Friction Test Machine
Final Design Report

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INTRODUCTION
Parker Aerospace is interested in collecting empirical data of friction coefficients for skewed roller bearings. A fixture will be designed, constructed and tested to allow a technician to quickly and easily measure the friction torque of a skew roller bearing with varying axial loads, geometries, lubricants, temperatures and rotational speeds in order to improve actuator efficiency.

BACKGROUND
Our sponsor, Parker Aerospace, wants to be able to measure friction coefficients for skewed roller bearings. In order to fully understand the project and all its intricacies, the team conducted research on all the different types of components we might expect to be using to induce axial loads, measure torque, generate rotational speed, and collect data. Previous Parker Senior Projects as well as currently existing torque measurement machines were also looked at and evaluated. A collection of the research conducted is shown in the contents of this section.

Skew Roller Bearings
Skewing is the motion of a roller as it turns about an axis normal to the roller interface. Skew monitoring is important for bearing design as it is an indirect measure of bearing life. [17] The shaft rotational speed has a significant effect on roller skew but the radial load has little effect. Through the research of skew bearings, the analysis of needle roller bearings on torque loss surfaced, and provided useful information for this project. [18] The friction coefficient is defined as a function of sliding velocity, and is used to describe the experimentally determined relationship between the skew angle and thrust force of the needle rollers. Figure 16, below, visually shows the needle roller's dynamics.

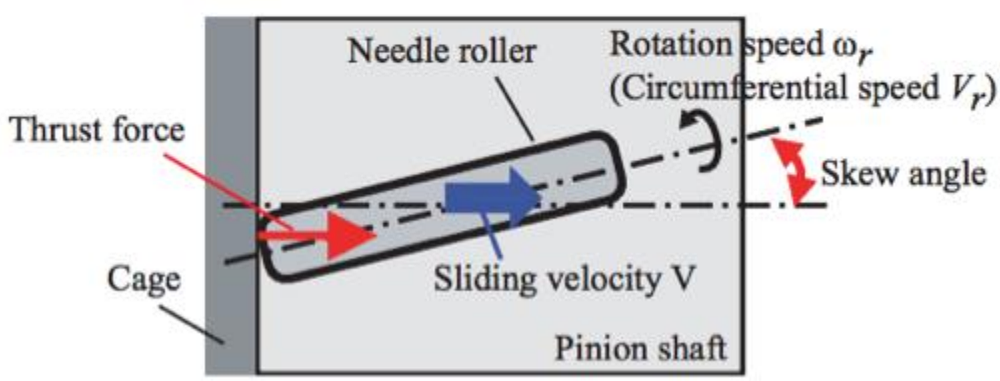


Figure 16. Skew Roller Dynamics [18]

Additional loss is caused by the thrust force generated by the skew of the rollers. They found that the cage pocket clearance needs to be small to lessen the friction loss of the bearing. The dynamic friction coefficient is influenced by surface roughness, pressure, rotational speed and lubrication.

Existing Designs

While there several machines out there that measure friction coefficients, most of those are using a linear force method rather than a torque force method. One such machine is the Lloyd Materials Testing FT1 friction testing machine shown in Figure 1.

Figure 1. Lloyd Materials Testing FT1 Friction Test Machine [1]

While this machine measures linear force to get friction rather than torsional force, there are still takeaways from this machine that can be applied to this project. This machine operates by applying a varying force in the x direction until slippage occurs to get the static friction, and then measures the kinetic friction when there is zero sliding acceleration along the plate. The way that the varying force is measured is through an axial load cell. The friction tester in this project will also need to measure axial load, but instead of varying the axial load, this project will vary rotational speed. This machine incorporates a data acquisition system that samples at a rate of 8 kHz, which is similar to the requirement imposed on this project.

