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Compact Electric Bi-directional Pulley Actuator

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Abstract

While significant work has been dedicated to the study of potential uses of in situ resources once mined, the methods to mine the raw materials has not been fully explored. This paper presents the first attempt at producing a drop in replacement for hydraulic cylinders on heavy construction equipment with an electromechanical actuator utilizing pulleys for the conversion of a single motor's rotational motion to bidirectional linear motion. By utilizing a twin block and tackle arrangement and paired spools, the system can generate bidirectional motion nearly identical to a hydraulic cylinder. More importantly, the use of rope builds in shock resistance to the system which is often a weak point for screw style actuators. Significant work remains to refine the concept into flight hardware, but the concept shows promise. The produced actuator can be applied or scaled for near direct replacement of hydraulic cylinders on terrestrial mining and construction equipment for faster development of off Earth use.

Introduction

There is a need for an abrasive resistant and temperature tolerant version of hydraulic cylinders to power the industrial size construction equipment needed for sustainable habitation away from Earth.

As NASA looks towards sustainable habitation of the Moon and Mars, the question of in situ resource use has become less of a question and more of a requirement. We simply cannot transport enough material from Earth to maintain human presence on these celestial bodies. In light of that, substantial effort has gone into the utilization of in situ resources and ways to gather those resources. However, the Phoenix Lander remains the leader in off Earth digging depth at 15cm and struggled with digging into ice, utilizing a scrapping action rather than a digging action to loosen material for analysis.[4] This is completely insufficient for a mining operation that needs to generate meaningful amounts of water ice and regolith. Therefore, more industrial size equipment needs to be developed to enable bulk material processing to support in situ utilization of resources.

The issue with hydraulics is that most hydraulics are powered through a central pump and hose network. They often run in the 21MPa range for operating pressure with a hydrocarbon based working fluid. The lunar surface has a temperature range of -233ºC to 123ºC[1], which makes it extremely difficult to use a single kind of hydraulic fluid without major variations in viscosity or even freezing. To compound this issue, the seals need to work in the lower temperatures, not outgas in vacuum, and resist the severe abrasion of the lunar regolith. One way to get around this issue is to use the hydraulic fluid as a heating fluid to keep all the fluids and seals within a consistent temperature range. However, this takes substantial power to continuously run, and does not address the regolith's abrasion. Additionally, the hydraulic fluid, seals, and tubing add a host of spare parts that would need to be replaced as the Lunar regolith cuts them apart, which would have to be shipped with the equipment due to infrequent resupply missions.

For this project, the goal is to build a linear actuator based on pulleys in the block and tackle style. By utilizing a twin block and tackle, bidirectional movement can be achieved. This will allow for the elimination of the hydraulic systems on construction equipment with the least amount of equipment design changes possible. The reason to develop a drop-in replacement for construction equipment is to enable the easier conversion of any construction equipment for use in space. This is critical in recycling the last few hundred years of construction equipment design experience which can not only save money and time, but also improve the reliability and efficiency of the equipment that gets developed.

By switching to electric drive, this design would eliminate the issues with exposed high-pressure hoses, eliminate high-pressure seals, and increase temperature resilience. Additionally, by keeping the original operating parameters of the selected hydraulic cylinder, the concept will be proven to work as a drop-in replacement for already designed specialty equipment.

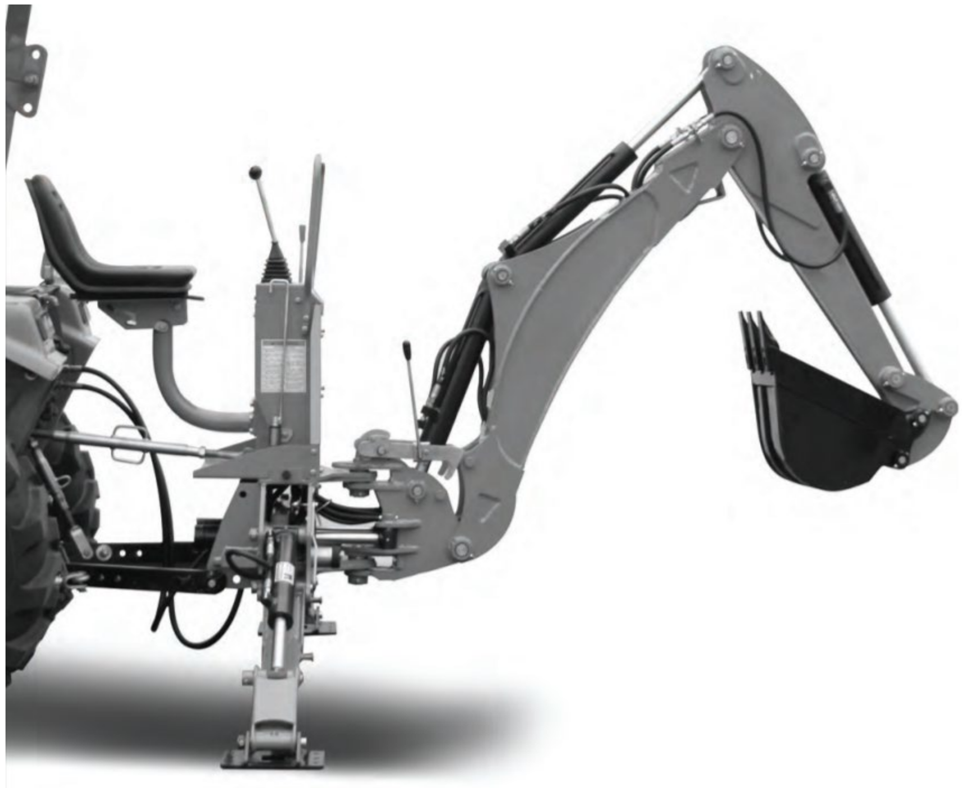

Fig. 1 Manufactured BK215 Arm Frame

As shown in Figure 1, the BK215 tractor mount excavator is a lightweight and affordable implement. As explored later in this report, it provides a good basis for the design due to available manufacturer specifications. The actuator was modeled from the hydraulic cylinder powering the dipper arm.

The reasoning behind the choice of pulleys is that they work in the same manner as hydraulics by trading distance traveled for greater force. This is not normally visible in hydraulic systems because the hydraulic pump is acting as an infinite cylinder, but the principle remains the same. Therefore, we can reduce the torque requirements for the motor which reduces the frictional losses on a worm gear reduction while also building in a natural brake similar to the effect of stopping the flow in a hydraulic cylinder.

Design Description

The project was focused primarily on the design, testing and applications of an electric double-pulley actuator and not necessarily on the design of the platform that it will be attached to, but investigation into the applicability of such high forces was also performed. By selecting the BK215, the team was then able to take the parameters surrounding the hydraulic components of the excavator arm and the actuator to match. The dipper cylinder was chosen as the baseline target due to it having the smallest mechanical space of the double acting cylinders.

Table 1: BK215 Hydraulic Specs

Cylinder	Retracted Length (inch)	Stroke (inch)	Mechanical Space (inch)	Force (lbf)
Boom	24.4	15.12	9.29	13779
Dipper	23.5	14.84	8.66	10148
Bucket	21.06	11.7	9.29	10148
Slew	15.6	8.78	6.89*	10148

*Slew cylinders are single direction only

Fig. 2 Excavator with Actuator attachment

Actuator

The mechanical aspect of the actuator is broken into a few main assemblies: the inner and outer sleeves, center carriage, spooling assembly, and pulley assembly. As for material, 304 stainless steel was chosen as the primary frame material due to a number of factors, despite its density and difficulty to machine. The two primary considerations were the ability to design for the fatigue limit so that the larger and heavier frame can remain operational indefinitely and its compatibility with cryogenic temperatures. Furthermore, most of the structure is loaded in compression and due to the tight packing of the components the strength to volume ratio was extremely important. While aluminum is often the default material for aerospace, its comparatively poor compressive strength and tendency to cold weld made it a poor fit for this application. While stainless steel is quite dense and thus the prototype extremely heavy, the strength of the steel allows for significant material removal in future development without compromising the frame’s strength. Aggressive weight reduction is recommended, as is working closely with the CNC operator to optimize the machining of the cut outs to match the machine and tool characteristics in order to avoid exponential cost increases due to the difficulties in machining 304.

Pulley Assembly

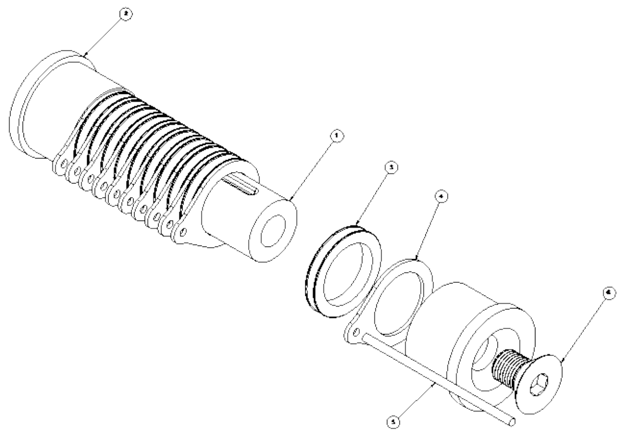

Fig. 3 Pulley assembly: Pulley shaft (1) Shaft Nut (2)
Pulley (3) Liner (4) Guide pin (5) Shaft bolt (6)

Pulley bearing selection

A few of the options investigated for the bearing selection include ball, roller, needle, thin, and plain. Due to the tight size constraints of the system, see mechanical space section, the pulleys are limited to around 1.375 inches outer diameter. This drove a majority of the design and limits the load capacities for bearings severely due to the size of the bearings in relation to the load exerted on the pulley. These are the specifications and calculations of bearing based on design and material. The formula of bearing load C_{eq} can be found in Reference [5], as shown below Table 2.

- Max RPM for pulley: 860 rpm
- K_a for light impact:1.5
- F_e load per pulley: 6300N
- $K_r L_R$ industry standard: $90\text{x}10^6$
- Life L_{10}:80 hours minimum

Table 2: Bearing options

Bearing	ID (mm)	Width	Max P (N/mm^2)	Max Velocity (m/s)	RPM	C_r
K5B0805	8	5	49	0.65	1241	3849
MDZB18-12*	18	5	49	0.65	690	6927
MBFS30-2.1	8	5	N/A	N/A	33,000	1990

*MBZD18-12 has a 12mm width when purchased, but calculations are for 5mm width after being cut to size.

$$C_{eq} = F_e K_a \left(\frac{L}{K_r L_R}\right)^{0.3}$$

$$C_{eq} = (6300)(1.5)\left(\frac{4.13}{90}\right)^{0.3} = 3750\text{N}$$

$$L = \text{rpm} * 60 * \text{hours} = 860 * 60 * 80 = 4.13\text{x}10^6$$

Through life analysis, it was determined that only plain bearings would fit the design envelope, while operating for 80 hours (L_{10}) with a 1400 pound-force radial load. The axial load is insignificant. The closest options were plain bearings made of bronze, but it made the pulley section very thin and difficult to manufacture. Therefore, the decision was made for the pulleys to be made from SAE954 bronze, with the pulley acting as its own bearing. These were remarkably easy to manufacture from bronze tube and simplified the parts count. While this isn't ideal for a long service life, the abrasive nature of the Lunar dust makes even two weeks of operation a significant achievement and the larger bearing surface expands the service life if the effects of the dust can be mitigated.

It should be noted that a major idea behind this design is for periodic rebuilds of the actuators inside a clean and pressurized habitat. This allows for the smaller and lightweight internal components like the rope and pulleys to be replaced while reusing the heavier frame elements.

Rope

Dyneema® rope is currently the best choice of material due to high strength, tight bend radius, and performance from testing on the International Space station[3]. It does not handle snags very well and the strength degrades significantly in the higher ranges of Lunar temperature, so it will need to be shaded from direct sun exposure and kept isolated from the Lunar dust as much as possible. The specific choice is currently Marlow EXCEL D12 MAX 78, 3mm diameter.[B] Dyneema® is recommended to have a minimum 10:1 bend diameter to rope diameter ratio, so a 1.25in minimum OD for the pulley groove fits this perfectly. Additionally, the 3mm diameter was the smallest that we could procure at the time, but at a 3500lb breaking strength for a 700lb design load there is ample safety factor built in for shock loads and abrasive wear.

Kevlar-49 is an alternative material to Dyneema® due to high strength, high temperature ceiling, and abrasion resistance. However, it is not the first choice because it requires a larger bend radius which is a critical factor in making the actuator work as a drop-in replacement.
Steel cable was not a suitable choice because of the very large bend radius needed and higher mass.

Pulley Support

The pulley shaft as seen in fig. 3 was made in two versions, one in 304 Stainless steel and the other in 17-4 stainless steel. The 304SS was cold rolled to a 73.9KSI yield strength and 203 Brinell hardness which is not ideal for shaft material, but a good control until better material is tested. The 17-4SS was used as a substitute for Stellite 6B since it is easier to machine and drastically cheaper. Stellite 6B could be an ideal material due to its low friction characteristics, hardness, and yield strength, but needs further research into cryogenic properties.[24] The shaft is 1in OD. While a nearly solid 0.75in OD shaft can theoretically hold the designed loads, in simulations failure occurred at the stress concentration where the shaft was supported by the shaft nuts. With a 1in OD shaft the stress concentration was acceptable and a larger 7/16in ID could be used for the grease reservoir and pre-threading pilot hole. The grease exits the shaft through a linear pattern of holes for distributing the grease under the pulleys. Grease is added through one of the shaft bolts(6) that has a grease zerk countersunk below the hex drive. While removing so much material from a ½-20 bolt does weaken it significantly, these only provide axial alignment. The shaft nuts (2) provide the radial support and transmit the load from the shafts to the inner sleeve or center carriage side plate. These are needed because the target load for the actuator exceeds the compressive yield strength of the3/8in plate thickness of the inner sleeve and center carriage at a 1in OD shaft. The shaft nuts increase the effective diameter for the load transfer to the plate while increasing the effective thickness for the load transfer from the shaft, and the3/16in deep step controls the axial alignment. Lastly the liners (4) are made from 1/16in thick PTFE sheet, die cut with a leather clicker, and works to reduce friction between adjacent pulleys but primarily works with the guide pin (5) to control the rope when the system is not under tension like during assembly. This is a critical element discovered in early 3D printed test assemblies because of the limited access to the center carriage during operation and the tendency for the rope to 'jumble' if not carefully controlled. A recommended improvement is to add retention capabilities to the guide pin, and one possibility for that is to use a long bolt.

Frame

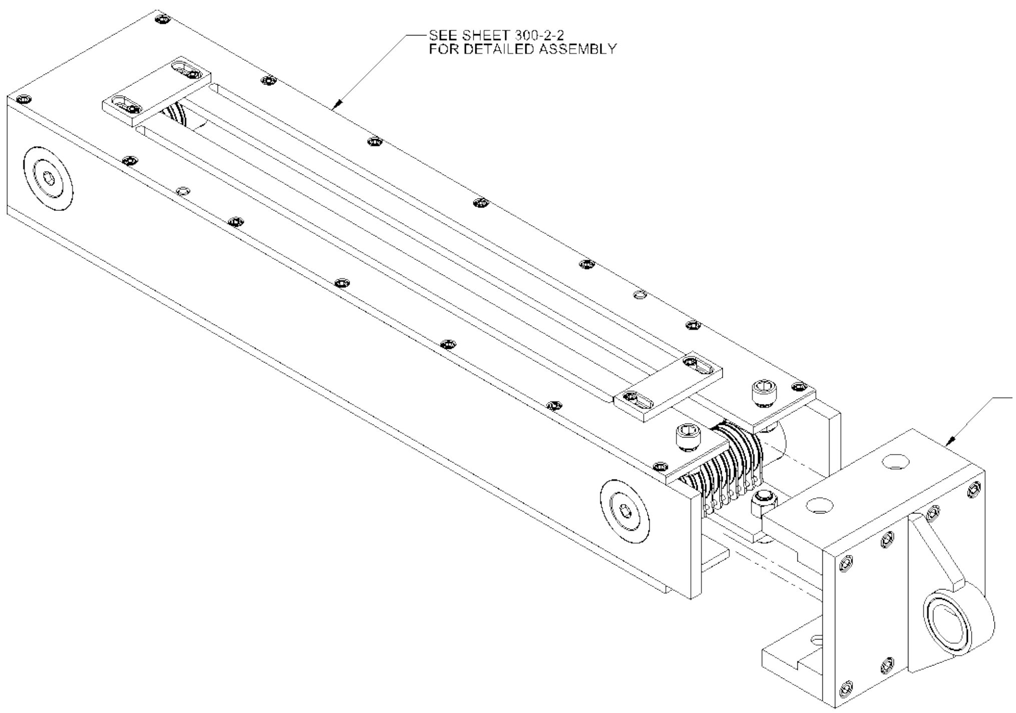

Fig 4 Outer sleeve assembly

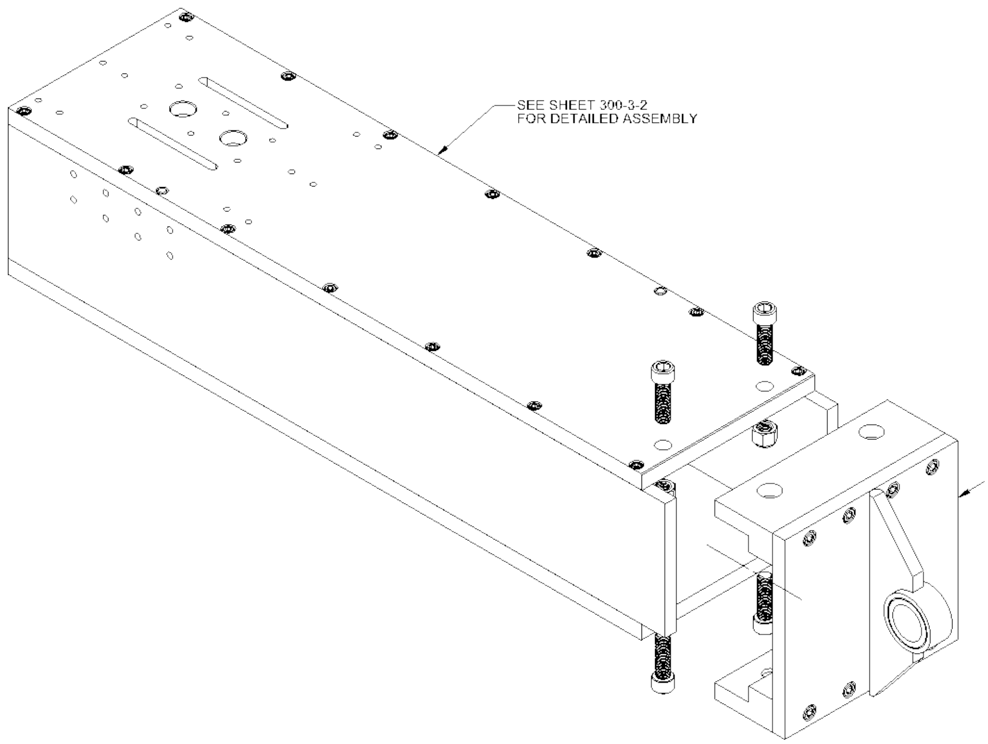

Fig 5 Outer sleeve assembly

There were a few major design deviations between the actuator as drawn, and the actuator as produced. First, the stainless 304 plate was replaced by A36 due to difficulties encountered during machining. Second, the plates were designed to bolt together as seen in figs 4 & 5, but tack welds were used in a similar frequency due to time limitations. Lastly, the plates used still had a mill scale and waterjet finish rather than being fully machined. All performance evaluation should account for the variances in material characteristics and manufacturing techniques.

The side plates are all 3/8in thick plate as a result of the high load being transferred from the shaft nuts to the plates, however this is overbuilt for most of the structure. As seen in fig 6, there is a machined counterbore for the shaft nut alignment. This complicates the manufacturing into a 2.5D operation when everything else is a 2D operation. Reducing the plate thickness and adding in a weld on thickener plate would simplify the manufacturing process and reduce the mass significantly. Additionally, it is recommended to skeletonize the frame through generative design and/or add maintenance holes in the outer sleeve for grease and removing the shaft nuts to allow for full removal of the pulley assemblies without detaching the center carriage from the outer sleeve. A similar mass to a hydraulic cylinder should be possible with these aggressive mass reductions, but it does increase the machining time beyond what this project could afford.

The inner sleeve shown in fig 4 has three long channels cut out of it which does reduce the strength; however, it has under 0.005in of deflection at full load which is more than acceptable for this application. The purpose of the wider middle channel is clearance for the center carriage attachment, and the narrower side channels for the two ropes to pass up from the pulleys to the spooling assembly. The channels must run nearly the entire length of the inner sleeve to maximize the travel distance.

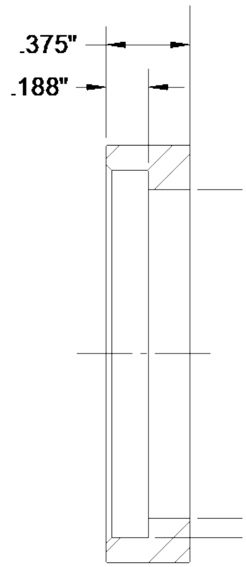

Fig 6 Sleeve counterbore for shaft nut

Center Carriage

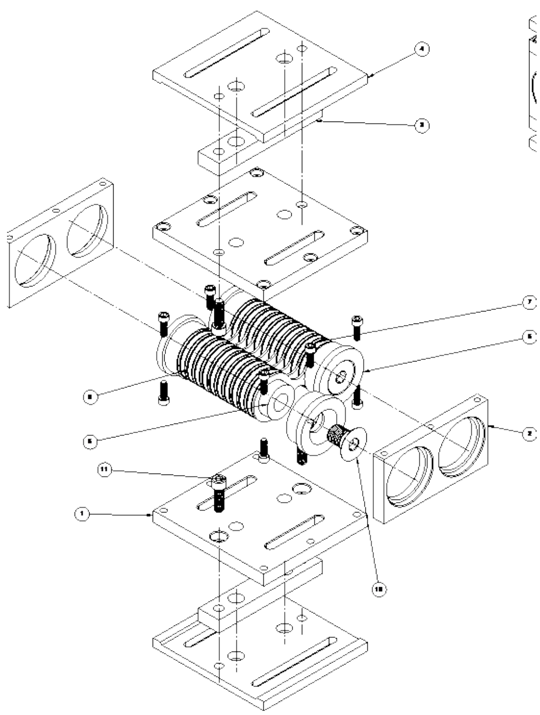

Fig 7 Center carriage assembly

The center carriage as shown in fig 7 is a critical element of the system, as it is the anchor point for both block and tackle assemblies and transmits the load to the outer sleeve enabling the bidirectional movement. The core box of the center carriage as designed is 304 stainless steel, but as previously mentioned the actual hardware ended up as A36 for machinability. On either side of the core box are two pieces of SAE954 bronze, which are the guides and wear surfaces for the inner sleeve to slide against. These are designed to be replaceable, similar to the pulleys and rope. The only major design change would be to reverse the ¼-28 bolt that holds the bronze to the core box as it is difficult to access once the rope is installed.

Spooling Assembly

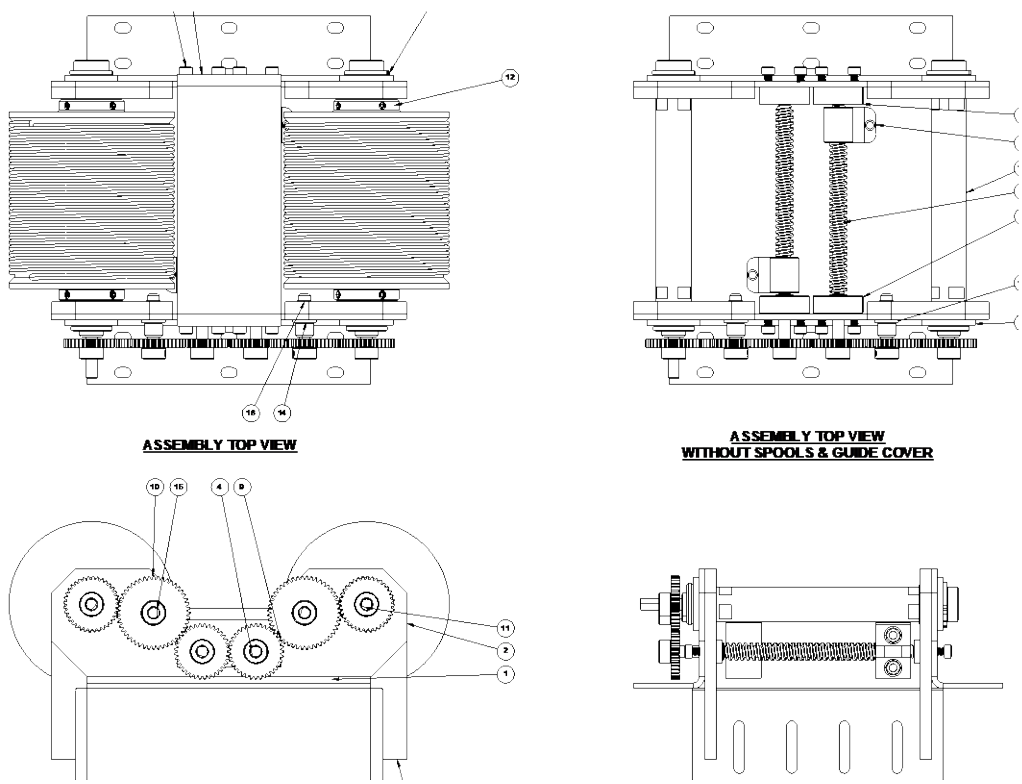

Fig 8 Spooling assembly

The spooling assembly as shown in fig 8 is a complex system to allow for a single motor to operate the system in both directions and reduce the error from using multiple motors. This design has substantial weaknesses, but was designed for the internal manufacturing capabilities available, present component supply, and some unknown issues that later arose. The fundamental design is a pair of timed spools that simultaneously spool and unspool at a consistent rate. While this design could be done with a single spool, the paired spool design was chosen to test the capability to enable uneven forces similar to actual hydraulic cylinders. This would be accomplished by having different sized block and tackles and using an appropriate ratio for the spools. The lead screws near the center control the spooling guides, which keeps the rope spooling evenly and organized on the spools in order to maximize the spool capacity while keeping only a single layer on the spool. This is critical, since two or more layers would change the effective spool diameter and cause the two pulley systems to be mistimed.

As mentioned this design had a number of compromises that weaken the system. Recommendations for the next iteration would be a fully enclosed gearbox that incorporates the motor’s worm reduction and gears better sized for the forces involved. A critical mistake that wasn’t caught in time was that while the gear train reversed the direction of rotation for the passive spool as designed, the location of the spool in relation to the spooling guide means the reversed rotation actually causes both spools to spool or unspool at the same time. This could be fixed by reducing the spooling guide to a single leadscrew and/or by using a chain drive inside the gearbox. Additionally, this design relied heavily on set screws, but future hardware should be made with keyways for better load transmission.

The frame for the spooling system as designed was made from 8 pieces, however we cut that down to two during production by using 3/8in plate with a bolt on flange. This preserved the ability to perform maintenance, while strengthening the face plates around the lead screws enabling better power transmission. The final design recommendation for the spooling assembly is to use hollow aluminum spools. Solid pieces of 304 stainless steel were used for ease of manufacture, but hollow tube with end plates is a much more efficient use of mass despite the added complexity. The multipart spool would also enable a more resilient clamping mechanism to hold the rope in place.

Electronics and thermals

The electrical system is built around the Brushless DC (BLDC) motor. To achieve parity with the original hydraulic system, a minimum of a 3kw BLDC motor is needed. This is based on the load of 11,000 lbs at 15.12in of extension over 6 seconds, or 3.13kw of power. In practice however, the Maxon EC90, 600W motor is the preferred choice for their experience with other challenging environments, but due to the international shipping situation from the pandemic a 2450W 6354 BLDC electric skateboard motor was used.

To control the BLDC motor, an electronic speed controller (ESC) with hall effect sensor capabilities is needed. The VESC[A] 6 MkV was the preferred ESC due to its open source design and easy programming through the companion application, VESC Tool. For input into the ESC, a simple servo tester will be used to provide PPM input and the ESC is programmed to use the input as the motor rpm target. This avoids the complexity of custom programming through an Arduino board, however the VESC does have the ability to be controlled and monitored through a control system. The VESC was set up to run in field oriented control (FOC) while monitoring the motor with the hall effect sensors. This enables the ESC to run the BLDC motor with greater efficiency and to automatically increase current to match the required torque to maintain the target rpm. Additionally the VESC has built in over current protection and temperature monitoring for a stall or overheating.

Connectors will be standard from the radio-controlled vehicle industry since most of the equipment is designed for versatility and modularity. The VESC to BLDC wires will be joined with 6mm bullet connectors, power will be distributed to the VESC with 10-gauge copper wire with XT90 connectors, and the hall effect sensors will connect to the VESC board through the surface mounted JST, 6 pin, 2.0mm pitch connector. While insufficient for flight hardware, this saves costs during development and keeps a wider supply chain available which helped to alleviate the numerous supply issues created by the pandemic.

