Final Technical Report

Scissor Jack Design & Analysis

Team 2

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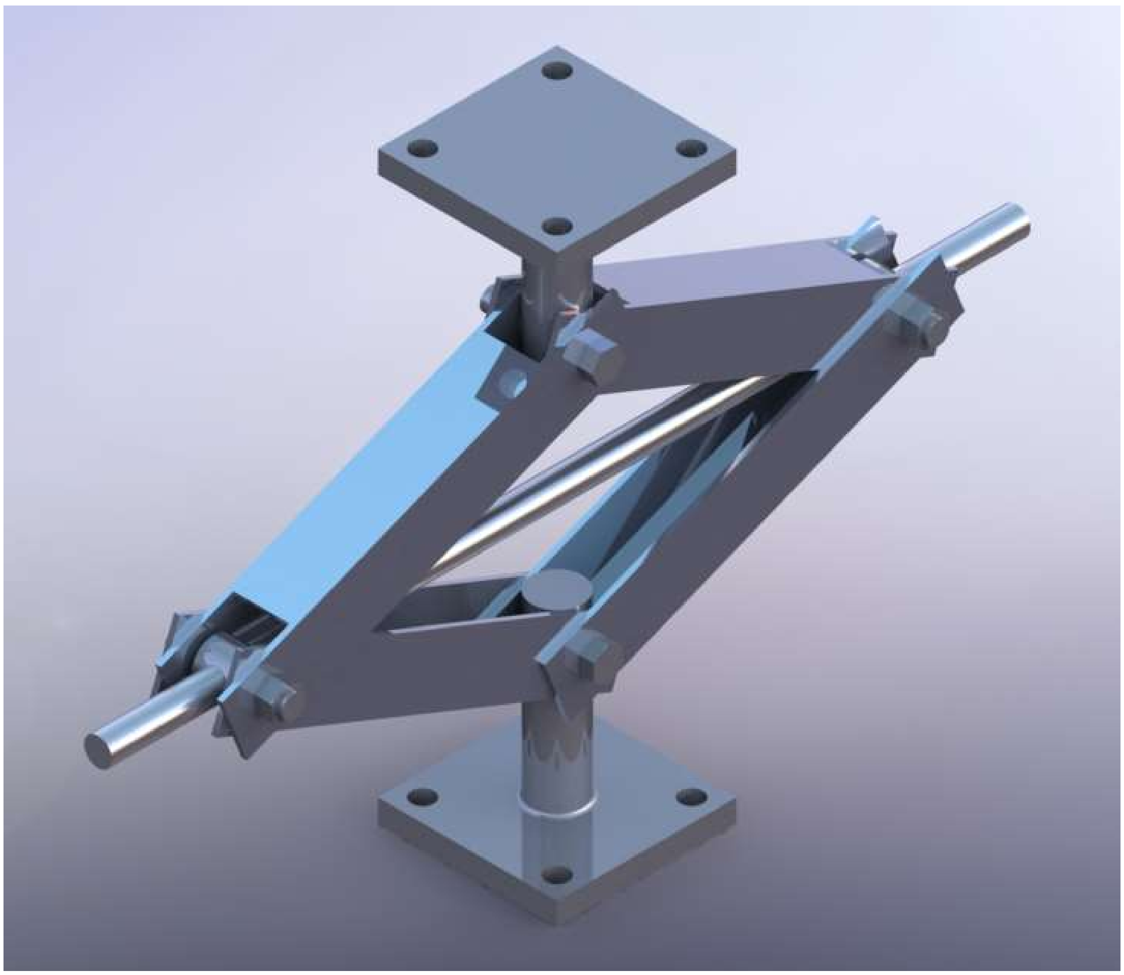


Executive Summary

For this project, we were asked to design a simple scissor jack to lift a 2000lb load with a minimum distance of 6 inches. The scissor jack was to be mounted on the ceiling to raise heavy objects. With these basic design needs in mind, the following design requirements and constraints were defined to increase the overall construction, functionality, and safety. The scissor jack was specified for use in a home garage. Therefore, it was to have a cycle life of 7,000 cycles to ensure that it was functional and long-lasting. It should also fit in a box smaller than 20” x 4” x 4” to ensure portability. All catastrophic failure modes were to have a safety factor of 3, and all non-catastrophic failure modes were to have a safety factor of 1.5 to ensure a safe and reliable product.

With these design requirements, it was decided that our model was to be constructed to optimize weight. By optimizing the weight of the overall scissor jack, we were able to explore numerous parameters simultaneously. Those worth mentioning are the cross-section of the diagonal members (square, circular, and u-channel geometry), the material of the diagonal, crossbar, and pin members (steel, aluminum), and the respective lengths of the members (diagonal and crossbar members).

After validating our model with hand calculations, a known scissor jack failure scenario, and variations in the parameters, it was time to run our final model. In doing so, we were able to successfully optimize the weight of our scissor jack given our numerous constraints. The numerically optimized parameters guided us to the closest combination of commercially available parts that minimized weight while not jeopardizing our other constraints. The results are listed as follows: The diagonal members were optimized to be made out of 1. 75” x 1.75” x 0.125” 6063-T5 Aluminum U Channels. The total distance between holes was 7.50” and the length of tearout was 0.875”. The steel crossbar was optimized to a diameter of 0.625” that was to be made solely out of 304 Stainless Steel. The steel bolts were optimized to a Grade 5 Mild Carbon Steel Bolt that was 0.5” in diameter and 2.625” long.

The most complex and least predictable portion of our design at this point was the joint pin. Therefore we chose to model this part in FEA to validate and confirm our model’s outputted safety factors. Our FEA analysis showed that our numerical model’s estimations were good within the scope of the failure modes the numerical model estimated (namely the axial and the bearing stresses). Our numerical mode did not account for pin bending. Upon comparison to hand calculations, the FEA predicted 50% larger maximum bending stress, and this was due to the proximity of the bearing forces to a stress concentrator fillet. But even with this oversight, our safety factor was close to the constraint and could easily be iterated upon to improve the stress distribution and improve safety.

In conclusion, we deemed our final design to be hyper-economic while also being satisfactory for our safety and functionality constraints. We recognize the weakness of our model with pin bending and thread strength. We reason, however, that our high safety factors protect against most unknown failure types. To build an ever more encompassing model, we recommend that future designs and iterations of our model should account for these other stresses and other real-world factors, as well as carefully account for stress concentrations.

Introduction

Objective

Our team was asked to design a scissor jack that was to be mounted on the ceiling and lift objects below it. We were given two primary constraints: it has to be able to lift a 2000lb object while also lifting it a minimum distance of 6 inches. The scenario, design requirements, parameters, and objective were to be defined by our team.

After creating a problem statement, the next step was to use our engineering skills and judgment to select appropriate design tools to help create a design model. This model was to take all of our inputs (design requirements, constraints, and parameters) and convert them to measurable outputs (lifting capability, operating performance, stress in the members, and safety factors for failure modes). More specifically, it was to be constructed so that it could optimize the objective of the project. The model then needed to be verified with numerous methods to ensure its validity (hand calculations, checking answers with a known scissor jack problem, and confirming trends by changing parameters to ensure that our engineering judgment was accurate).

The final step was to run our model with an optimized solver (one of the requirements of the project was to choose an objective to optimize). We chose to minimize the overall weight, while still maintaining the required safety factors. By running the optimized solver, our team was able to design the ideal scissor jack for our problem statement according to static and fatigue failure. After this, we also verified its static failure behavior using Finite Element Analysis (FEA).

Design Requirements

Our scissor jack was designed for use in a hobbyist's home garage to lift heavy car parts. It was to lift parts that weighed a maximum of 2,000 lbs and a minimum distance of 6 inches. Provided below is our list of requirements for a successful design:

1. Sustain a 7,000-cycle life
2. Fit in a box that is 20" x 4" x 4"
3. Catastrophic failure mode safety factors greater than 3.0
    - Diagonal member axial
    - Diagonal member tear out
    - Crossbar member axial
    - Crossbar member buckling
    - Pin shear
    - Diagonal member axial fatigue
    - Diagonal member tear-out fatigue
    - Crossbar member axial fatigue
    - Pin shear fatigue
4. Non-catastrophic failure mode safety factors greater than 1.5
    - Diagonal member bearing
    - Crossbar member bearing
    - Pin bearing
    - Diagonal member bearing fatigue

- Crossbar member bearing fatigue
- Pin bearing fatigue

These design constraints helped us design a functional scissor jack that can lift car parts twice a day for up to 10 years; is compact and easy to handle and operate; and is safe by accounting for human safety and the likelihood of large, unexpected loads. Our overall objective was to minimize its weight while ensuring all other design requirements were met.

Methods

Key Failure Modes

The scissor jack was intended to be mounted on the ceiling to raise heavy objects. This design scenario meant that the diagonal members would be in tension, the crossbar member in compression, and the pins in shear and bending. In this scenario, our team decided to identify catastrophic and non-catastrophic failure modes and apply a fixed safety factor to each respective class of failure. After some discussion, we also safely assumed a perfectly homogenous, two-dimensional, fully ductile structure.

Our team was most concerned with catastrophic failure due to the high risk of injury these pose to customers. Regarding the diagonal members, we were most concerned with axial and tear-out failure around the pins. We knew that the joint is where scissor jacks tend to fail and where the smallest stress areas are. Concerning the crossbar, we were most concerned with the buckling failure of the entire member. The crossbar member was to support the entire load and operate the scissor jack, meaning that its performance and reliability were crucial. For the pins, we were most concerned with shear failure. The pins are what transfer the weight from the diagonal to the crossbar members, which can greatly exceed the applied force of the load. They also provide rigidity to the scissor jack. If the scissor jack was to fail through any of the methods, it would be catastrophic. The scissor jack would collapse suddenly and jeopardize the safety of the operator. As such, a safety factor of 3.0 was factored in for the diagonal axial, diagonal tearout, crossbar axial, crossbar buckling, and pin shear static and fatigue failure modes.

The next step was to identify non-catastrophic failure modes. In this particular scenario, we clarified that for failure to be non-catastrophic, we would be able to observe its failure propagation long before it suddenly fails. For the scissor jack, this was most likely to happen with bearing failure around the pins or with fatigue failure. It was concluded that if this failure was observed, the operator could return the scissor jack for a warranty replacement long before any physical danger was present. As such, a safety factor of 1.5 was factored in for the diagonal bearing, crossbar bearing, and pin bearing static and fatigue failure modes.

Variable	Value										
d_cb	0.625		**Other Calculations**			**Stresses**	**Stress (psi)**	**n_y**	**n_f**		**Constraints**
l_d	7.750		h_prime	9.500		diagonal_tear	6459.38	3.25	3.39		n > 3
w_d	1.500	**l_d Constraint**	theta	0.660		diagonal_axial	6526.32	3.22	3.36		n > 3
t_d	0.125	3.1	l_cb	12.247		diagonal_bearing	13052.63	1.61	2.82		n > 1.5
l_tearout	0.875		A_pin	0.196		crossbar_axial	8404.31	3.71	11.43		n > 3
d_pin	0.500		A_cb	0.307		pin_shear_cb	11372.42	8.09	7.92		n > 3
Sy_cb	31200	Stainless	A_d	0.250		pin_shear_d	7196.29	12.78	12.52		n > 3
Sy_d	21000		l_pin	2.625		pin_bearing_d	13052.63	7.05	11.06		n > 1.5
Sy_pin	92000		A_square_bulk	0.688	0.01						
E_cb	29000000		A_channel_bulk	0.531	0.01	**Buckling**	**Pcr**	**F**	**n**		
E_d	10000000					crossbar_buckling	7969.34	2578.41	3.09		n > 3
Sut_cb	73200						Johnson				
Sut_d	27000		**Weight**			diagonal_buckling_sq	14292.31	1631.58	8.76		n > 3
Sut_pin	120000		Crossbar Weight	1.818	0.01		Johnson				
rhocb	0.289		Sq. Diagonal Weight	2.082	0.01	diagonal_buckling_c	11069.56	1631.58	6.78		n > 3
rhod	0.098		C. Diagonal Weight	1.717	0.01		Johnson				
rhopin	0.284		Pin Weight	0.586	0.01						
isALCB	FALSE		Sq. Total Weight	**4.485**							
isALD	TRUE		C. Total Weight	**4.120**							
isALPin	FALSE										
			Forces						3.05		
			F	2000.000		theta (deg)	37.80		2.66		
			F_cb	2578.410							
			F_d	1631.579							

Figure 1. A screen capture of our Excel model. Highlighted in green are the yielding and fatigue safety factor outputs.

Appendix A-1 shows a black box diagram of our model. Figure 1 shows our fully realized analytical model as a spreadsheet in Excel. The leftmost column defines our variable dimensions and material constants. The middle-left section shows the weight calculation of our model, as well as other intermediate calculations. The middle-right section displays the stresses for each stress type, the static safety factors, and the fatigue safety factors. Lastly, the far-right displays the safety factor constraints that the Excel Solver used to optimize our model.

The failure modes that were not fully addressed in our analytical model were pin bending, pin thread failure, and diagonal buckling. The diagonals were only expected to be in tension, which negates the need for buckling. Regardless, our analytical model still calculated the buckling safety factor in the diagonal members if ever loaded in compression, though these were not used to expressly optimize the design. Pin bending is addressed exclusively by our FEA model due to its complexity, and further discussion of this will be in the FEA section. Threaded failure was not addressed as it was presumed out of the scope of this project.

Forces in the Structure

The forces in the structure were found using static analysis. A two-dimensional free-body diagram of the scissor jack was drawn, and trigonometric relationships were applied to find the force in the diagonal and crossbar members as shown in Figure 2; the results are shown in Table 1.

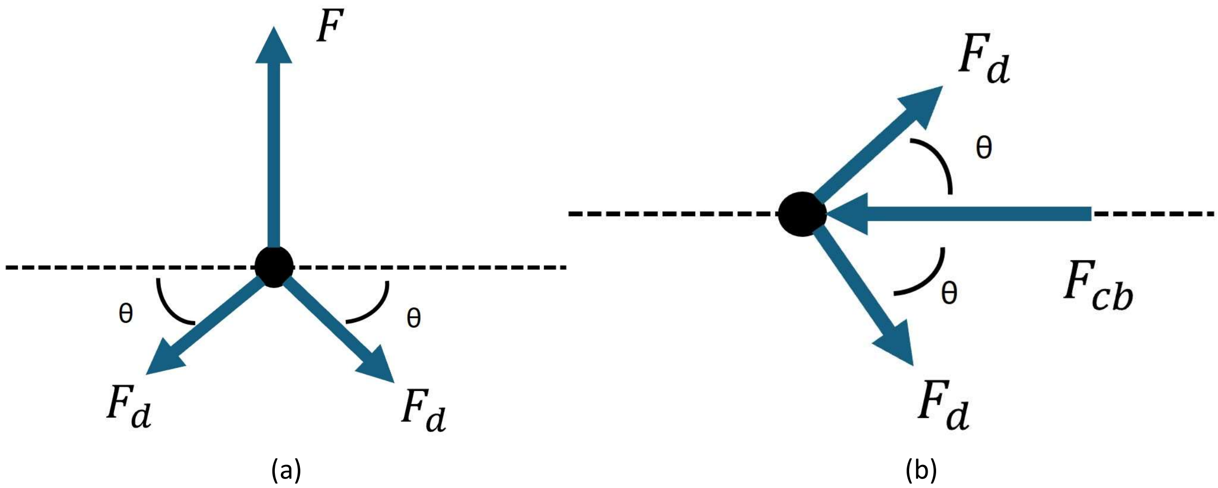


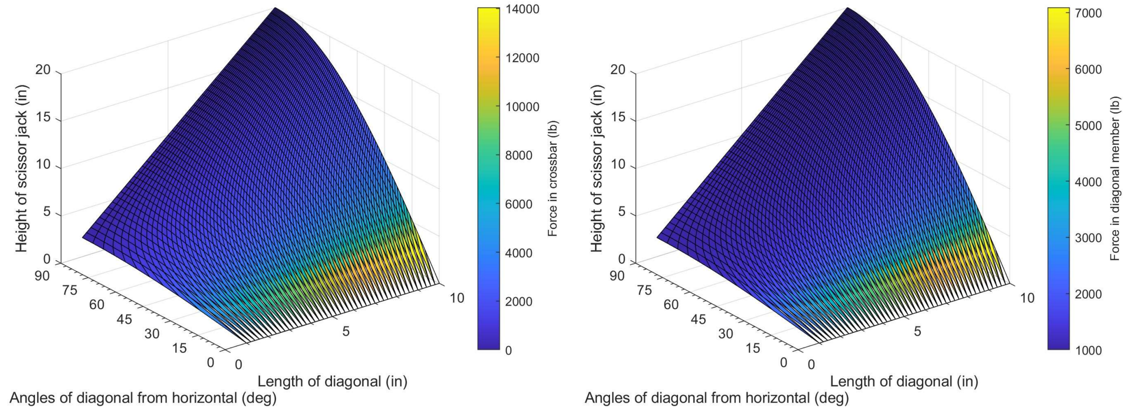
Figure 2. A simple free-body diagram of the forces at (a) the top and bottom diagonal pinned joints and (b) the left and right crossbar pinned joints.