Torque Cells

<u>Reaction Torque Sensors vs. Rotational Torque Sensors</u>

There are two major types of torque sensors used to make torque measurements, reaction torque cells and rotational torque cells. Reaction torque sensors utilize Newton's Third Law of Motion: "For every action there is an equal and opposite reaction." The reaction torque sensor remains stationary and measures the torque that is resisting the rotational motion, known as the "reaction torque." One of the benefits of this style of torque sensor is that the torque sensor does not have to rotate. This avoids the problem of having to make an electrical connection to a sensor that is constantly rotating, which makes the devices simpler and cheaper than rotational torque sensors. One drawback of a reaction torque sensor is that these sensors typically have to take other types of loads, such as axial loads from the weight of the motor. Sometimes this leads to the sensor being oversized and as such, the sensors tend to be less sensitive. Figure 2 is an illustration of the position of a typical reaction torque sensor.

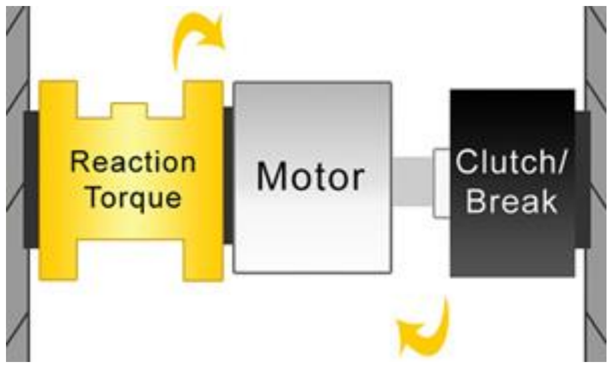


Figure 2. Reaction Torque Sensor Position [2]

Rotational Torque sensors are also known as inline sensors, and measurements are made by placing the sensor between torque carrying components. One of the main benefits of this type of sensor is that the sensor can be placed as closely as possible to the torque interested in being measured. This eliminates one of the problems of the reaction torque sensors in that there are not as many/no extraneous loads to interfere with the measurement. Inline torque sensors are usually the only way to measure dynamic torque accurately.

Since the rotational torque sensors are attached to rotating components, there needs to be a way to connect them to non-rotating electronics. There are four main ways that this is accomplished: slip rings, rotary transformers, infrared, and FM transmitters. Due to the more complicated nature of these sensors, rotational torque sensors tend to be much more expensive than reactionary torque sensors.

The most commonly used device is the slip ring as shown in Figure 3. This is the simplest and cheapest solution can be used in a wide variety of applications, but has the drawback of having parts that have limited life spans. The basic idea is that the sensor has a set of rotating conductive rings that contact a series of stationary brushes to transmit the measurement to an electronic measurement device. One drawback of this design is that the machine has to be able to overcome the drag that the brushes impose on the system.

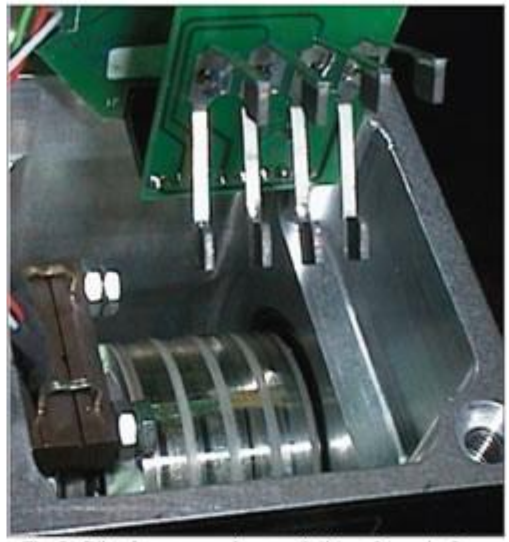


Fig.2. Slip ring example consisting of a set of conductive rings that rotate with the sensor and a series of brushes that contact the rings and transmit sensor signals.

