Date: 4/21/2022	File Name: CenterPivotCollar_wFlange
Material: Plain Carbon Steel	Size: **A** · Scale: 1:8 · Rev. 3



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ELMVAC Inc.
101 Thornton Dr., Suite 3
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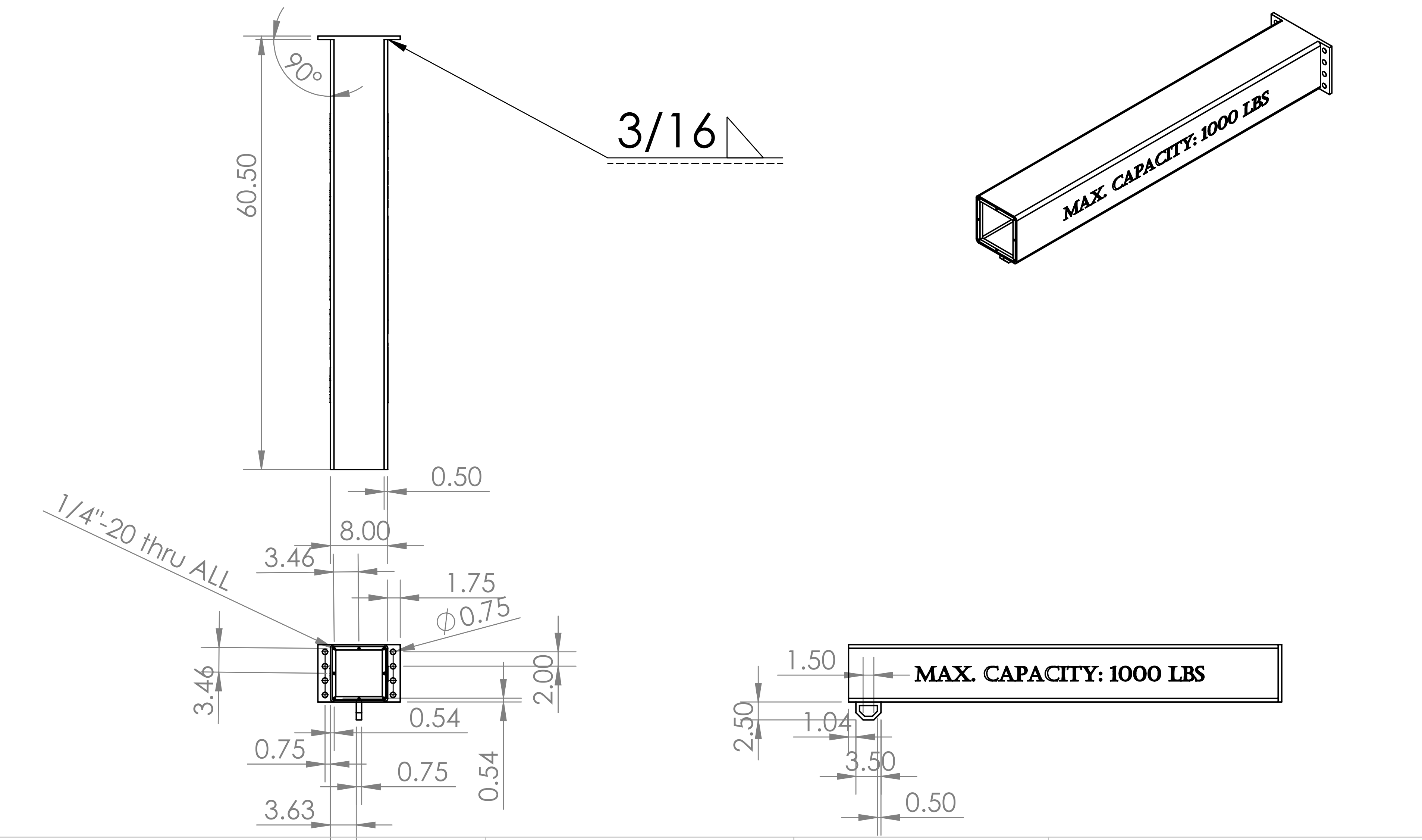
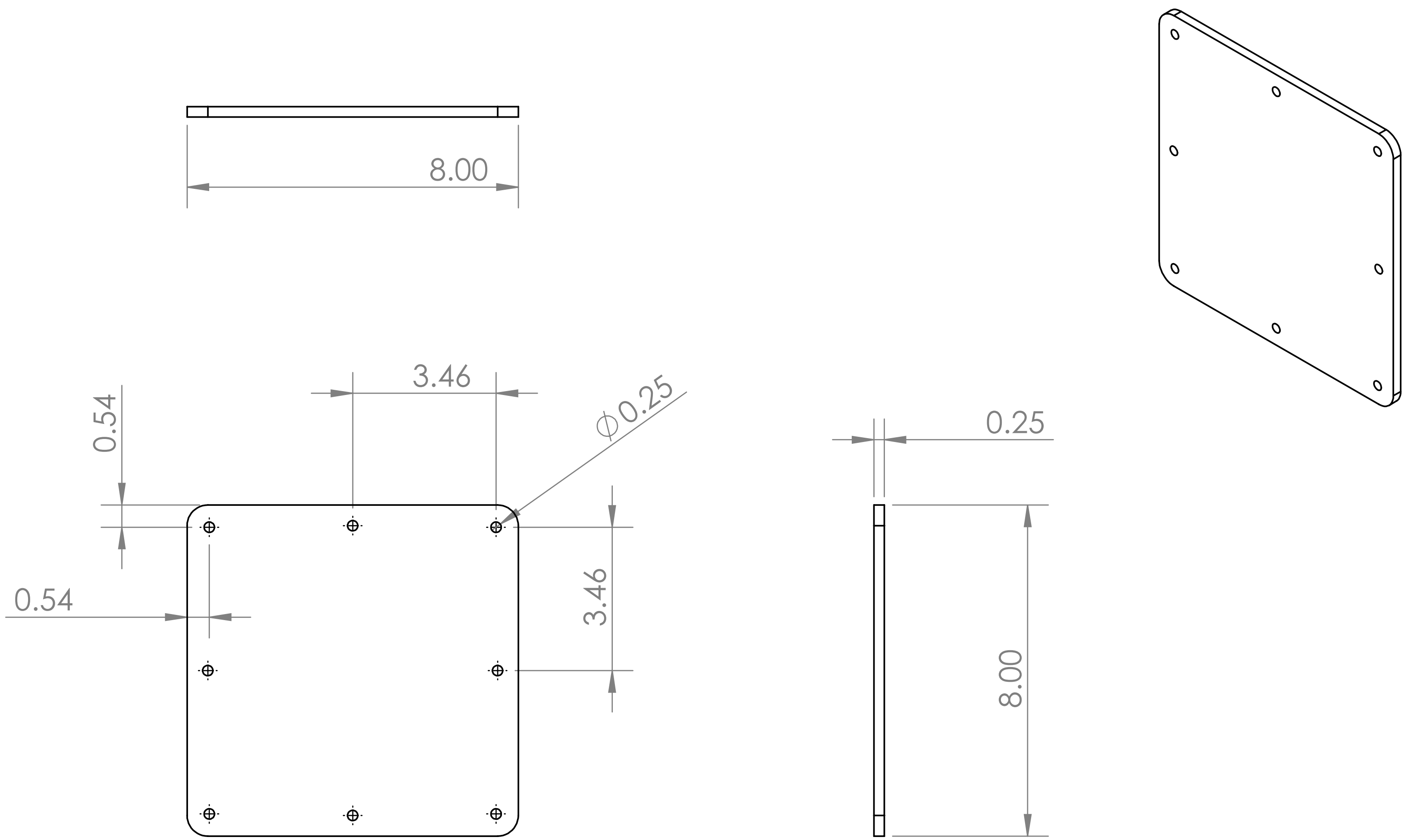
Unless Otherwise Specified
Dimensions in Inches