Table 1. A summary of the forces in our scissor jack, for an applied tensile load, F, of 2000lb.

	θ	37.8°- Model	37.8°- Hand-calculated
$\boldsymbol{F_d(\theta)}$ (lbf)	$\frac{F}{2}csc(\theta)$	1631	1631
$\boldsymbol{F_{cb}(\theta)}$ (lbf)	$Fcot(\theta)$	2578	2578

Table 1 summarizes the forces in the scissor jack as a function of theta, the angle of the diagonal member to the horizontal. However, our constraints only specify a necessary height displacement of the object in tow, which is a function of both the operating angles and the length of the diagonal members. To better visualize this complex relationship, MATLAB was used to plot the trend between dimensions, changes in height, and forces, as shown in Figures 3a and 3b.

It is important to note that, for a given member length, the forces in the members increase as the horizontal angle becomes shallower. Additionally, the crossbar has a higher internal force than the diagonal member at lower angles (< 40**°**). Assuming our scissor jack starts operating at 90° (fully extended), the Excel model was able to alter the length of the diagonal member to achieve 6 inches of vertical movement while minimizing the stresses in the members. Accordingly, the operating angles of the scissor jack were determined to be 90°-37.8° for 6 inches of vertical displacement by the Excel Solver. The forces in the members are detailed in Table 1. These values were verified with hand calculations as shown in Appendix A-2.



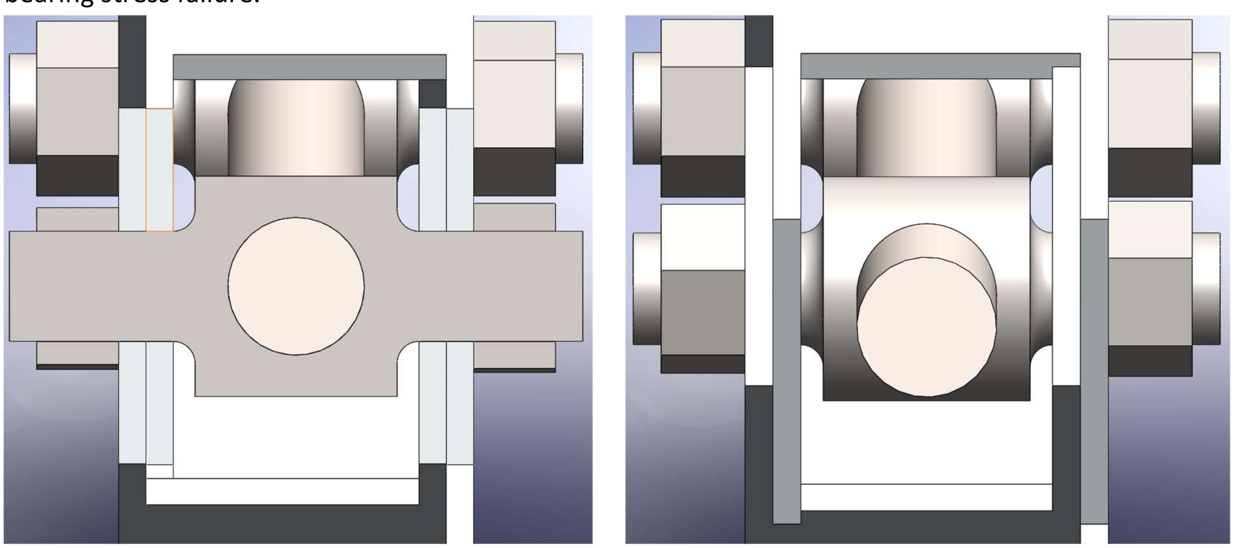
(a) (b)

Figure 3. Surface plots show the force in the (a) crossbar and (b) diagonal member as a function of height and operating angle.

Failure Prediction Methods

Appropriate failure modes were found using static and fatigue failure analysis. The first step was to visualize the diagonal and crossbar pinned joint, which was where failure was most likely to occur. Figure 4 shows (a) a cross-sectional view and (b) a peripheral view of the joints. The joints were modeled as four plates with a pin through them, with the crossbar being threaded through the pin.

Hand sketches were then drawn, and equations for failure were carefully applied to identify and solve for all relevant failure modes. These equations were found in the class textbook *Mechanical Engineering Design* by Shigley, and are summarized in Table A-3 in the Appendix. These equations were ultimately used in our numerical model to analyze the effects of buckling, axial, tearout, pin shear, and bearing stress failure.



(a) (b)

Figure 4. A screen capture of the crossbar connecting joint in our scissor jack. Depicted on the left is a cross-sectional view (a) and depicted on the right is a peripheral view (b) of the joint.

After analyzing the static failure modes, the next step was to analyze the fatigue failure modes. As shown in Table A-3 in the appendix, the endurance strength of the material was determined using marin factors to modify experimental data to calculate a fatigue safety factor. The surface factor was adjusted to fit cold rolled and machined surfaces, using the following equation $k_a = 2.00(S_{UT})^{-0.217}$. The size factor was disregarded due to the axial loading nature of each of our failure modes. The resources available to us on how to treat shear loading were unclear when determining fatigue properties. Discussing this issue with our professor, we decided that a load factor, k_c , of .85 should be applied in all of the failure cases being considered by our numerical model. This is because of the axial nature of each failure mode being considered (for example, tearout being caused by an axial load). Furthermore, engineering judgment led us to also convert the shear to an equivalent Von Mises stress in order to compute the safety factors directly. We also defined the reliability of our design as 90%, creating a load factor $k_e$equal to 0.897. The temperature and miscellaneous Marin factors were safely set to one, assuming normal operating conditions in a room-temperature environment.

Additionally, we adapted the fatigue model to predict both aluminum and steel performance. The material property-determining resources available to us are steel-specific, such as the equations to determine k_a and fS_{ut}, which are functions of $S_{ut,steel}$. Accordingly, we estimated aluminum's "equivalent steel tensile strength", $S_{ut,eq}$, as it relates to these equations. This was done by recognizing the similarity between the equations defining S_e in steel and S_f in aluminum, shown in Figure 5. A parallel was drawn between the 48 ksi cutoff value in aluminum and the 200 ksi cutoff value in steel, creating an estimated equivalency ratio of 200:48 as it relates to determining k_a and fS_{ut} for aluminum. Through this method, we were able to estimate the fatigue performance of aluminum, despite having fewer material-specific resources. To conclude the construction of our analytical model, we verified the equations found in Table A-3 in the appendix with the hand calculations found in Appendix A-5. This verification is discussed in more detail in the next section.

Steels:

$$S'_e = 0.5\, S_{UT}\ (S_{UT} < 200\ ksi\ (1400\ MPa))$$

$$S'_e = 100\ ksi\ \left(S_{UT} \geq 200\ ksi\ (1400\ MPa)\right)$$

Corrected value:

$$S_e = K_a K_b K_c K_d K_e K_f S'_e$$

Aluminum

$$S'_f = 0.4\, S_{UT}\ (S_{UT} < 48\ ksi\ (330\ MPa))$$

$$S'_f = 19.2\ ksi\ \left(S_{UT} \geq 48\ ksi\ (330\ MPa)\right)$$

Corrected Value:

$$S_f = K_a K_b K_c K_d K_e K_f S'_f$$

Figure 5. A screen capture showing the endurance strength equations for steel and fatigue strength equations for aluminum.

After applying the correct failure modes to the scissor jack, we ran through our model and tested numerous materials. After our exploration, we decided to make the diagonals out of 6063 T-5

Aluminum, the pins and scissor jack mounts out of 1018 Steel, and the crossbar out of 304 Stainless Steel. For calculating the weight of the entire assembly, standard density values of 0.289 $\frac{lb}{in^3}$ were used for 304 Stainless Steel, 0.284 $\frac{lb}{in^3}$ for 1018 Steel, and 0.098 $\frac{lb}{in^3}$ for 6063 T-5 Aluminum. The moduli of elasticity that were used are $2.9\ x\ 10^7$ psi for 304 Stainless Steel and $1.0\ x\ 10^7$ psi for 6063-T5 Aluminum.

Design Approach

Before constructing our analytical model, we employed engineering decision-making to narrow our search space. Our design consists of three main parts: the diagonal member, the crossbar, and the pin connections. For each of these, we decided to explore the possibility of selecting either aluminum or stainless steel. Both have good corrosion resistance and are fairly commonplace to find and use in manufacturing. Aluminum provides the advantage of being lighter per unit of mass, but steel is stronger per unit of mass, presenting a unique trade-off worthy of exploration.

The next engineering decision involved the cross-section type. Our crossbar, once manufactured, would be a leadscrew by the necessity of movement. For simplicity, we modeled it as a circular bar without threading. Our pin connections also needed to be primarily circular, as only this shape would allow turning around an axis of rotation. Hence, the primary decision-making was applied to the cross-section of the diagonal member. Many possible shapes were eliminated due to their complexity (hexagon, triangle, etc), leaving further consideration for only circular, square (hollow or solid), and u-channel cross-sections. A circular cross-section was eliminated for the reason that such a cross-section would make the assembly complex and bulky, as connections between diagonal members would have to be made tangentially rather than flush with a face. Solid square cross-sections were eliminated due to the extra weight, extra material, and the more complicated asymmetric geometry that would be required around the pin connections if only one bar were to be used for each side of the parallelogram-shaped scissor jack. This left only the u-channel and hollow square cross-section to be considered. To achieve the assembly shown in Figure 13, each of these cross-sections would be shaved at the pin connections into two parallel plates, allowing them to mesh with the adjacent diagonal members in the assembly. While both of these cross-section options were considered and compared throughout the design and exploration process, the u-channel became the cross-section of choice due to its lighter weight, lack of interference with other diagonal members as the scissor jack assembly shortens, and the indifference in safety factors at the pin connections due to the exact geometry (i.e. two parallel plates) at those locations.

After employing engineering decision-making, we desired to validate our Excel model before exploring the design space. We first tested some scaling by doubling individual parameters and making sure that it scaled properly or as expected. After that preliminary confirmation, we wanted to validate against previously known values. We inputted the values from a previous project, "Static Analysis of a Scissor Jack," and confirmed that the values given by our model resembled those in the assignment as seen in Table 2. It is worth noting that although there are slight differences due to our assumptions and rounding, our model was proven to be accurate and consistent. There were a few failure modes that the previous project did not account for (such as buckling), so we decided to confirm these failure modes with hand calculations. We also confirmed our model's ability to estimate the weight. These hand calculations are shown in Appendix A-4. Lastly, we verified our model with two additional sets of input parameters, confirming the robustness of our model, and the ability to change the material of each part individually. The results of these hand calculations are also shown in Appendix A-5.

Table 2. Table comparing values from a previous project, “Static Analysis of a Scissor Jack,” to that of our model with the same inputs.

	Diagonal Member Tearout (psi)	Diagonal Member Axial (psi)	Diagonal Member Bearing (psi)	Cross Bar Bearing (psi)	Pin Shear (psi)	Pin Bearing (psi)
Previous Scissor Jack Project Values	4714	2177	8709	27793	9603	27793
Our Model	4684	2163	8654	27494	9543	27494

With a verified model, we were now able to iterate numerous parameters and explore the design space. Our model allowed us to change the geometry of the crossbar (diameter), diagonal (length, width, thickness), and pin (diameter); the material (and respective material properties) of the crossbar, diagonal, and pin; as well as the location of the hole and pin connection (tear out distance) in the diagonal members. As discussed previously, our model also allowed for the exploration of a square, circular (crossbar), and u-channel cross-section. Incorporating varied materials, the model also allowed for the exploration of any steel or aluminum.

Our model was then optimized (both numerically for dimensions and manually for cross sections and material) with the aid of the Excel Solver to minimize the weight of the scissor jack while staying within some given constraints. These constraints included our previously prescribed constraints (lifting range, required load, and safety factors) while also including general geometric constraints to keep the jack practical. We researched commercially available parts whose cross-section, material, and dimensions reassembled our model-optimized values. We then selected numerous similar commercial parts and inputted them back into our model. In general, this meant rounding the optimized decimals up to the next standard dimension (next highest ⅛” if less than an inch, and the next highest ¼” otherwise). We then elected the lightest design option that remained within our devised restraints. In summary, our model allowed us to simultaneously explore numerous design parameters that helped us to design a lightweight scissor jack that met all of our design requirements and is easily manufacturable.

Finite Element Analysis Methods

By using FEA analysis, we were able to answer numerous questions about our design and our numerical model. These questions revolved around the most complex part of our design: the pin connection between the diagonal members and the crossbar. Because of its intricate geometry, there were uncertainties in the accuracy of our numerical model in this section. These questions were:

- Is the estimated pin bearing and crossbar axial stress accurate?
- Are there any other failure modes that we are not considering?

- How impactful is the inclusion/exclusion of stress concentration factors?

To answer these questions, we used ANSYS workbench to model our pin design according to the geometry optimized in our numerical solver. Our pin had a total length of 2.625", a diameter of 0.5" on the edges, and a 1" diameter in the middle. A .1" fillet merging those two cross-sections was also added to improve manufacturability. Our pin connector was modeled with the same material as the Grade 5 bolts used elsewhere - S_y = 92ksi and S_{ut} = 120 ksi. To model the loads on this part, the inner circular face was fixed in place while, bearing loads were added to account for the stresses induced by the four flanges at the end of the diagonal members, as shown in Figure 6. The resulting four bearing forces were applied to the crossbar pin, adjacent to the fillets on both sides. Each region had the same thickness as the flanges on the connecting diagonal members. These were inputted in the correct direction of 37.8°, with the magnitude of each being half of F_d. The compressive load felt in the crossbar-pin connection was automatically accounted for in our model due to the fixed boundary condition set at that location.