Figure 3. Slip Ring Torque Sensor Example [3]

Another way to accomplish the problem of stationary sensors attached to rotating machinary is through a rotary transformer as shown in Figure 4. This style attempts to overcome the two major drawbacks of the slip ring discussed above. This style uses a rotary transformer coupling to transmit power and then send that signal to the sensor. This is accomplished by externally providing an AC excitation voltage, and using two rotating coils to transform the signal. This eliminates the issue of wear on the rotating coils, as well as the drag from the brushes. However, one drawback of the design is that slight misalignment between the coils of the transformer could lead to errors in measurement as well as unwanted noise.

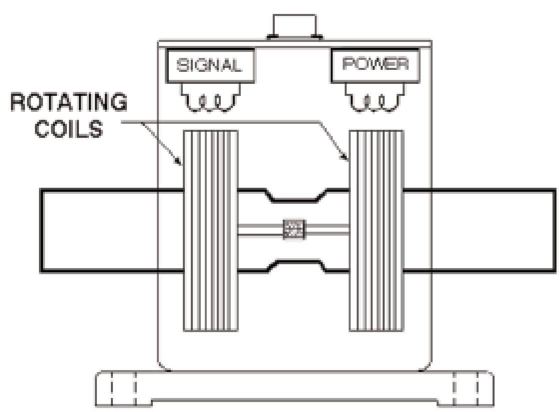


Fig. 3. the rotary transformer system uses a rotary transformer coupling to transmit power to rotating sensor.

Figure 4. Rotary Transformer Sensor [3]

The next solution is the infrared torque sensor as shown in Figure 5. This method is similar to the rotary transformer in that it avoids contact of the signal surfaces to the rotary component, but the output signal is transmitted via infrared light to stationary receiver diodes. Since the output signal is a digital signal rather than an analog one, infrared sensors have fewer noise issues.

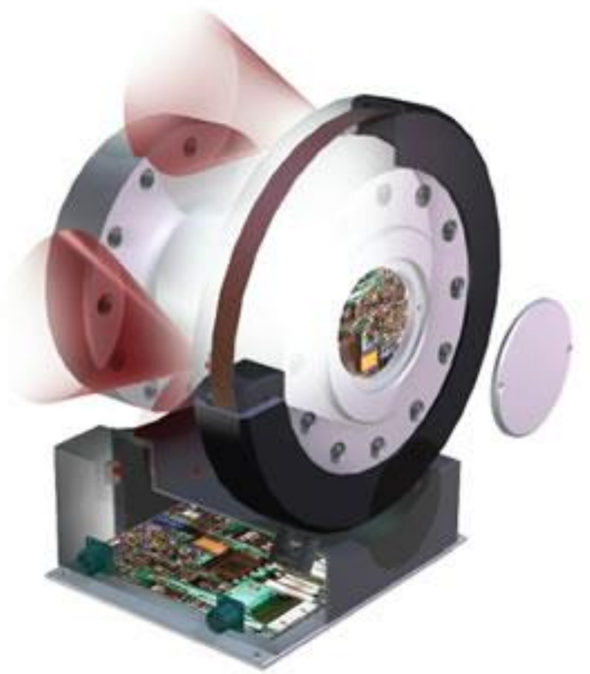


Fig. 4. IR torque sensor utilizing contact-less method of getting a torque signal from a rotating sensor back to the stationary world.

Figure 5. Infrared Torque Sensor [3]

The last style of sensor is the FM Transmitter as shown in Figure 6. This style uses applied strain gauges along the device, and then the measurement system can be clamped on to the outside of the shaft the torque is being measured from. This transmitter converts its signal to a digital form and then transmits it to an FM receiver connected to a data acquisition system that converts it to an analog voltage. This style is very easy to install and also very versatile, however the transmitter on the shaft does need its own power source (usually a 9V battery) so it is not used in any long term testing applications.