Tolerances
Two Places: +/- 0.01
Three Places: +/- 0.003
Angular: +/- 1 Deg

Drawn By: Lautaro Visuara	Title: Primary Boom
Date: 4/21/2022	File Name: PrimaryBoom
Material:	Size: **A**	Scale: 1:20	Rev.



Dpt. of Research & Development ELMVAC Inc. 101 Thornton Dr., Suite 3 Burlington VT, 05401	**Unless Otherwise Specified** Dimensions in Inches Tolerances Two Places: +/- 0.01 Three Places: +/- 0.003 Angular: +/- 1 Deg	Drawn By: Lautaro Visuara	Title: Secondary Boom	
		Date: 4/21/2022	File Name: SecondaryBoom	
		Material:	Size: **A**	Scale: 1:20 / Rev.



Dpt. of Research & Development
ELMVAC Inc.
101 Thornton Dr., Suite 3
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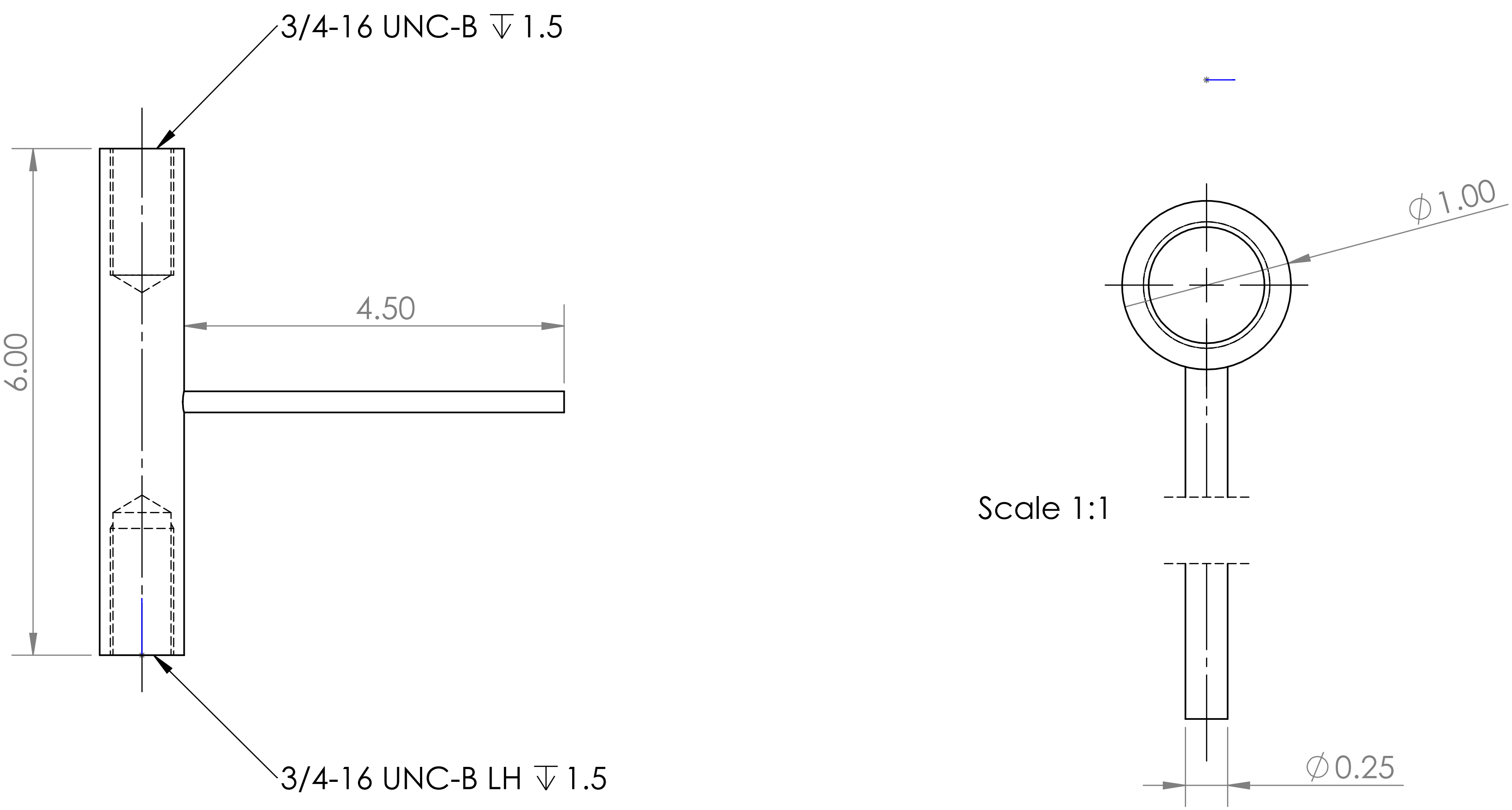
Unless Otherwise Specified
Dimensions in Inches