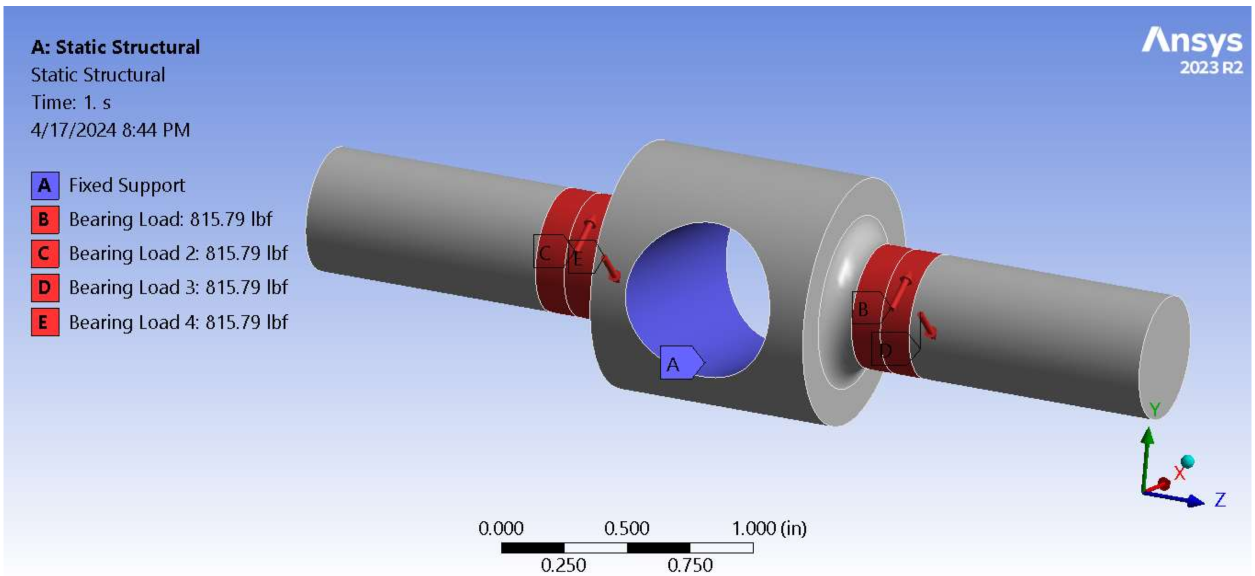


Figure 6. A screen capture of the modeled crossbar pin in Ansys Workbench. Shown are all four locations of bearing stresses.

Having applied the appropriate bearing stresses and displacement boundary conditions, Finite Element Analysis was used to solve for the von Mises stresses across the pin. The results are shown in Figure 7.

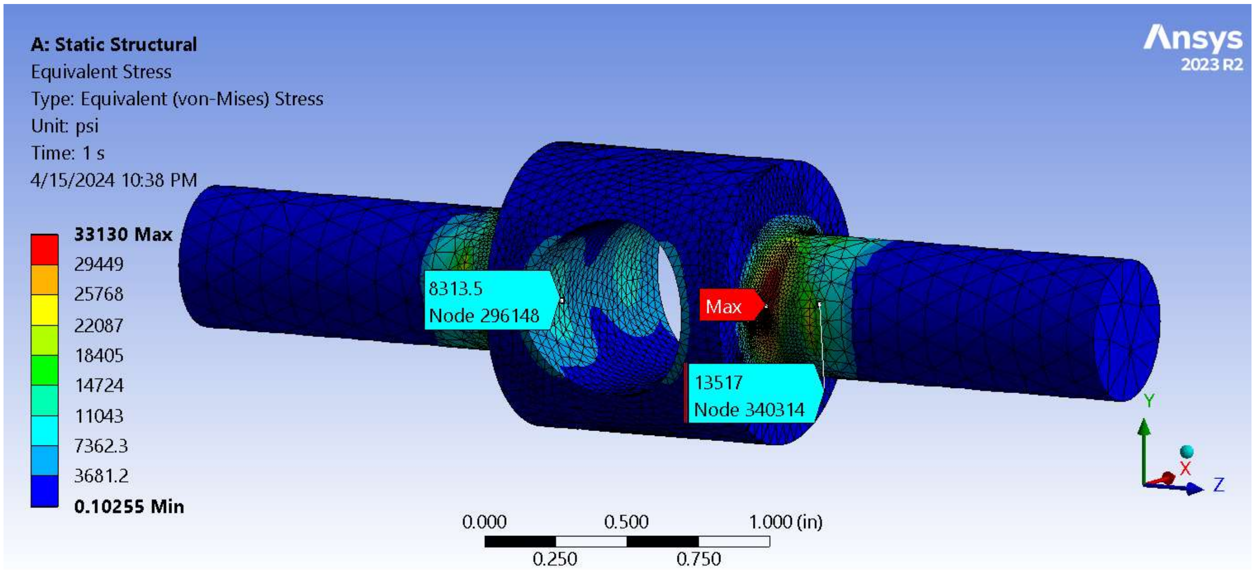
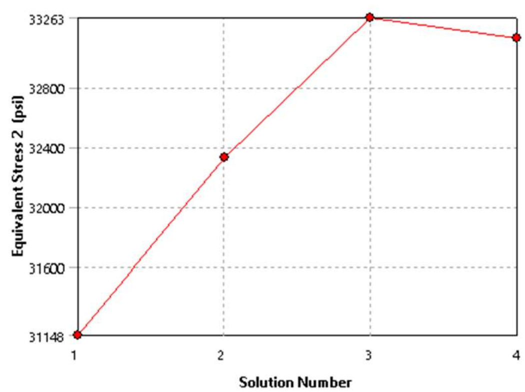


Figure 7. A plot of the equivalent von Mises stresses in the crossbar pin.

Our results predicted an 8.4 ksi axial stress at the pin-crossbar connection and a 13 ksi bearing stress at each pin-diagonal connection. Figure 7 confirms these numbers in each location, answering one of our main questions. The general trend of the stress profile is also as expected, following that the stresses on the ends of the pin are close to zero, there is a stress concentration at the filet, and there is the most stress farthest from the yz bending neutral axis plane. Additionally, upon doing a convergence study, as shown in Figure 8, the maximum stress was determined to be 33.1 ksi.



	Equivalent Stress 2 (psi)	Change (%)	Nodes	Elements
1	31148		41511	27644
2	32330	3.723	114367	80612
3	33263	2.8452	248834	178783
4	33130	-0.40162	425579	306912

Figure 8. A convergence plot showing the maximum equivalent stress in the crossbar pin.

Our numerical model did not predict this high of stress in the pin. We predict that this higher stress is due to the presence of bending in the pin, which our numerical model does not account for. Hand-calculating this pin bending scenario (Appendix A-6) estimated a maximum bending stress of 20.42 ksi, roughly 50% less than the FEA-calculated stress. To determine the true cause behind this discrepancy, we theorized that the proximity of the crossbar hole to the fillet created a compounding effect on the two stress concentrations. This was tested with the boundary conditions shown in Figure 9 (ie the same forces in Figure 6). The back, round face (not shown) was marked as fixed. As can be seen in Figure 10, this hypothesis was proved incorrect, as the maximum stress is the same as in Figure 7.

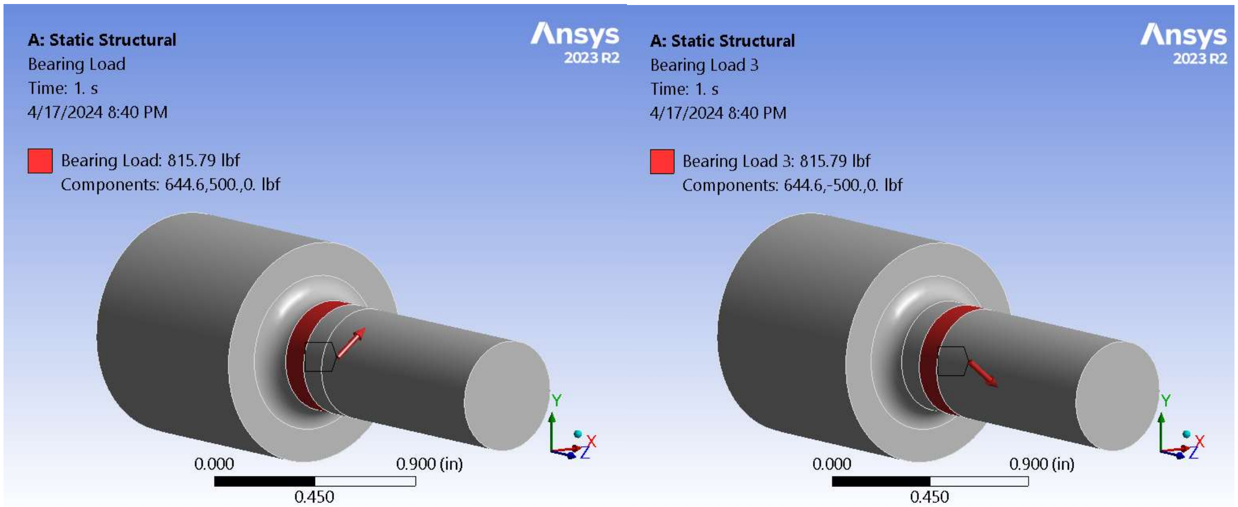


Figure 9. The boundary conditions for the pin with no inner hole.

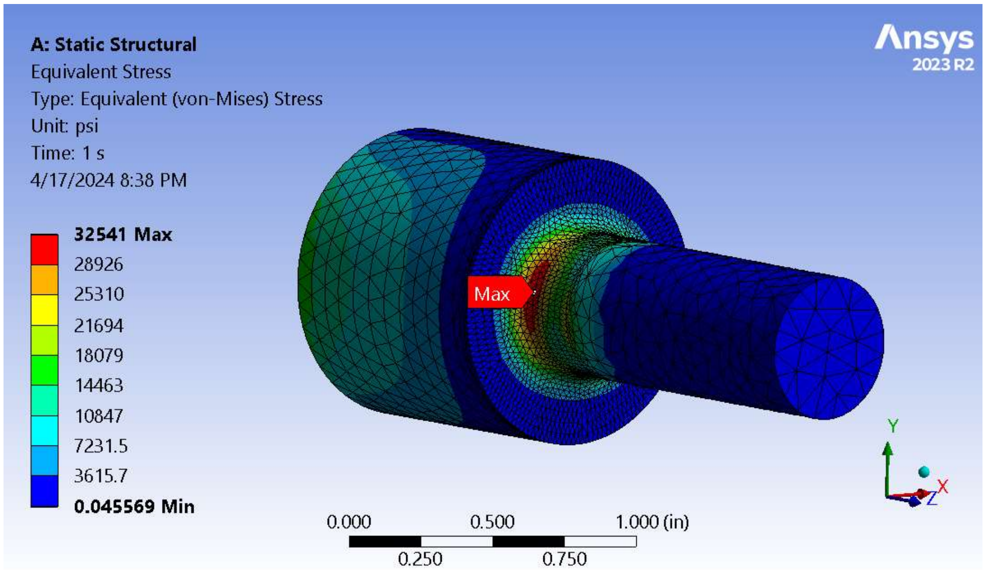


Figure 10. The stress plot for the pin with no inner hole.

Our subsequent hypothesis suggested that the positioning of the filet in Figure 6 might be too proximal to the point where various forces were applied, thereby disrupting the expected distribution of bending stresses. To test this hypothesis, the bearing stresses were removed, and forces were added to the ends of the pin (Figure 11). These forces were calculated to produce the same moment at the base of the fillet as did the bearing stresses. Those calculations are shown in Appendix A-6.

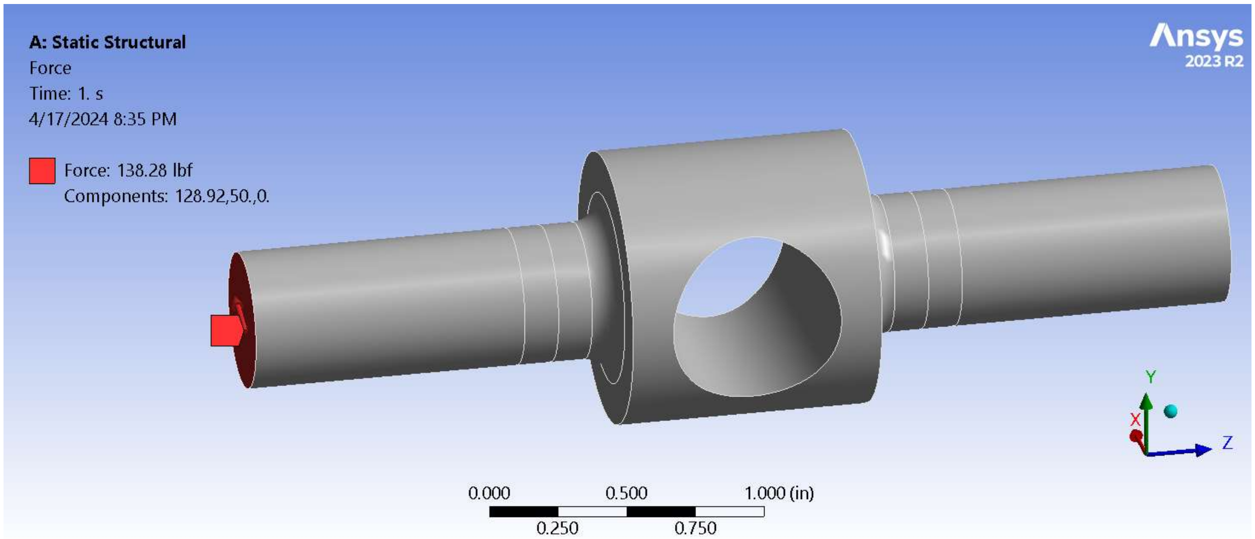


Figure 11. Forces added to the left side of the pin which generated the same moment at the base of the fillet as did the bearing stresses. These forces were also added on the right side of the pin.

As shown in Figure 12, the maximum bending stress in the pin is exactly what we calculated in Appendix A-6. This proves our hypothesis correct – Figure 7 is indeed the correct stress plot, and the discrepancy in maximum forces between hand calculations (A-6) and the FEA model (Figure 7) is due to the proximity of the load application and the fillet. Therefore, our FEA model is indeed correct and represents a robust analysis.

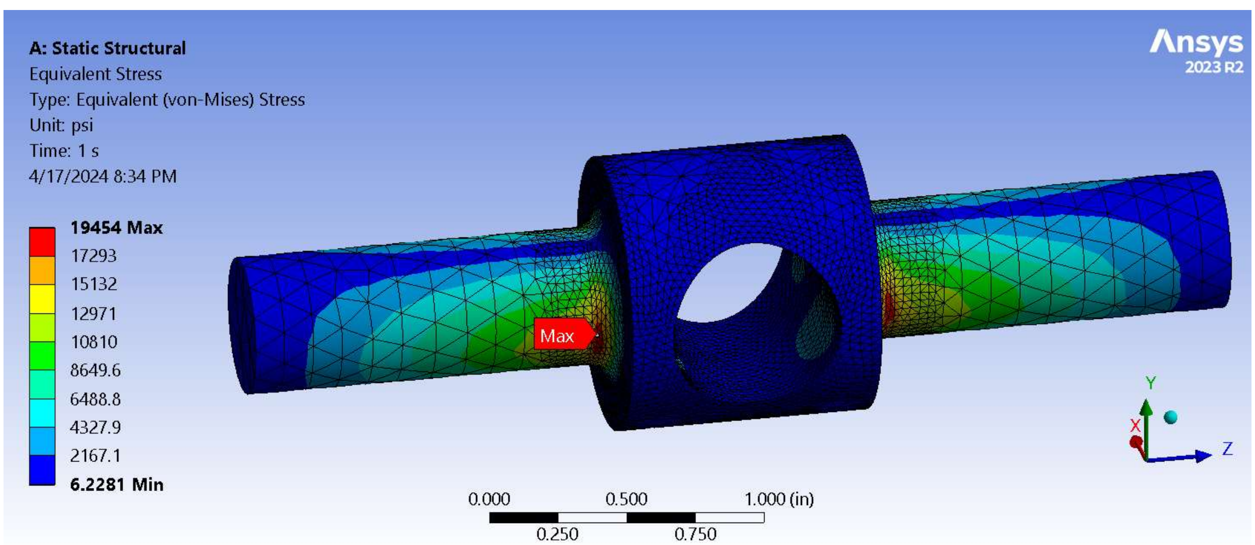


Figure 12. The stress plot of the pin with forces applied to the ends of the pin rather than as bearing forces.