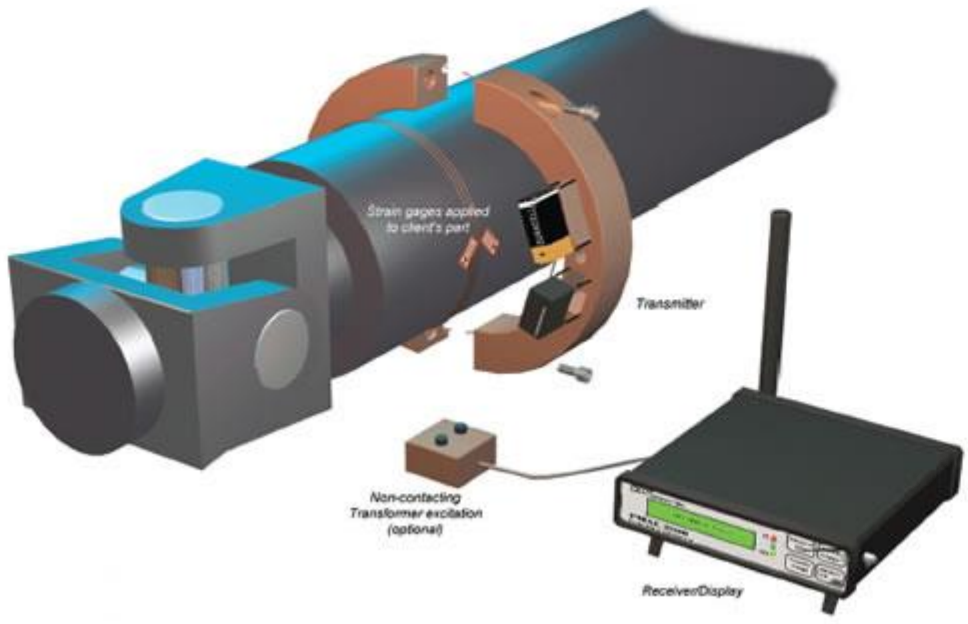


Fig. 5. FM transmitter devices connect force or torque sensors to remote data-acquisition systems.

Figure 6. FM Transmitter torque sensor [3]

Axial Load Application

A vital component necessary for the friction testing machine is the application of a 5000 pound axial load on the skew roller plate. The friction-torque of the skew roller bearing is related to the applied force normal to the bearing. Data needs to be collected to quantify the friction-torque and axial force relationship via experimentation. To induce an applied axial load, a linear actuator can be used. There are many types of linear actuators that can accomplish an axial force, including mechanical actuators, hydraulic actuators, pneumatic actuators, piezoelectric actuators, electro-mechanical actuators, and linear motors.

Mechanical Actuators

Mechanical linear actuators utilize leadscrews to create linear motion from rotational motion. Types of screw mechanical actuators include screw jacks, ball screws, and roller screws. These mechanical actuators all involve the rotation of a nut around a screw to create linear motion, but vary by specific mechanisms. Ball screw and roller screw actuators are more expensive due to the high efficiency of rotational to linear energy conversion. Low friction screw actuators, such as ball screws, are not necessary to satisfy the axial load requirement. A simple manual screw jack is cost-effective and capable of satisfying the axial load requirement. A 3-ton, or 6000 pound, manual screw jack can be purchased for approximately $50. [4] Figure 7, below, depicts a ball screw actuator (left) and a manual screw jack (right).

Ball Screw Actuator [5]

Manual Screw Jack [6]

Figure 7. Mechanical Screw Actuators

Another mechanical actuator type is wheel and axle actuators. A common type of wheel and axle actuator is a rack and pinion configuration, shown in Figure 8. The principle behind wheel and axle actuators is to create mechanical advantage using a member in tension, such as a rope, that unwinds from a wheel joined to a drum or shaft. These actuators commonly require a higher driving torque than screw actuators and have lower efficiency than ball screw and roller screw actuators. For the application in this project, screw actuators are likely preferred to wheel and axle actuators due to cost and simplicity.