For heat sink design, the team has taken inspiration from the design used on the Lunar Rover.[12] The rover utilized a box of wax which was configured to the motor through thermal straps. The 1100 Aluminum thermal straps were to convey heat from the battery, through the radiator, and to the fusible mass tank of paraffin wax. Even with the 87% efficiency of a Maxon EC90 motor, we will still need to dissipate ~80W of heat when at full power. That converts to

4.8kJ/minute of heat and based on just paraffin's heat of fusion (solid to liquid) ~150kj/kg[5], a full 8 hour day of continuous operation without cooling would require 15.4kg of paraffin. In practice however, the needed amount is much lower because the motor wouldn't be operating at full power the entire time, and thus wouldn't require the maximum amount of paraffin because of the reduction in excess heat generated. This is an amount of paraffin that could easily be stored in the frame of the excavator arm and could be reused for keeping critical electrical components warm when not in use. Due to the custom nature of thermal straps,[13] for bench testing purposes a twin finned heat sink pulls heat from the motor face and the rear face of the gearbox through the aluminum adapter plate. This would perform better with thermal compound; however, heat buildup in the motor has not been an issue yet.

Dust protection

Since regolith is the main enemy of any mechanical system on the Moon, it quickly became a primary focus for reliability. While not tested with simulant in this project, a folded paper Tyvek[10][11] bellows is proposed for excluding dust from the system entirely. This could also improve the performance of ball screw actuators and should be studied further.

Counterweight

As stated in the description, this study needed to account for the practicality of such high forces particularly because the force required to excavate material isn't dependent on the weight of the material being removed. Most excavators rely on heavy counterweights which are both impractical to launch and 1/6th as effective due to the Moon's lower gravitational pull. From the manufacturers data on the BK215 the maximum moment generated from the arm at maximum extension was calculated as shown in *Fig 9*.

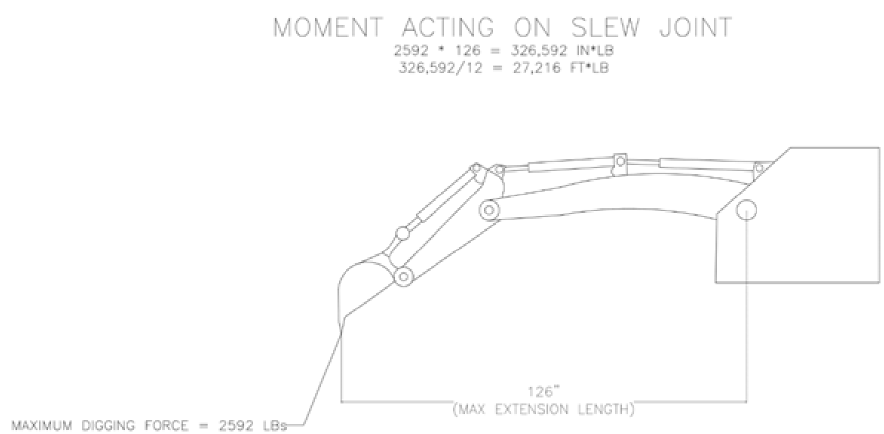

Fig 9 Moment Generated by BK215

To counteract this moment, we designed a chassis/counterweight combo that works as a bucket which can be filled with in situ mined regolith assuming a 1.5g/cm^3 density. To counteract the digging forces for terrestrial testing a capacity of 1.3m^3 is needed, similar in size to a small utility trailer. For flight hardware a larger 7.9m^3 capacity is needed, which is similar in size to the capacity of a commercial dump truck. Therefore, it is recommended to pair an excavator arm with a dump bed for early missions. This would enable transport and digging with one vehicle. Higher packing densities or melting regolith into solid blocks would aid in decreasing the size of the counterweight needed and increase the effectiveness and versatility of dedicated equipment.

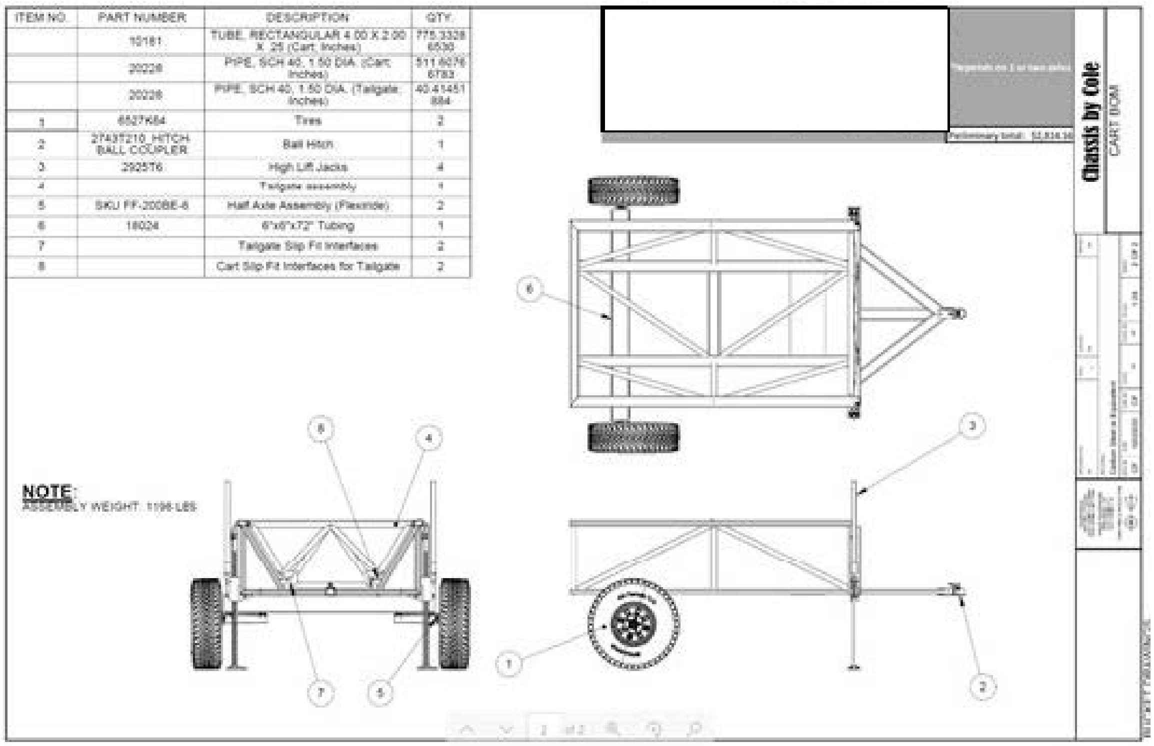

Fig 10 Counterweight/Chassis

Manufacturing Parts

Due to the pandemic and numerous weather events during the year, manufacturing was particularly challenging. Outside of a few components, most parts had to be made internally due to high demand for local labor repairing the numerous chemical plants that suffered weather damage during the year. To help alleviate this, personal equipment was used were possible. Most turning was completed on a 7x12 mini-lathe, Atlas TH42, or Jet 14in lathe, and milling was completed on a Jet knee mill or HAAS mini-Mill. Some components, like the leadscrews in fig 11, were slightly modified off the shelf components to save on machine time. As previously mentioned, keyways are a better solution for power transmission, however the broaching and keyway cuts were outside of the team’s comfort level to manufacture.

Fig 11 Spooling assembly shafts with bushings and gears

The shafts for the spooling assembly were all turned from 304 stainless steel, and other than poor chip breaking all three lathes handled the material with ease when cutting with DCMT-21.51 carbide inserts. As these were the first components made, the ease of machining on even a small machine like the 7x12 mini-lathe lulled us into a false sense of security which will be discussed later.

One of the few components that we managed to outsource was the pulleys. These were ideal for an outside CNC shop to manufacture the significant number of pulleys needed. The pulley liners were cut with a leather clicker die and gasket cutter, but a custom clicker die would improve repeatability and precision. The pulley shafts were cut on the Jet 14in lathe for the higher precision offered by the larger machine, then polished with red rouge and a benchtop polishing wheel. When assembled like shown in fig 12, the pulley liners reduce the friction from the minor thrust loads and maintain rope organization when tension is lost on the system.

The spools were made from solid 304 stainless steel bar stock to simplify manufacturing, despite weighing approximately 10lbs each. As mentioned previously, moving to a multipart design for the spools would save a considerable amount of weight by making them hollow, and would enable a proper clamping mechanism for the rope. The surface finish for the spool recess was left with slight ridges, and this seemed to help hold the rope in place during spooling and unspooling operations.

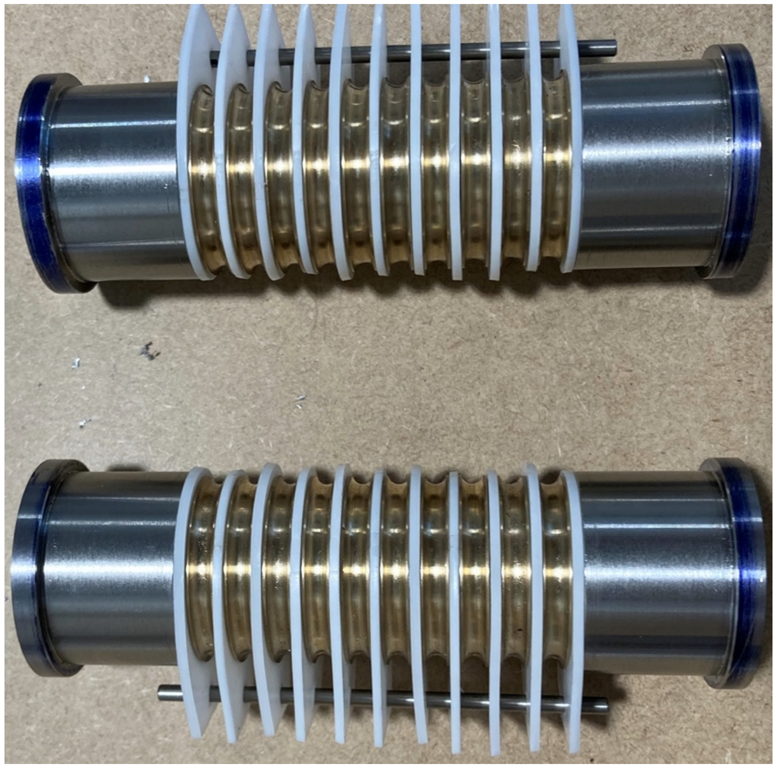

Fig 12 Pulley assembly detail

Fig 13 End of inner sleeve with pulley assembly detail in the 10x10 pulley configuration

The main issue during production occurred when trying to machine the 304 stainless steel plates for the inner and outer sleeve. Partly due to the sharp edge radius of the endmills, the lack of rigidity in the available machinery, and the required deep plunge cuts to start the channels in the sleeves, it proved impossible to machine the plates from 304 stainless steel within the available time. It was decided to switch to A36 steel in order to get the various steel plates cut. While aluminum would be easier to cut, the team has experience with welding steel, and using tack welds in place of the designed bolts to assemble the inner and outer sleeve saved valuable time by eliminating dozens of blind taps. Even with the comparatively low precision of the tack welds, the pulley assemblies fit perfectly as seen in fig 13.

The bronze guide plates did not present a challenge like the 304 stainless steel did and machined quite easily allowing for the precise adherence to the design drawings. The large bronze plates from fig 14 and fig 18 provided a good running surface for the inner sleeve while also joining the center carriage to the outer sleeve. Fig 15 shows the inner and outer sleeve assembled with the bronze plate aligned with the bolt holes of the outer sleeve. During the testing phase it was found that the bolts don’t have enough clearance to be removed while the spooling assembly is installed. By reducing the width of the bronze guides seen in fig 16 and reducing the diameter of the bolt heads, it was possible to make the bolts removable while the spooling assembly was still installed. While the build has plenty of needed improvements already identified, the assembly and disassembly is already simple and quick to accomplish.

Fig 14 Center carriage running in the inner sleeve

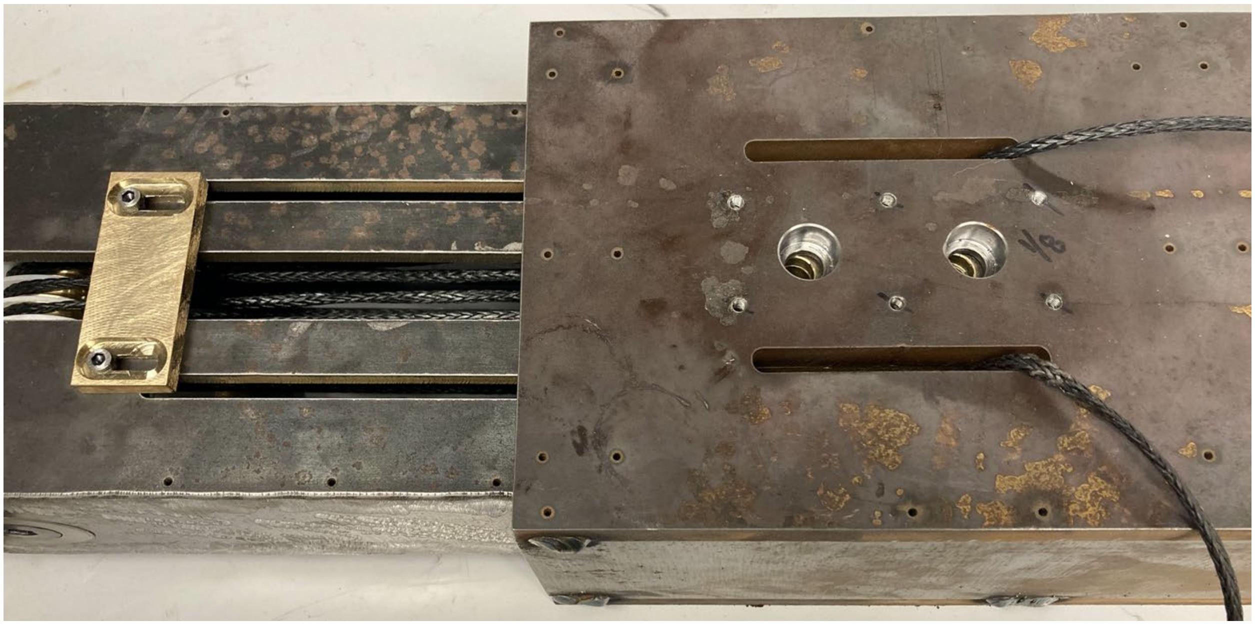

Fig 15 Inner and outer sleeve

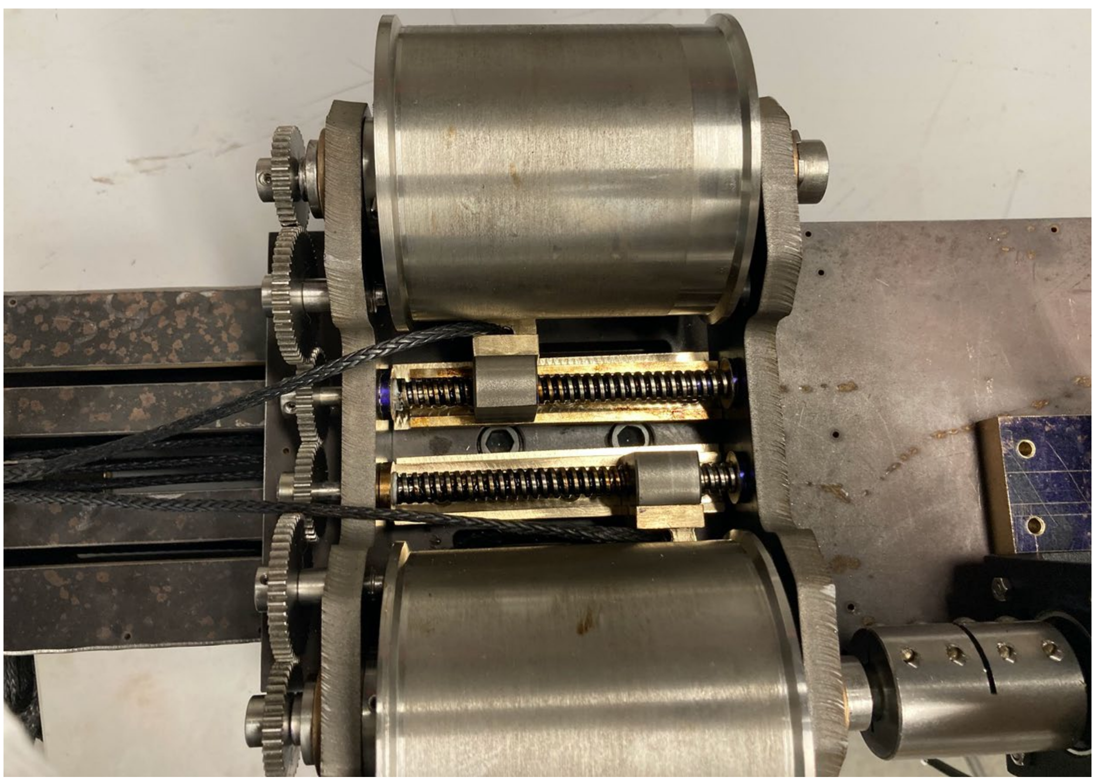

Fig 16 Spooling Assembly

Results

Motor Torque Test setup:
Mechanical setup:
Smooth pin located 12in from motor shaft center. A fishing scale's hook was positioned around the pin. Motor was activated and pulls against the fishing scale.

Electrical setup:
ESC was 6.6 VESC, 65A max at 12v
Power supplies were twin 12V 30A run in parallel for 12v 60A supply
Speed control input for the ESC was provided by a servo tester potentiometer.

Results:
2.68lb
2.19lb
2.10lb
Minimum 2.1lb force at 12in from center of rotation. **24in-lb** is the maximum reliable torque for this motor.
11000/20 = 550lbs @ 1.625in => 894in-lb
894/24 = 37.24 minimum gear reduction (50:1 used)

Motion Testing

During the first round of motion testing it was discovered that there was insufficient clearance between the bronze rope guides and the main spools. The lack of clearance is likely due to the rope twisting during this first test, as well as the possibility of the spools being slightly oversized. The available measuring tools could only measure the ends of the spool and the depth of the recess, but the diameter of the spool's recessed center couldn't be measured directly. To resolve this and enable testing, new bronze rope guides were made that lowered the bronze outcropping to make to gap sufficiently large. Replacing the bronze rope guides was not easy as it requires removal of the spools, so extending the bronze rope guides over the top of the guide nut for an easily accessible bolt is highly recommended. This also simplifies reassembly as the bronze rope guides have a very small through hole for the rope which is difficult to access while installed.

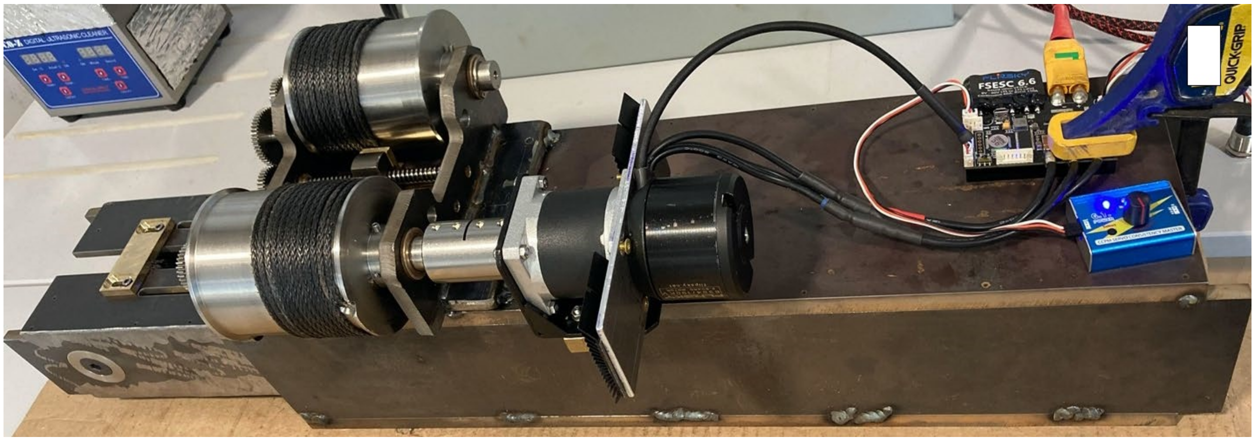

Fig 17 Full initial assembly

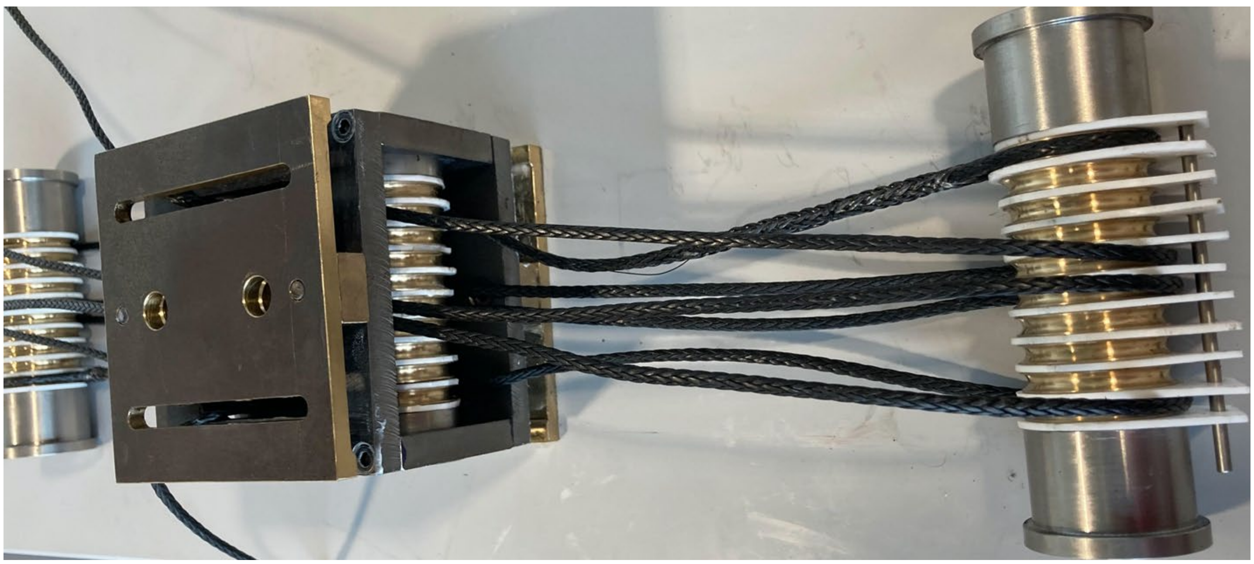

Fig 18 4x4 pulley re-configuration

The second round of testing presented an overlooked issue that based on the gear pattern and rope/spooling orientation, the spools loosened or tightened at the same time. This is easily fixed in a redesign, but it did unfortunately require that one spool to be disconnected for testing. The next issue that this round of testing presented was that the pulleys generated too much friction to all function at the same time. It was noticed during assembly that the rope's slack could be removed from only 4 pulleys at a time, but it was hoped that the higher forces from the motor would enable the rest of the pulleys to also be engaged. However, this did not happen and in one block and tackle set the first rope length could have full tension while the last rope length had no tension. It is suspected that larger clearances are needed for the bearing surface than is recommended for journal bearings, or a bimetallic pulley is needed to decrease or eliminate the pulleys pinching the shaft when under load. In order to work around this issue, the pulleys were restrung in a 4x4 configuration as shown in Fig 18. This did enable full movement of the system, and allowed for gathering data regarding the power consumption of the system.

Due to the numerous areas for power consumption between the power supply and the actuator movement, power consumption data was gathered with incremental additions to the load. This should allow for better targeting of inefficient elements.

Table 3: Power usage from extension

Action	Voltage (V)	Amperage (A)	Power consumption (w)
Idle	24.16	0.06	1.4
Spooling w/o rope	24.15	0.84	20.2
Full extension (no load)	24.14	1.81	43.6

The full extension test took 47 seconds to travel 13.88in at 24.14v. Selected screenshots of the test video are presented in fig 19 below. This spooling rate shows a motor rpm of 638 which is equivalent to 3.36v of a possible 44.4v for the 190kv motor. More extensive troubleshooting of the ESC and motor settings/behavior should be investigated to ensure the motor is performing properly as the observed rpm is unusually low. An extension time of just 6.6 seconds was expected given the motor, gear ratio, and pulley configuration. While slow performance could be explained by high loads, the ESC and motor were only consuming 43.6 watts out of an available 1569 watts and during stall tests the ESC would only go into overcurrent protection around 500 watts. Therefore, the motor had plenty of power available and should have operated at a higher rpm than observed.

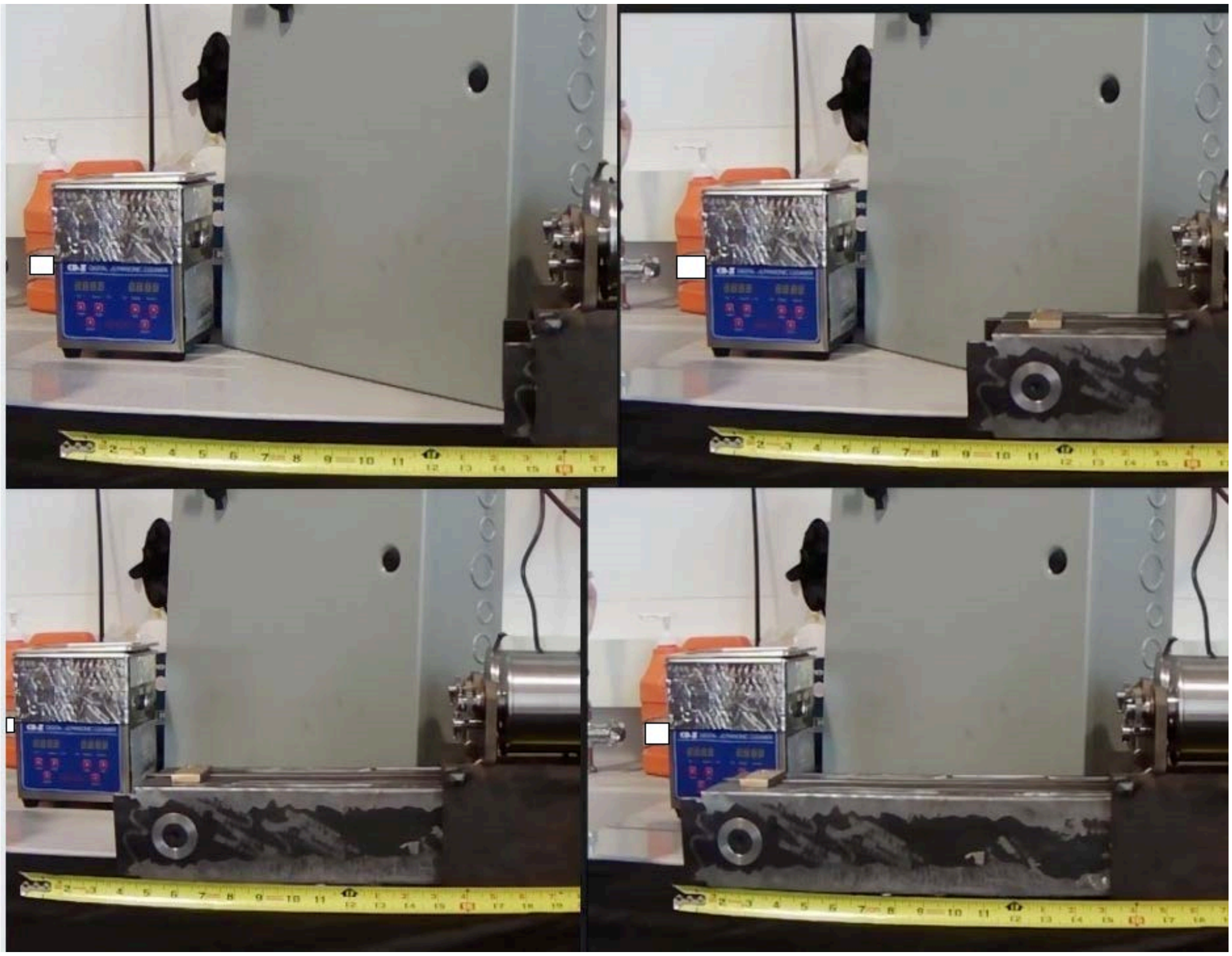

Fig 19 Extension test: Screenshots from the video taken of the extension test.
Upper left: 0in@0:00 Start of test; Upper right: 4.75in@0:16; Lower left: 10in@0:34; Lower right: 13.88in@0:47 End of test

CONCLUSIONS AND RECOMMENDATIONS

The design mostly met our goal of replacing the hydraulic cylinder, however the design deviates from the target in several areas, some more important others. Overall however, the design's dust protection looks to be applicable to any linear actuator, the chassis/counterweight design study enables full force digging even in Luna's low gravity, and the pulley system looks extremely promising as a mechanical actuator option even with the needed improvements. The more specific recommendations for individual assemblies or components are listed with each component in the design description, but the system review is below.