Having verified our FEA model, the maximum stress was confirmed to be 33.1 ksi. Comparing this stress to our safety factor constraints, this results in a 2.8 static failure safety factor and a 2.74 fatigue failure safety factor when plugged into our numerical model. While these are below the minimum of 3 for catastrophic failure, we are confident that through more iterations, an improved safety factor could be realized. Methods would include increasing the fillet radius, decreasing the changes in cross-sectional areas, and largening the distance between the bearing forces and the fillet. Despite this, our FEA model confirmed that our numerical model was accurate within the prediction scope of the numerical analysis (axial compression and bearing stress). Hence, our results appear to be reliable, and our engineering approach was justifiable.

Results

Final Design Description

Our final design is pictured in Figure 13 and consists of four U-channel diagonal members, a threaded solid circular crossbar, two specially designed pins, two commercial bolts, and two attachment pieces. The diagonals are made out of 6063 T-5 Aluminum, the pins and scissor jack mounts are made out of 1018 Steel, and the crossbar is made out of 304 Stainless Steel. The standard density values of 0.289 $\frac{lb}{in^3}$ were used for 304 Stainless Steel, 0.284 $\frac{lb}{in^3}$ for 1018 Steel, and 0.098 $\frac{lb}{in^3}$ for 6063 T-5 Aluminum, and the moduli of elasticity of $2.9\ x\ 10^7$ psi was used for for 304 Stainless Steel and $1.0\ x\ 10^7$ psi was used for 6063-T5 Aluminum.

As shown in Figure 13, the diagonal members have cutouts where they meet the side joints that reduce each of the two joining diagonal members to just two parallel plates. The diagonals then have a hole put through these plates, where the pin can be inserted through both members, joining them together. The pin also has a threaded hole perpendicular to its main axis where the crossbar is inserted. The top and bottom joints have similar assemblies, however, a commercial bolt is used instead of the special pin. This bolt goes through both diagonal members as well as the roof or part attachment. The dimensions of the rounded-up, commercially available components are summarized in Table 3.

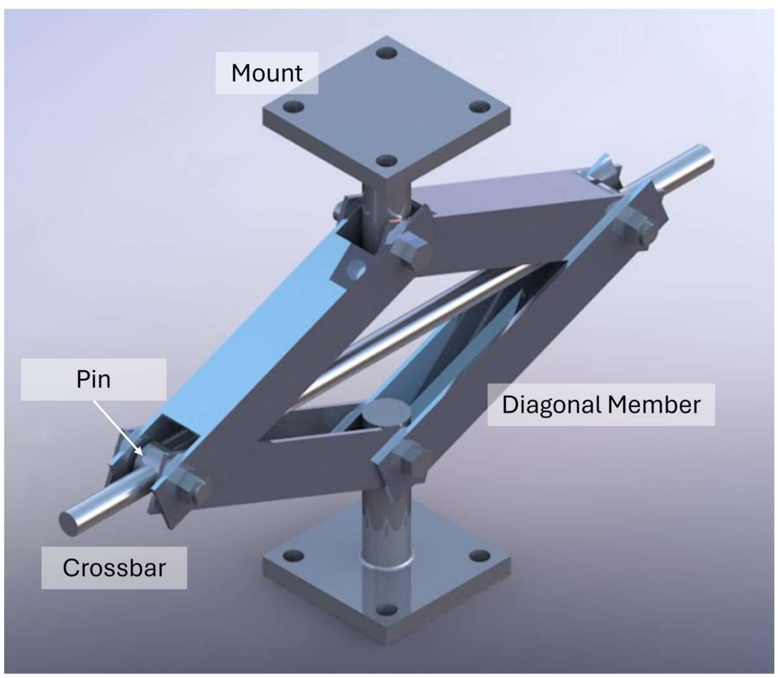


Figure 13. A render of the final, optimized design, of the scissor jack.

Table 3. The final, optimized, and commercially available dimensions of our scissor jack. All units are in inches.

Diameter of Crossbar (in)	Length of Diagonal (in)	Width of Diagonal (in)	Thickness of Diagonal (in)	Length of Tearout (in)	Diameter of Pin (in)
0.625	7.750	1.500	0.125	0.875	0.500

Prediction of design performance

Table 4. A table highlighting our design performance. Bolded are the calculated outputs for our final design at the shortest point of its 6" travel.

Location of Failure	Stress (psi)	Yielding Safety Factor	Fatigue Safety Factor	Safety Factor Constraint
Diagonal Tearout	6459	**3.25**	**3.39**	n > 3
Diagonal Axial	6526	**3.22**	**3.36**	n > 3
Diagonal Bearing	13053	**1.61**	**2.82**	n > 1.5
Crossbar Axial	8404	**3.71**	**11.43**	n > 3
Pin Shear Crossbar	11372	**8.09**	**7.92**	n > 3
Pin Shear Diagonal	7196	**12.78**	**12.52**	n > 3
Pin Bearing Diagonal	13052	**7.05**	**11.06**	n > 1.5
Crossbar Buckling	2578.41 (lbf)	**3.09**	—	n > 3
Weight of Total Scissor Jack (lbf)			**4.12**	

As shown in Table 4, our final design exceeded all of our design constraints and safety factors, the closest one being the diagonal yield safety factor. These safety factors are calculated at the scissor jack's most compact position in its 6" travel. This point corresponds to the largest stress, consistent with the trends described in Figures 3a and 3b. This was estimated to be 1.61, which is only .11 higher than our designated 1.5 safety factor required of non-catastrophic failure modes. We are confident that the failure modes predicted by our numerical model are above the specified safety factors. Further iteration would be needed to improve our pin-bending safety factor.

Additionally, with the use of aluminum and stainless steel for the majority of the design, our scissor jack boasts good corrosion resistance. These materials are also widely used, allowing for ease of manufacturability and processing (further coatings, paint, etc).

Discussion

The final design we selected had many advantages and disadvantages. The design met all requirements that were set, while the main advantage was its ease of modeling and testing. With few complex or custom parts, our design can be tested, modeled, and manufactured easily. Our model did not need advanced numerical solutions while our FEA analysis was efficient and reliable due to its simplicity. Our design also requires little manufacturing or assembly. The bulk of the jack is constructed of commercially available parts with minor machining required to get their final function. This makes the jack not only cheaper to purchase but also to manufacture and assemble.

As mentioned previously, two of the essential design questions regarded the best cross-section for the diagonal members and the optimal material for each member. A U-channel cross-section was selected. U-channel diagonal members allow adjacent diagonals to mesh as they're rotated about the pins, as opposed to square members whose inward-jack-facing faces would collide as the scissor jack shortens. Additionally, the geometry at the pin joints is unaffected by the choice of cross-section throughout the rest of the member. Hence, a U-channel was selected for its lower weight without a subsequent decrease in strength.

Interestingly, our model predicts that the lightest optimized design would include aluminum diagonal members, steel pins, and aluminum lead screws, weighing around 3.25 lbs. This is in comparison to our optimized design weighing 4.12 lbs. The decision to use a stainless steel lead screw was because of its 1) lower price and 2) greater availability. Through a detailed search of the internet, very few aluminum lead screws were found, and those available were much more expensive and/or specialized than their steel counterparts. Therefore, considering cost and availability, a stainless steel lead screw was chosen, despite the slight increase in overall optimized weight.

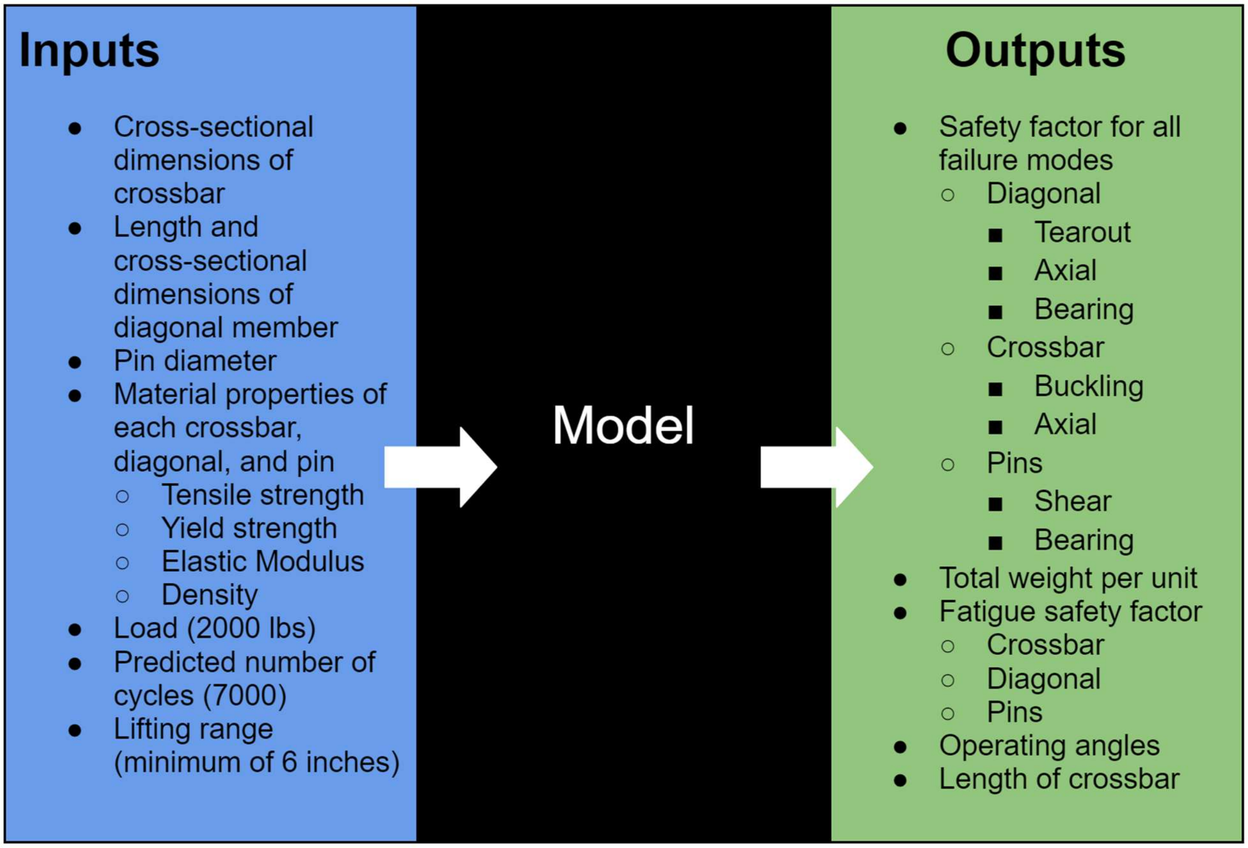
Our design has some weak points, however. Simple assumptions may cause slight deviations between our calculated results and its actual performance. Assumptions of a perfectly homogenous, two-dimensional, fully ductile structure only negligibly change our predicted values (See our FEA portion and stress concentrations). The biggest weakness however is the neglect of thread stress and pin bending in the joint. These are very difficult to model and test, and therefore large safety factors were used to compensate for their ambiguity. In actuality, these failure types are catastrophic and fail at an unknown load. This may lead to unexpected failure during the lifetime of the part. An improved model would account for these additional failure modes as well.

Conclusions

In conclusion, our design is both lightweight and very safe. We guarantee that this will be the cheapest available scissor jack for the design constraints given. We recommend using the scissor jack only at or below the maximum load of 2000 lbs. This would eliminate any ambiguity in the thread strength and pin bending that were identified as weaknesses in our model and estimations. In future models, we recommend implementing these stresses into the model and FEA analysis. The larger safety factor of 3 should also be applied to these stresses as we deem them catastrophic. We also recommend widening the scope of the model to include the attachments. All kinds of stresses (bearing, shear, axial, tear out, etc.) at these locations should then be considered and constrained with the appropriate safety factors. This would provide a more encompassing estimation of both the jack's safety and functionality.

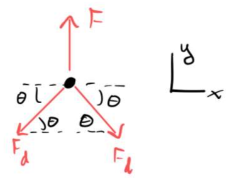
Appendix Outline

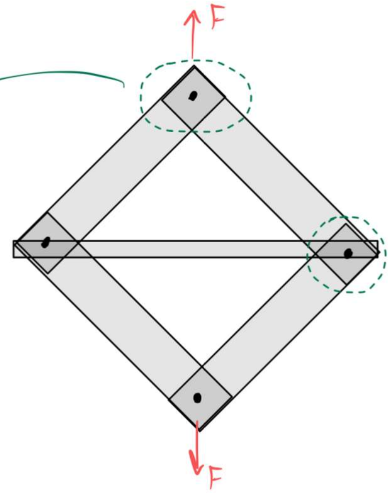
A-1 - Blackbox Diagram of Model

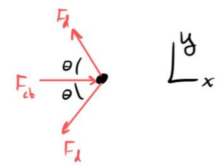


A-2 - Forces in Members

Force Verification







$\sum F_y$: $F = 2 F_d \sin\theta$

$$F_d = \frac{F}{2\sin\theta}$$

$$F_d = \frac{2000}{2(\sin 37.8°)}$$

$$\boxed{F_d = 1631 \text{ lb}}$$

$\sum F_x$: $F_{cb} = 2 F_d \cos\theta$

$$F_{cb} = 2\left(\frac{F}{2\sin\theta}\right)\cos\theta$$

$$F_{cb} = F \cot\theta$$

$$F_{cb} = 2000 \cot(37.8°)$$

$$\boxed{F_{cb} = 2578 \text{ lb}}$$

Table A-3 - Key Equations

Table A-3. A summary of the static failure and fatigue failure equations that were used in our failure prediction analysis. For the crossbar alone, the buckling equations for Euler or Johnson Buckling were used, depending on part dimensions, to predict the buckling failure safety factor.

Static Failure Equations

Diagonal Tearout:

$$\sigma_{vm} = \frac{\frac{F_d}{2}}{(2)(l_{tearout})(t_d)}(\sqrt{3})$$

Diagonal Axial: $\sigma_{vm} = \frac{\frac{F_d}{2}}{(t_d)(w_d - d_{pin})}$

Diagonal Bearing: $\sigma_{vm} = \frac{\frac{F_d}{2}}{(d_{pin})(t_d)}$

Crossbar Axial: $\sigma_{vm} = \frac{F_{cb}}{\pi(\frac{d_{pin}}{2})^2}$

Crossbar Bearing: $\sigma_{vm} = \frac{\frac{F_{cb}}{2}}{(d_{pin})(t_{cb})}$

Pin Shear (Crossbar): $\sigma_{vm} = \frac{\frac{F_{cb}}{2}}{\pi(\frac{d_{pin}}{2})^2}(\sqrt{3})$

Pin Shear (Diagonal): $\sigma_{vm} = \frac{\frac{F_d}{2}}{\pi(\frac{d_{pin}}{2})^2}(\sqrt{3})$

Fatigue Failure Equations

Fatigue Strength (for cycles given):

$S(N) = aN^b$, where $a = \frac{S_m}{10^{3*b}}$ and $b = -\frac{1}{log(N_1)-log(N_2)} log(\frac{fS_{UT}}{S_e})$

Buckling Failure Equations

$k = \sqrt{\frac{I}{A}}$, and $(\frac{l}{k})_1 = \sqrt{\frac{2\pi^2 CE}{S_y}}$

Euler's Buckling:

$P_{cr} = \frac{C\pi^2 EI}{l^2}$, or $\frac{P_{cr}}{A} = \frac{C\pi^2 E}{(l/k)^2}$

Johnson's Bucking:

$\frac{P_{cr}}{A} = S_y - (\frac{S_y}{2\pi}\frac{l}{k})^2 \frac{1}{CE}$, where $\frac{\boldsymbol{l}}{\boldsymbol{k}} \leq (\frac{\boldsymbol{l}}{\boldsymbol{k}})_{\mathbf{1}}$

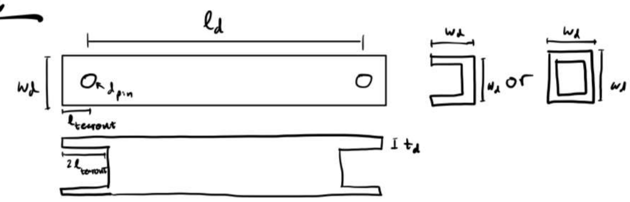
A-4 - Scissor Jack Assignment Hand-Calculations

Below is our model output for the dimensions of the Scissor Jack class assignment. The results are summarized in Table 2. Below is our numerical model output, along with the hand calculations verifying the outputs. Both safety factors and weight were confirmed.