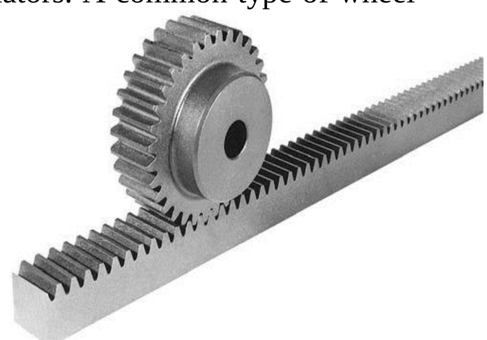


Figure 8. Rack and Pinion Actuator [7]

Hydraulic Actuators

Hydraulic cylinders are devices comprised of a hollow cylinder filled with a hydraulic fluid and a piston inserted into it to create a linear displacement along the axis of the piston. A common manual hydraulic actuator is a hydraulic car jack. Figure 9 displays a typical 3-ton hydraulic car jack, which is priced at about $80. [8]

Figure 9. Hydraulic Car Jack [8]

The two types of hydraulic cylinders are tie rod and welded body cylinders. Tie rod hydraulic cylinders use threaded steel rods to hold the end caps of the cylinder together to the cylinder body. An advantage of tie rod cylinders is the ability to be taken apart for maintenance. Additionally, tie rod cylinder dimensions have been standardized. Welded body cylinders have the end caps directly welded to the cylinder body. Welded body cylinders are typically narrower and shorter and is the most common configuration for portable hydraulic cylinders. An 8-ton manual welded body cylinder is priced at approximately \$40. [9] Figure 10 shows the two styles of hydraulic cylinders, tie rod and welded body.

Tie Rod Hydraulic Cylinder [10] Welded Body Hydraulic Cylinder [11]

Figure 10. Tie Rod and Welded Body Hydraulic Cylinders

An important consideration in cylinder selection and design is the mounting configuration. There are several common mounting methods currently in use and the cylinder mounting factors in to the performance of the cylinder. Some of the common mounting methods include the use of flange mounts, side-mounted cylinders, centerline lug mounts, and pivot mounts. Fixed mounts along the centerline of the cylinder are the most effective for the transfer of force.

Pneumatic Actuators

Pneumatic cylinders work under the same principles as hydraulic cylinders, except they utilize compressed air in lieu of hydraulic fluid. Pneumatic cylinders require a device such as an air compressor to provide the linear force. Air compressors can be large, noisy, and require a power source, which should be considered when evaluating the benefit of using one. An 8-ton pneumatic cylinder can be purchase for about \$100, shown below in Figure 11. [12]

Figure 11. Pneumatic Cylinder [12]

The most common types of pneumatic cylinders are single-acting cylinders (SAC) and double-acting cylinders (DAC). SAC work by providing compressed air to drive the piston in a single direction, and a spring to drive the piston back in the opposite direction after being compressed. SAC have limited extension range due to the space of the spring in the cylinder and a smaller possible force due to the push-back force of the spring. DAC, on the other hand, use the force of air for both the extension and retraction of the piston. DAC do not limit the extension range and are capable of producing more force than SAC, but the piston is more prone to failure.

Electro-Mechanical Actuators

Electro-mechanical actuators implement a mechanical actuator that converts rotational motion to linear motion with an electric motor. A common configuration is a DC or stepper motor providing rotational torque and speed with a lead screw to produce linear motion and force. The advantage of using electro-mechanical actuation is that the process can be automated and it is reliable and versatile. For the purpose of this project, automation is not necessary and only adds unnecessary complexity and cost. An example of an electro-mechanical actuation system is depicted in Figure 12.