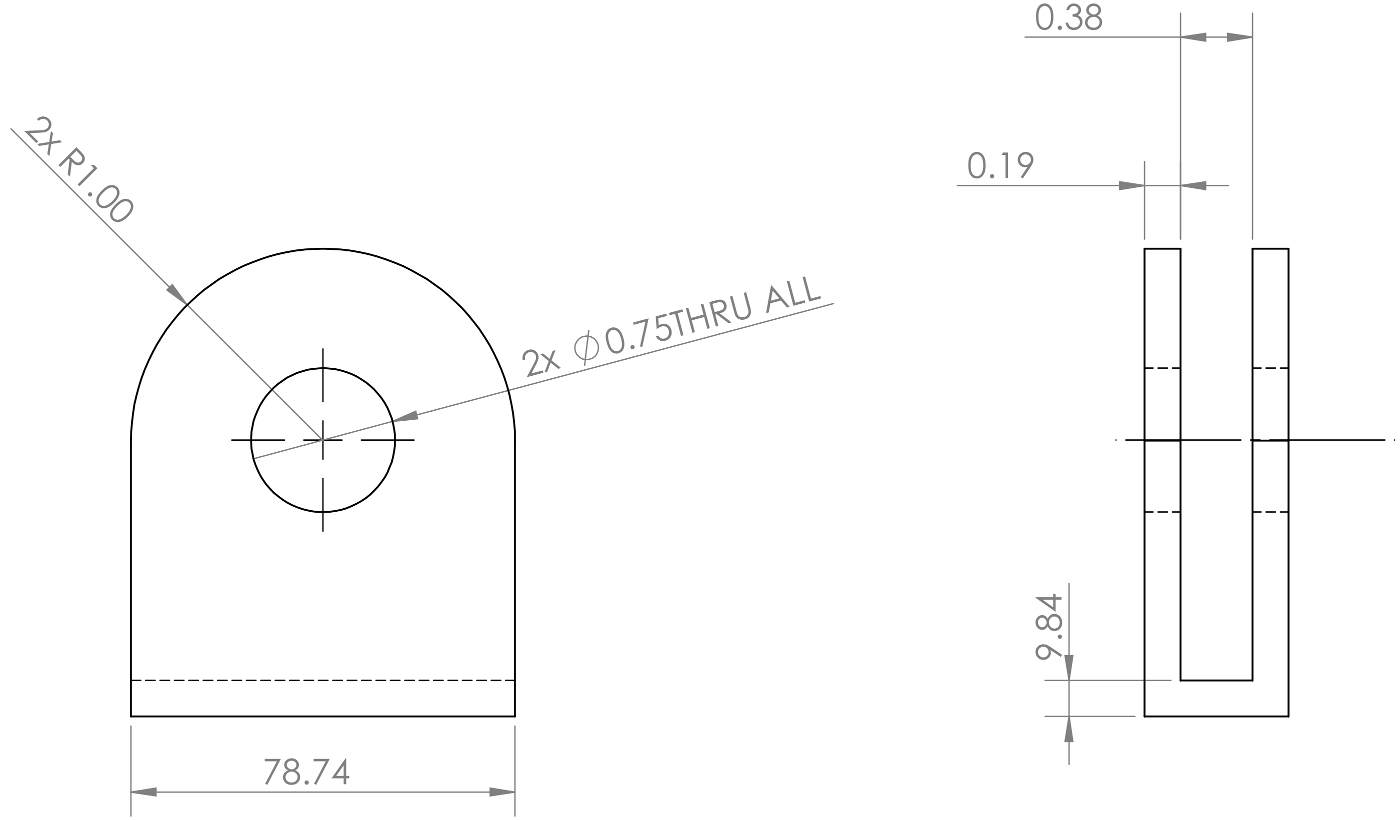
Tolerances

Two Places: +/- 0.01
Three Places: +/- 0.003
Angular: +/- 1 Deg

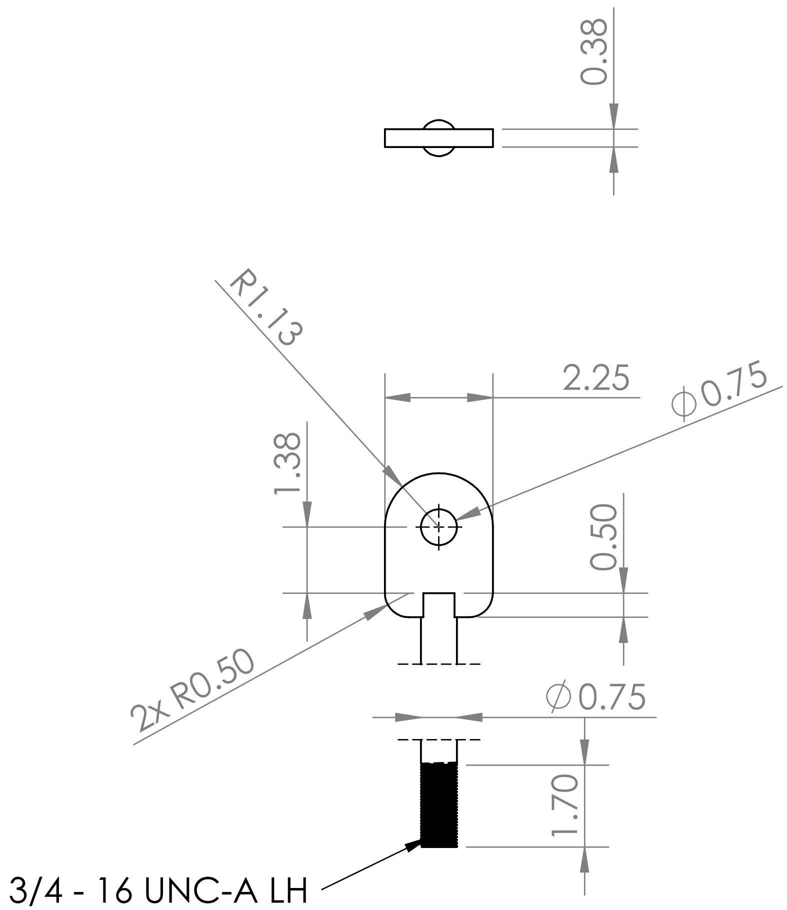
Drawn By: Lautaro Visuara	Title: Endcap
Date: 4/21/2022	File Name: SecondaryBoomEndCap
Material:	Size: **A** / Scale: 1:4 / Rev.

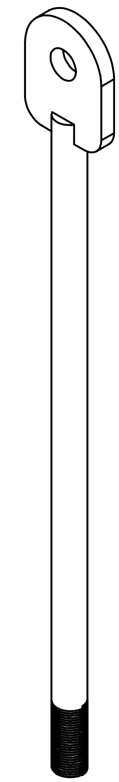


Dpt. of Research & Development ELMVAC Inc. 101 Thornton Dr., Suite 3 Burlington VT, 05401	**Unless Otherwise Specified** Dimensions in Inches Tolerances Two Places: +/- 0.01 Three Places: +/- 0.003 Angular: +/- 1 Deg	Drawn By: W Labbe	Title:		
		Date: 4/22/2022	File Name: Turnbuckle		
		Material: AISI 1020 HR	Size: **A**	Scale: 1:2	Rev.