Variable	Value								
d_cb	0.250		**Other Calculations**			**Stresses**	**Stress (psi)**	**n_y**	**n_f**
t_cb	0.060		h_prime	11.000		diagonal_tear	4684.41	3.42	5.91
w_cb	0.500	**l_d Constraint**	theta	0.704		diagonal_axial	2163.64	7.39	12.79
l_d	8.500	3.1	l_cb	12.961		diagonal_bearing	8654.55	1.85	5.29
w_d	1.250		A_pin	0.049		crossbar_bearing	27494.05	1.16	3.37
t_d	0.125		A_cb	0.049		crossbar_axial	16803.13	1.90	5.51
l_tearout	0.400		A_cb_pin	0.030		pin_shear_cb	14551.93	9.62	7.56
d_pin	0.250		A_d	0.250		pin_shear_d	9543.00	14.67	11.52
Sy_cb	32000.000		l_pin	2.750		pin_bearing_cb	27494.05	5.09	6.31
Sy_d	16000.000		A_square_bulk	0.563	0.01	pin_bearing_d	8654.55	16.18	20.05
Sy_pin	140000.000		A_channel_bulk	0.438	0.01				
E_cb	30000000.000					**Buckling**	**Pcr**	**F**	**n**
E_d	30000000.00		**Weight**			crossbar_buckling	337.94	824.82	0.41
Sut_cb	70000.000		Crossbar Weight	0.307	0.01		Euler		
Sut_d	35000.000		Sq. Diagonal Weight	1.875	0.01				
Sut_pin	150000.000		C. Diagonal Weight	1.499	0.01				
rhocb	0.284		Pin Weight	0.153	0.01				
rhod	0.098		Sq. Total Weight	**2.335**					
rhopin	0.284		C. Total Weight	**1.959**					
IsALCB	FALSE								
IsALD	TRUE		**Forces**						
IsALPin	FALSE		F	700.000		theta (deg)	40.32		
			F_cb	824.822					
			F_d	540.909					

Weight

Diagonal



Square: $W = 4\rho_d\left[\left(w_d^2-(w_d-2t_d)^2\right)(l_d+2l_{tearout}) - 4\left(\frac{\pi}{4}d_{pin}^2\, t_d\right) - 4\left(2l_{tearout}(w_d-2t_d)t_d\right)\right]$

$\boxed{W = 1.88\ \text{lb}}$

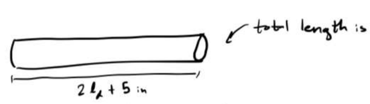
Channel: $W = 4\rho_d\left[\left(2w_dt_d+(w_d-2t_d)t_d\right)(l_d+2l_{tearout}) - 4\left(\frac{\pi}{4}d_{pin}^2\, t_d\right) - 2\left(2l_{tearout}(w_d-2t_d)t_d\right)\right]$

$\boxed{W = 1.499\ \text{lb}}$

Total Weight:

- Square Design: $\boxed{W = 2.335\ \text{lb}}$
- C-Channel Design: $\boxed{W = 1.959\ \text{lb}}$

Crossbar



$W = \rho_{cb}\left(\frac{\pi}{4}d_{cb}^2\right)(2l_d + 5\text{ in})$

$\boxed{W = 0.307\ \text{lb}}$

Pins

$W = 4\rho_{pin}\left(\frac{\pi}{4}d_{pin}^2\right)(d_{cb} + 4t_d + 2)$

$\boxed{W = .153\ \text{lb}}$

Cross bar Buckling

Assumptions:

- homogenous material under static loading
- For a pinned-pinned connection

$$C = 1$$

Determine Johnson or Euler

$$I_{CB} = \frac{\pi D^4}{64} = \frac{\pi (.25'')^4}{64}$$

$$I_{CB} = 1.91747\,E\text{-}4 \text{ in}^4$$

Now find Area

$$A_{CB} = \frac{\pi}{4} D^2 = \frac{\pi}{4} (.25 in)^2$$

$$A_{CB} = .049 \text{ in}^2$$

Find K

$$K = \sqrt{\frac{I_{CB}}{A_{CB}}}$$

$$K = .0625$$

Now Find $\left(\frac{l}{K}\right)_1$

$$\left(\frac{l}{K}\right)_1 = \sqrt{2}\,\pi \sqrt{\frac{C\,E}{S_y}} = \pi \sqrt{\frac{(2)(1)(30 \times 10^6 \frac{lbs}{in})}{140{,}000\,psi}}$$

$$\left(\frac{l}{K}\right)_1 = 65.037$$

$$\frac{l}{K} = \frac{12.961 in}{.0625 in} = 207.376$$

$$\left(\frac{l}{K}\right)_1 < \frac{l}{K}$$

Use Euler Criteria

$$P_{cr} = \left(\frac{C \pi^2 E}{(l/K)^2}\right) A$$

$$P_{cr} = \boxed{337.9667 \text{ lbs}}$$

Scissor Jack Assignment Verification

Safety Factors

$S_e' = 14$ ksi
$k_a = .6784$
$k_b = 1$ (axial)
$k_c = .85$ (axial)
$k_d = 1$
$k_e = .847$ (k_c)
$k_f = 1$

$\Big\} S_e = 7.241$ ksi

Diagonal – $S_y = 16$ ksi, $S_{ut} = 35$ ksi (aluminum) $\longrightarrow$ $S_{ut_{en}} = 145.83$ ksi $\therefore f = .7984$

$$b = \frac{1}{\log 10^3 - \log 5\cdot 10^8} \log\left(\frac{35 \cdot 0.7984}{7.24}\right)$$

$b = -0.1029$ $\quad a = \frac{.7984(35)}{(10^3)^{-.217}} = 56.8878$

$S_f(N = 7000) = a(7000)^b = 22.873$

Tearout: $\sigma_{vm} = 4684.4$ psi $= 2\sigma_a = 2\sigma_m$

$n_y = \frac{16000}{4684.4} = \boxed{3.42}$ $\quad n_f = \left(\frac{2342.2}{22873} + \frac{2342.2}{35000}\right)^{-1} = \boxed{5.91}$

Axial: $\sigma_{vm} = 2163.6$ psi $= 2\sigma_a = 2\sigma_m$

$n_y = \frac{16000}{2163.6} = \boxed{7.39}$ $\quad n_f = \left(\frac{1081.8}{22873} + \frac{1081.8}{35000}\right) = \boxed{12.79}$

Bearing: $\sigma_{vm} = 8654.5$ psi (compression) $= 2\sigma_a = 2\sigma_m$

$n_y = \frac{16000}{8654.5} = \boxed{1.85}$ $\quad n_f = \frac{S_f}{\sigma_a} = \boxed{5.29}$

Crossbar – $S_y = 32$ ksi, $S_{ur} = 70$ ksi $\xrightarrow{\text{(steel)}}$ $f = .898$

$S_f' = 35$ ksi
$k_a = .7458$
$k_b = 1$
$k_c = .85$
$k_d = 1$
$k_e = .847$
$k_f = 1$

$\Big\} S_f = 21.22$

$b = -\frac{1}{3}\log\left(\frac{f S_{ut}}{S_f}\right) = -0.157$

$a = \frac{f S_{ut}}{(10^3)^b} = 186.05$

$S_f = a(7000)^b = 46.29$ ksi

Bearing: $\sigma_{vm} = 27494$ psi (compression) $= 2\sigma_a = 2\sigma_m$

$n_y = \frac{32000}{27494} = \boxed{1.16}$ $\quad n_f = \frac{46291}{13747} = \boxed{3.37}$

Axial: $\sigma_{vm} = \frac{F_{cb}}{\pi/4\, d_{pin}^2} = 27494$ psi (compression) $= 2\sigma_a = 2\sigma_m$

$n_y = \frac{32000}{27494} = \boxed{1.90}$ $\quad n_f = \frac{46291}{27494} = \boxed{5.51}$

Pin – $S_y = 140$ ksi, $S_{ur} = 150$ ksi $\xrightarrow{\text{(steel)}}$ $f = .795$

$S_e' = 75$ ksi
$k_a = .674$
$k_b = 1$
$k_c = .85$
$k_d = 1$
$k_e = .847$
$k_f = 1$

$\Big\} S_e = 38.56$ ksi

$b = -\frac{1}{3}\log\left(\frac{f S_{ut}}{S_e}\right) = -0.163$

$a = \frac{f S_{ut}}{(10^3)^b} = 369.06$

$S_f(7000) = 86.78$ ksi

Shear (crossbar): $\sigma_{vm} = 14551.9$ psi $= 2\sigma_a = 2\sigma_m$

$\sigma_{vm} = \sqrt{3}\left(\frac{F_{cb}/2}{\frac{\pi}{4} d_{pin}^2}\right) = 14551.9$ psi $\quad n_y = \frac{140000}{14551.9} = \boxed{9.62}$ $\quad n_f = \left(\frac{7276}{86780} + \frac{7276}{150000}\right)^{-1} = \boxed{7.56}$

Shear (Diagonal): $\sigma_m = 9543$ psi $= 2\sigma_a = 2\sigma_m$

$n_y = \frac{140000}{9543} = \boxed{14.67}$ $\quad n_f = \left(\frac{4771.5}{86780} + \frac{4771.5}{150000}\right)^{-1} = \boxed{11.52}$

Bearing (crossbar): $\sigma_m = 27494$ (compression) $= 2\sigma_a = 2\sigma_m$

$n_y = \frac{140000}{27494} = \boxed{5.09}$ $\quad n_f = \frac{86780}{13747} = \boxed{6.31}$

Bearing (Diagonal): $\sigma_m = 8654.5$ psi (compression) $= 2\sigma_a = 2\sigma_m$

$\sigma_m = \frac{F_d}{2 d_{pin} t_d} = 8654.5$ $\quad n_y = \frac{140000}{8654.5} = \boxed{16.18}$ $\quad n_f = \frac{86780}{4327.3} = \boxed{20.05}$

A-5 - Numerical Model Additional Hand-Calculations

1st Hand Verification. Below is the model output, and then the hand-calculated values. Safety factors were confirmed.

Variable	**Value**									
d_cb	1.00		**Other Calculations**				**Stresses**	**Stress (psi)**	**n_y**	**n_f**
t_cb	0.75		h_prime	7.500			diagonal_tear	7794.23	8.10	9.59
w_cb	1.50	**l_d Constraint**	theta	0.589			diagonal_axial	6923.08	9.11	10.80
l_d	6.75	3.1	l_cb	11.225			diagonal_bearing	12857.14	4.91	9.45
w_d	2.00		A_pin	0.385			crossbar_bearing	2850.79	10.87	16.05
t_d	0.10		A_cb	0.785			crossbar_axial	3811.22	8.13	12.00
l_tearout	1.00		A_cb_pin	1.200			pin_shear_cb	6735.95	7.97	7.57
d_pin	0.70		A_d	0.260			pin_shear_d	4050.58	13.26	12.60
Sy_cb	31000.00		l_pin	3.400			pin_bearing_cb	2850.79	18.84	29.82
Sy_d	63100.00		A_square_bulk	0.760	0.01		pin_bearing_d	12857.14	4.18	6.61
Sy_pin	53700.00		A_channel_bulk	0.580	0.01					
E_cb	10000000.00						**Buckling**	**Pcr**	**F**	**n**
E_d	30000000.00		**Weight**				crossbar_buckling	20493.06	2993.33	6.85
Sut_cb	35000.00		Crossbar Weight	1.417	0.01			Johnson		
Sut_d	97200.00		Sq. Diagonal Weight	5.744	0.01					
Sut_pin	63800.00		C. Diagonal Weight	4.772	0.01					
rhocb	0.10		Pin Weight	1.486	0.01					
rhod	0.28		Sq. Total Weight	**8.647**						
rhopin	0.28		C. Total Weight	**7.675**						
isALCB	TRUE									
isALD	FALSE		**Forces**							3.05
isALPin	FALSE		F	2000.000			theta (deg)	33.75		2.66
			F_cb	2993.326						
			F_d	1800.000						

Hand Calc Verification Round 1

Diagonal – $S_y = 63.1$ ksi, $S_{ut} = 97.2$ (Steel) $\rightarrow f = 0.853$

$S_e' = 48.6$
$k_a = .7408$
$k_b = 1$
$k_c = .85$
$k_d = 1$
$k_e = .847$
$k_f = 1$

$\Big\}\ S_e = 27.45$ ksi

$b = -\frac{1}{3}\log\left(\frac{.853(97.2)}{27.45}\right) = -.16$

$a = \frac{.853(97.2)}{10^{3b}} = 250.44$

$S_f(7000) = 60.73$ ksi

Tearout:

$$\sigma_{vm} = \frac{F_d/2}{2L_{tearout}t_d}\sqrt{3} = \boxed{7794.2\text{ psi}}$$

$$n_y = \frac{63100}{7794.2} = \boxed{8.10} \qquad n_f = \left(\frac{3897}{60729} + \frac{3897}{97200}\right)^{-1} = \boxed{9.59}$$

Axial:

$$\sigma_{vm} = \frac{F_d/2}{t_d(w_d - d_{pin})} = \boxed{6923.1}\text{ psi}$$

$$n_y = \frac{63100}{6923.1} = \boxed{9.11} \qquad n_f = \left(\frac{3461}{60729} + \frac{3461}{97200}\right)^{-1} = \boxed{10.80}$$

Bearing:

$$\sigma_{vm} = \frac{F_d/2}{d_{pin}t_d} = \boxed{12857\text{ psi}} \quad (\text{compression})$$

$$n_y = \frac{63100}{6923.1} = \boxed{4.91} \qquad n_f = \frac{60729}{6428.6} = \boxed{9.45}$$

Crossbar – $S_y = 31$ ksi, $S_{ut} = 35$ ksi, $E = 10\times10^6$ psi (Aluminum) $\rightarrow S_{UTP} = 145.83 \rightarrow f = .798$

$S_f' = 14$
$k_a = .678$
$k_b = 1$
$k_c = .85$
$k_d = 1$
$k_e = .847$
$k_f = 1$

$\Big\}\ S_f = 7.24$

$b = \frac{1}{3 - \log 5\cdot10^8}\log\left(\frac{fS_{ut}}{S_f}\right) = -0.103$

$a = \frac{fS_{ut}}{10^{3b}} = 56.89$

$S_f(7000) = 22.87$ ksi

Bearing:

$$\sigma_{vm} = \frac{F_{cb}/2}{d_{pin}t_{cb}} = 2850\text{ psi} \quad (\text{compression})$$

$$\boxed{n_y = \frac{31000}{2850} = 10.87} \qquad \boxed{n_f = \frac{22870}{1425.2} = 16.05}$$

Axial:

$$\sigma_{vm} = \frac{F_{cb}}{\frac{\pi}{4}d_{pin}^2} = 3811\text{ psi} \quad (\text{compression})$$

$$\boxed{n_y = \frac{31000}{3811} = 8.13} \qquad \boxed{n_f = \frac{22870}{3811/2} = 12.0}$$