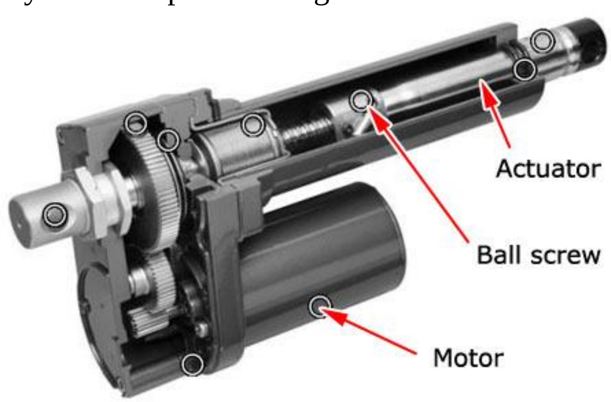


Figure 12. Electro-Mechanical Actuator [13]

Linear Motors

A linear motor is an electric motor that bypasses the need to convert rotational motion to linear motion through the use of a mechanical actuator, such as a lead screw. The linear motor is an electric motor, except the components are laid out in a straight line, direction creating linear motion. The motor moves repeatedly across of the length of its actuator. Linear motors are capable of achieving high speeds, however, are not capable of producing high loads, such as what is required for the friction testing machine. A linear motor is likely a poor choice for the production of the axial load across the skew rollers. An image of a linear motor is shown below, in Figure 13.

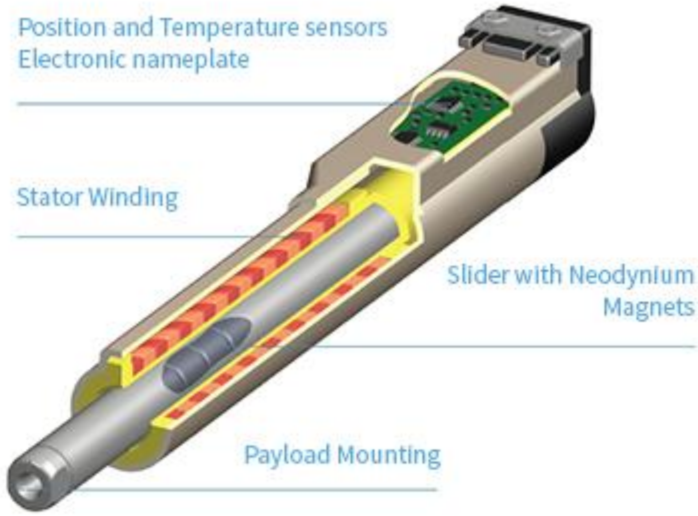


Figure 13. Linear Motor [14]

Axial Load Measurement

The most common and widely used method of accurately measuring a linear force is via the use of a load cell. A load cell is a transducer that creates a voltage signal of magnitude that is proportional to the force placed on it. Load cells are calibrated to output the measured force given the output voltage. Load cells are capable of use with any data acquisition system because of its electric signal output. Some types of loads cells include hydraulic, pneumatic, and strain-gauge load cells.

Strain-gauge load cells are the most common type of load cell. For an axial load measurement, compression load cells are commonly used. A compression load cell can be placed in series with the loading actuator to reliably measure the applied load. Load cells can be quite expensive, thus the preferred method of acquisition is to borrow one from the project sponsor. A typical 5000 pound capacity compression load cell can run for approximately $600, shown in Figure 14. [15]

Figure 14. Compression Load Cell [15]

Motors

In order to test the skew roller plates, a motor must be capable of producing at least 600 in-lb of torque and 200 RPM. A common way of achieving these requirements is to use a simple magnetic electric motor. A simple two-pole DC electric motor is capable of satisfying the requirements. When choosing a motor for the project, it is essential to keep in mind that the system must run off a 115 VAC standard wall outlet. Thus, it is necessary to implement a variable DC supply voltage to allow for the input of different DC voltages to the motor. This allows the ability to run a DC motor at different speeds, since the speed is a function of the input voltage to the motor.