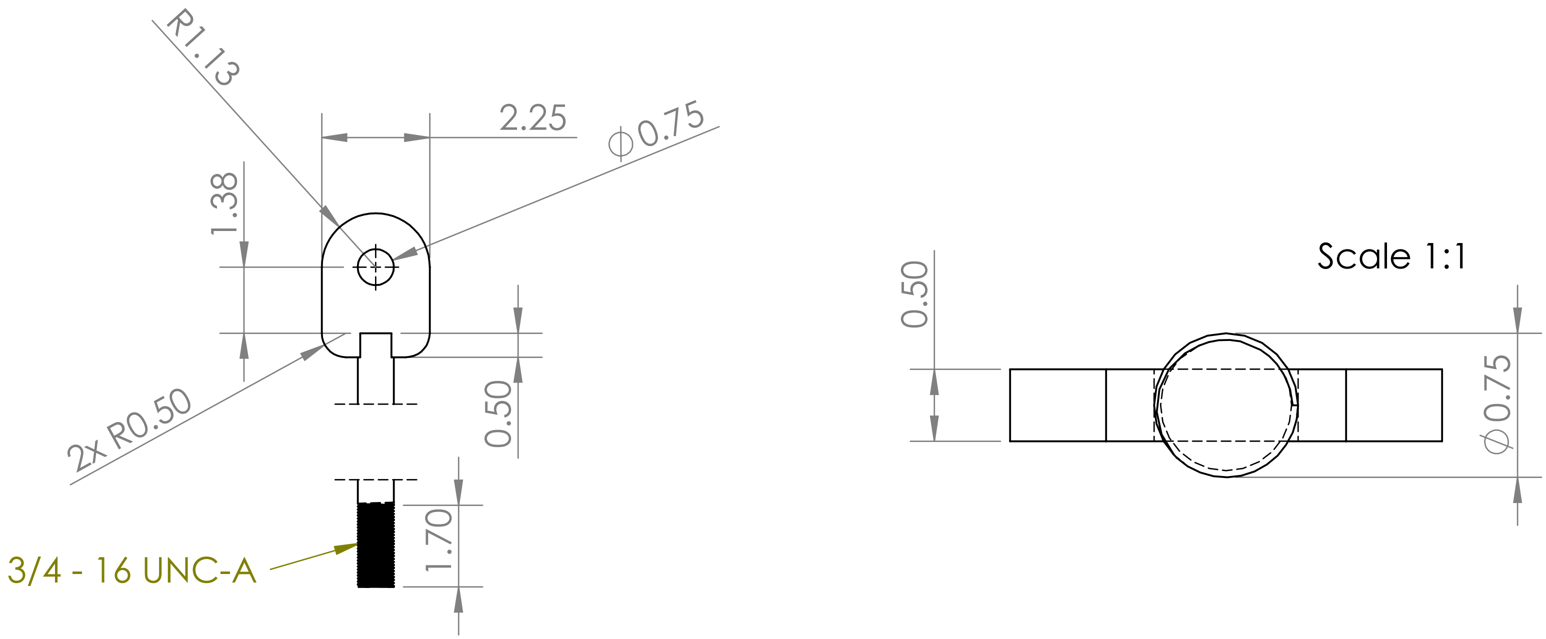


Dpt. of Research & Development ELMVAC Inc. 101 Thornton Dr., Suite 3 Burlington VT, 05401	**Unless Otherwise Specified** Dimensions in Inches <u>Tolerances</u> Two Places: +/- 0.01 Three Places: +/- 0.003 Angular: +/- 1 Deg	Drawn By: W Labbe	Title: Boom Bracket		
		Date: 4/22/2022	File Name: Bracket - Boom		
		Material: AISI 1020 HR	Size: **A**	Scale: 1:1	Rev.





Dpt. of Research & Development ELMVAC Inc. 101 Thornton Dr., Suite 3 Burlington VT, 05401	**Unless Otherwise Specified** Dimensions in Inches <u>Tolerances</u> Two Places: +/- 0.01 Three Places: +/- 0.003 Angular: +/- 1 Deg	Drawn By: W Labbe	Title: Lower Half Brace	
		Date: 4/22/2022	File Name: Brace LH	
		Material: AISI 1020 HR	Size: **A**	Scale: 1:2	Rev.



Dpt. of Research & Development
ELMVAC Inc.
101 Thornton Dr., Suite 3
Burlington VT, 05401

Unless Otherwise Specified
Dimensions in Inches

Tolerances

Two Places: +/- 0.01
Three Places: +/- 0.003
Angular: +/- 1 Deg

Drawn By: W Labbe		Title: Transport Brace	
Date: 4/22/2022		File Name: Brace	
Material: AISI 1020 HR	Size: **A**	Scale: 1:2	Rev.