Buckling

$$S_r = \frac{4R_{cb}}{d_{cb}} = 44.9$$

$$\left(\frac{l}{k}\right)_1 = \sqrt{\frac{2\pi^2E}{S_y}} = 76.79 < S_r \therefore \text{Johnson}$$

$$P_{cr} = .785\left(31000 - \frac{1}{10\times10^6}\left(\frac{31000\cdot44.9}{2\pi}\right)^2\right) = \boxed{20473\ \text{lb}}$$

$$\boxed{n = \frac{P_{cr}}{F_{cb}} = 6.85}$$

Pin – $S_y = 53.7$ ksi, $S_{ut} = 63.8$ ksi (Steel) $\rightarrow f = .9$

$S_e' = 31.9$
$k_a = .811$
$k_b = 1$
$k_c = .85$
$k_d = 1$
$k_e = .847$
$k_f = 1$

$\Big\}\ S_e = 19.741$

$b = -\frac{1}{3}\log\left(\frac{fU_t}{S_e}\right) = -.155$

$a = \frac{fU_t}{10^{3b}} = 167$

$S_f(7000) = 42.5$ ksi

Shear (crossbar):

$$\sigma_{vm} = \frac{F_{cb}/2}{A_{pin}}\sqrt{3} = \boxed{6735.9\text{ psi}}$$

$$n_y = \frac{53700}{6735.9} = 7.97 \qquad n_f = \left(\frac{3368}{42500} + \frac{3368}{63800}\right)^{-1}$$

$$\boxed{n_f = 7.57}$$

Shear (Diagonal):

$$\sigma_{vm} = \frac{F_d/2}{A_{pin}}\sqrt{3} = \boxed{4050\text{ psi}}$$

$$n_y = \frac{53700}{4050} = \boxed{13.26} \qquad n_f = \left(\frac{2025}{42500} + \frac{2025}{63800}\right)^{-1} = \boxed{12.60}$$

Bearing (Crossbar):

$$\sigma_{vm} = \frac{F_{cb}/2}{d_{pin}t_{cb}} = \boxed{2851\text{ psi}}$$

$$n_y = \frac{53700}{2851} = \boxed{18.84}$$

$$n_f = \frac{42500}{1425.5} = \boxed{29.82}$$

Bearing (Diagonal):

$$\sigma_{vm} = \frac{F_d/2}{d_{pin}t_d} = \boxed{12857.14}$$

$$n_y = \frac{53700}{12857} = \boxed{4.18}$$

$$n_f = \frac{42500}{6428.5} = \boxed{6.61}$$

2nd Hand Verification. Below is the model output, and then the hand-calculated values. Safety factors were confirmed.

Variable	Value									
d_cb	1.00		**Other Calculations**				**Stresses**	**Stress (psi)**	**n_y**	**n_f**
t_cb	0.10		h_prime	18.000			diagonal_tear	14433.76	3.72	3.53
w_cb	0.60	**l_d Constraint**	theta	0.848			diagonal_axial	33333.33	1.61	1.53
l_d	12.00	3.1	l_cb	15.875			diagonal_bearing	26666.67	2.01	3.19
w_d	0.45		A_pin	0.049			crossbar_bearing	35276.68	0.88	2.72
t_d	0.10		A_cb	0.785			crossbar_axial	2245.78	13.89	42.79
l_tearout	0.40		A_cb_pin	0.070			pin_shear_cb	31118.49	1.29	1.12
d_pin	0.25		A_d	0.040			pin_shear_d	23523.37	1.70	1.48
Sy_cb	31200.00		l_pin	3.400			pin_bearing_cb	35276.68	1.13	1.62
Sy_d	53700.00		A_square_bulk	0.140	0.01		pin_bearing_d	26666.67	1.50	2.14
Sy_pin	40000.00		A_channel_bulk	0.115	0.01					
E_cb	29000000.00						**Buckling**	**Pcr**	**F**	**n**
E_d	29000000.00		**Weight**				crossbar_buckling	21811.88	1763.83	12.37
Sut_cb	73200.00		Crossbar Weight	6.469	0.01			Johnson		
Sut_d	63800.00		Sq. Diagonal Weight	1.923	0.01		diagonal_buckling_sq	5218.59	1333.33	3.91
Sut_pin	45000.00		C. Diagonal Weight	1.604	0.01			Johnson		
rhocb	0.28		Pin Weight	0.065	0.01		diagonal_buckling_c	4623.98	1333.33	3.47
rhod	0.28		Sq. Total Weight	**8.456**				Johnson		
rhopin	0.10		C. Total Weight	**8.138**						
isALCB	FALSE									
isALD	FALSE		**Forces**							3.05
isALPin	TRUE		F	2000.000			theta (deg)	48.59		2.66
			F_cb	1763.834						
			F_d	1333.333						

Hand Calc Verification Round 2

Diagonal – $S_y = 53.7\text{ksi},\ S_{ut} = 63.8$ (Steel) $\rightarrow f = 0.909$

$S_e' = 31.9$
$k_a = .811$
$k_b = 1$
$k_c = .85$
$k_d = 1$
$k_e = .897$
$k_f = 1$

$\Big\}\ S_e = 19.74\text{ ksi}$

$b = -\frac{1}{3}\log\left(\frac{.853(47.2)}{27.45}\right) = -.15$

$a = \frac{.853(47.2)}{10^{3b}} = 167$

$S_f(7000) = 42.5$

Tearout:

$$\sigma_{vm} = \frac{F_d/2}{2 l_{tearout} t_d}\sqrt{3} = \boxed{14433\text{ psi}}$$

$$n_y = \frac{53700}{14433} = \boxed{3.72} \qquad n_f = \left(\frac{7216.5}{42500} + \frac{7216.5}{63800}\right)^{-1} = \boxed{3.53}$$

Axial:

$$\sigma_{vm} = \frac{F_d/2}{t_d(w_d - d_{pin})} = \boxed{33333\text{ psi}}$$

$$n_y = \frac{53700}{33333} = \boxed{1.61} \qquad n_f = \left(\frac{16666}{42500} + \frac{16666}{63800}\right)^{-1} = \boxed{1.53}$$

Bearing:

$$\sigma_{vm} = \frac{F_d/2}{d_{pin} t_d} = \boxed{26667}$$

$$n_y = \frac{53700}{26667} = \boxed{2.01} \qquad n_f = \frac{42500}{13333} = \boxed{3.19}$$

Crossbar – $S_y = 31.2\text{ksi},\ S_{ut} = 73.2\text{ksi},\ E = 29\times10^6\text{ psi}$ (Steel) $\rightarrow f = .89$

$S_e' = 36.6$
$k_a = .787$
$k_b = 1$
$k_c = .85$
$k_d = 1$
$k_e = .897$
$k_f = 1$

$\Big\}\ S_e = 21.98$

$b = \frac{1}{-3}\log\left(\frac{fS_{ut}}{S_e}\right) = -.158$

$a = \frac{fS_{ut}}{10^{3b}} = 143.9$

$S_f(7000) = 48.05$

Bearing:

$$\sigma_{vm} = \frac{F_{cb}/2}{d_{pin} t_{cb}} = \boxed{35277\text{ psi}}$$

$$n_y = \frac{31200}{35277} = \boxed{0.88} \qquad n_f = \frac{48050}{17639} = \boxed{2.72}$$

Axial:

$$\sigma_{vm} = \frac{F_{cb}}{\frac{\pi}{4}d_{pin}^2} = \boxed{2245.8\ (\text{compression})}$$

$$n_y = \frac{31200}{2245.8} = \boxed{13.89} \qquad n_f = \frac{48050}{1122.9} = \boxed{42.79}$$

Buckling

$$S_r = \frac{4 l_{cb}}{d_{cb}} = 63.5$$

$$\left(\frac{l}{k}\right)_1 = \sqrt{\frac{2\pi^2 E}{S_y}} = 135.5 < S_r \therefore \text{Johnson}$$

$$P_{cr} = .785\left(31000 - \frac{1}{10\times10^6}\left(\frac{31000\cdot 44.4}{2\pi}\right)^2\right) = \boxed{21811}$$

$$n = \frac{P_{cr}}{F_{cb}} = \boxed{12.37}$$

Pin – $S_y = 40\text{ksi},\ S_{ut} = 45\text{ksi}$ (Aluminum) $\rightarrow S_{utp} = 187.5 \rightarrow f = .778$

$S_f' = 18$
$k_a = .642$
$k_b = 1$
$k_c = .85$
$k_d = 1$
$k_e = .897$
$k_f = 1$

$\Big\}\ S_f = 8.82$

$b = -\frac{1}{3}\log\left(\frac{fU_T}{S_e}\right) = -0.105$

$a = \frac{fU_T}{10^{3b}} = 72.3$

$S_f(7000) = 28.522$

Shear (crossbar):

$$\sigma_{vm} = \frac{F_{cb}/2}{A_{pin}}\sqrt{3} = \boxed{31118\text{ psi}}$$

$$n_y = \frac{40000}{31118} = \boxed{1.29} \qquad n_f = \left(\frac{15559}{28522} + \frac{15559}{45000}\right)^{-1}$$

$$\boxed{n_f = 1.12}$$

Shear (Diagonal):

$$\sigma_{vm} = \frac{F_d/2}{A_{pin}}\sqrt{3} = \boxed{23523\text{ psi}}$$

$$n_y = \frac{40000}{23523} = \boxed{1.70} \qquad n_f = \left(\frac{11762}{28522} + \frac{11762}{45000}\right)^{-1} = \boxed{1.48}$$

Bearing (Crossbar):

$$\sigma_{vm} = \frac{F_{cb}/2}{d_{pin} t_{cb}} = \boxed{35277\text{ psi}}$$

$$n_y = \frac{40000}{35277} = \boxed{1.13}$$

$$n_f = \frac{28522}{17638.5} = \boxed{1.62}$$

Bearing (Diagonal):

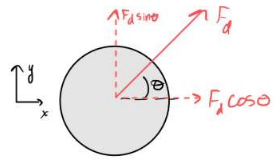
$$\sigma_{vm} = \frac{F_d/2}{d_{pin} t_d} = \boxed{26667\text{ psi}}$$

$$n_y = \frac{40000}{26667} = \boxed{1.5}$$

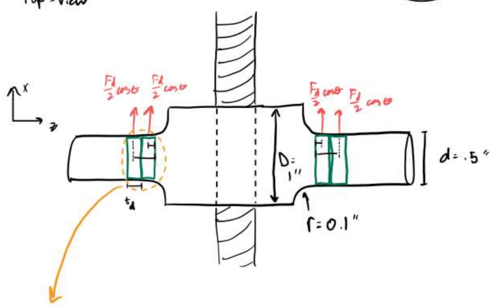
$$n_f = \frac{28522}{13333} = \boxed{2.14}$$

A-6 - Bending Stresses Hand-Calculations

Pen Bending



Top-View



$r/d = .2$

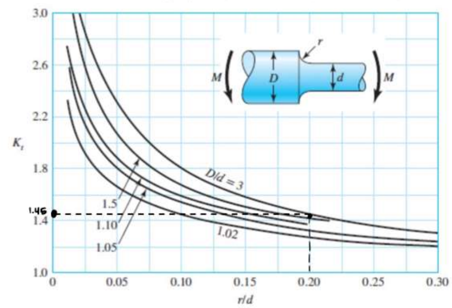
$D/d = 2$

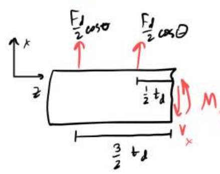
$F_d = 1632$ lbf

$t_d = .125''$

Figure A-15-9

Round shaft with shoulder fillet in bending. $\sigma_0 = Mc/I$, where $c = d/2$ and $I = \pi d^4/64$.





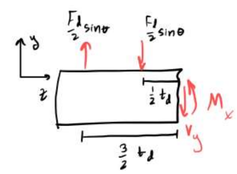
$$M_y = \frac{F_d}{2}\cos\theta\left(\tfrac{3}{2}t_d\right) + \frac{F_d}{2}\cos\theta\left(\tfrac{1}{2}t_d\right)$$

$$M_y = \frac{F_d}{2}\cos\theta\,(2t_d)$$

$$M_y = F_d\cos\theta\, t_d$$

$$M_y = 161.15 \text{ lbf in}$$

$$V_x = F_d\cos\theta = 1289 \text{ lbf}$$



$$M_x = \frac{F_d}{2}\sin\theta\left(\tfrac{3}{2}t_d\right) - \frac{F_d}{2}\sin\theta\left(\tfrac{1}{2}t_d\right)$$

$$M_x = \frac{F_d}{2}\sin\theta\, t_d$$

$$M_x = 62.5 \text{ lbf in}$$

$$V_y = 0 \text{ lbf}$$

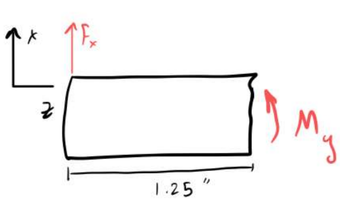
$$M_{max} = \sqrt{M_x^2 + M_y^2} = 172.84 \text{ lbf in}$$

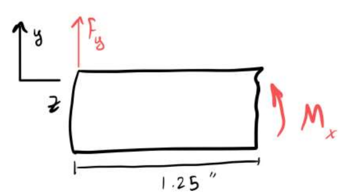
$$\sigma_0 = \frac{Mc}{I} = \frac{M(d/2)}{\pi d^4/64} = \frac{32M}{\pi d^3}$$