A cheap and rugged option for a DC motor is to implement a power drill motor with a variable supply voltage. Drill motors are relatively inexpensive compared to other small electric motors. The disadvantages of using a drill motor include that the speed control is inaccurate and they cannot accurately run at varying speeds. However, the drill motor is capable of running at differing speeds with the variable power supply and the speed will be measured with a separate sensor, making the drill motor a good option for this project. Drill Motors can be purchased for around $30. [16] A drill motor is shown in Figure 15. [16]

Figure 15. Drill Motor [16]

Data Acquisition System (DAQ)

Data acquisition, DAQ, is the process of measuring an electrical or physical phenomenon such as voltage, current, temperature, pressure or sound with a computer. A DAQ system consists of sensors, DAQ measurement hardware and a computer with programmable software. Pressure, load and torque sensors consist of designed structures that perform in a predictable and repeatable manner when a force is applied. That force is translated into a signal voltage by the resistance change of the strain gauges, organized in an electrical circuit, and applied to the load cell structure. A change in resistance indicates the degree of deflection and, therefore, the load on the structure. [19]

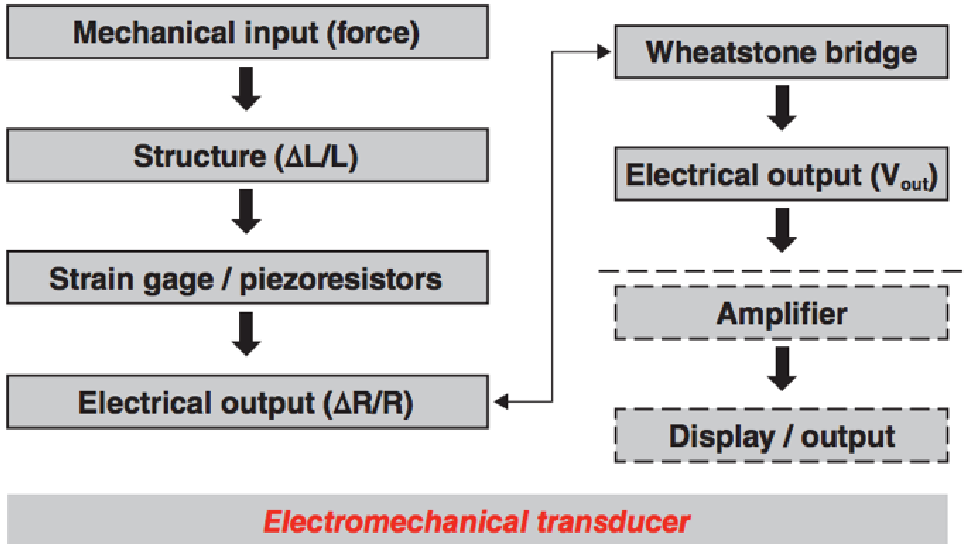


Figure 17. Electromechanical Transducer [19]

From conversations with the creator of a data acquisition program used at Cal Poly, we have learned that an efficient option for collecting data in our situation is with a simple system that we can program ourselves. As suggested by our sponsor, programming a CompactRIO using LabVIEW seems to be a simple solution with software that we can learn and adapt to the data channels we will be monitoring. [20] The project team for the original Parker friction test machine project selected a National Instruments DAQ system compatible with their sensors, programmable with LabVIEW, and easily connected to any computer by USB. [21] They did run into a problem with the temperature monitoring using the myRIO-1900, so possibly another version of similar hardware would be appropriate for our use.

Previous Senior Projects

As a portion of our research, we analyzed the friction test machine senior project that Parker sponsored previously at Cal Poly [21]. Although the problem definition differs a bit from ours, looking into viscous drag generated by bearings and seals, there are some helpful ideas and suggestions we can take to guide our project in a more successful way. Some of the main takeaways from reviewing the final report are the data acquisition method, the torque sensor selection, rotational speed monitoring, safety concerns regarding the thermal chamber, and vibration awareness when designing and manufacturing the fixture.