While the width exceeds that of the hydraulic cylinder, this does not impact the actuator's ability to fit onto the BK215. The length and stroke are the more important dimensional deviations; however, the prototype did achieve 93% of the targeted stroke while only exceeding the retracted length by 10.6%. This would reduce the range of motion for the equipment, but not by enough to be a major concern. More pressing concerns are the mass and pulley efficiency. At 117lbs the current prototype is grossly overweight, but it has significant room for mass reduction without reducing the strength. With more time and machinery resources it should be possible to reduce the mass by 60-80% without changing materials through skeletonizing the frame and/or cutting in an Iso-grid pattern. Changing the material to aluminum could see even further reduction in mass, however that would exclude the ability to design for the fatigue limit and add complexity with additional wear plates and load spreaders due to aluminum's lower compressive strength and weakness to cold welding.

The primary issues going forward are the quantity of pulleys needed for an actuator and reducing the operating friction of the pulleys. The original design was for a 10x10 pulley arrangement for a 20x force amplification, however even without load the pulley friction exceeded the system's ability to get consistent tension across the whole assembly. It was so bad that one side could be loose while the other side was at the full tension the motor could pull. Therefore, we reduced the active pulley count to a 4x4 arrangement which did produce smooth motion. It is recommended to investigate alternate pulley clearances, designs, and/or arrangements to resolve this issue. Large crane operators often exceed even the original 10x10 pulley arrangement without issue, so there is a solution. More work into understanding the motor/esc interaction is needed since there was an obvious disconnect between the supplied voltage and the observed rpm and power consumption.

Further research needs to cover some of the following areas. Full low temperature properties for Dyneema®® and other ropes/cables. As research focuses more on lightweight tension systems there needs to be a wider body of research with which to inform decisions. A greater focus into high compressive/tensile strength steel replacements for cryogenic conditions would also be welcome. Titanium and aluminum alloys can replace steel for many things, but not highly loaded shafts or pins where high strength steel is the best option. Stellite looks to be a good candidate, but there is a lack of low temperature research since most of its current applications are for high temperatures and the material cost is too high for most applications to use it for anything other than hard facing. A comprehensive review of commercially available blasting media[8] would also be valuable as more research focuses on larger construction and building techniques. In particular, a massive indoor facility could allow for full scale testing of equipment and techniques for construction and mining with freight and personnel doors. Essentially a cold, low humidity, dusty version of the Neutral Buoyancy Lab and Space Vehicle Mockup Facility.

Recommendations for future development have a couple of angles to approach. It would still be extremely useful to demonstrate an entire hydraulic system using multiple actuators. More extensive testing would also be beneficial, specifically endurance testing to establish and modify problematic areas of wear. While this system is already at the extreme upper end of the power and force desired by proposed NASA missions, an even larger system based on chain would be good to explore. The tighter turn radius of the chain and higher abrasion resistance could enable truly massive equipment, but the high weight penalty is a major detraction for early missions where mass is extremely important.

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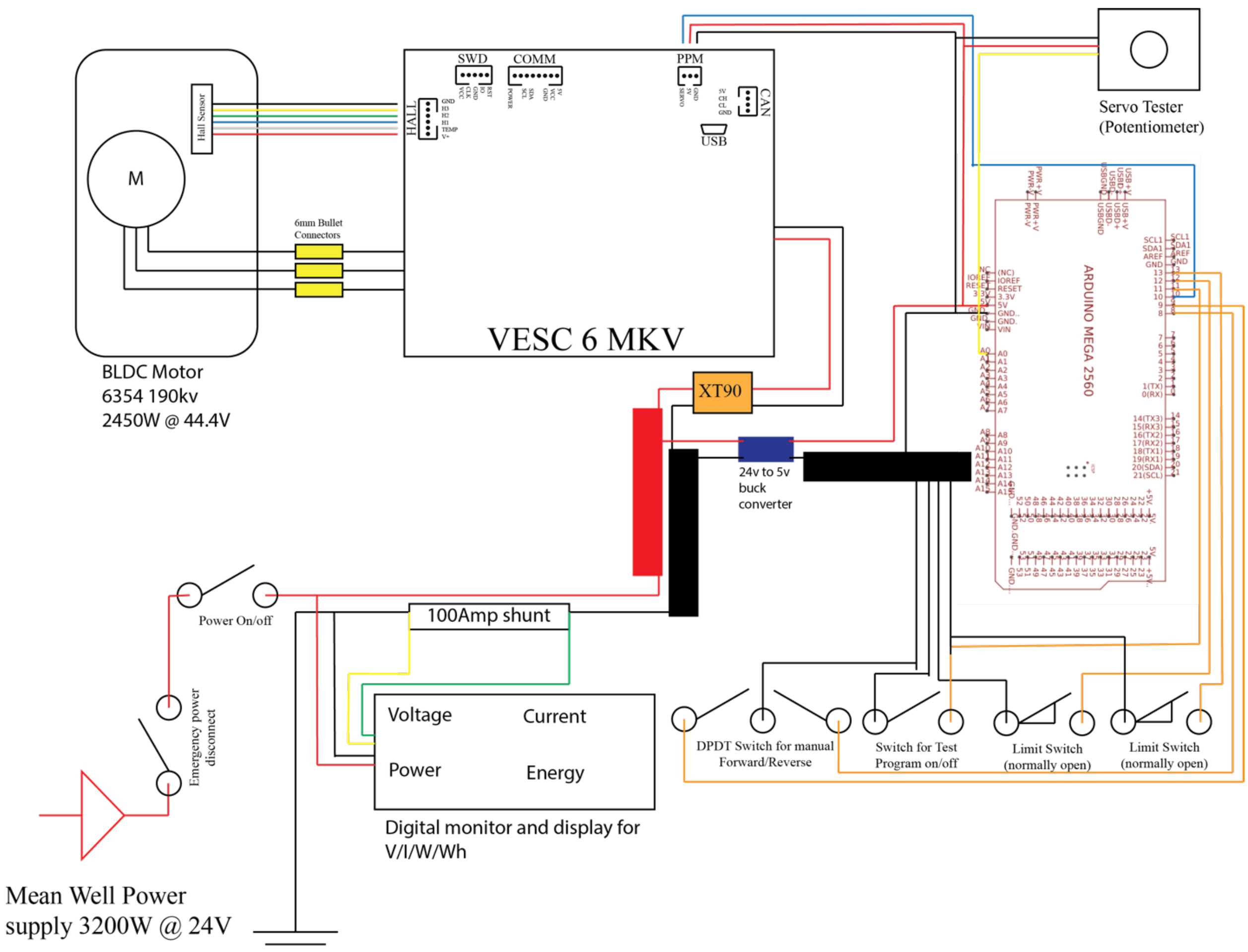
APPENDICES

Appendix A: Electrical plan



Appendix B
(1) Dyneema® Rope
(2) Marlow EXCEL D12 MAX 78, 3mm diameter
(3) https://www.velasailingsupply.com/marlow-excel-d12-max-78-3mm-black/

Marlow DATASHEET ®

D12 MAX SK75 SK78

D12 MAX SK75/SK78 is premium, light weight, high strength braided rope produced from DSM Dyneema's SK75 or SK78 fibres. The SK78 product has superior resistance to creep (permanent elongation resulting from extended time under load).

MATERIAL

MANUFACTURED FROM DYNEEMA SK75 (D12 SK75 ONLY): Light weight / High strength / Low stretch / Good resistance to chemicals and UV

MANUFACTURED FROM DYNEEMA SK78 (D12 SK78 ONLY): Lowest creep HMPE fibre / Light weight / High strength / Low Stretch / Good resistance to chemicals and UV

MATERIAL: Dyneema SK75 or Dyneema SK78 (Ultra high modulus Polyethylene)
STRAND: Single twisted (large strands are double twisted)
BRAID: 12 Strand Braided
COATING: Colour coated with 'ArmourCoat' Polyurethane
PRE-STRETCH: Heat set and pre-stretched

CONSTRUCTION

TWISTED FIBRE CONSTRUCTION: Improved abrasion resistance / Firmer round rope, aids handling and works better with cleats and jammers

12 STRAND BRAIDED CONSTRUCTION: Pitch (lay) optimised to work with the yarn twist / Improved strength / Easier to splice / Flexible product

HEAT SET AND PRE-STRETCHED: Improved strength / Lower stretch / Easier to splice, as residual stress removed from fibres

COATING OPTIONS

ARMOURCOAT POLYURETHANE COATED: Improves abrasion resistance and durability / Increases friction, aids handling & works better with cleats and jammers

PROPERTIES

RELATIVE DENSITY: 0.97 (floats)
CHEMICAL RESISTANCE: Excellent resistance to most chemicals

N.B. KNOTS WILL SIGNIFICANTLY REDUCE THE STRENGTH OF ANY ROPE. THIS PRODUCT WILL TYPICALLY RETAIN APPROXIMATELY 30% OF ITS STRENGTH IF TERMINATED WITH A KNOT. THE EXACT

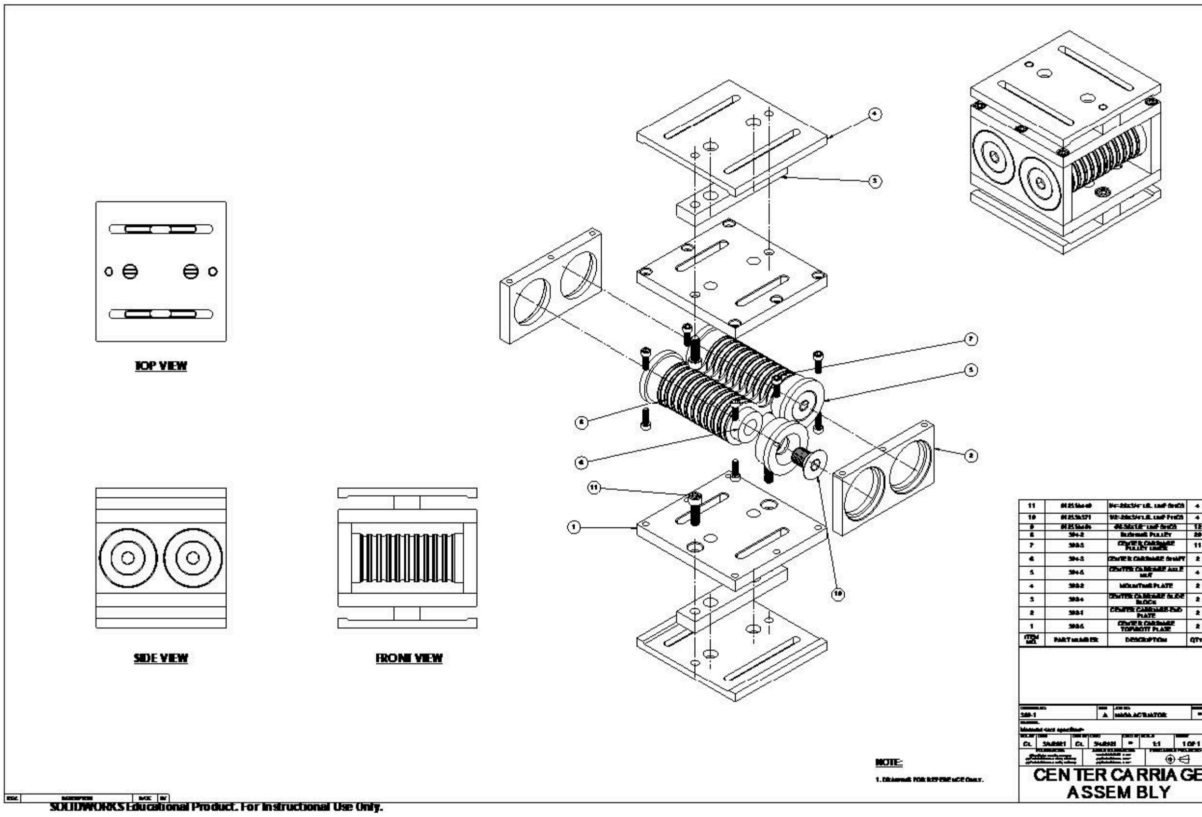
PERFORMANCE

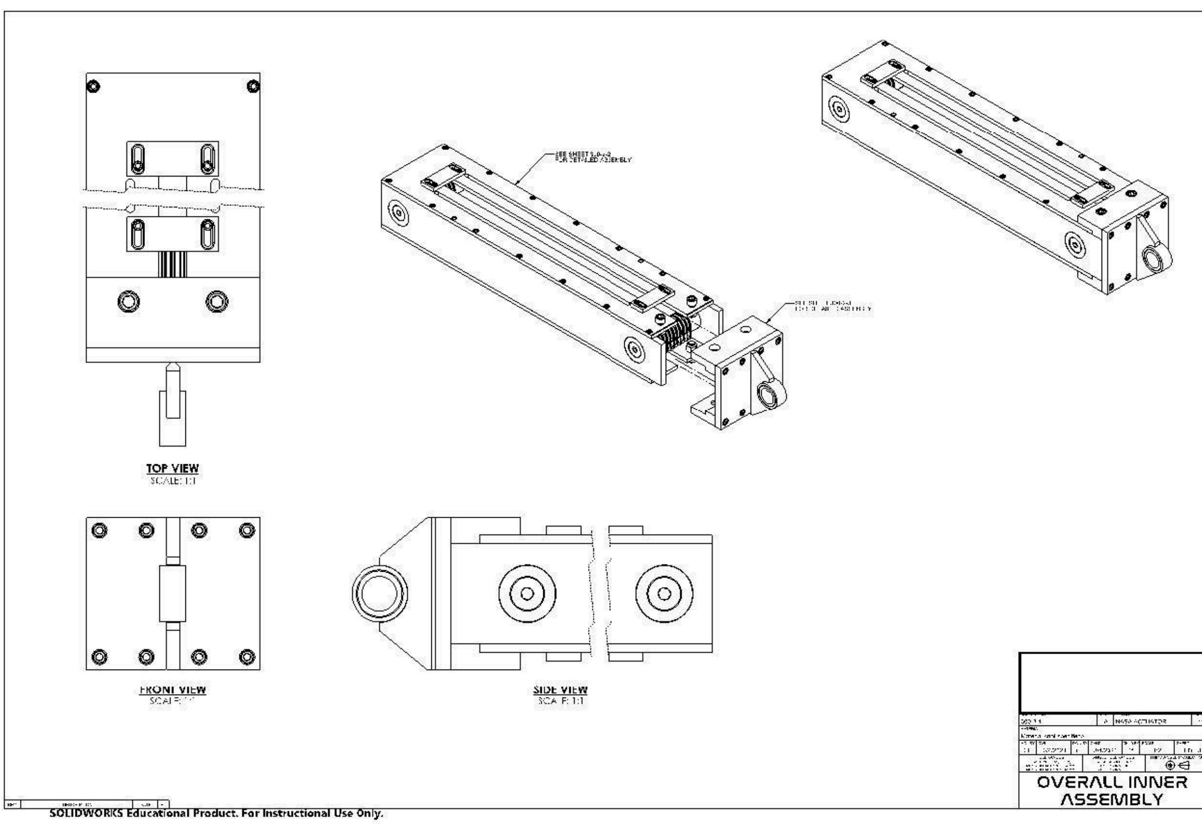
Note: D12 MAX is tested by Marlow with spliced eye termination.

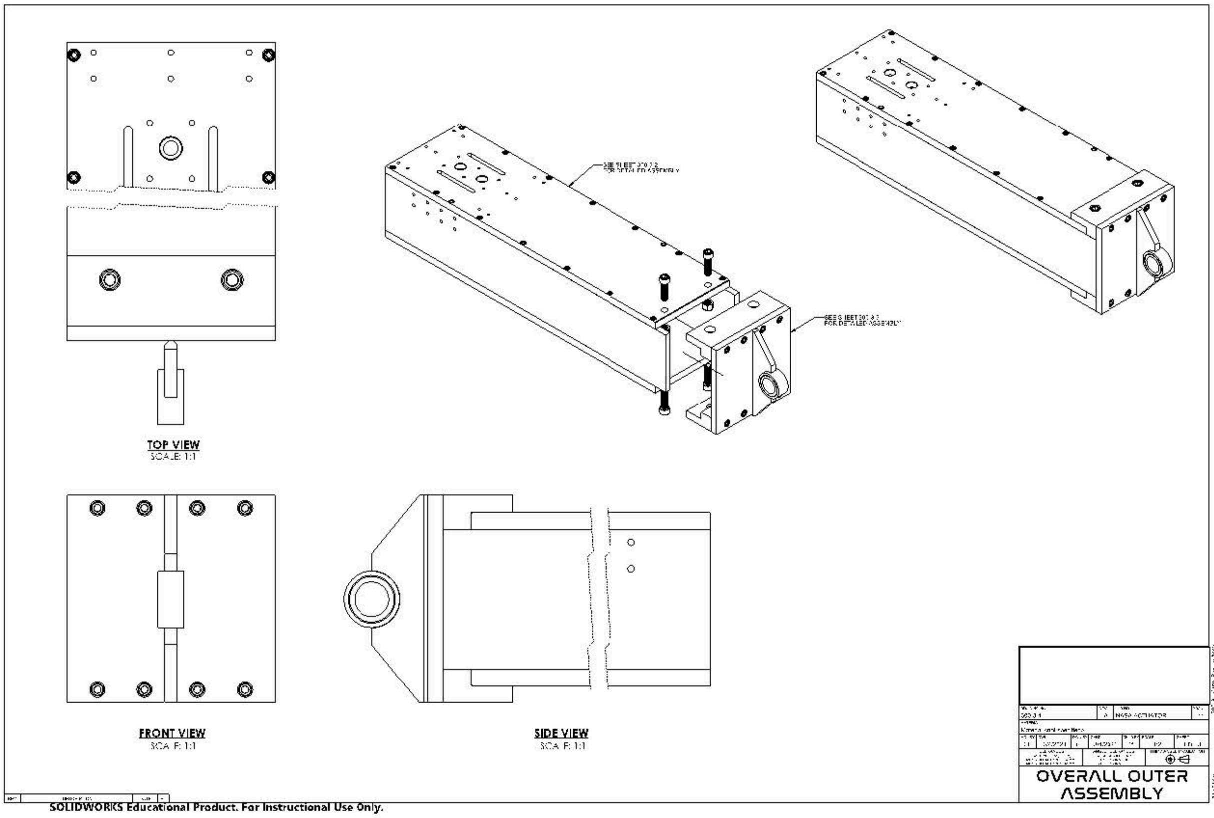
DIAMETER		MASS		AVERAGE STRENGTH			MIN STRENGTH		
mm	Inch	g/m	lb/100 ft	kg	lb	kN	kg	lb	kN
2.5	3/32	4.5	0.30	1010	2220	9.9	935	2060	9.2
3	1/8	6.8	0.46	1510	3320	14.8	1400	3090	13.7
4	5/32	11.1	0.74	2480	5460	24.3	2300	5080	22.6
5	3/16	15.6	1.05	3200	7050	31.4	2980	6560	29.2
6	7/32	22.3	1.50	4570	10100	44.8	4250	9380	41.7
7	1/4	35.6	2.39	7510	16600	73.7	6980	15400	68.5
8	5/16	44.5	2.98	9390	20700	92.1	8730	19200	85.6
9	3/8	54	3.62	10800	23800	106	10000	22100	98.1
10	13/32	63	4.22	12600	27800	124	11700	25800	115
11	7/16	75.5	5.06	15100	33300	148	14100	31000	138
12	15/32	90	6.04	18000	39700	177	16700	36900	164
13	1/2	107	7.18	21100	46600	207	19700	43400	193
15	9/16	134	8.99	26400	58300	259	24600	54200	241
17	11/16	184	12.34	32900	72400	323	30600	67211	300

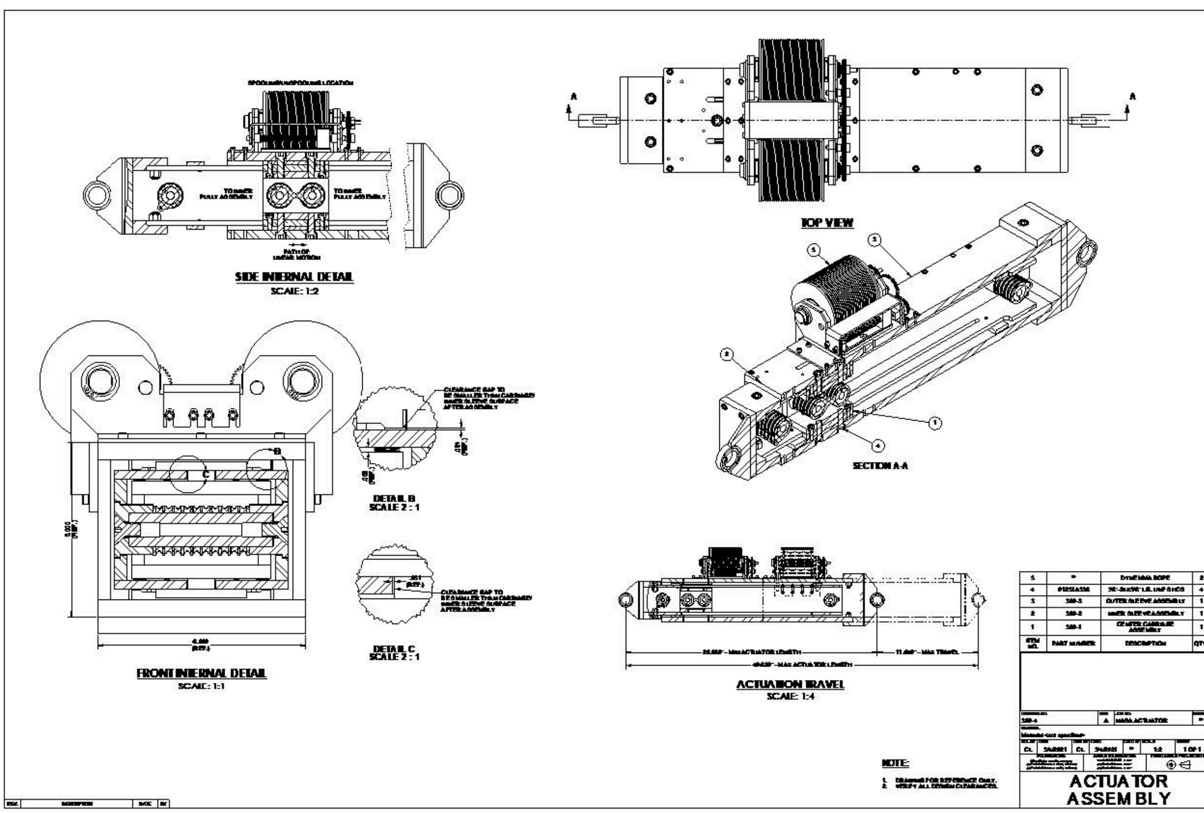
Appendix C
(1) Excavator Arm
(2) BK215
(3) https://www.palletforks.com/3-point/backhoes/bk215-3-point-hitch-backhoe-fits-cat-1-or-2/122500.html
http://titanmanuals.com/pdfs/BK215.pdf

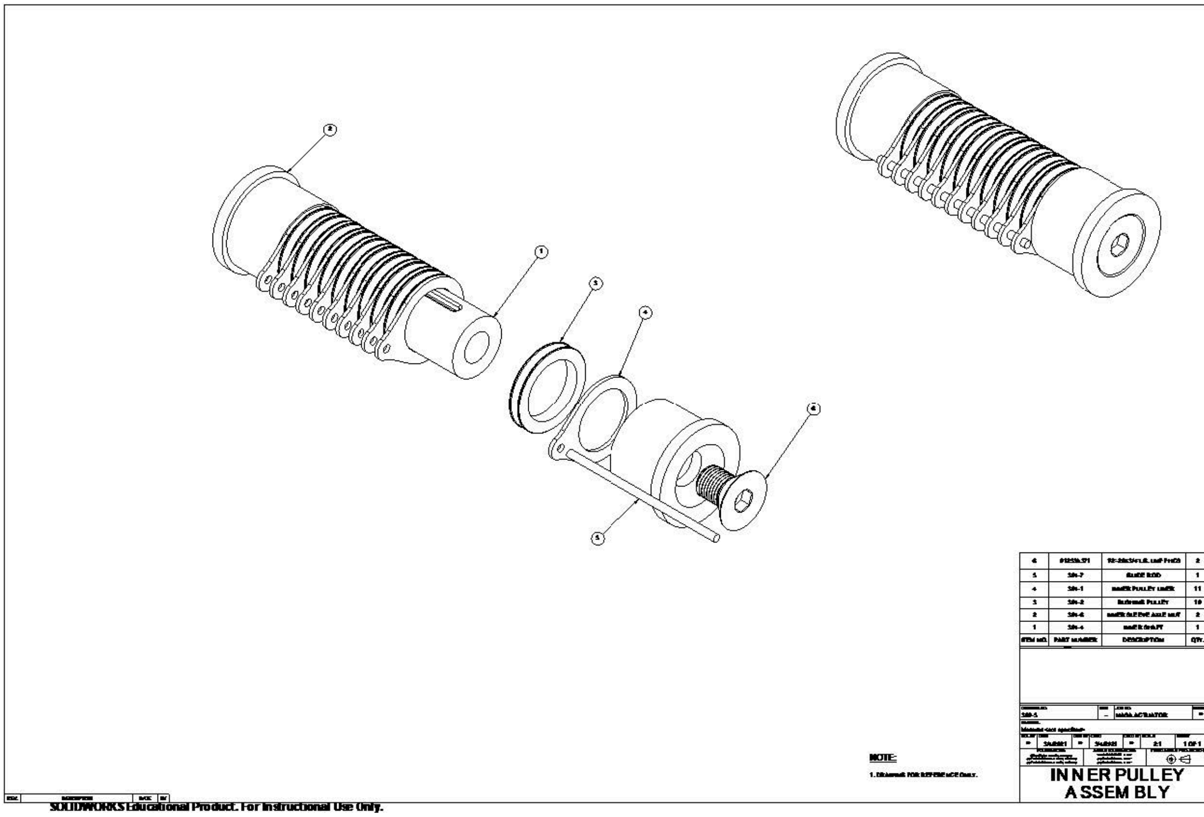
Appendix D: Assembly and part drawings

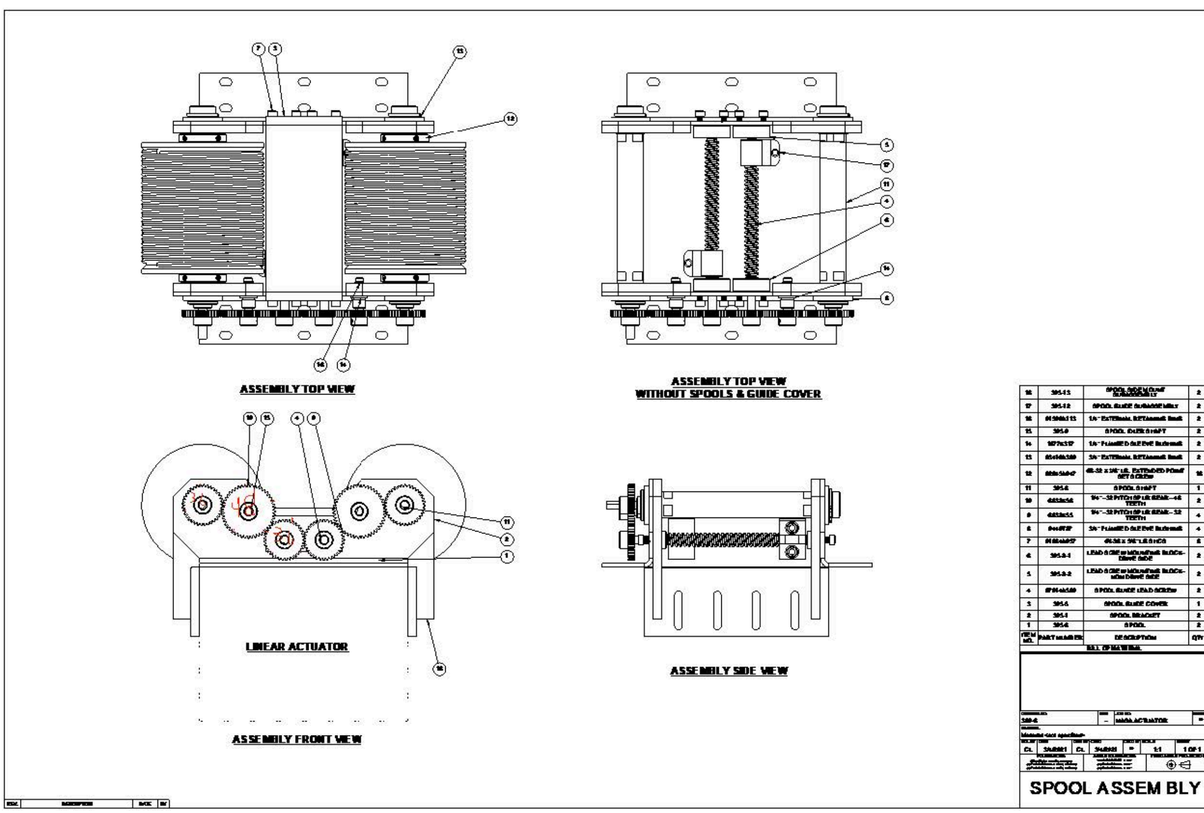

TOP VIEW
SIDE VIEW
FRONT VIEW
NOTE:
CENTER CARRIAGE ASSEMBLY
SOLIDWORKS Educational Product. For Instructional Use Only.