$$\sigma_0 = 14.08 \text{ ksi}$$

$$\boxed{\sigma_{max} = K_t\sigma_1 = 20.42 \text{ ksi}}$$

Equivalent Moments





$$F_x(1.25) = 161.15$$

$$\boxed{F_x = 128.92 \text{ lbf}}$$

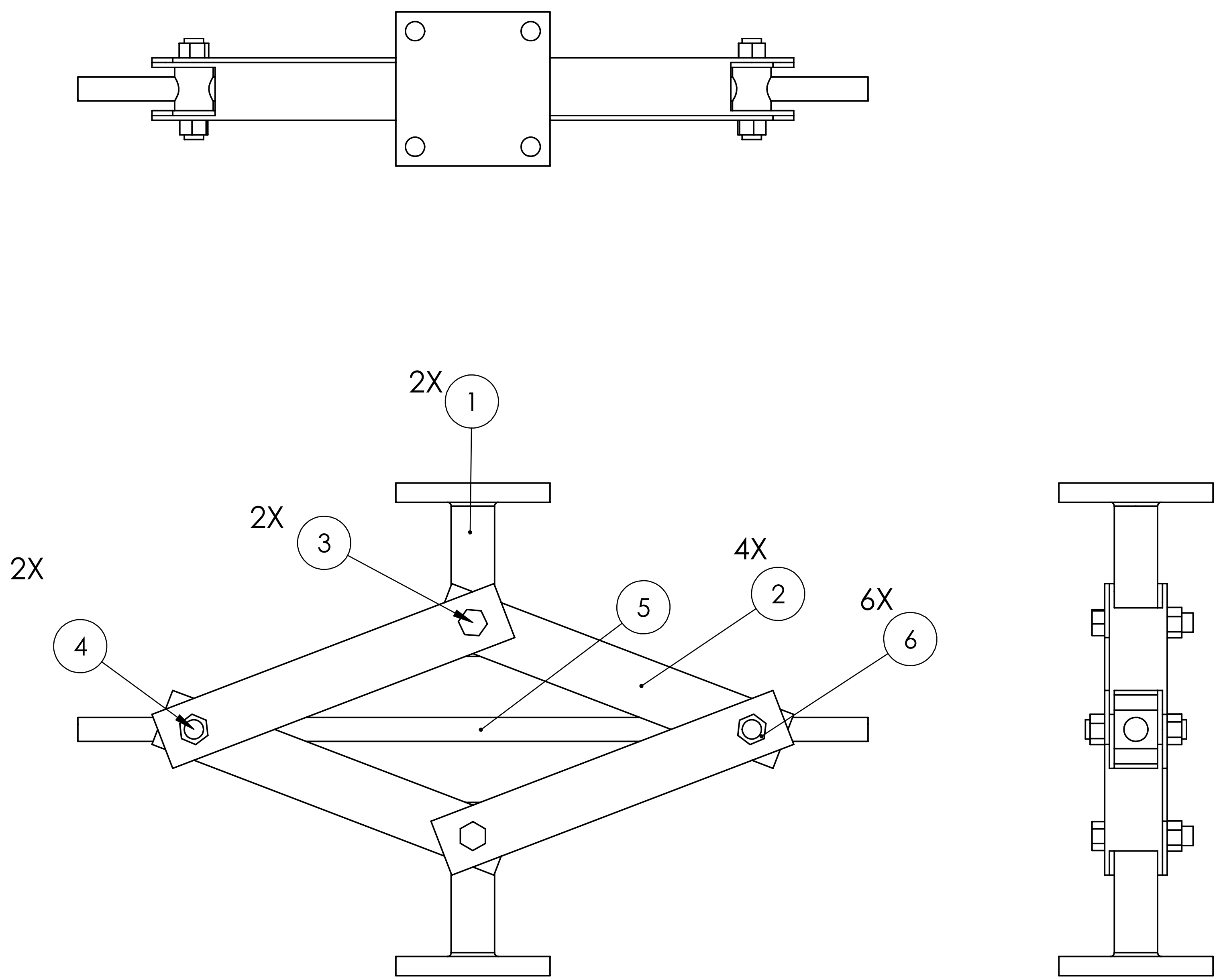
$$F_y(1.25) = 62.5$$

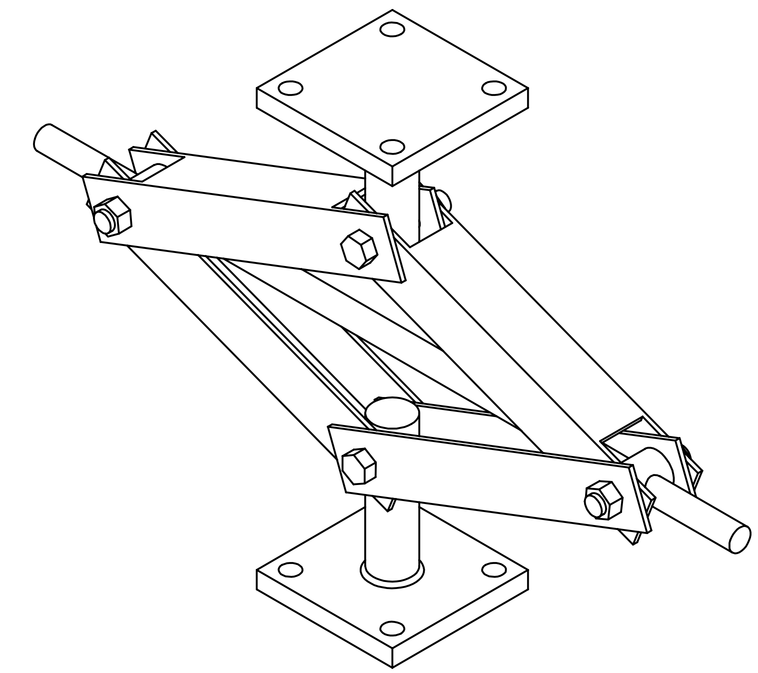
$$\boxed{F_y = 50 \text{ lbf}}$$

A-7 - Assembly Model, Bill of Materials, and Part Drawings

Below is a packet containing the assembly view, bill of materials, and part drawings of our final designed scissor jack. Please recognize that the diagonal members and crossbar pin are to be custom-made. As such, stock material will be purchased and the parts will be post-processed to meet the geometric specifications of our scissor jack. Please see Table A-8 in the appendix for available stock parts.

It was also specified that because we modeled the bolts and nuts as simple pins during our static failure and fatigue analysis, they would only be added to simulate a finished product and enhance visual appeal. As such, the bolts and nuts were custom-made in SOLIDWORKS and used to simulate an assembled, finished scissor jack. Although these bolts and nuts would be feasible and work well with our scissor jack, readily available bolts and nuts would be more practical and economical. Please see Table A-8 in the appendix for these available stock parts.





ITEM NO.	PART NUMBER	DESCRIPTION	MATERIAL	QTY.
1	P-005S	SCISSOR JACK MOUNT	1018 STEEL	2
2	P-001S	DIAGONAL MEMBER	6063-T5 ALUMINUM	4
3	P-003S	DIAGONAL BOLT	1018 STEEL - GRADE 5 BOLT	2
4	P-004S	CROSSBAR PIN	1018 STEEL	2
5	P-002S	CROSSBAR LEAD SCREW	304 STAINLESS STEEL	1
6	P-007S	NUT	1018 STEEL - GRADE 5 NUT	6

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
ANGULAR: ±1°
ONE PLACE DECIMAL ±0.1
TWO PLACE DECIMAL ±0.03
THREE PLACE DECIMAL ±0.005

INTERPRET GEOMETRIC TOLERANCING PER:

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	G. HAWS	15 APR 2024
CHECKED	J. WADE	15 APR 2024
ENG APPR.	G. HAWS	15 APR 2024
MATERIAL		
FINISH		

COMMENTS:

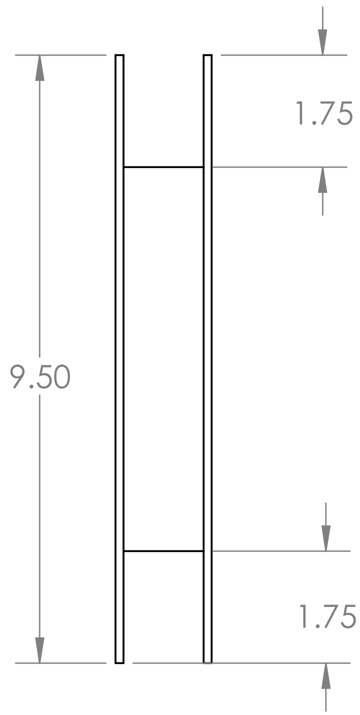
TEAM 2

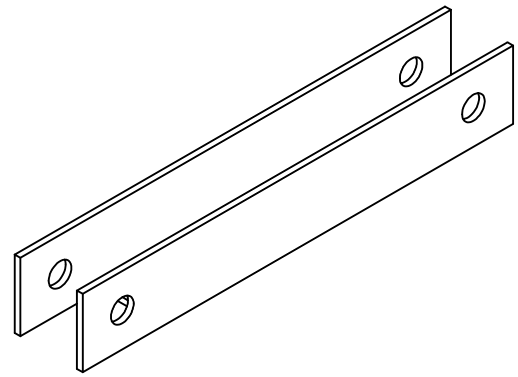
TITLE:

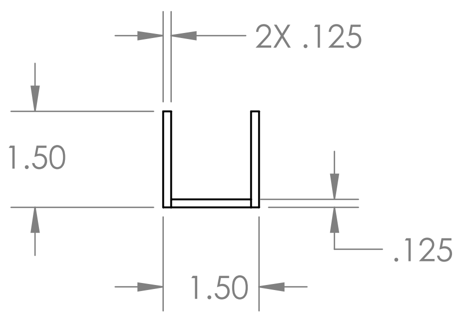
SCISSOR JACK ASSEMBLY VIEW

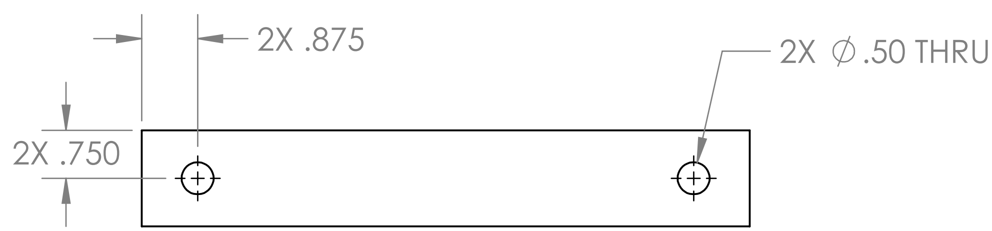
SIZE **B** PART NO. REV 1

SCALE: 1:4 DWG. NO. 001 SHEET 1 OF 1

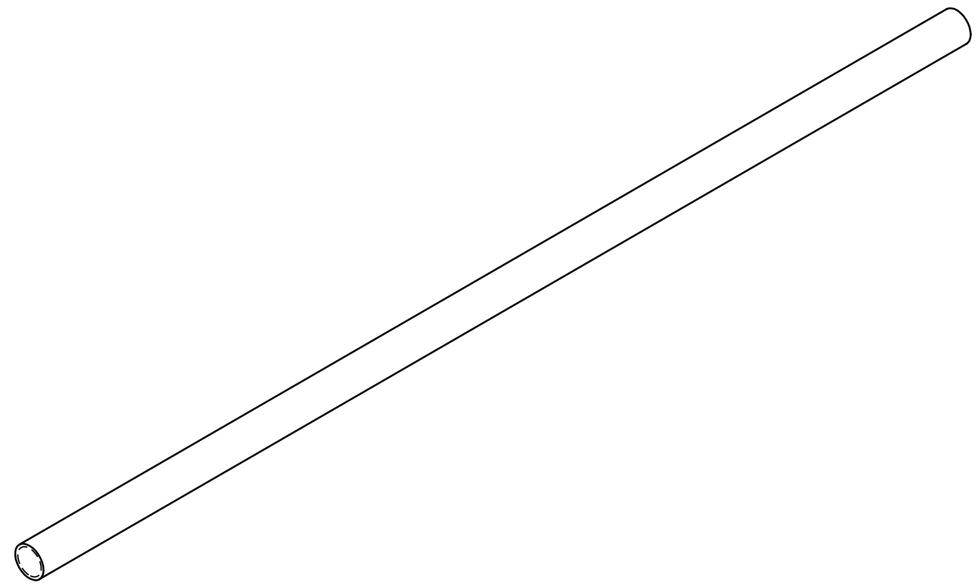


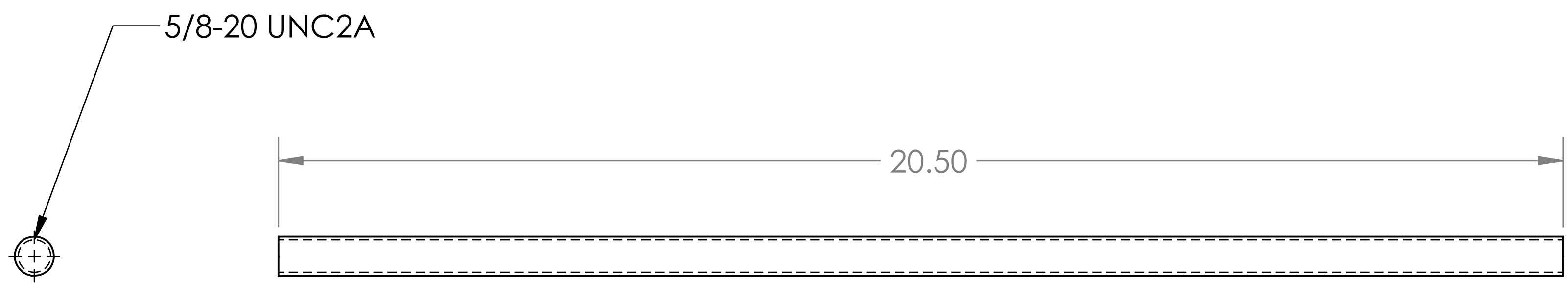




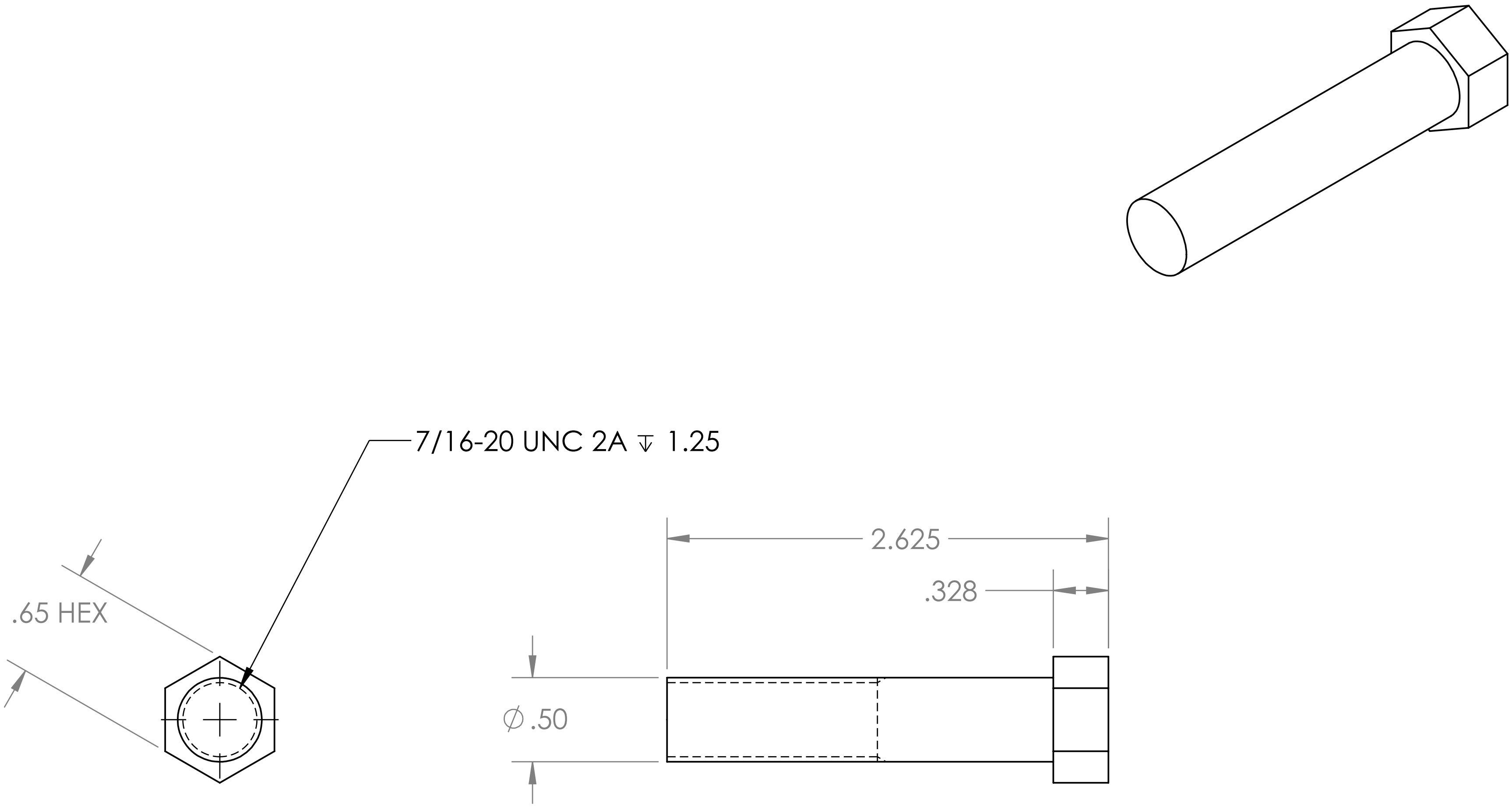


UNLESS OTHERWISE SPECIFIED:		NAME	DATE	TEAM 2
DIMENSIONS ARE IN INCHES TOLERANCES: ANGULAR: ±1° ONE PLACE DECIMAL ±0.1 TWO PLACE DECIMAL ±0.03 THREE PLACE DECIMAL ±0.005	DRAWN	G. HAWS	15 APR 2024	TITLE:
	CHECKED	J. WADE	15 APR 2024	DIAGONAL MEMBER
	ENG APPR.	G. HAWS	15 APR 2024	
	MATERIAL	6063 - T5 ALUMINUM		
INTERPRET GEOMETRIC TOLERANCING PER:	FINISH	POLISHED		
	COMMENTS:			SIZE A · PART NO. P-001S · REV 1
DO NOT SCALE DRAWING				SCALE: 1:3 · DWG. NO. 001 · SHEET 1 OF 1