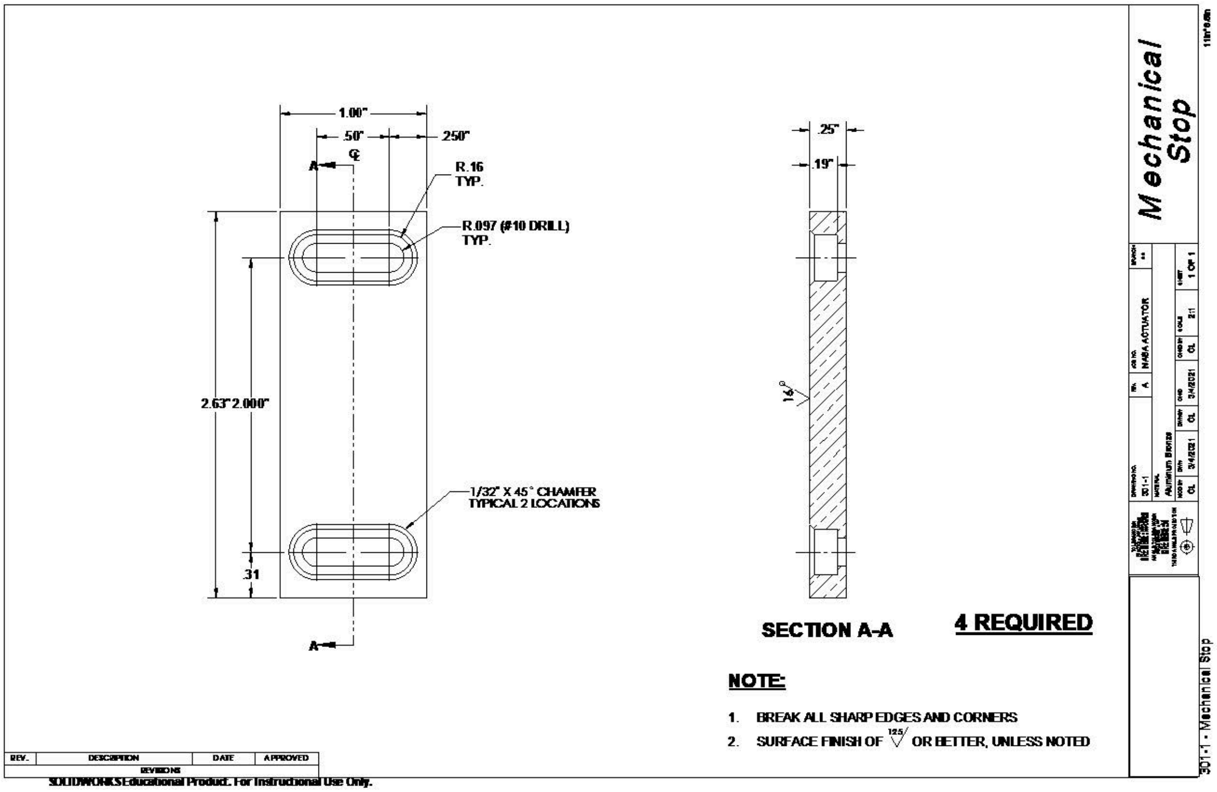

TOP VIEW
SCALE 1:1
FRONT VIEW
SIDE VIEW
SCALE 1:1
OVERALL INNER ASSEMBLY
SOLIDWORKS Educational Product. For Instructional Use Only.


TOP VIEW
SCALE 1:1
FRONT VIEW
SCALE 1:1
SIDE VIEW
SCALE 1:1
OVERALL OUTER
ASSEMBLY
SOLIDWORKS Educational Product. For Instructional Use Only.

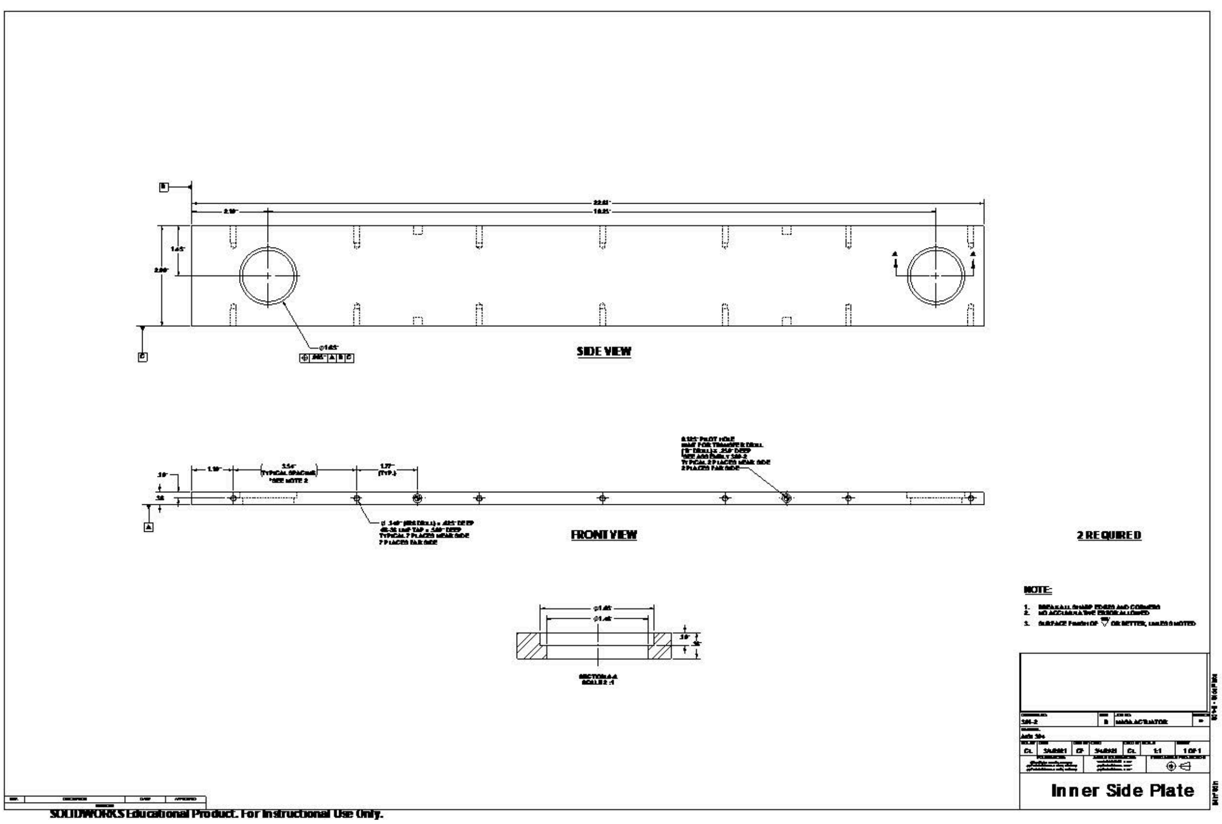

SIDE INTERNAL DETAIL
SCALE: 1:2
TOP VIEW
SECTION A-A
FRONT INTERNAL DETAIL
SCALE: 1:1
DETAIL B
SCALE 2 : 1
DETAIL C
SCALE 2 : 1
ACTUATION TRAVEL
SCALE: 1:4
NOTE:
ACTUATOR
ASSEMBLY


NOTE:
INNER PULLEY ASSEMBLY
SOLIDWORKS Educational Product. For Instructional Use Only.

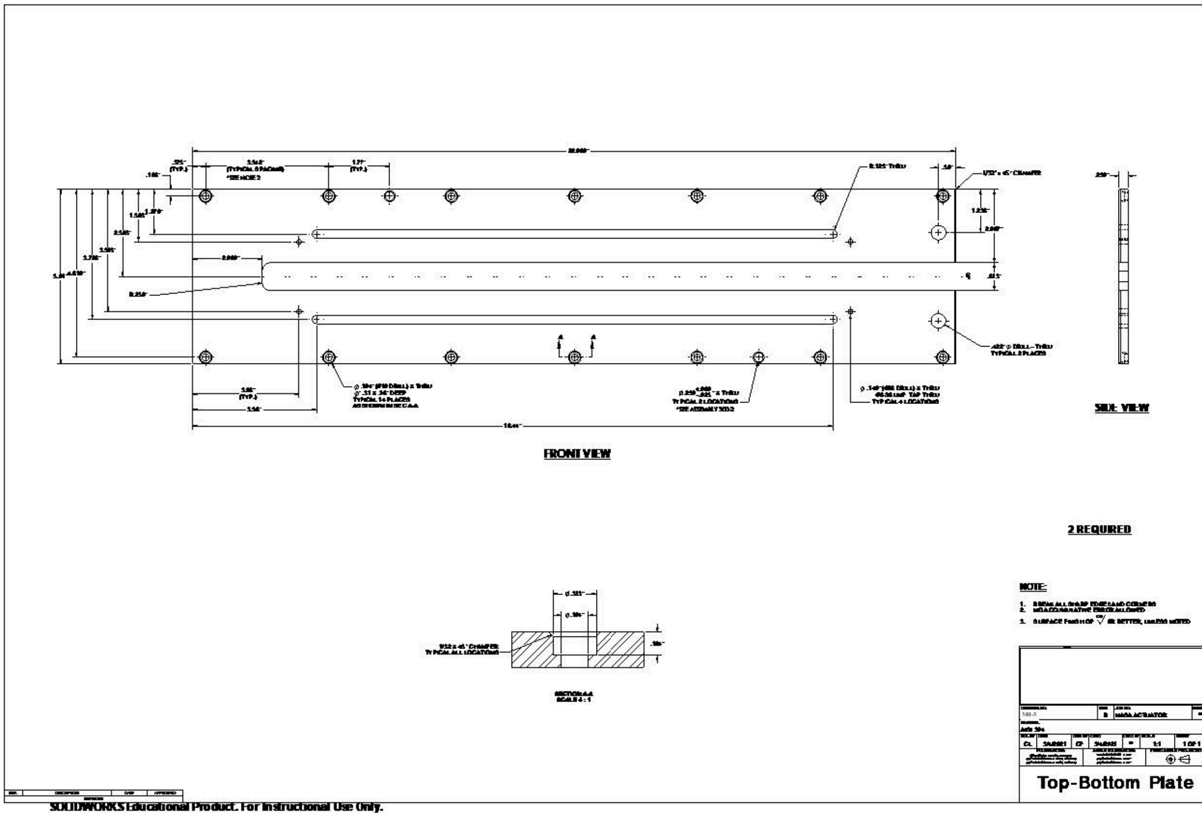

ASSEMBLY TOP VIEW
ASSEMBLY TOP VIEW WITHOUT SPOOLS & GUIDE COVER
LINEAR ACTUATOR
ASSEMBLY FRONT VIEW
ASSEMBLY SIDE VIEW
SPOOL ASSEMBLY

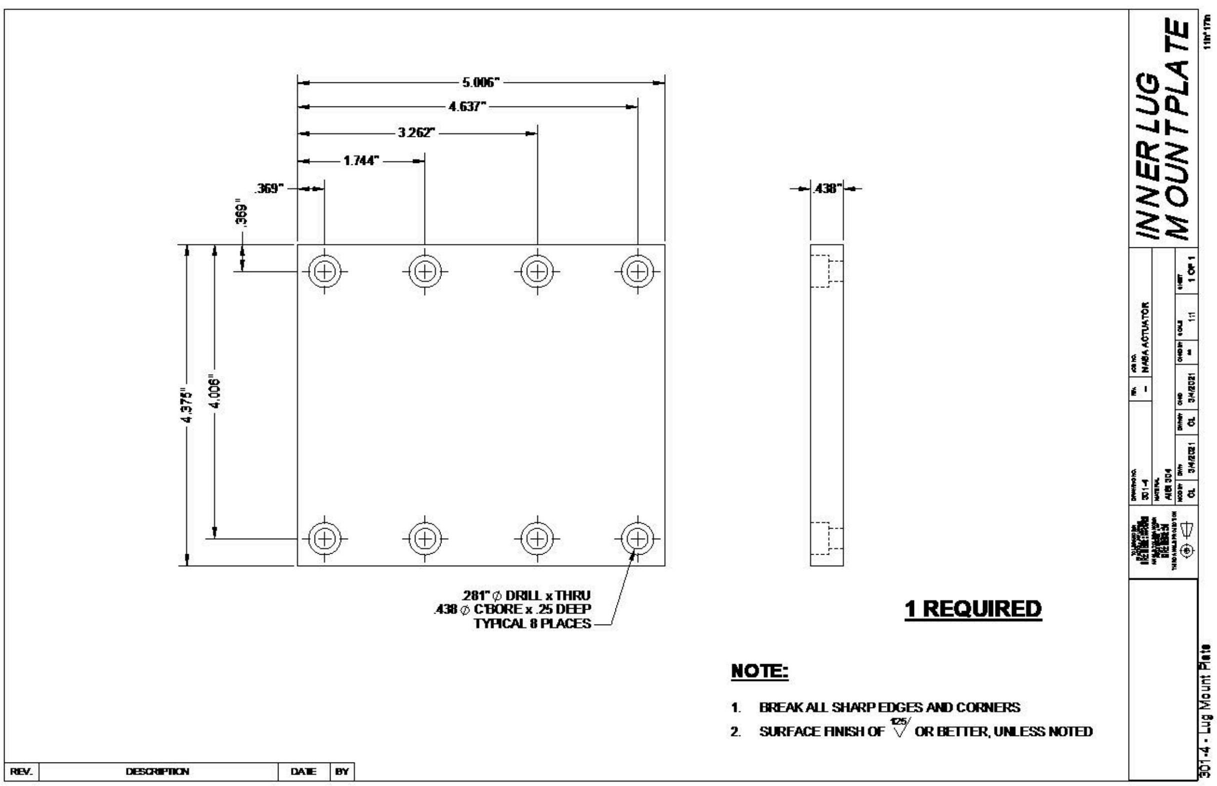


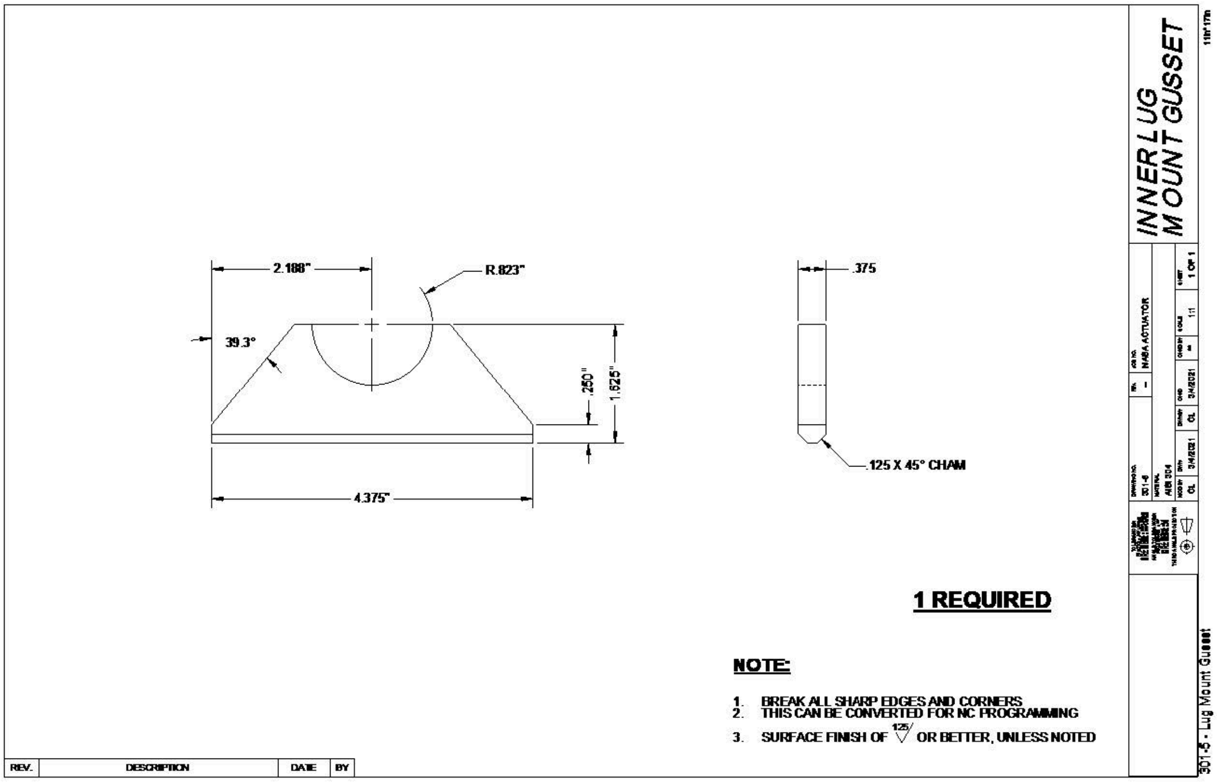
1.00"
.50"
.250"
.25"
.19"
R.16
TYP.
R.097 (#10 DRILL)
TYP.
2.63" 2.000"
1/32" X 45° CHAMFER
TYPICAL 2 LOCATIONS
.31
16
A
SECTION A-A
4 REQUIRED
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
2. SURFACE FINISH OF 125 OR BETTER, UNLESS NOTED
REV.
DESCRIPTION
DATE
APPROVED
REVISIONS
SOLIDWORKS Educational Product. For Instructional Use Only.
Mechanical Stop
NASA ACTUATOR
Aluminum Bronze
301-1
CL
3/4/2021
2:1
1 OF 1
301-1 - Mechanical Stop

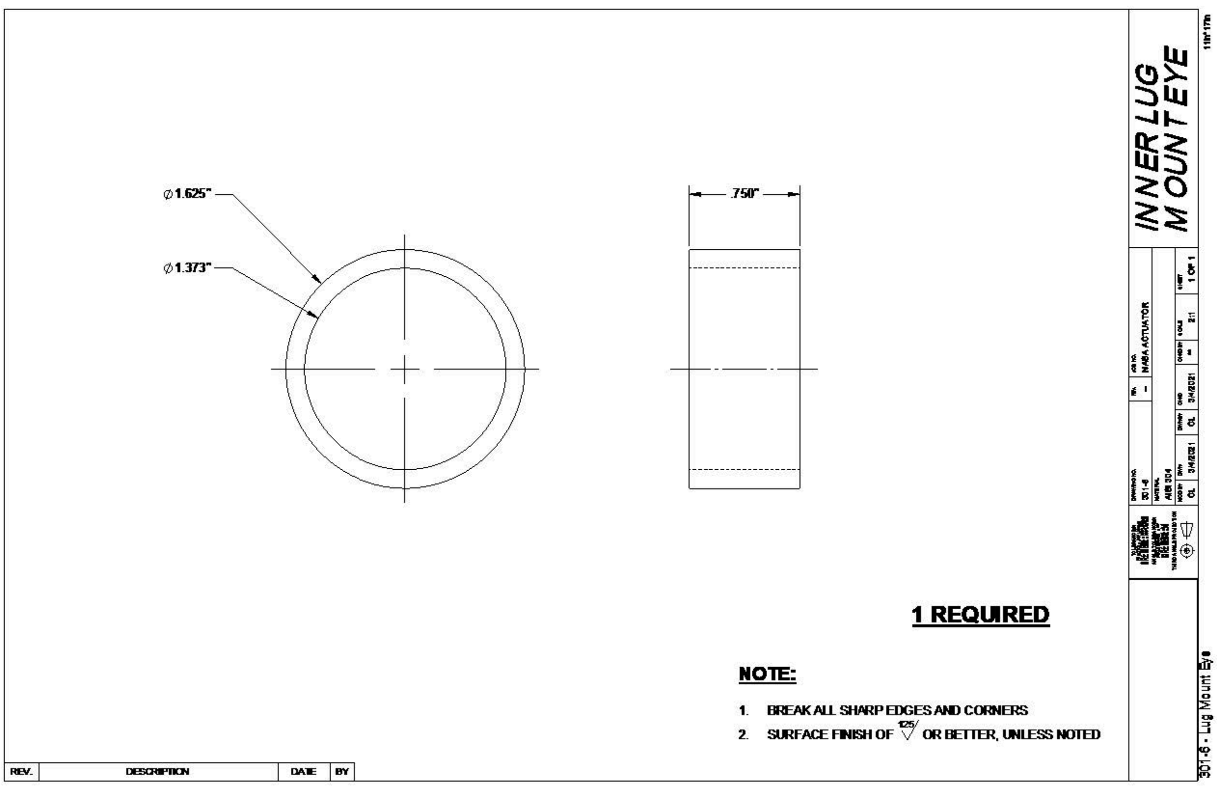


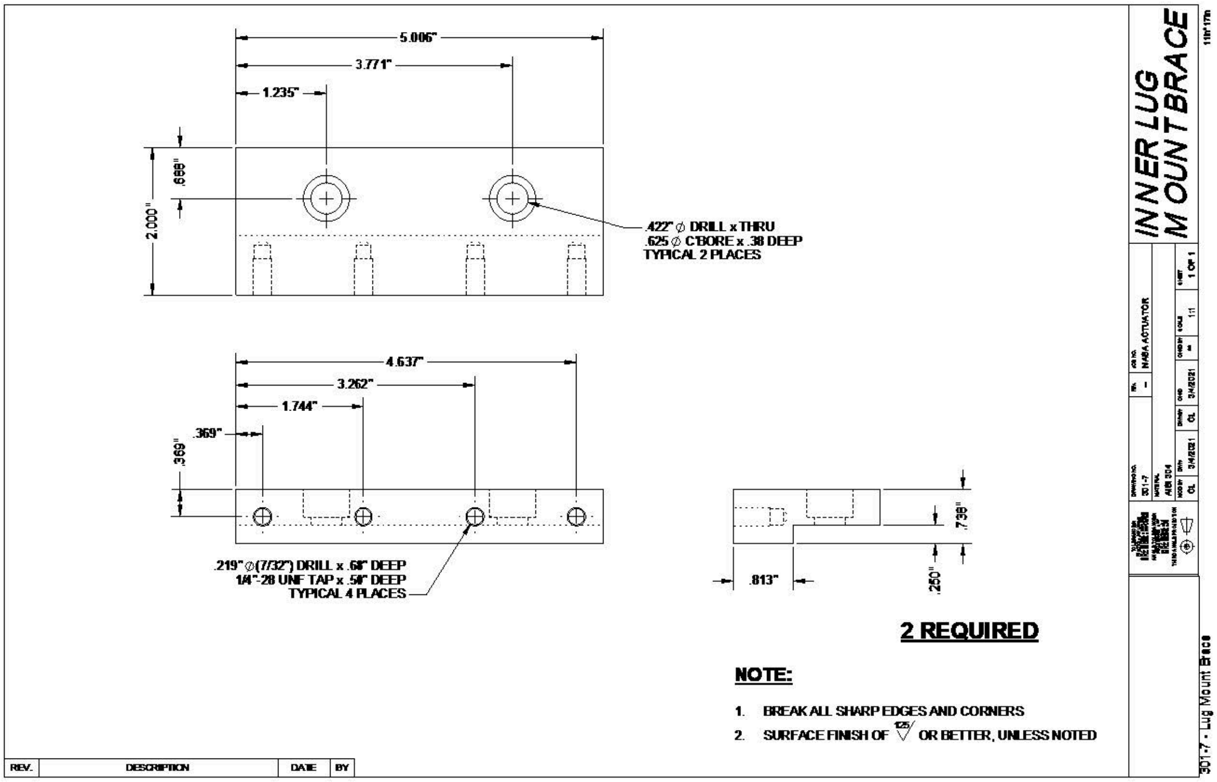
SIDE VIEW
FRONT VIEW
2 REQUIRED
NOTE:
NASA ACTUATOR
1 OF 1
Inner Side Plate
SOLIDWORKS Educational Product. For Instructional Use Only.

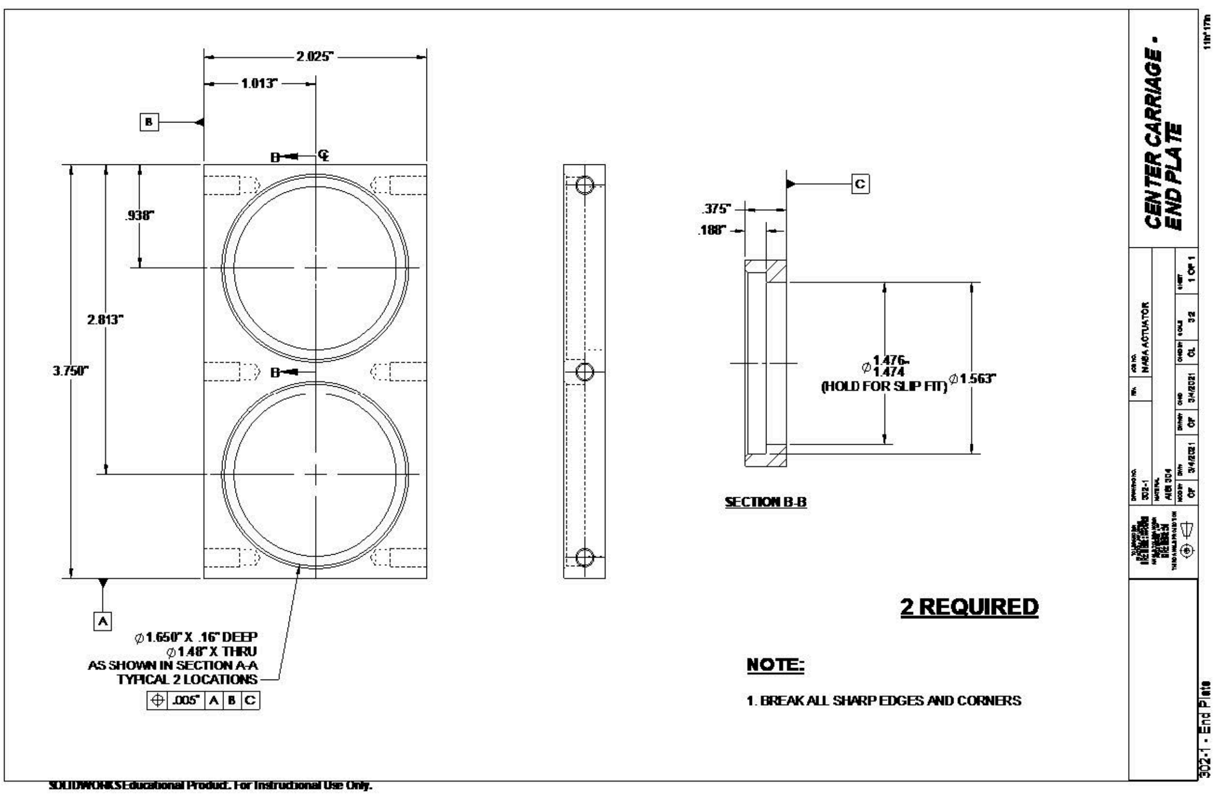

FRONT VIEW
SIDE VIEW
2 REQUIRED
NOTE:
Top-Bottom Plate

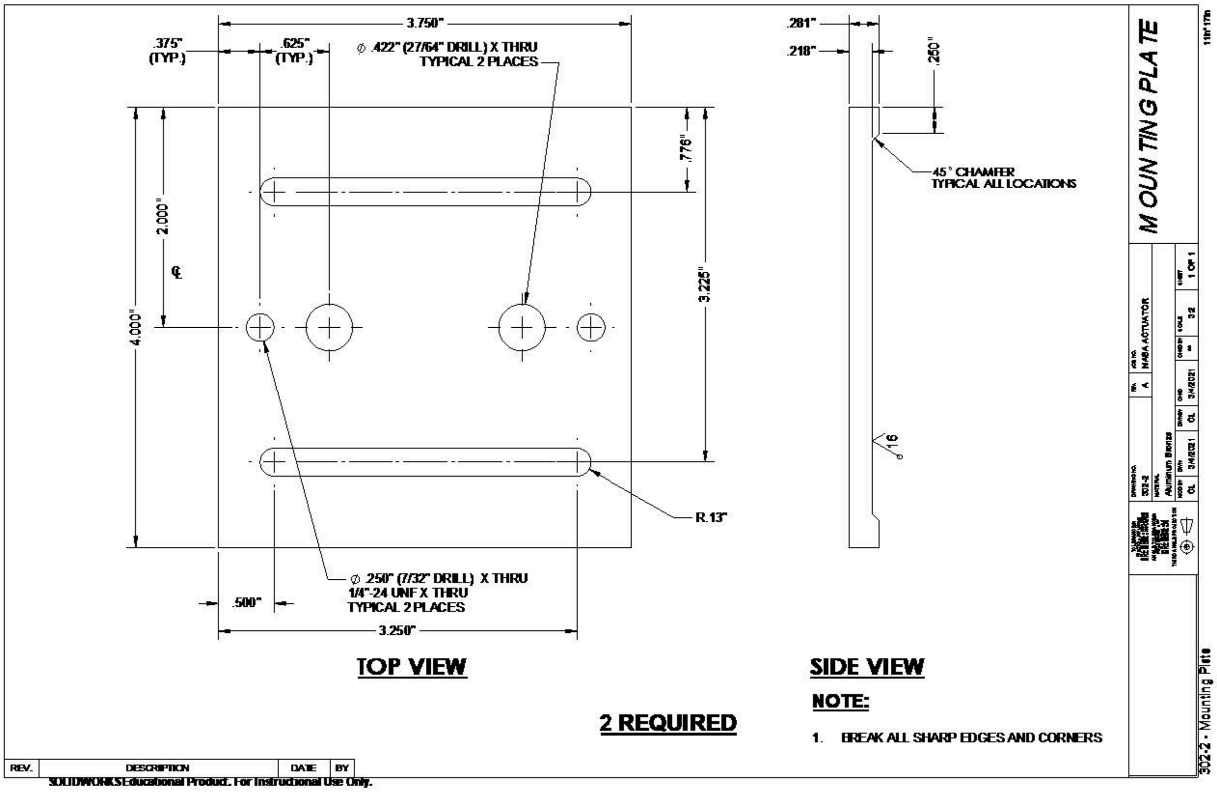

5.006"
4.637"
3.262"
1.744"
.369"
.369"
4.375"
4.006"
.438"
.281" ⌀ DRILL x THRU
.438 ⌀ C'BORE x .25 DEEP
TYPICAL 8 PLACES
1 REQUIRED
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
2. SURFACE FINISH OF 125 OR BETTER, UNLESS NOTED
INNER LUG MOUNT PLATE
NASA ACTUATOR
301-4
AISI 304
1:1
1 OF 1
3/4/2021
REV.
DESCRIPTION
DATE
BY
301-4 - Lug Mount Plate
11in*17in

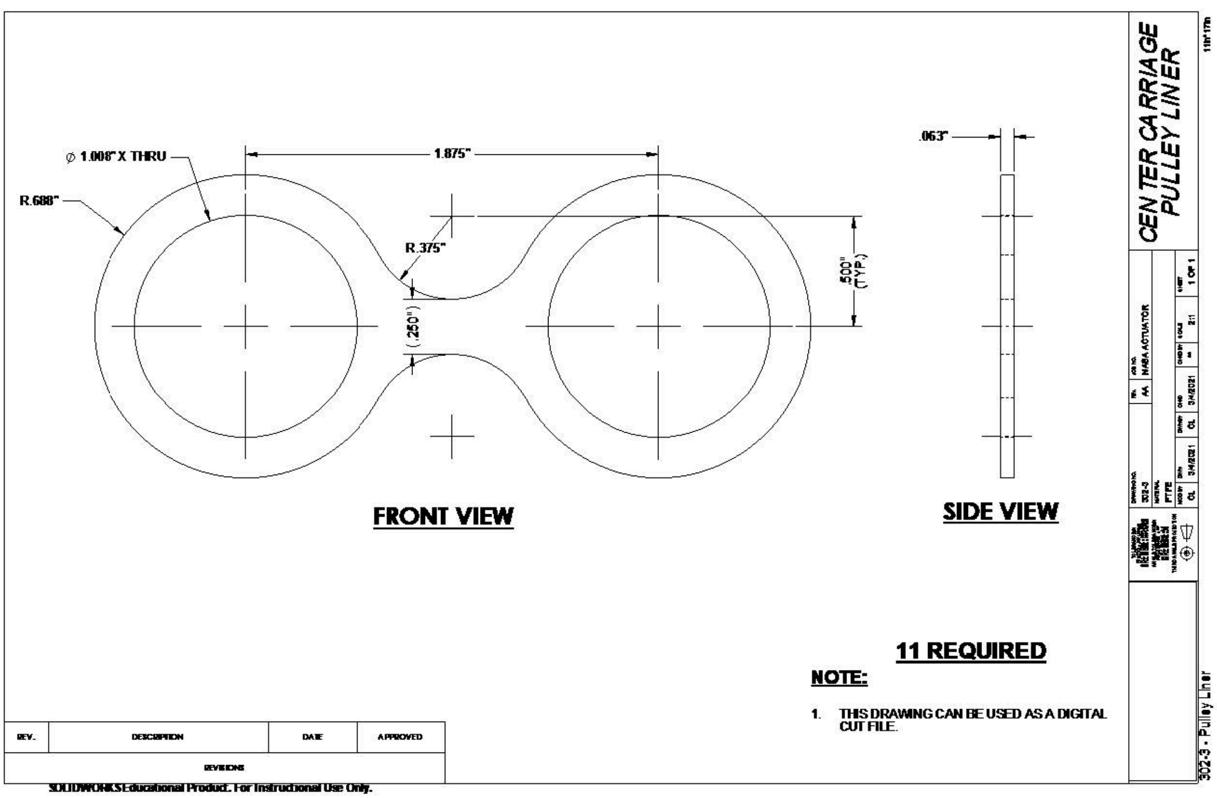

INNER LUG MOUNT GUSSET
2.188"
R.823"
39.3°
.250"
1.625"
4.375"
.375
.125 X 45° CHAM
1 REQUIRED
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
2. THIS CAN BE CONVERTED FOR NC PROGRAMMING
3. SURFACE FINISH OF 125 OR BETTER, UNLESS NOTED
NASA ACTUATOR
1:1
1 OF 1
AISI 304
REV.
DESCRIPTION
DATE
BY
301-5 - Lug Mount Gusset

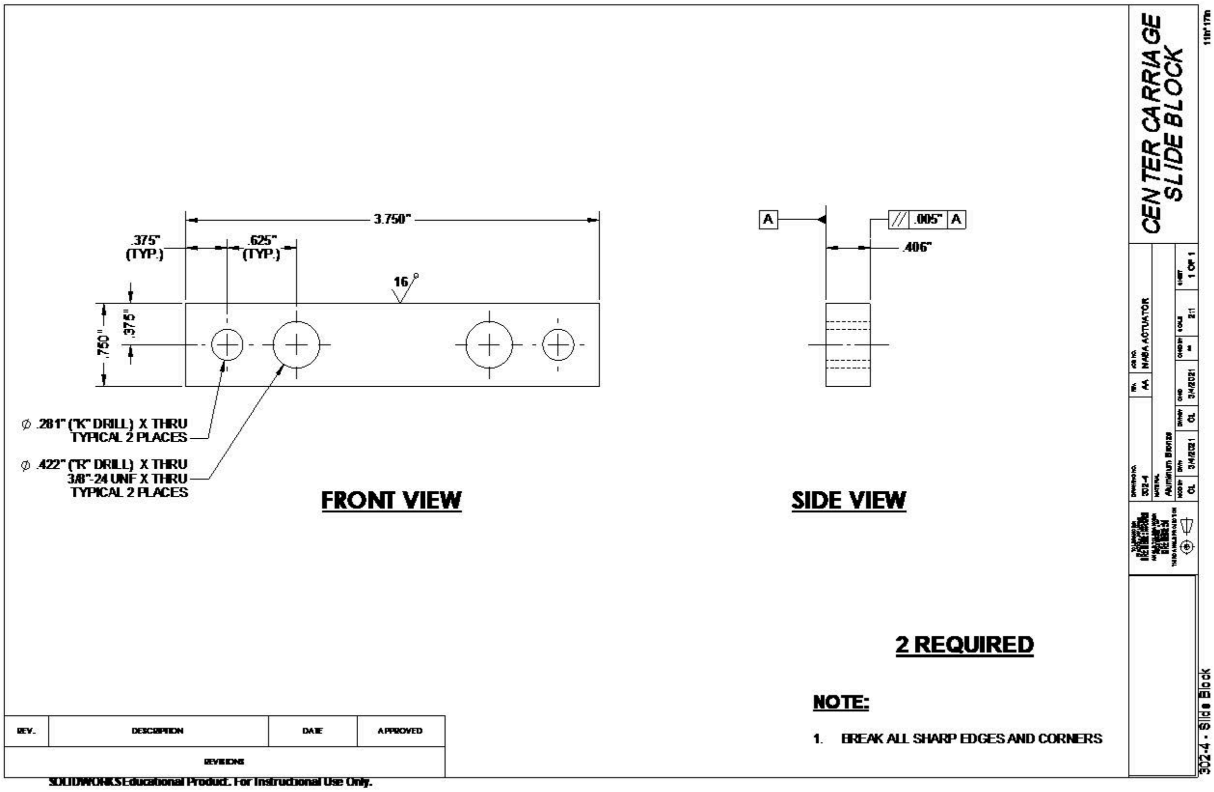

INNER LUG MOUNT EYE
Ø1.625"
Ø1.373"
.750"
1 REQUIRED
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
2. SURFACE FINISH OF 125 OR BETTER, UNLESS NOTED
NASA ACTUATOR
2:1
1 OF 1
AISI 304
REV.
DESCRIPTION
DATE
BY
301-6 - Lug Mount Eye

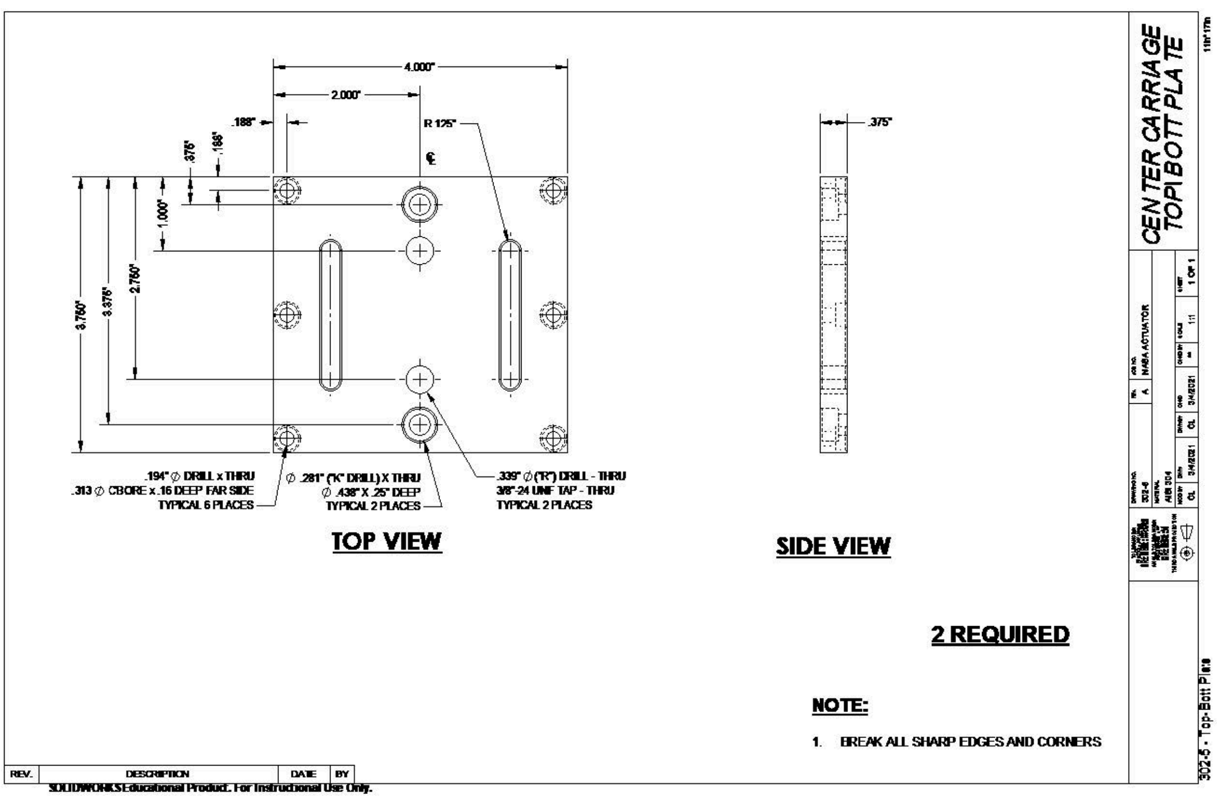

5.006"
3.771"
1.235"
2.000"
.688"
.422" ⌀ DRILL x THRU
.625 ⌀ C'BORE x .38 DEEP
TYPICAL 2 PLACES
4.637"
3.262"
1.744"
.369"
.369"
.219" ⌀ (7/32") DRILL x .68" DEEP
1/4"-28 UNF TAP x .50" DEEP
TYPICAL 4 PLACES
.813"
.738"
.250"
2 REQUIRED
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
2. SURFACE FINISH OF 125 OR BETTER, UNLESS NOTED
INNER LUG MOUNT BRACE
NASA ACTUATOR
301-7
AISI 304
1:1
1 OF 1
3/4/2021
301-7 - Lug Mount Brace
REV.
DESCRIPTION
DATE
BY

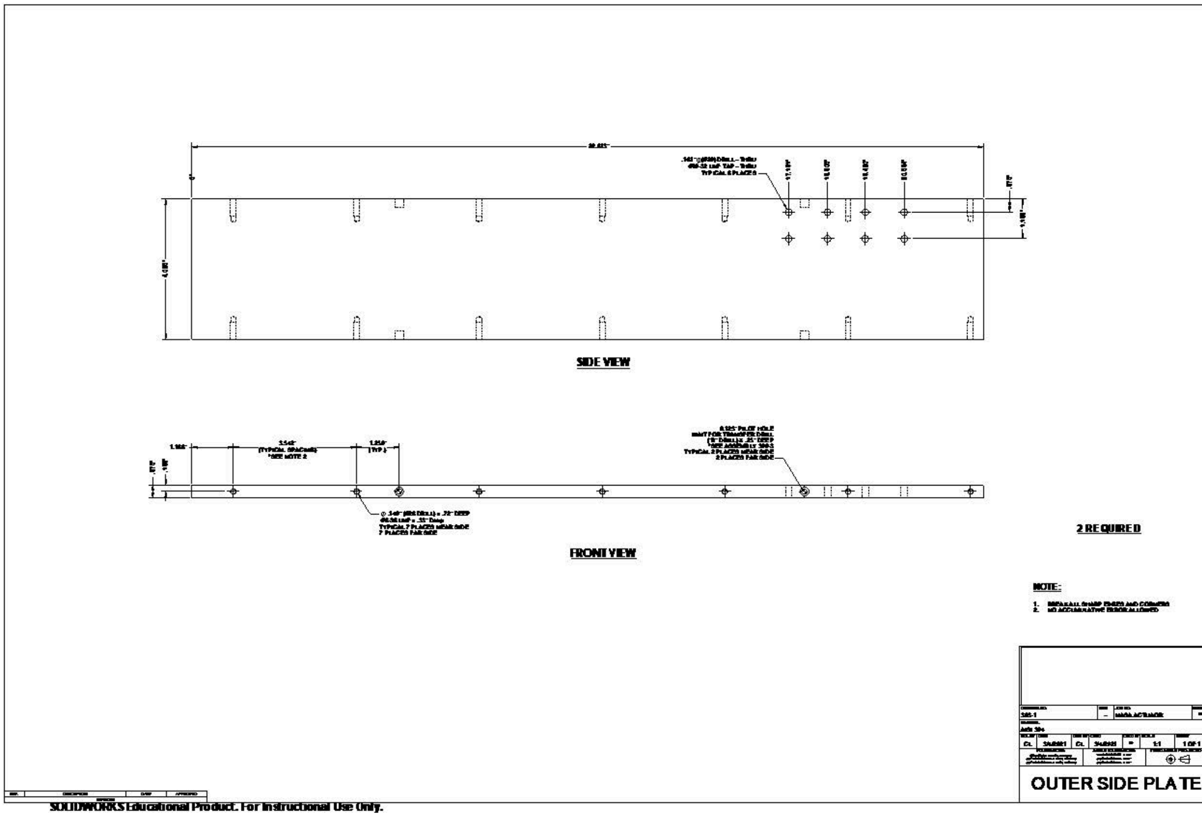

2.025"
1.013"
B
B
C
.938"
2.813"
3.750"
B
A
⌀1.650" X .16" DEEP
⌀1.48" X THRU
AS SHOWN IN SECTION A-A
TYPICAL 2 LOCATIONS
.005" A B C
.375"
.188"
C
⌀1.476
1.474
(HOLD FOR SLIP FIT)
⌀1.563"
SECTION B-B
2 REQUIRED
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
CENTER CARRIAGE - END PLATE
NASA ACTUATOR
302-1
AISI 304
3:2
1 OF 1
3/4/2021
302-1 - End Plate
SOLIDWORKS Educational Product. For Instructional Use Only.

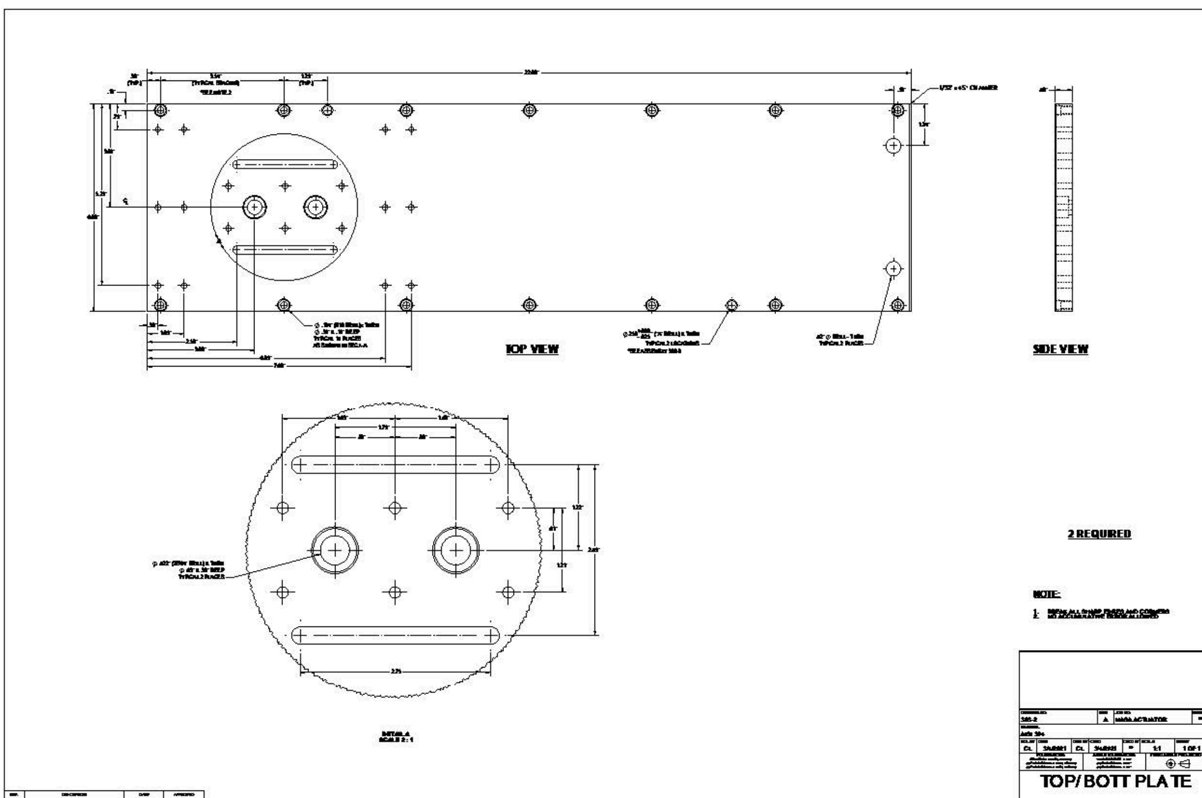

3.750"
.375" (TYP.)
.625" (TYP.)
∅ .422" (27/64" DRILL) X THRU TYPICAL 2 PLACES
.776"
2.000"
4.000"
3.225"
R.13"
∅ .250" (7/32" DRILL) X THRU 1/4"-24 UNF X THRU TYPICAL 2 PLACES
.500"
3.250"
TOP VIEW
.281"
.218"
.250"
45° CHAMFER TYPICAL ALL LOCATIONS
16
SIDE VIEW
2 REQUIRED
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
REV.
DESCRIPTION
DATE
BY
MOUNTING PLATE
302-2 - Mounting Plate
SOLIDWORKS Educational Product. For Instructional Use Only.

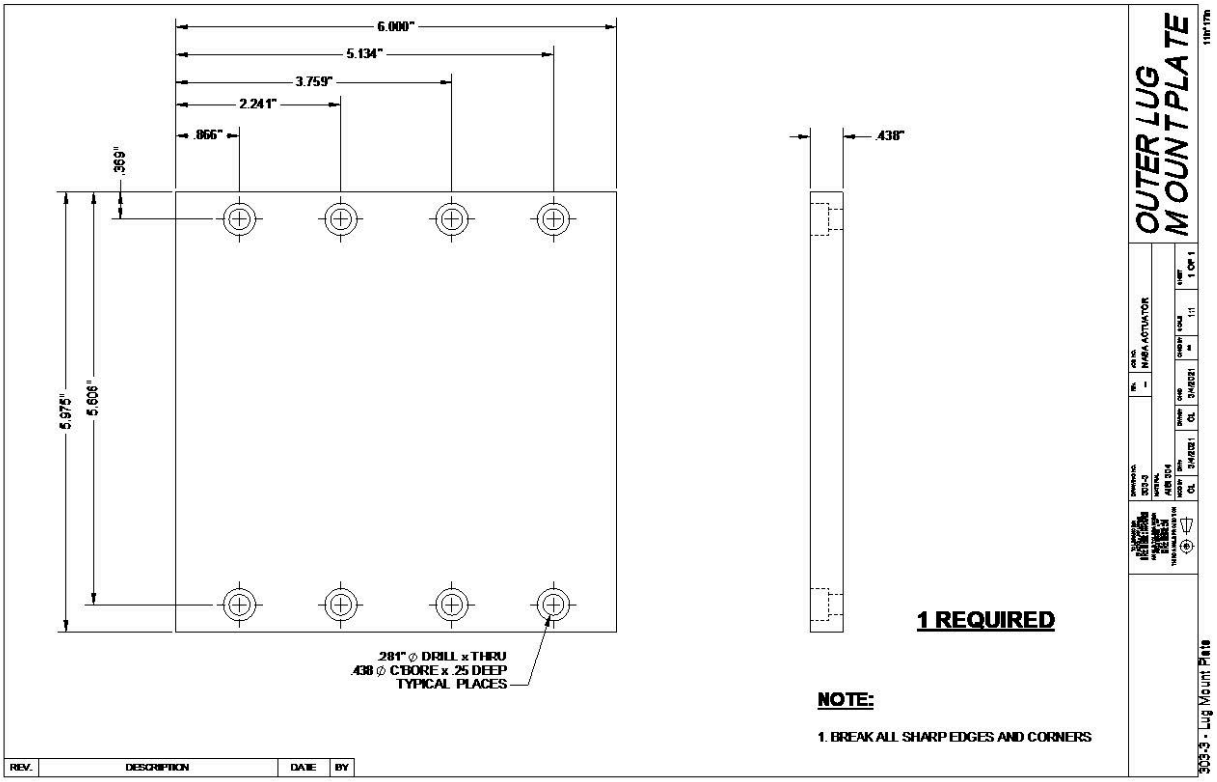

∅ 1.008" X THRU
1.875"
R.688"
R.375"
(.250")
.500" (TYP.)
.063"
FRONT VIEW
SIDE VIEW
11 REQUIRED
NOTE:
1. THIS DRAWING CAN BE USED AS A DIGITAL CUT FILE.
REV.
DESCRIPTION
DATE
APPROVED
REVISIONS
CENTER CARRIAGE PULLEY LINER
302-3 - Pulley Liner
SOLIDWORKS Educational Product. For Instructional Use Only.

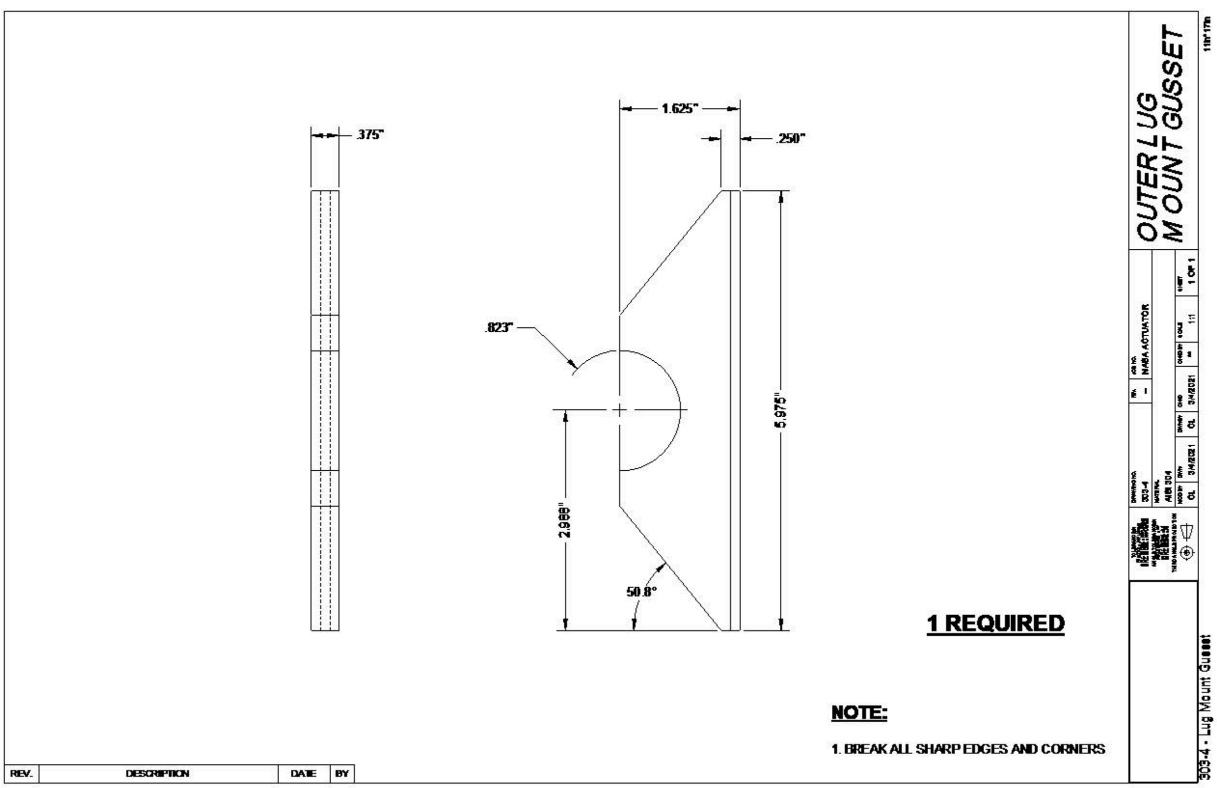

CENTER CARRIAGE
SLIDE BLOCK
3.750"
.375" (TYP.)
.625" (TYP.)
16
.750"
.375"
A
.005" A
.406"
⌀ .281" ("K" DRILL) X THRU
TYPICAL 2 PLACES
⌀ .422" ("R" DRILL) X THRU
3/8"-24 UNF X THRU
TYPICAL 2 PLACES
FRONT VIEW
SIDE VIEW
2 REQUIRED
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
REV.
DESCRIPTION
DATE
APPROVED
REVISIONS
NASA ACTUATOR
302-4
1 OF 1
302-4 - Slide Block
SOLIDWORKS Educational Product. For Instructional Use Only.

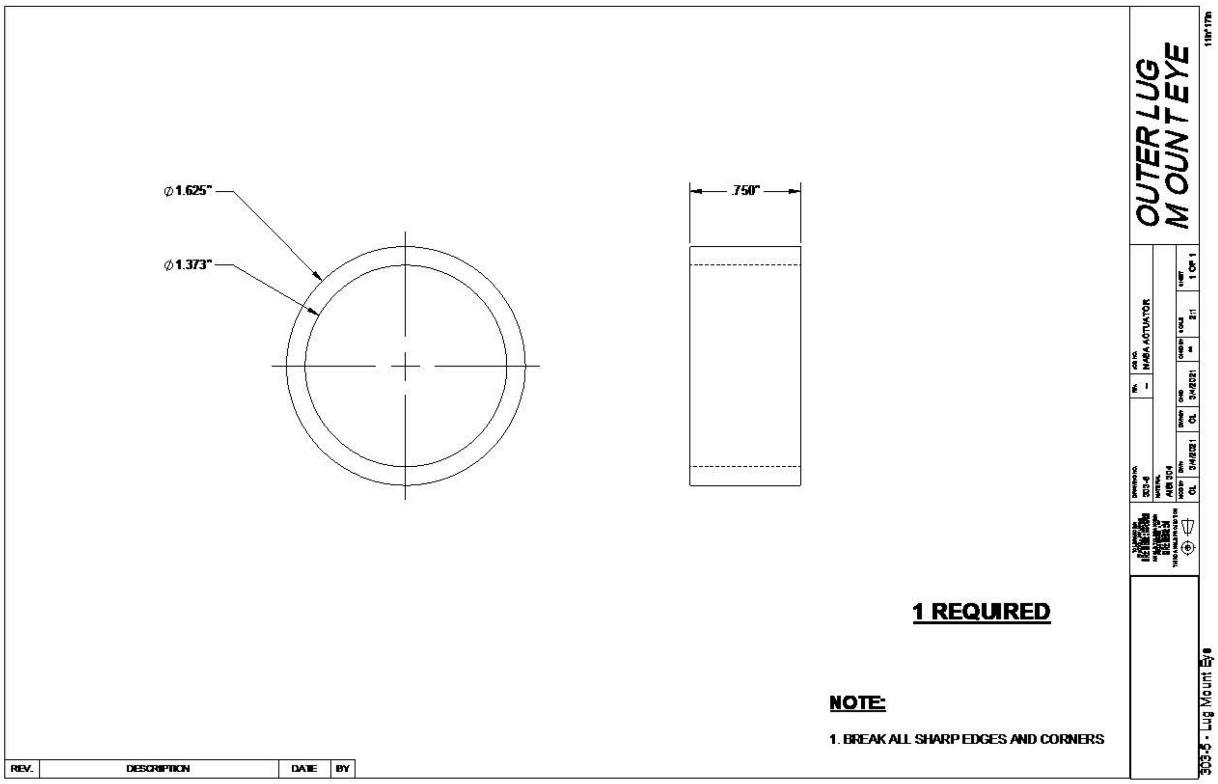

CENTER CARRIAGE
TOP/BOTT PLATE
4.000"
2.000"
.188"
R.125"
.375"
3.750"
3.375"
2.750"
1.000"
.194" ⌀ DRILL x THRU
.313 ⌀ CBORE x .16 DEEP FAR SIDE
TYPICAL 6 PLACES
⌀ .281" ("K" DRILL) X THRU
⌀ .438" X .25" DEEP
TYPICAL 2 PLACES
.339" ⌀ ("R") DRILL - THRU
3/8"-24 UNF TAP - THRU
TYPICAL 2 PLACES
TOP VIEW
SIDE VIEW
2 REQUIRED
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
REV.
DESCRIPTION
DATE
BY
NASA ACTUATOR
302-5
AISI 304
1 OF 1
302-5 - Top-Bott Plate
SOLIDWORKS Educational Product. For Instructional Use Only.