UNLESS OTHERWISE SPECIFIED:		NAME	DATE	TEAM 2
DIMENSIONS ARE IN INCHES TOLERANCES: ANGULAR: ±1° ONE PLACE DECIMAL ±0.1 TWO PLACE DECIMAL ±0.03 THREE PLACE DECIMAL ±0.005	DRAWN	G. HAWS	15 APR 2024	TITLE:
	CHECKED	J. WADE	15 APR 2024	CROSSBAR LEAD SCREW
	ENG APPR.	G. HAWS	15 APR 2024	
	MATERIAL	304 STAINLESS STEEL		
INTERPRET GEOMETRIC TOLERANCING PER:	FINISH	POLISHED		
	COMMENTS:			SIZE A · PART NO. P-002S · REV 1
DO NOT SCALE DRAWING				SCALE: 1:3 · DWG. NO. 001 · SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
ANGULAR: ±1°
ONE PLACE DECIMAL ±0.1
TWO PLACE DECIMAL ±0.03
THREE PLACE DECIMAL ±0.005

INTERPRET GEOMETRIC TOLERANCING PER:

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	G. HAWS	15 APR 2024
CHECKED	J. WADE	15 APR 2024
ENG APPR.	G. HAWS	15 APR 2024
MATERIAL	1018 STEEL	
FINISH	POLISHED	

COMMENTS:

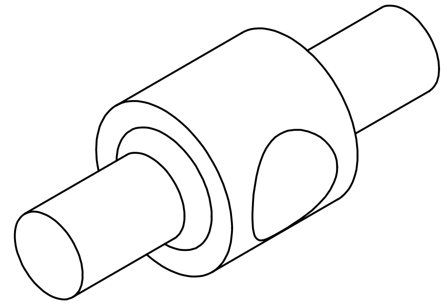
TEAM 2

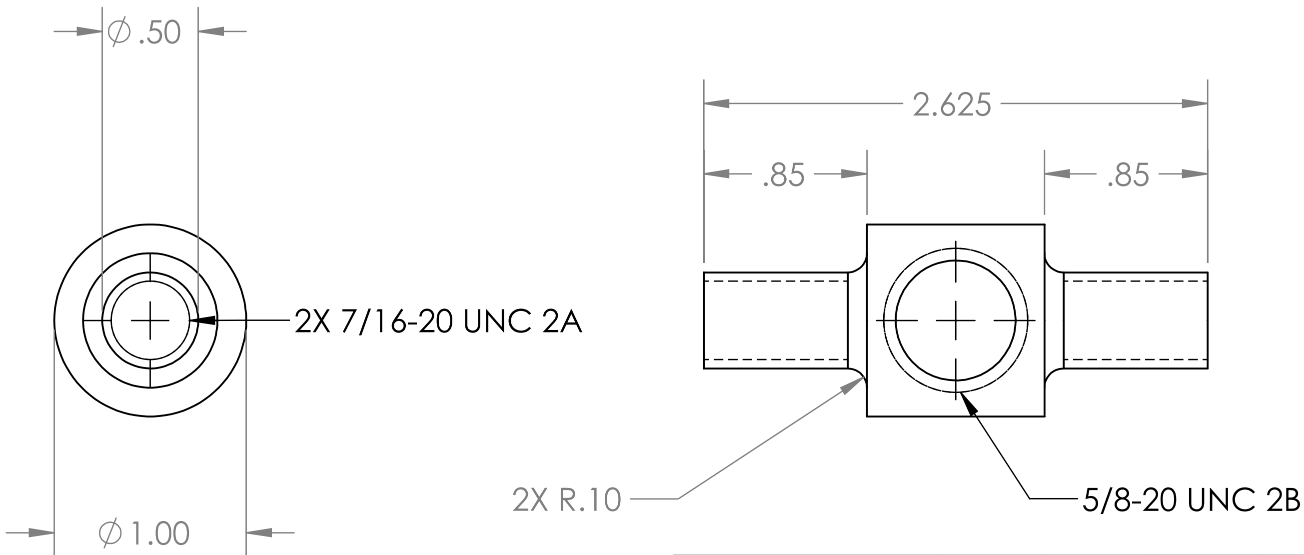
TITLE:

DIAGONAL BOLT

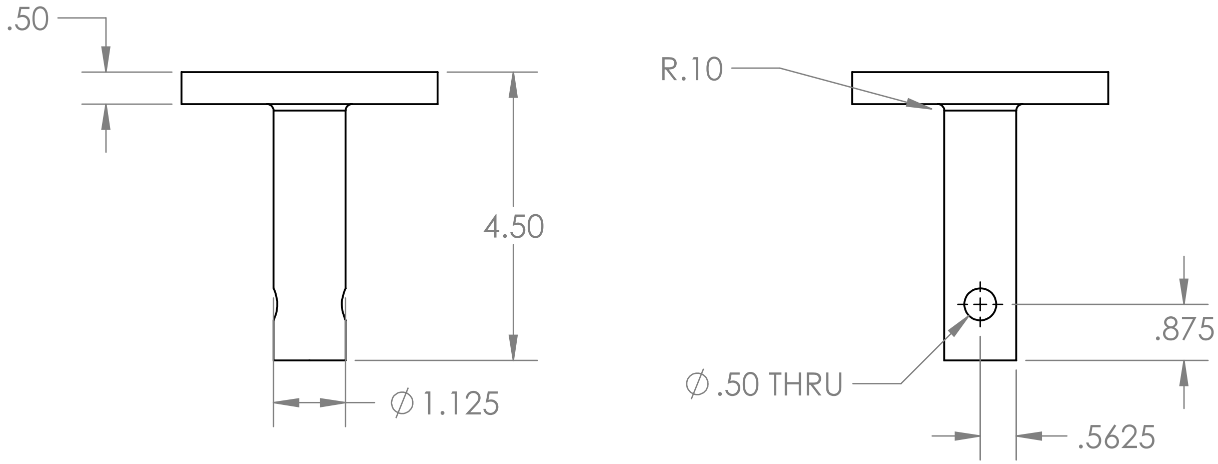
SIZE	PART NO.	REV
A	P-003S	1

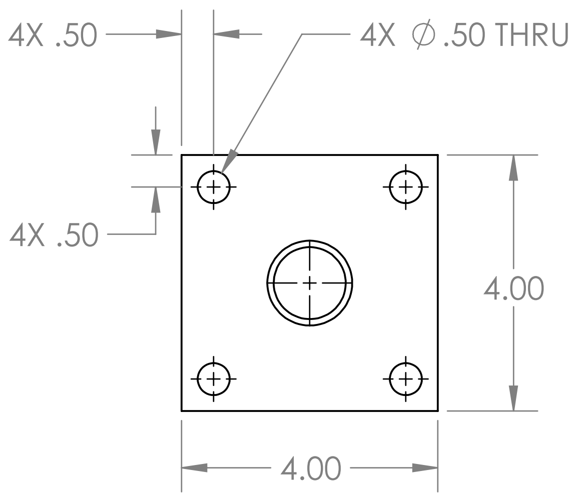
SCALE: 1:1 | DWG. NO. 001 | SHEET 1 OF 1

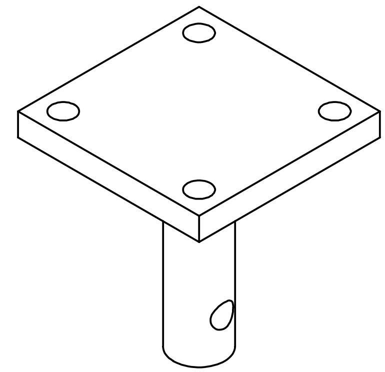




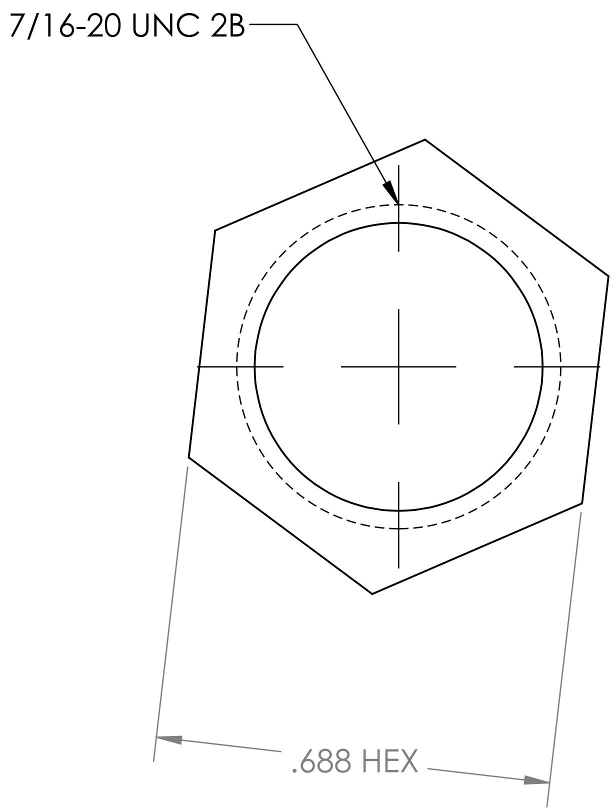
UNLESS OTHERWISE SPECIFIED:		NAME	DATE	TEAM 2
DIMENSIONS ARE IN INCHES TOLERANCES: ANGULAR: ±1° ONE PLACE DECIMAL ±0.1 TWO PLACE DECIMAL ±0.03 THREE PLACE DECIMAL ±0.005	DRAWN	G. HAWS	15 APR 2024	TITLE:
	CHECKED	J. WADE	15 APR 2024	CROSSBAR PIN
	ENG APPR.	G. HAWS	15 APR 2024	
	MATERIAL	1018 STEEL		
INTERPRET GEOMETRIC TOLERANCING PER:	FINISH	POLISHED		
	COMMENTS:			SIZE A · PART NO. P-004S · REV 1
DO NOT SCALE DRAWING				SCALE: 1:3 · DWG. NO. 001 · SHEET 1 OF 1

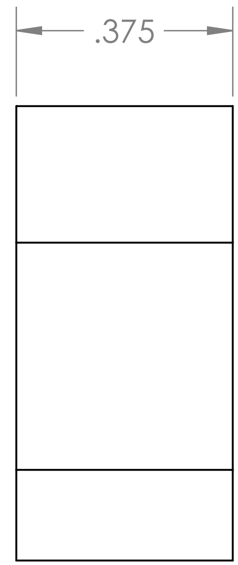


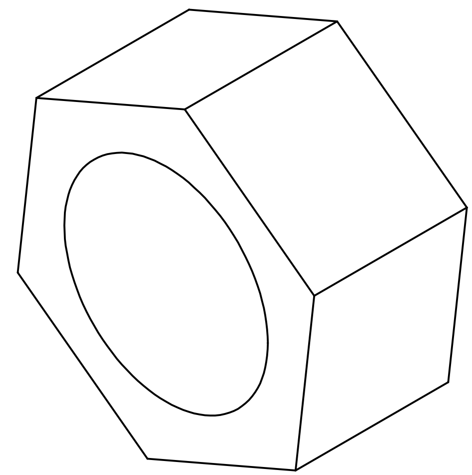




UNLESS OTHERWISE SPECIFIED:		NAME	DATE	TEAM 2
DIMENSIONS ARE IN INCHES TOLERANCES: ANGULAR: ±1° ONE PLACE DECIMAL ±0.1 TWO PLACE DECIMAL ±0.03 THREE PLACE DECIMAL ±0.005	DRAWN	G. HAWS	15 APR 2024	TITLE:
	CHECKED	J. WADE	15 APR 2024	SCISSOR JACK MOUNT
	ENG APPR.	G. HAWS	15 APR 2024	
	MATERIAL	1018 STEEL		
INTERPRET GEOMETRIC TOLERANCING PER:	FINISH	POLISHED		
	COMMENTS:			SIZE A · PART NO. P-005S · REV 1
DO NOT SCALE DRAWING				SCALE: 1:3 · DWG. NO. 001 · SHEET 1 OF 1







UNLESS OTHERWISE SPECIFIED:		NAME	DATE	TEAM 2
DIMENSIONS ARE IN INCHES TOLERANCES: ANGULAR: ±1° ONE PLACE DECIMAL ±0.1 TWO PLACE DECIMAL ±0.03 THREE PLACE DECIMAL ±0.005	DRAWN	G. HAWS	15 APR 2024	TITLE:
	CHECKED	J. WADE	15 APR 2024	NUT
	ENG APPR.	G. HAWS	15 APR 2024	
	MATERIAL	1018 STEEL		
INTERPRET GEOMETRIC TOLERANCING PER:	FINISH	POLISHED		
	COMMENTS:			SIZE A
				PART NO. P-007S
				REV 1
DO NOT SCALE DRAWING				SCALE: 3:1
				DWG. NO. 001
				SHEET 1 OF 1

Table A-8 - Catalog of Purchased Parts

Table A-8. A catalog of parts that are available for purchase. It is important to note that two prices were gathered for each specific part, which shows that we explored numerous options. For the diagonal members, crossbar pin, and crossbar (lead screw), stock material will be purchased and post-processed to meet the geometric requirements of our scissor jack. For this project, it was suggested that we ignore the cost and weight of the scissor jack attachment. As such, it has been disregarded.

Member	**Source**	**Part #**	**Description**	**Price In Bulk**
Diagonal Member	OnlineMetals	**7016**	1.5" x 1.5" x 0.125" Aluminum Channel 6063-T52 Extruded Architectural	$64.94/8ft
	McMaster-Carr	**9001K59**	1.5" x 1.5" x 0.125" Architectural 6063 Aluminum U-Channels	$71.51/8ft
Crossbar Pin	McMaster-Carr	**8920K231**	1" Low-Carbon Steel Rods (Stock)	$17.88/1ft
	MetalsDepot	**R21**	1" 1018 Cold Finish Steel Round (Stock)	$11.01/1ft
Crossbar	McMaster-Carr	**89535K11**	⅝" Cold Worked 304 Stainless Steel Rod (Stock)	$22.34/2ft
	OnlineMetals	**80**	0.625" Stainless Round Bar 304 Annealed Cold Finish (Stock)	$23.55/2ft
Bolt	McMaster-Carr	**91247A358**	½ "-20 x 2 ¾ " Medium-Strength Grade 5 Steel Hex Head Screws	$12.46/10 Bolts
	TannerBolt	**50F275HCS5Z**	½ "-20 x 2 ¾ " Grade 5 Hex Head Cap Screws, Fine Thread, Steel, Zinc Plated	$24.40/50 Bolts
Nut	McMaster-Carr	**95462A525**	½"-20 x ¾" Medium-Strength Steel Hex Nuts—Grade 5	$24.52/100 Nuts
	BoltDepot	**2579**	Hex nuts, Zinc plated grade 5 steel, ½ "-20	$0.30/Nut

Table A-9 - Estimated Work Done by Members

Table A-9. A summary of the estimated work done by each team member. It is important to recognize that the work was shared evenly and each team member contributed to the best of their ability given their circumstances.

Team Member	Estimated Percentage of Work Done
Garrett Haws	25%
James Wade	25%
Luke Severson	25%
Calvin Clawson	25%