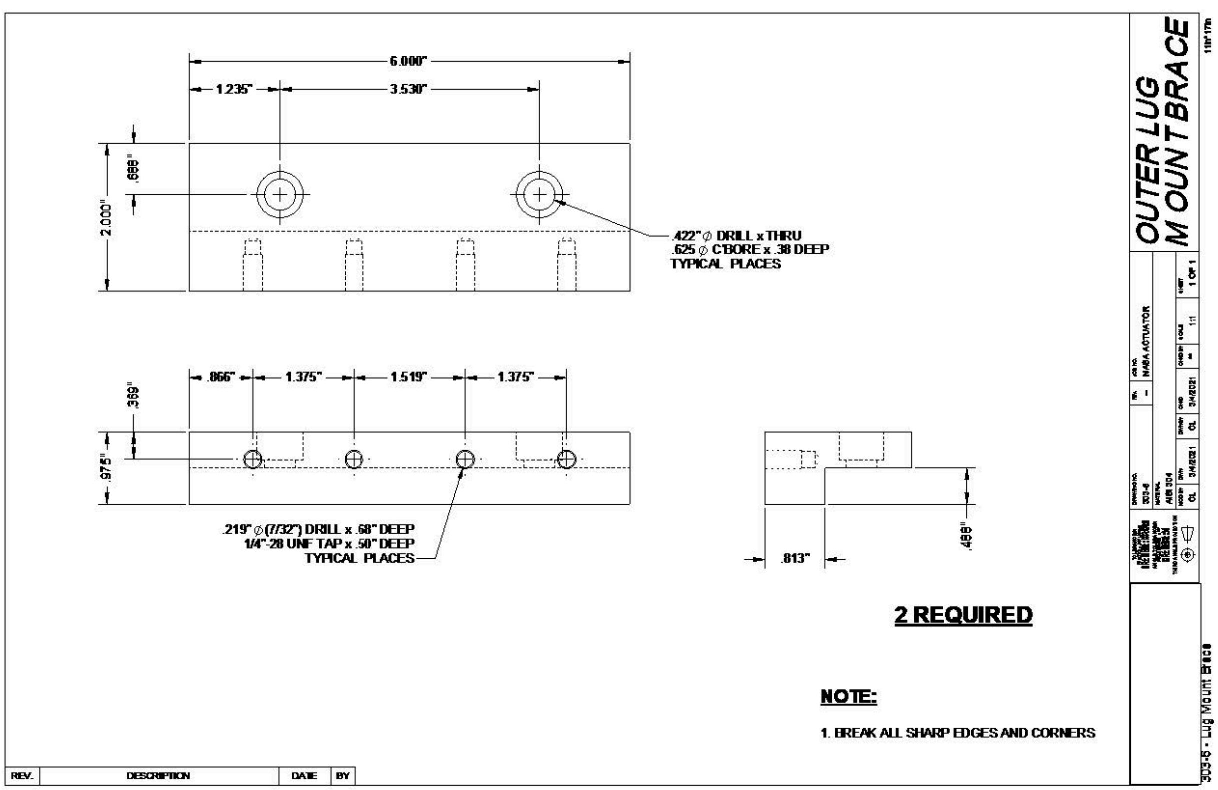

SIDE VIEW
FRONT VIEW
2 REQUIRED
NOTE:
OUTER SIDE PLATE

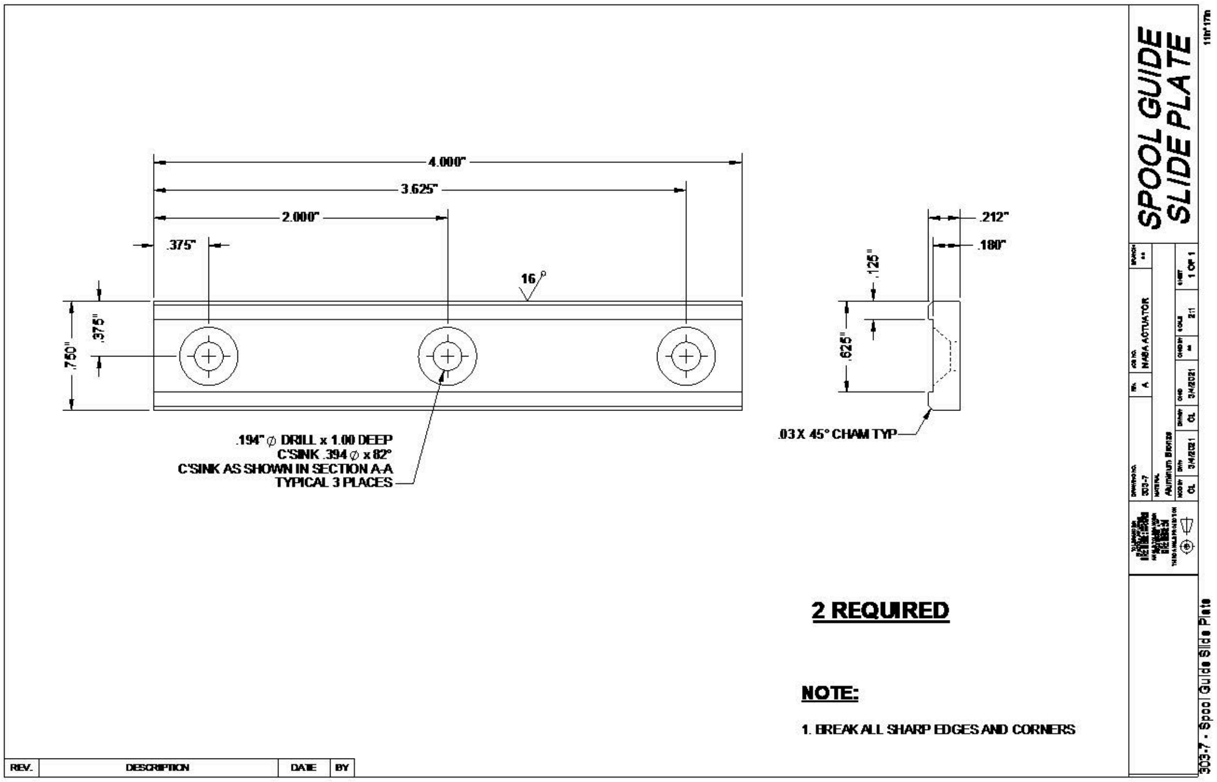

TOP VIEW
SIDE VIEW
2 REQUIRED
NOTE:
TOP/ BOTT PLATE


6.000"
5.134"
3.759"
2.241"
.866"
.369"
.438"
5.975"
5.606"
.281" ∅ DRILL x THRU
.438 ∅ C'BORE x .25 DEEP
TYPICAL PLACES
1 REQUIRED
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
REV.
DESCRIPTION
DATE
BY
OUTER LUG
MOUNT PLATE
NASA ACTUATOR
AISI 304
303-3 - Lug Mount Plate

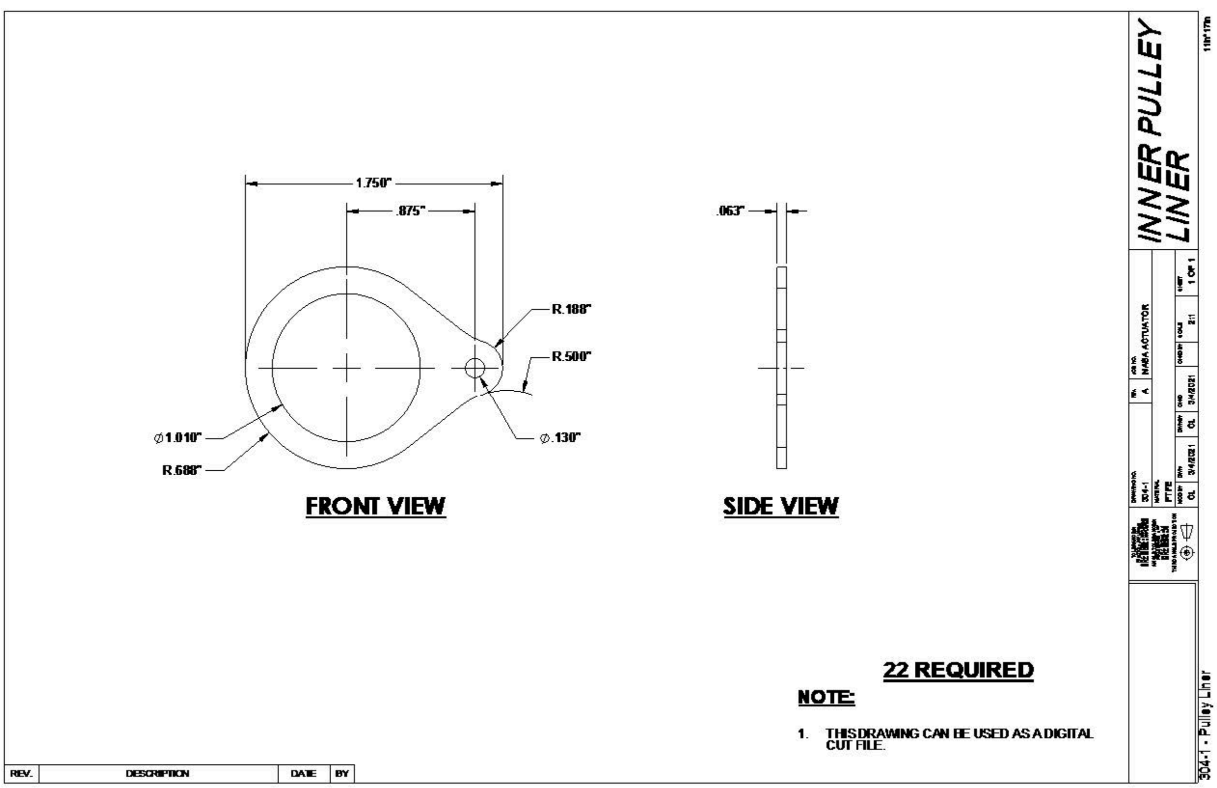

.375"
1.625"
.250"
.823"
5.975"
2.988"
50.8°
1 REQUIRED
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
REV.
DESCRIPTION
DATE
BY
OUTER LUG
MOUNT GUSSET
NASA ACTUATOR
AISI 304
303-4 - Lug Mount Gusset


OUTER LUG
MOUNT EYE
11in" 17in
Ø1.625"
Ø1.373"
.750"
1 REQUIRED
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
NASA ACTUATOR
303-5
AISI 304
1 OF 1
303-5 - Lug Mount Eye
REV.
DESCRIPTION
DATE
BY

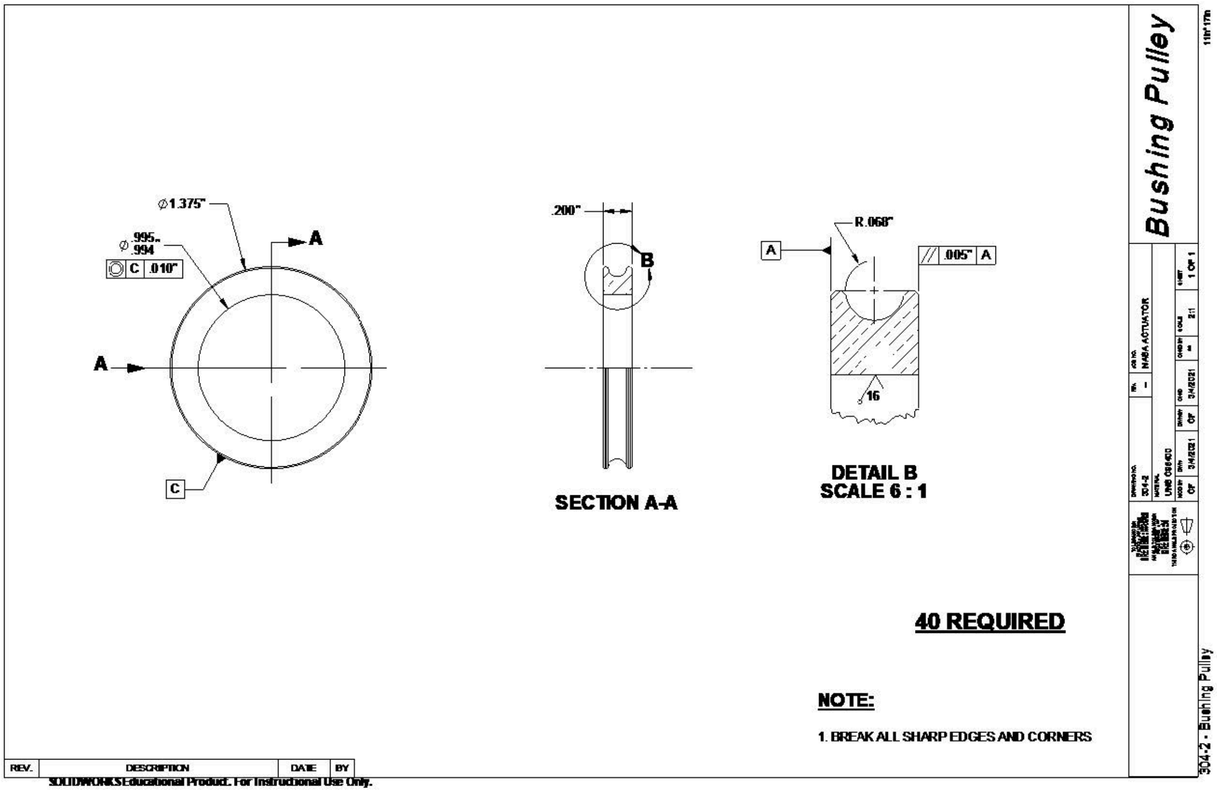

OUTER LUG
MOUNT BRACE
6.000"
1.235"
3.530"
.688"
2.000"
.422" Ø DRILL x THRU
.625 Ø C'BORE x .38 DEEP
TYPICAL PLACES
.866"
1.375"
1.519"
1.375"
.369"
.975"
.219" Ø (7/32") DRILL x .68" DEEP
1/4"-28 UNF TAP x .50" DEEP
TYPICAL PLACES
.813"
.488"
2 REQUIRED
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
NASA ACTUATOR
303-6
AISI 304
1 OF 1
303-6 - Lug Mount Brace
REV.
DESCRIPTION
DATE
BY

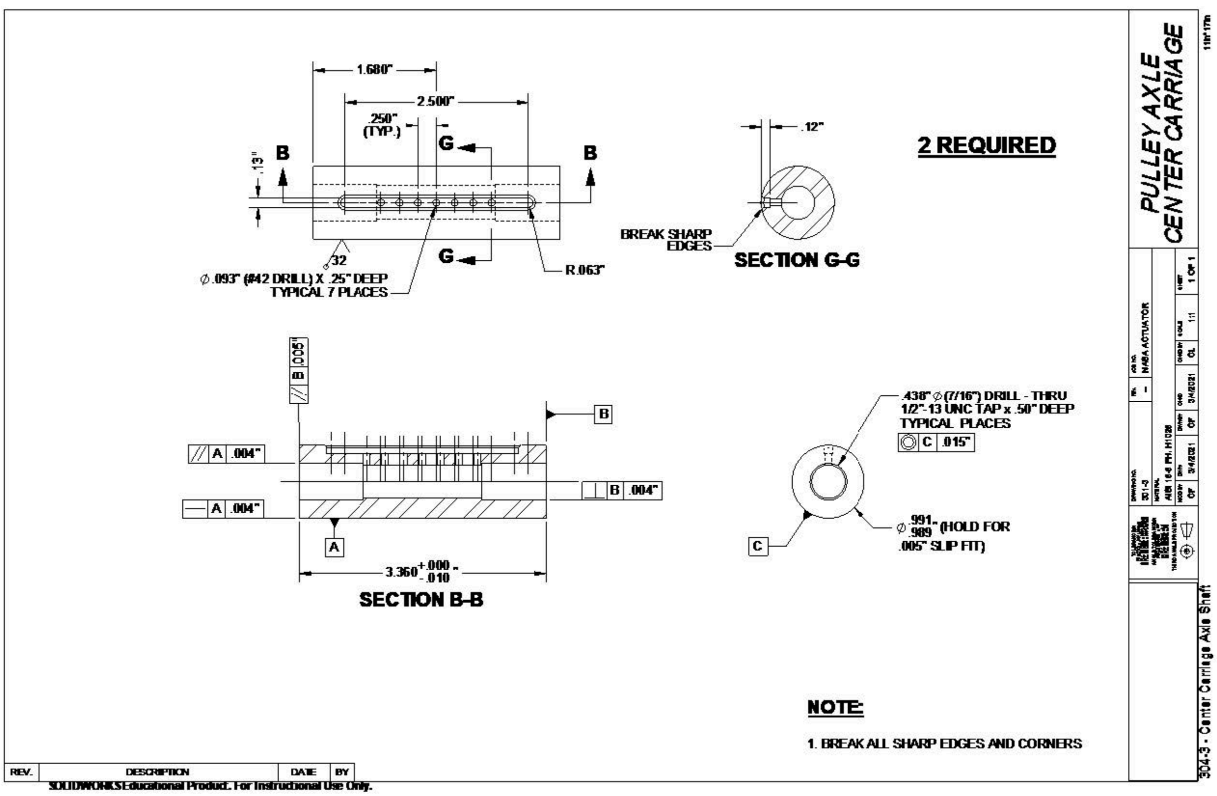


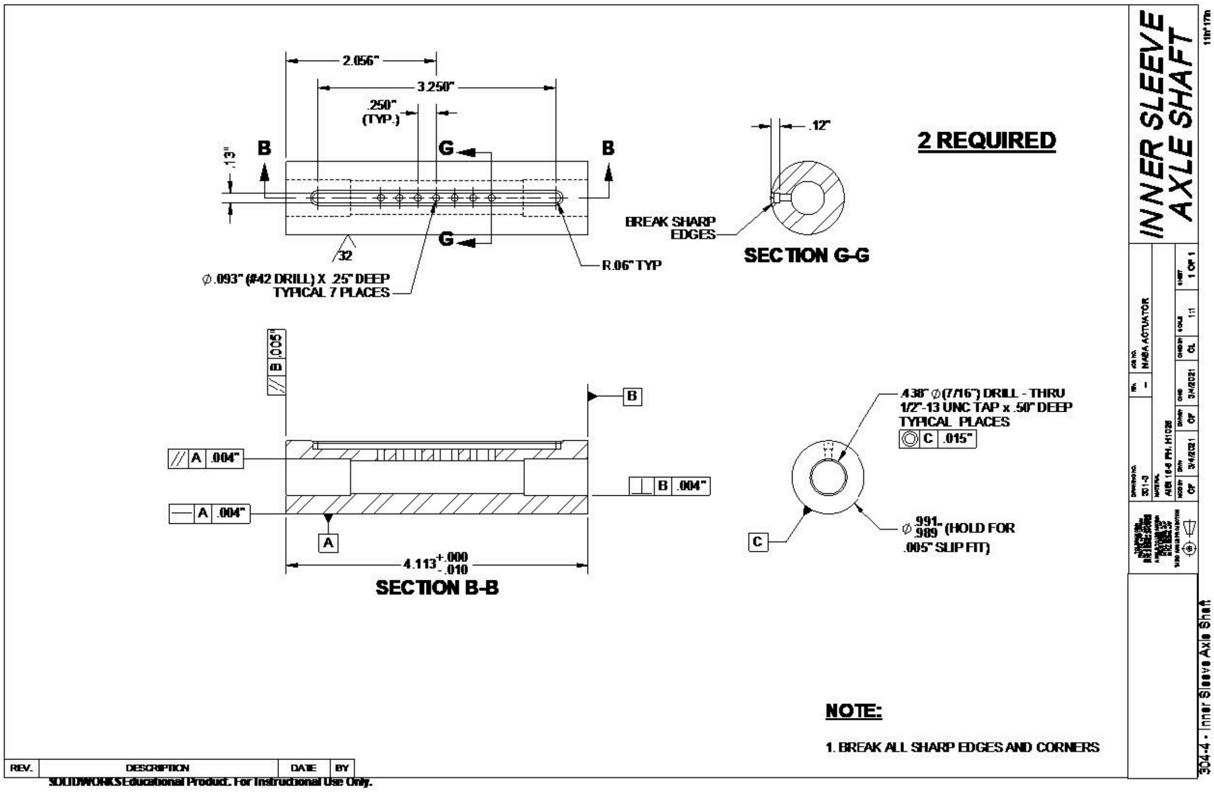
4.000"
3.625"
2.000"
.375"
16
.750"
.375"
.212"
.180"
.125"
.625"
.03 X 45° CHAM TYP
.194" ⌀ DRILL x 1.00 DEEP
C'SINK .394 ⌀ x 82°
C'SINK AS SHOWN IN SECTION A-A
TYPICAL 3 PLACES
2 REQUIRED
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
REV.
DESCRIPTION
DATE
BY
SPOOL GUIDE SLIDE PLATE
NASA ACTUATOR
303-7
Aluminum Bronze
1 OF 1
303-7 - Spool Guide Slide Plate

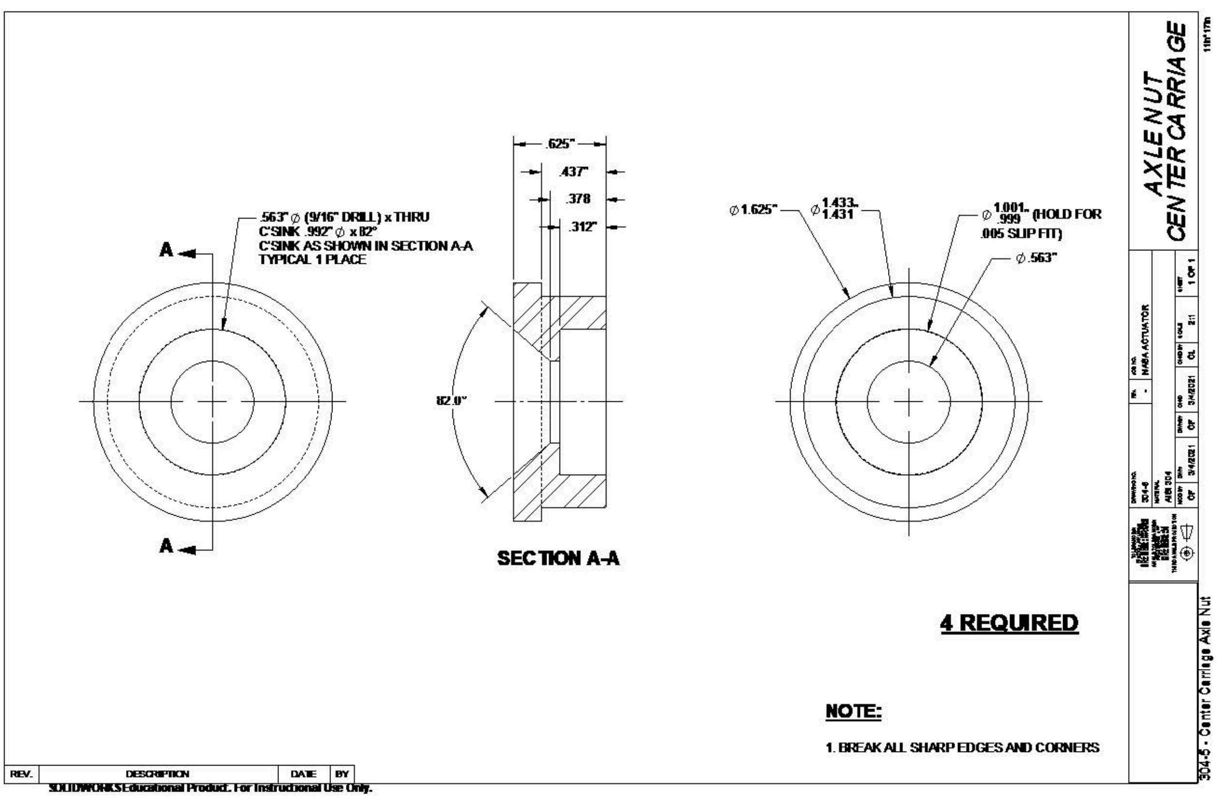


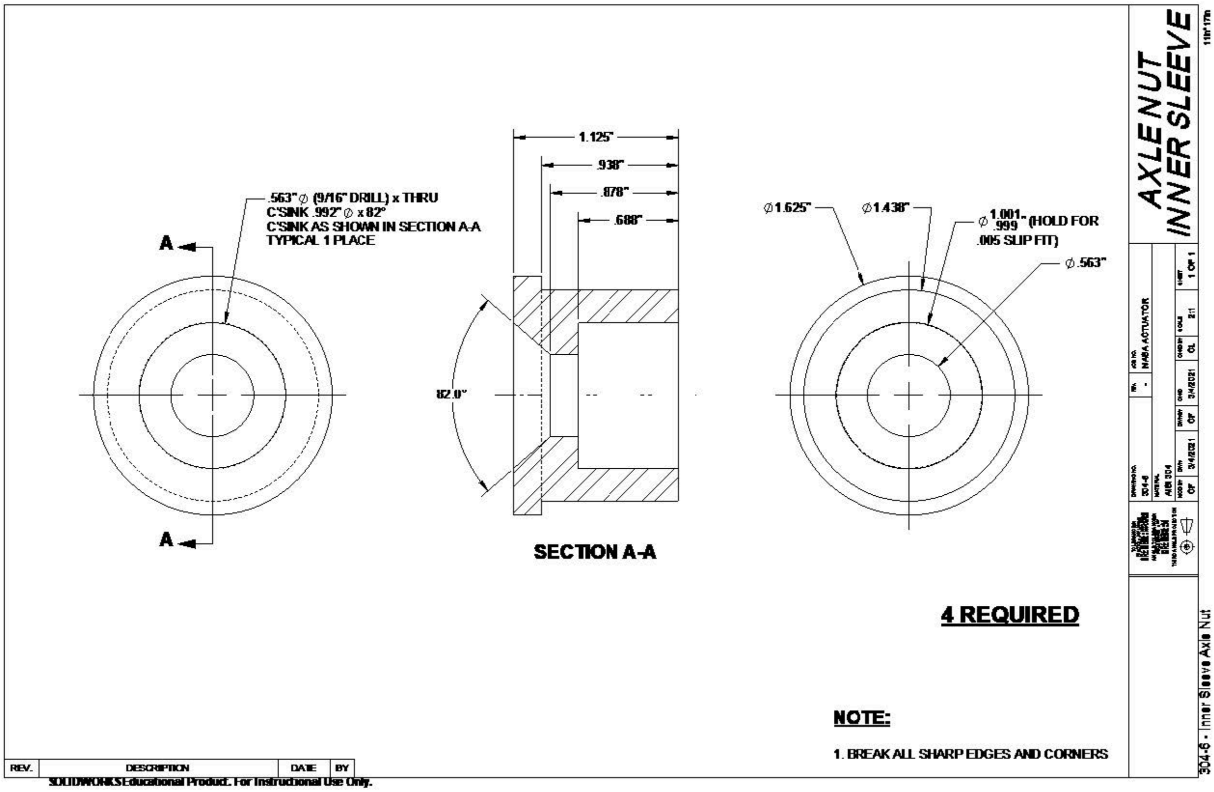
1.750"
.875"
.063"
R.188"
R.500"
⌀1.010"
R.688"
⌀.130"
FRONT VIEW
SIDE VIEW
22 REQUIRED
NOTE:
1. THIS DRAWING CAN BE USED AS A DIGITAL CUT FILE.
REV.
DESCRIPTION
DATE
BY
INNER PULLEY LINER
NASA ACTUATOR
304-1
PTFE
1 OF 1
304-1 - Pulley Liner

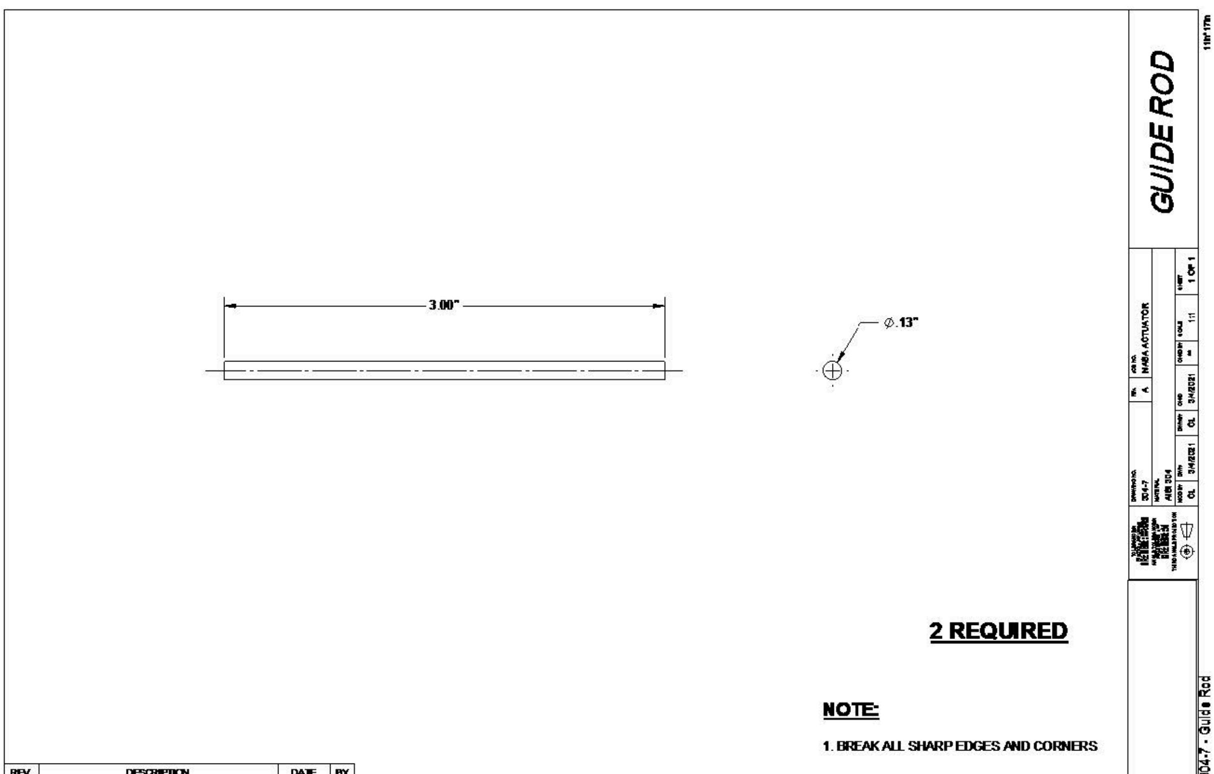

Bushing Pulley
Ø1.375"
Ø .995 .994
.010"
.200"
R.068"
.005"
SECTION A-A
DETAIL B
SCALE 6 : 1
40 REQUIRED
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
REV.
DESCRIPTION
DATE
BY
SOLIDWORKS Educational Product. For Instructional Use Only.
NASA ACTUATOR
304-2 - Bushing Pulley

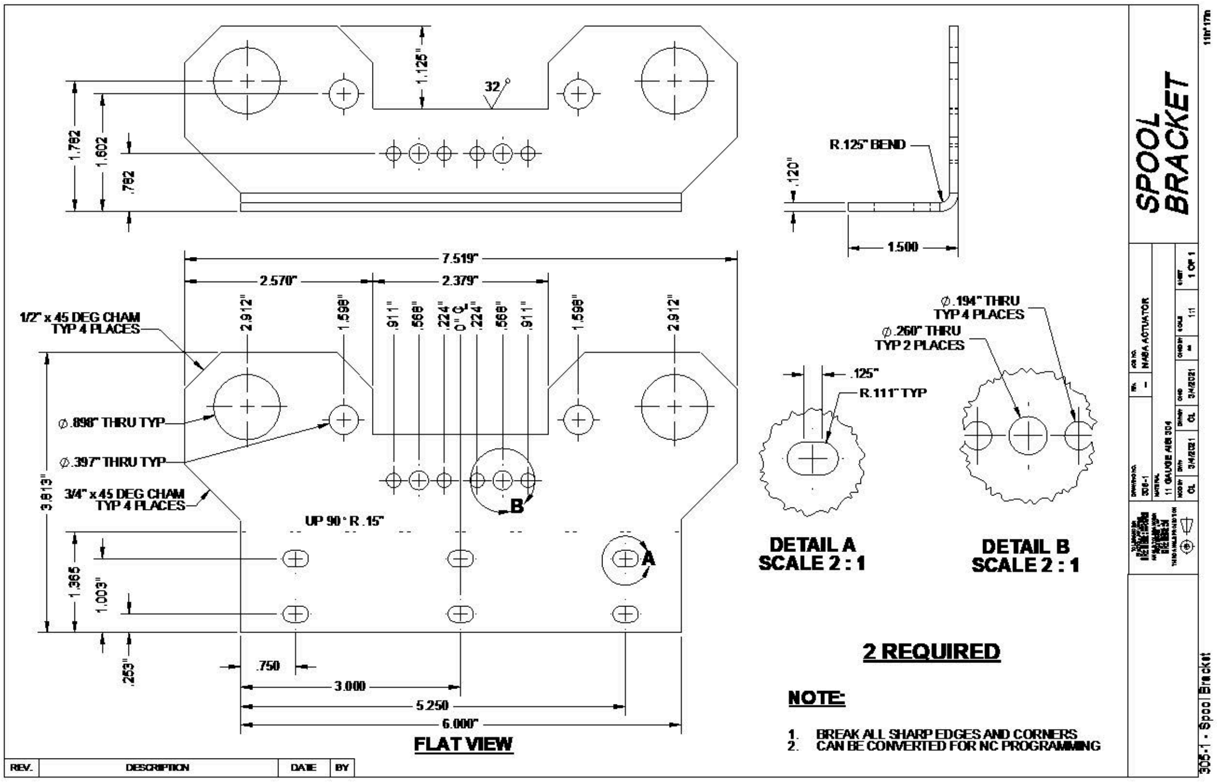

PULLEY AXLE
CENTER CARRIAGE
1.680"
2.500"
.250" (TYP.)
.13"
.12"
2 REQUIRED
BREAK SHARP EDGES
SECTION G-G
Ø.093" (#42 DRILL) X .25" DEEP
TYPICAL 7 PLACES
R.063"
.005"
.004"
.004"
.438" Ø (7/16") DRILL - THRU
1/2"-13 UNC TAP x .50" DEEP
TYPICAL PLACES
.015"
.004"
Ø .991 .989 (HOLD FOR .005" SLIP FIT)
3.360 +.000 -.010
SECTION B-B
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
REV.
DESCRIPTION
DATE
BY
SOLIDWORKS Educational Product. For Instructional Use Only.
NASA ACTUATOR
304-3 - Center Carriage Axle Shaft

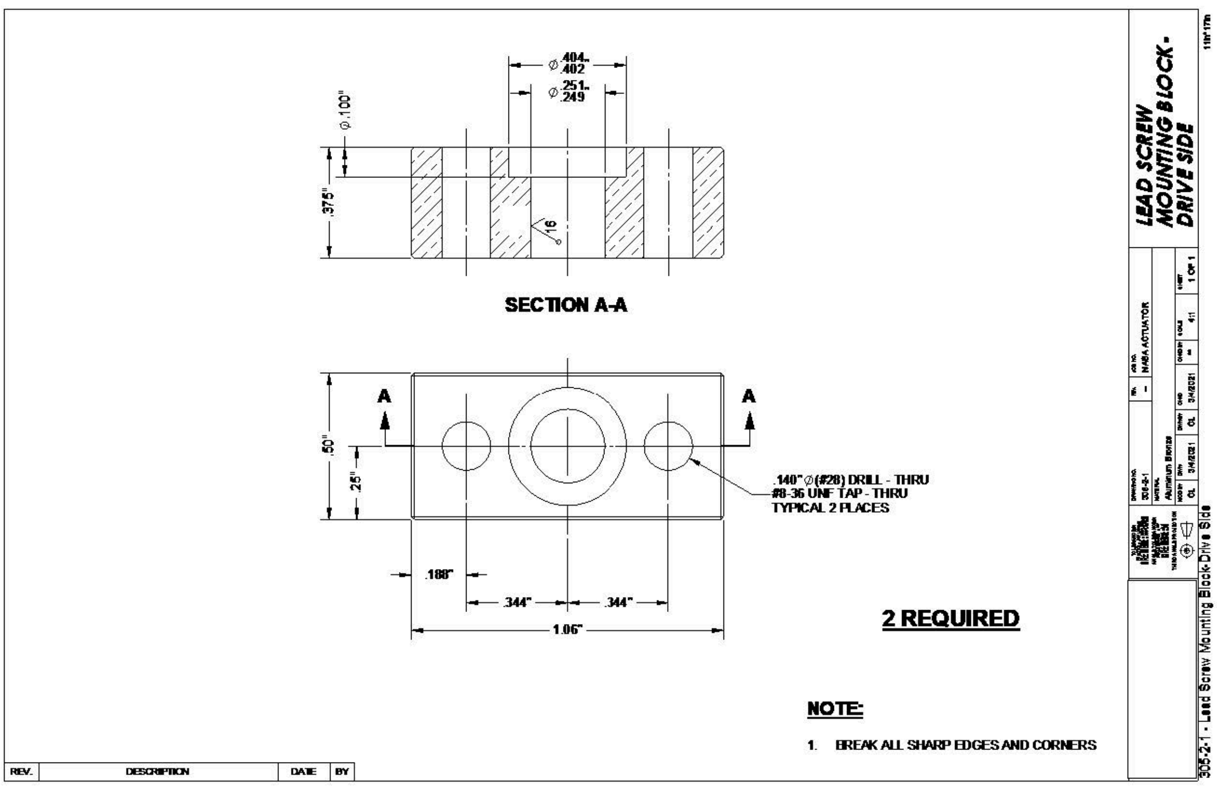

INNER SLEEVE AXLE SHAFT
2 REQUIRED
2.056"
3.250"
.250" (TYP.)
.13"
B
G
.12"
BREAK SHARP EDGES
SECTION G-G
R.06" TYP
32
Ø .093" (#42 DRILL) X .25" DEEP TYPICAL 7 PLACES
B .005"
A .004"
B .004"
A .004"
A
.438" Ø (7/16") DRILL - THRU 1/2"-13 UNC TAP x .50" DEEP TYPICAL PLACES
C .015"
C
Ø .991-.989 (HOLD FOR .005" SLIP FIT)
4.113 +.000 -.010
SECTION B-B
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
NASA ACTUATOR
1 OF 1
REV.
DESCRIPTION
DATE
BY
SOLIDWORKS Educational Product. For Instructional Use Only.
304-4 - Inner Sleeve Axle Shaft

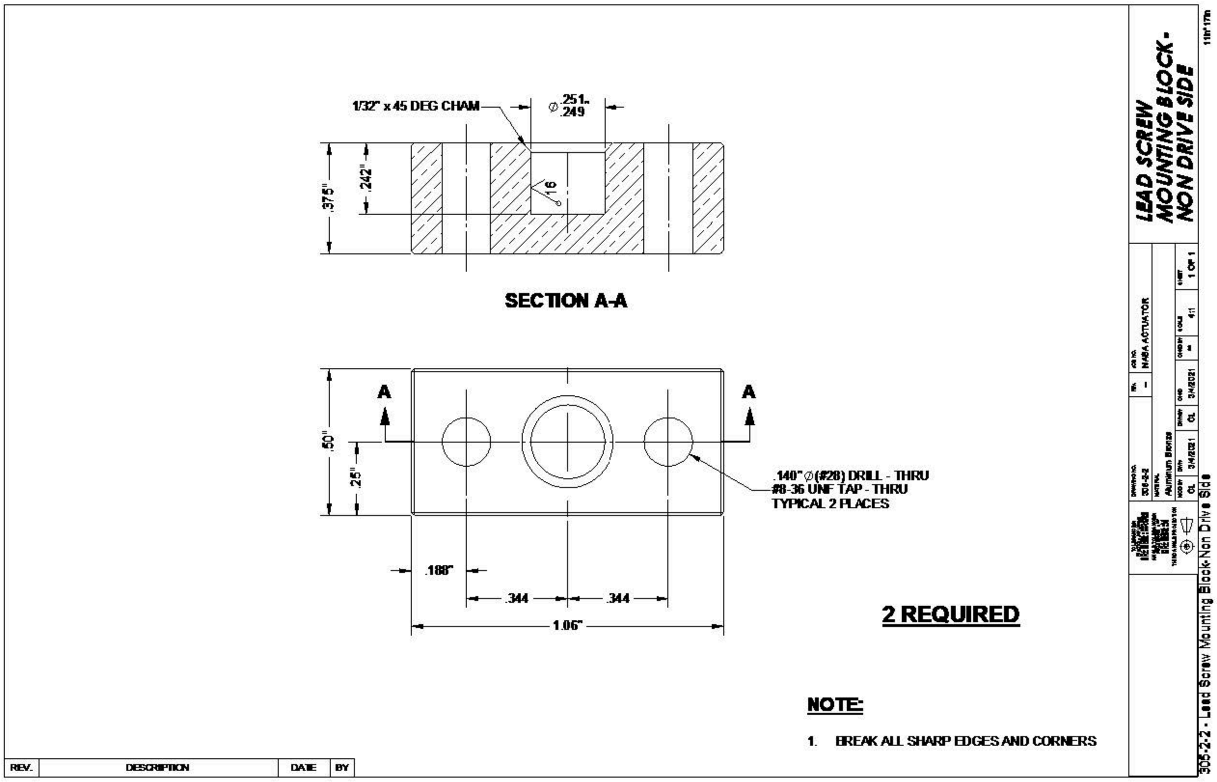

AXLE NUT CENTER CARRIAGE
.625"
.437"
.378"
.312"
A
.563" Ø (9/16" DRILL) x THRU C'SINK .992" Ø x 82° C'SINK AS SHOWN IN SECTION A-A TYPICAL 1 PLACE
82.0°
SECTION A-A
Ø 1.625"
Ø 1.433-1.431
Ø 1.001-.999 (HOLD FOR .005 SLIP FIT)
Ø .563"
4 REQUIRED
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
NASA ACTUATOR
1 OF 1
REV.
DESCRIPTION
DATE
BY
SOLIDWORKS Educational Product. For Instructional Use Only.
304-5 - Center Carriage Axle Nut

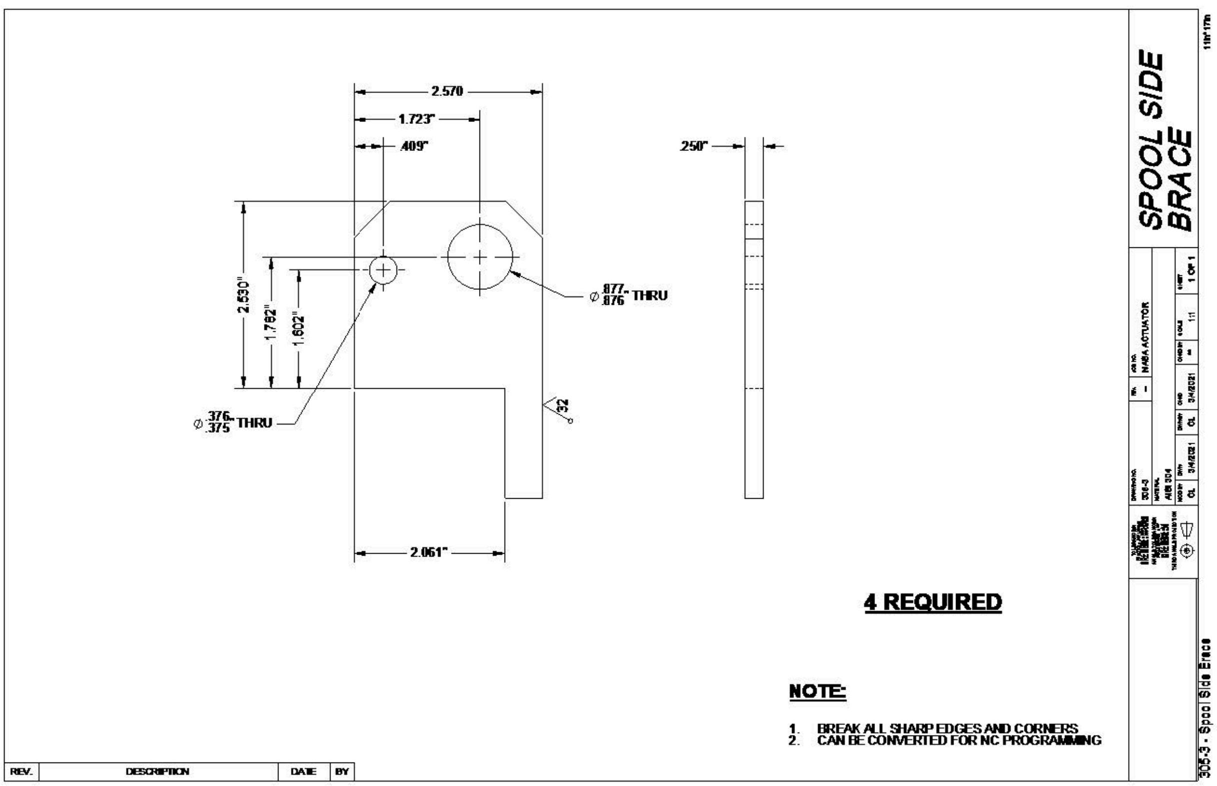

AXLE NUT INNER SLEEVE
1.125"
.938"
.878"
.688"
.563" ⌀ (9/16" DRILL) x THRU
C'SINK .992" ⌀ x 82°
C'SINK AS SHOWN IN SECTION A-A
TYPICAL 1 PLACE
A
A
82.0°
SECTION A-A
⌀1.625"
⌀1.438"
⌀ 1.001 .999 " (HOLD FOR .005 SLIP FIT)
⌀ .563"
4 REQUIRED
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
NASA ACTUATOR
304-6
AISI 304
1 OF 1
REV.
DESCRIPTION
DATE
BY
SOLIDWORKS Educational Product. For Instructional Use Only.
304-6 - Inner Sleeve Axle Nut

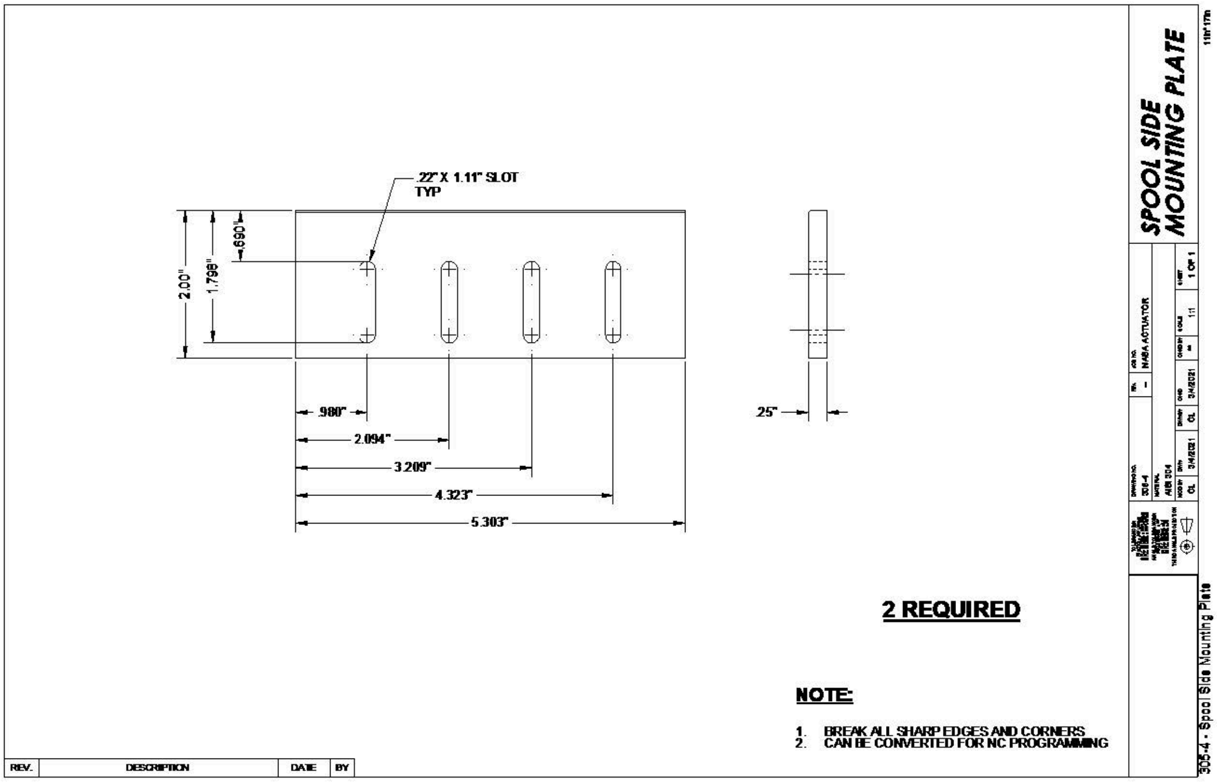

GUIDE ROD
3.00"
⌀.13"
2 REQUIRED
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
NASA ACTUATOR
304-7
AISI 304
1 OF 1
REV.
DESCRIPTION
DATE
BY
SOLIDWORKS Educational Product. For Instructional Use Only.
304-7 - Guide Rod


SPOOL BRACKET
2 REQUIRED
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
2. CAN BE CONVERTED FOR NC PROGRAMMING
FLAT VIEW
DETAIL A
SCALE 2 : 1
DETAIL B
SCALE 2 : 1
R.125" BEND
1/2" x 45 DEG CHAM
TYP 4 PLACES
3/4" x 45 DEG CHAM
TYP 4 PLACES
Ø.898" THRU TYP
Ø.397" THRU TYP
Ø.194" THRU
TYP 4 PLACES
Ø.260" THRU
TYP 2 PLACES
R.111" TYP
UP 90° R .15"
7.519"
2.570"
2.379"
1.500
6.000"
5.250
3.000
.750
305-1 - Spool Bracket

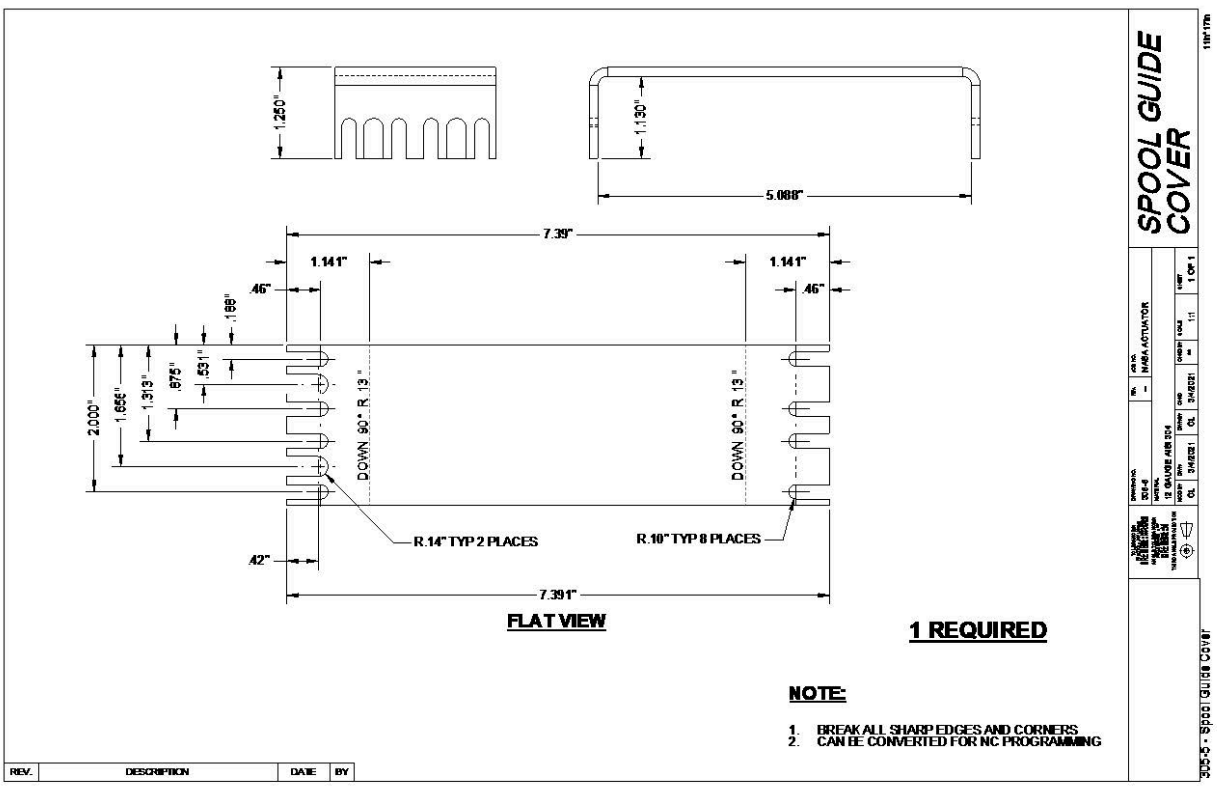

LEAD SCREW MOUNTING BLOCK - DRIVE SIDE
SECTION A-A
2 REQUIRED
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
.140" Ø(#28) DRILL - THRU
#8-36 UNF TAP - THRU
TYPICAL 2 PLACES
1.06"
.344"
.188"
305-2-1 - Lead Screw Mounting Block-Drive Side


1/32" x 45 DEG CHAM
Ø .251 .249
.375"
.242"
16
SECTION A-A
A
A
.50"
.25"
.140" Ø (#28) DRILL - THRU
#8-36 UNF TAP - THRU
TYPICAL 2 PLACES
.188"
.344
.344
1.06"
2 REQUIRED
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
REV.
DESCRIPTION
DATE
BY
LEAD SCREW MOUNTING BLOCK - NON DRIVE SIDE
NASA ACTUATOR
305-2-2
Aluminum Bronze
1 OF 1
305-2-2 - Lead Screw Mounting Block-Non Drive Side

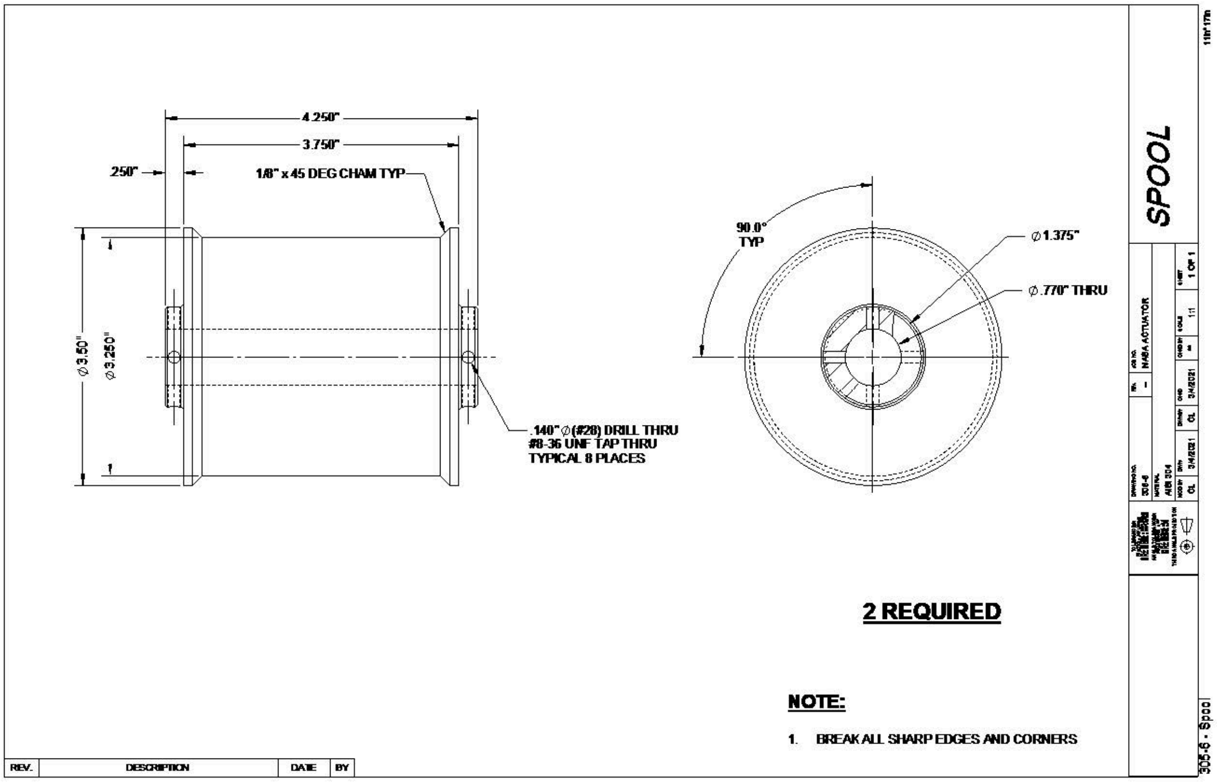

2.570
1.723"
.409"
.250"
2.530"
1.762"
1.602"
Ø .877 .876 THRU
Ø .376 .375 THRU
32
2.061"
4 REQUIRED
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
2. CAN BE CONVERTED FOR NC PROGRAMMING
REV.
DESCRIPTION
DATE
BY
SPOOL SIDE BRACE
NASA ACTUATOR
305-3
AISI 304
1 OF 1
305-3 - Spool Side Brace

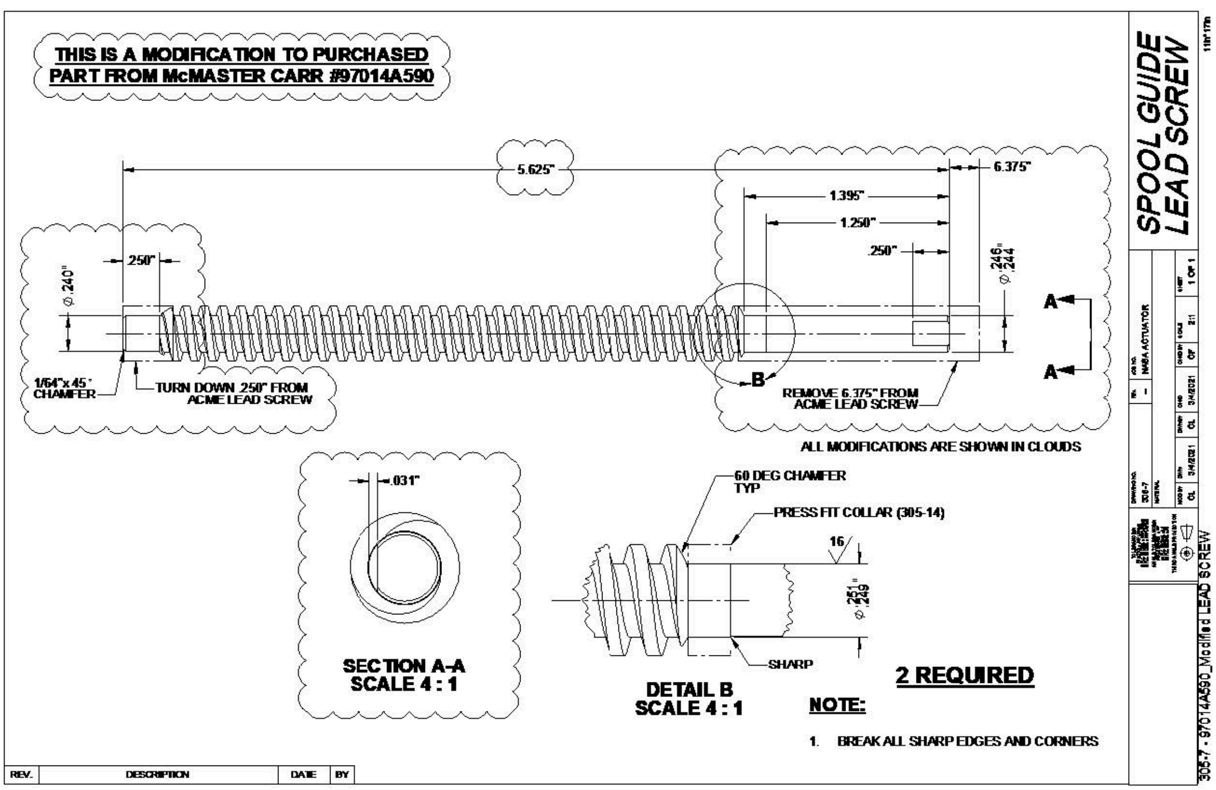


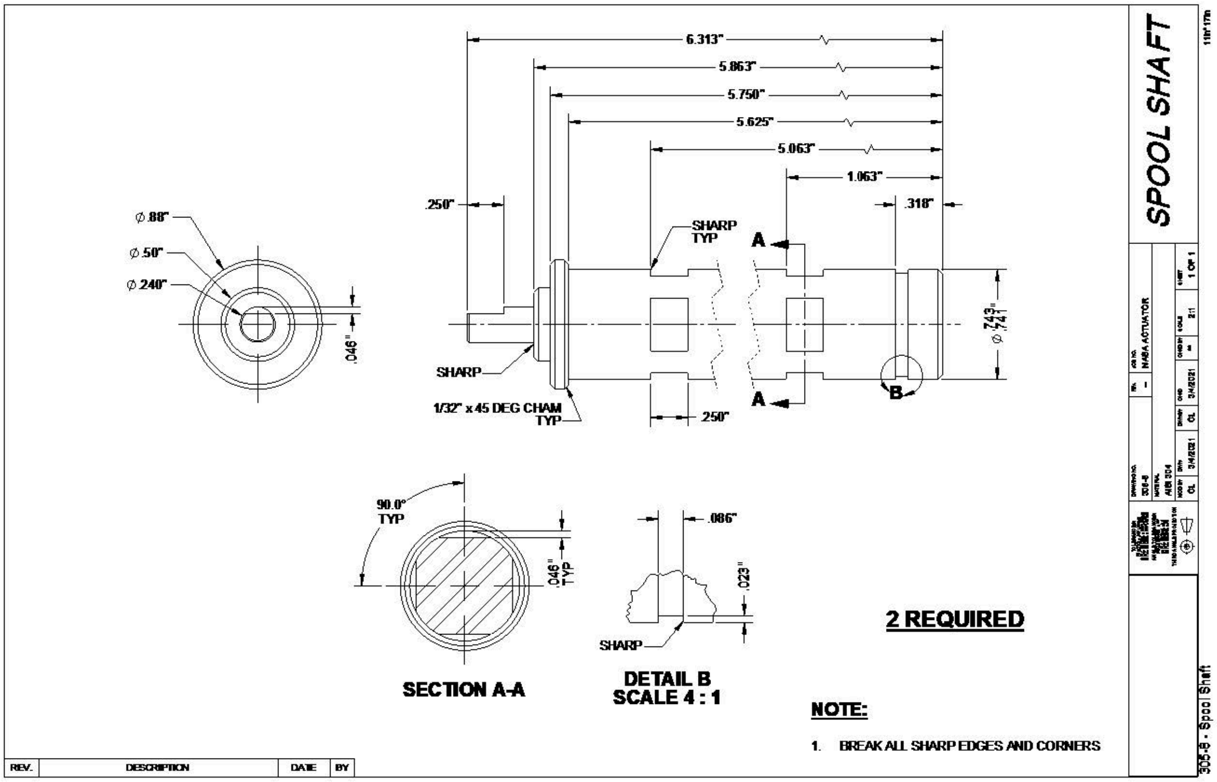
SPOOL SIDE MOUNTING PLATE
.22" X 1.11" SLOT TYP
2.00"
1.798"
.690"
.980"
2.094"
3.209"
4.323"
5.303"
.25"
2 REQUIRED
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
2. CAN BE CONVERTED FOR NC PROGRAMMING
REV.
DESCRIPTION
DATE
BY
NASA ACTUATOR
305-4
AISI 304
1:1
1 OF 1
3/4/2021
305-4 - Spool Side Mounting Plate
11in x 17in

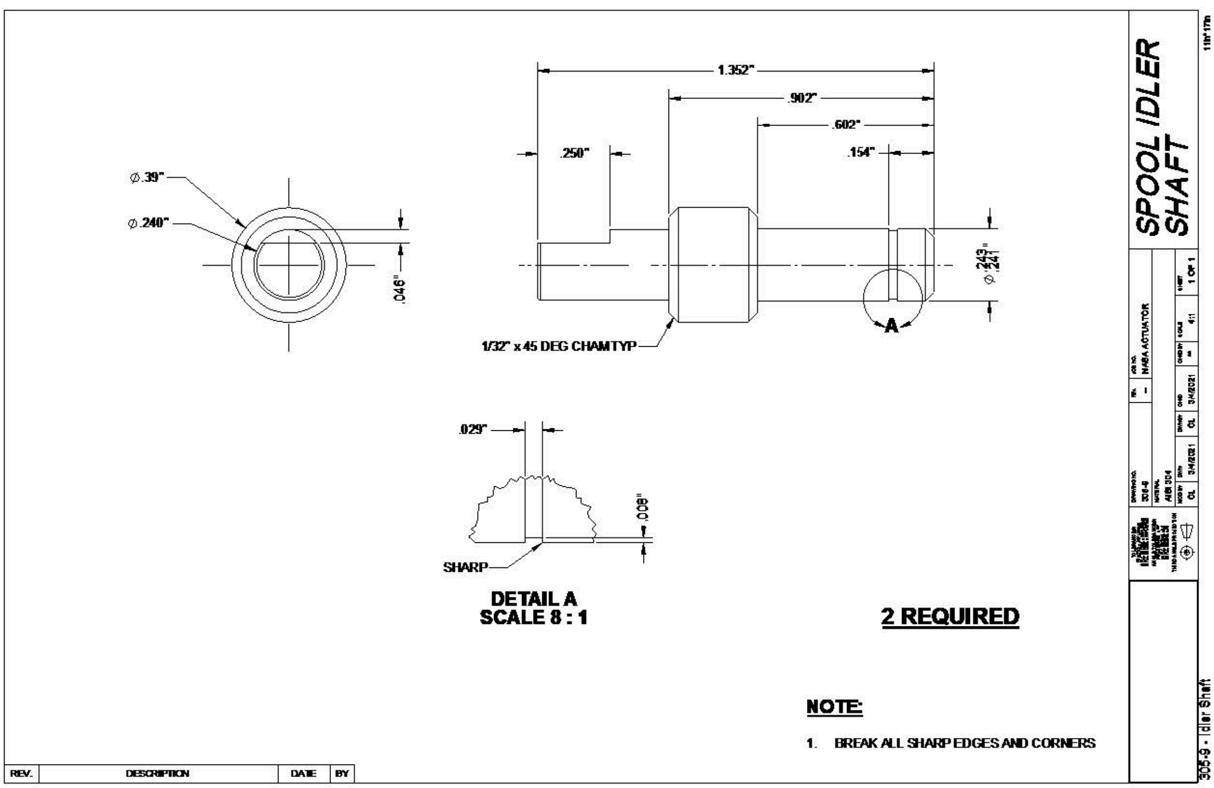


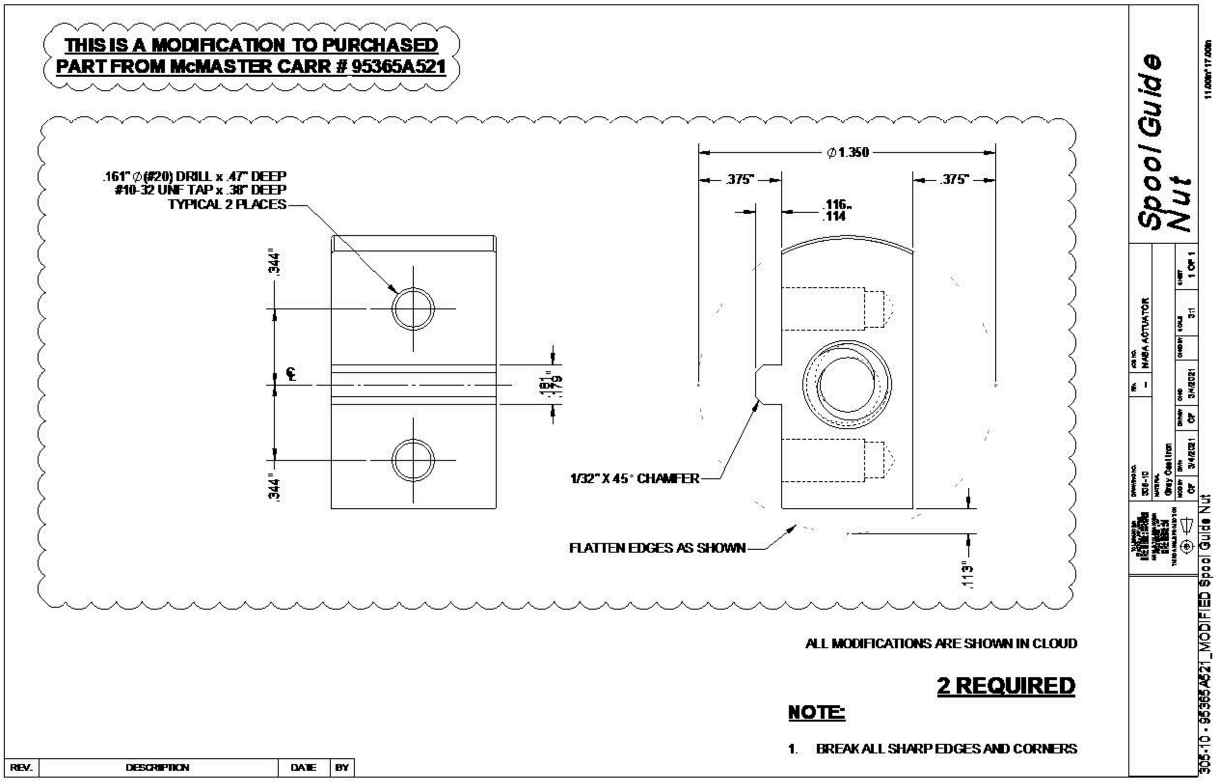
SPOOL GUIDE COVER
1.250"
1.130"
5.088"
7.39"
1.141"
1.141"
.46"
.46"
.188"
.531"
.875"
1.313"
1.656"
2.000"
DOWN 90° R.13"
DOWN 90° R.13"
R.14" TYP 2 PLACES
R.10" TYP 8 PLACES
.42"
7.391"
FLAT VIEW
1 REQUIRED
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
2. CAN BE CONVERTED FOR NC PROGRAMMING
REV.
DESCRIPTION
DATE
BY
NASA ACTUATOR
305-5
12 GAUGE AISI 304
1:1
1 OF 1
3/4/2021
305-5 - Spool Guide Cover
11in x 17in


4.250"
3.750"
.250"
1/8" x 45 DEG CHAM TYP
Ø 3.50"
Ø 3.250"
.140" Ø (#28) DRILL THRU
#8-36 UNF TAP THRU
TYPICAL 8 PLACES
90.0° TYP
Ø 1.375"
Ø .770" THRU
SPOOL
2 REQUIRED
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
REV.
DESCRIPTION
DATE
BY
305-6 - Spool

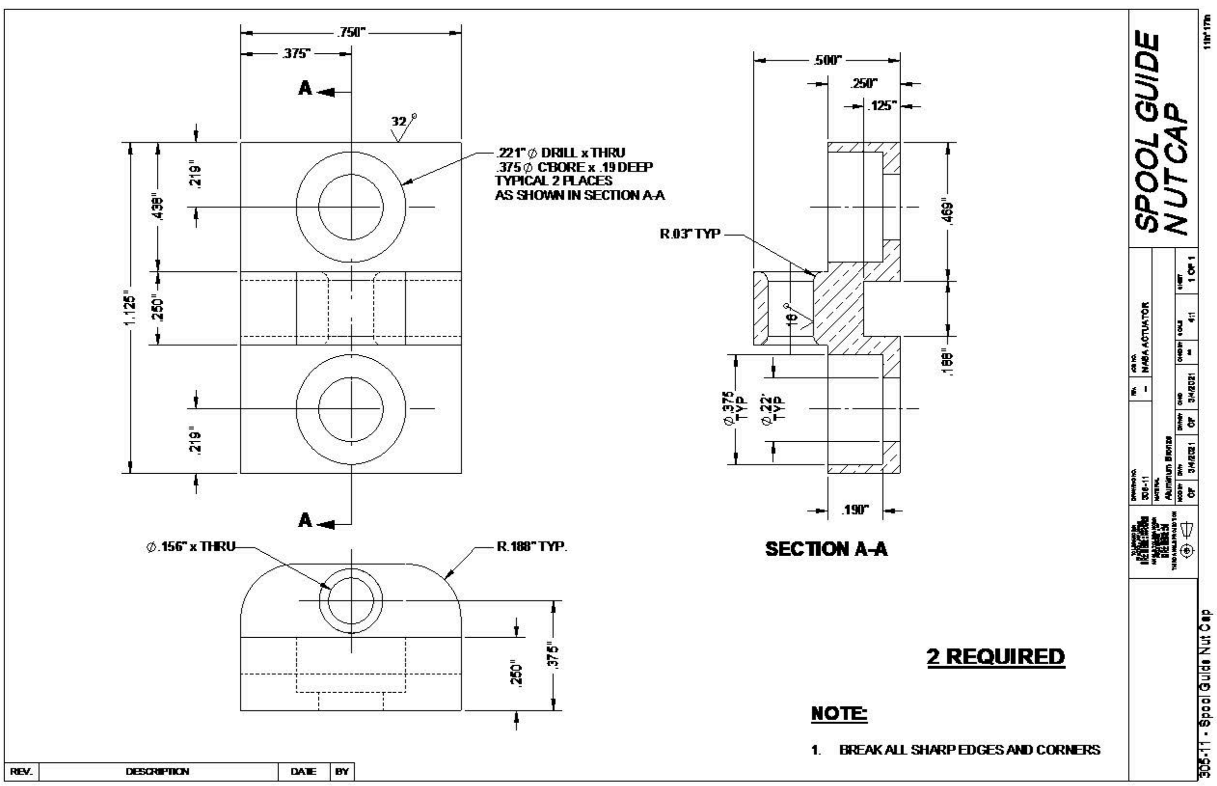

THIS IS A MODIFICATION TO PURCHASED
PART FROM McMASTER CARR #97014A590
5.625"
6.375"
1.395"
1.250"
.250"
.250"
Ø .240"
1/64" x 45° CHAMFER
TURN DOWN .250" FROM ACME LEAD SCREW
REMOVE 6.375" FROM ACME LEAD SCREW
ALL MODIFICATIONS ARE SHOWN IN CLOUDS
.031"
SECTION A-A
SCALE 4 : 1
60 DEG CHAMFER TYP
PRESS FIT COLLAR (305-14)
SHARP
DETAIL B
SCALE 4 : 1
2 REQUIRED
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
SPOOL GUIDE LEAD SCREW
REV.
DESCRIPTION
DATE
BY
305-7 - 97014A590_Modified LEAD SCREW


SPOOL SHAFT
6.313"
5.863"
5.750"
5.625"
5.063"
1.063"
.250"
.318"
SHARP TYP
SHARP
1/32" x 45 DEG CHAM TYP
.250"
Ø .88"
Ø .50"
Ø .240"
.046"
Ø .743" .741"
90.0° TYP
.046" TYP
.086"
.023"
SECTION A-A
DETAIL B SCALE 4 : 1
2 REQUIRED
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
REV.
DESCRIPTION
DATE
BY
NASA ACTUATOR
AISI 304
3/4/2021
2:1
1 OF 1
305-8 - Spool Shaft


SPOOL IDLER SHAFT
1.352"
.902"
.602"
.250"
.154"
Ø .39"
Ø .240"
.046"
Ø .243" .241"
1/32" x 45 DEG CHAM TYP
.029"
.008"
SHARP
DETAIL A SCALE 8 : 1
2 REQUIRED
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
REV.
DESCRIPTION
DATE
BY
NASA ACTUATOR
AISI 304
3/4/2021
4:1
1 OF 1
305-9 - Idler Shaft

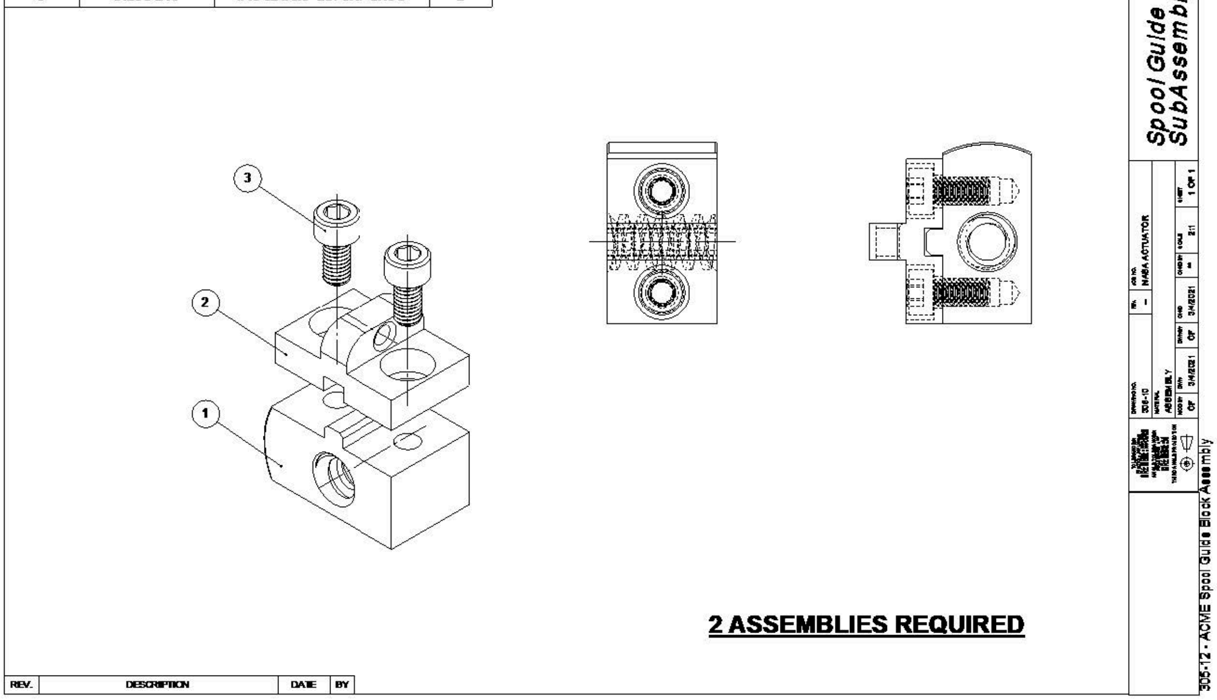


THIS IS A MODIFICATION TO PURCHASED PART FROM McMASTER CARR # 95365A521
.161" ∅(#20) DRILL x .47" DEEP
#10-32 UNF TAP x .38" DEEP
TYPICAL 2 PLACES
∅1.350
.375"
.116
.114
1/32" X 45° CHAMFER
FLATTEN EDGES AS SHOWN
.113"
ALL MODIFICATIONS ARE SHOWN IN CLOUD
2 REQUIRED
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
Spool Guide Nut
NASA ACTUATOR
305-10 - 95365A521_MODIFIED Spool Guide Nut

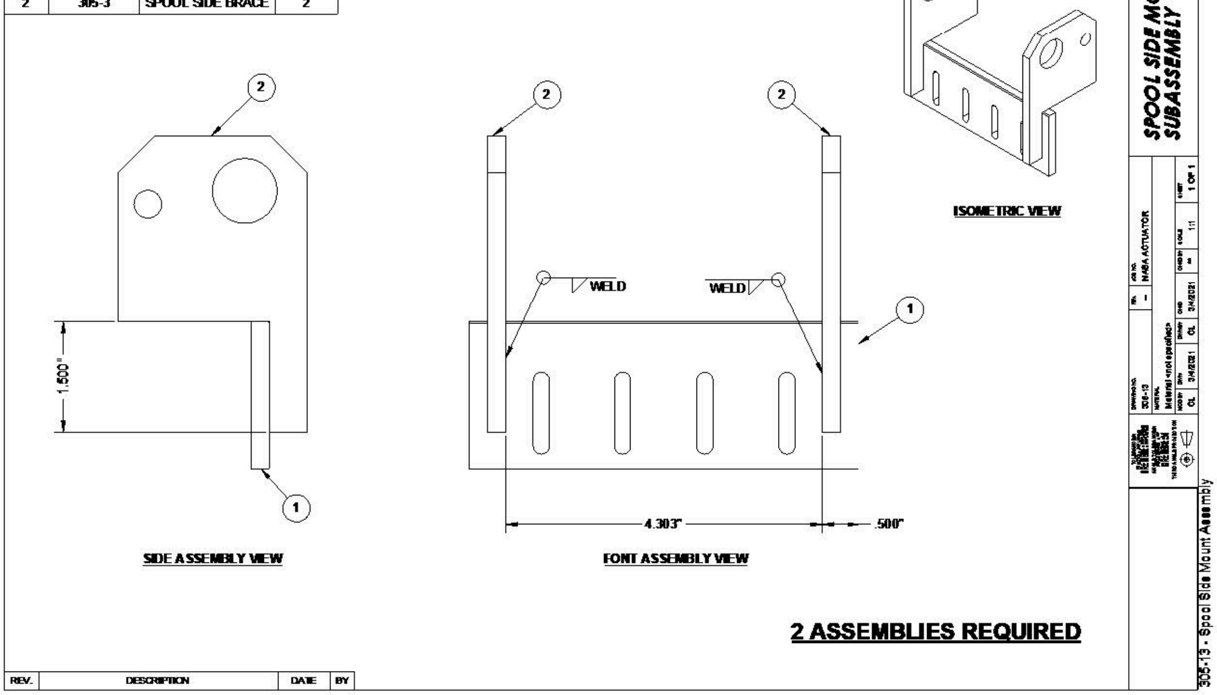


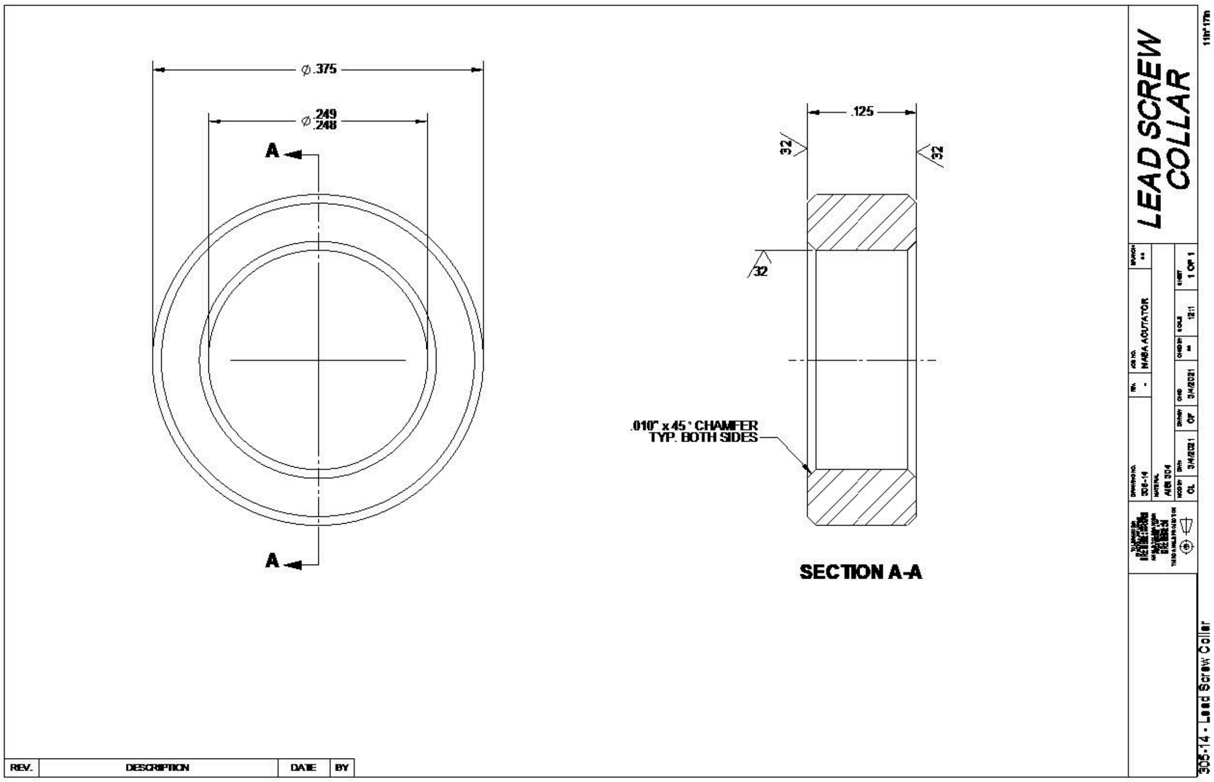
.750"
.375"
A
32
.221" ∅ DRILL x THRU
.375 ∅ C'BORE x .19 DEEP
TYPICAL 2 PLACES
AS SHOWN IN SECTION A-A
.219"
.438"
1.125"
.250"
∅.156" x THRU
R.188" TYP.
.375"
.500"
.125"
R.03" TYP
.469"
.188"
∅.375 TYP
∅.22' TYP
.190"
SECTION A-A
2 REQUIRED
NOTE:
1. BREAK ALL SHARP EDGES AND CORNERS
SPOOL GUIDE NUT CAP
NASA ACTUATOR
305-11 - Spool Guide Nut Cap

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	305-10	Modified Spool Guide Nut	1
2	305-11	SPOOL GUIDE NUT CAP	1
3	91251A340	#10-32 x 3/8" LG. UNF SHCS	2



BILL OF MATERIAL			
ITEM NO.	PART NUMBER	Description	QTY.
1	305-4	SPOOL SIDE MOUNTING PLATE	1
2	305-3	SPOOL SIDE BRACE	2





LEAD SCREW COLLAR
SECTION A-A
.010" x 45° CHAMFER TYP. BOTH SIDES
305-14 - Lead Screw Collar
NASA ACUATOR
AISI 304
1 OF 1
REV.
DESCRIPTION
DATE
BY