The Futek reaction torque sensor chosen for the previous project is similar to the tubular design that would be appropriate for our application. It is compact and has high torque capacities that would be necessary for the engineering specifications. Also, to identify rotational speed, their choice of using a Hall Effect sensor is the same simple, cost effective method that our group had previously deduced. Looking into the structure of the machine, stability and safety are both serious concerns. When brainstorming and designing our machine we will be careful to minimize machine vibration that could alter data measurements or shorten the lifetime of the machine. Also, when designing the thermal chamber we will keep the operator's safety our priority when selecting materials.

OBJECTIVES

The primary goal for the Friction Testing Machine project is to design, build, and test a fixture that can quickly and accurately collect relevant data during the testing of skew roller cages. We will attempt to fulfill the customer needs as thoroughly as possible as we understand them. The initial customer requirements list provided by the project sponsor, Parker, is shown below.

Customer Requirements List

1. The fixture must be stable during operation, and self-standing when placed on a floor or tabletop
2. 24 x 24 inch base maximum, 36 inch height maximum
    a. Control and data systems not included. may be remote desktop computer/RIO etc
3. The fixture shall be able to generate an axial load across the skew rollers of 5000 lb.
    a. Screw jack, Hydraulic or pneumatic cylinders ok
    b. A load sensor shall be provided to measure/verify load
    c. Manual Load control is acceptable
4. The fixture shall be able to generate a torque of 600 in-lb
    a. A High Output Drill motor, Motor and Gearbox, motor and Pulley arrangement is acceptable
    b. A torque sensor shall be provided to measure/verify torque
5. The fixture shall be able to generate the torque in item 4 above at a speed of 0 to 200 rpm
    a. A High Output Drill motor, Motor and Gearbox, motor and Pulley arrangement is acceptable
    b. A speed sensor shall be provided to measure/verify speed
    c. Manual Speed control is acceptable
6. A data acquisition system shall be provided to capture the sensor data and output this data to Text, ASCII or other digital format for post processing
    a. Control and data systems may be remote desktop computer/RIO etc
7. The Fixture shall be designed to be able to submerge the skew roller section under oil
8. A box (plywood or similar) shall be provided around the skew assembly to allow the introduction of hot and / or cold air (preferably the sensors will be outside the box)
9. Roller Bearings will be provided by Parker
10. Thrust Washer (if needed) shall be High Carbon Steel, through hardened to HRc 58 min. Ground to meet finish, flatness and parallelism
11. Cost goal <$5000 (note some sensors and possibly motor/controllers may be available from Parker stock)
12. The maximum weight of the fixture shall be no more than 150 lbs
13. The geometry of the fixture shall provide adequate gripping for two people to carry
14. Fixture must be able to run on standard 115 Vac 20 amp max wall power

The customer requirements list displayed above is a starting point from which the project engineering specifications can be derived. The method used for formulating the engineering specifications is the Quality Function Deployment (QFD) method, which is depicted in the following section. Appendix B and Table 1 show the engineering specifications developed for the Friction Test Machine (FTM) project.

Table 1. Engineering Specifications

Spec. #	Parameter Description	Requirement or Target (units)	Tolerance	Risk	Compliance
1	Weight	150 lb	Max	L	A, T
2	Size	24in x 24in x 80in	Max	L	A, T
3	Geometry	Places to grip		L	A
4	Axial Load	5000 lb	Min	M	A, T
5	Apply/Measure Torque	600 in.lb	Min	H	A, T
6	Rotational Speed	0 to 200 rpm		M	A, T
7	DAQ	1KHz sample rate. 4 channels	Min	H	A, T
8	Thermal Chamber	-65°F to 160°F		M	A, T
9	Power	115 VAC 20 Amp	Max	M	A, T
10	Cost	$5000	Max	M	A
11	Submersible	Bearings 100% submerged	N/A	M	A, T
12	Flange sample	Parker unit used	N/A	L	I
13	Skew roller bearings	Parker bearings used	N/A	L	I
14	Stability	Stability during test	N/A	L	A, T

The engineering specification table shown above, which was developed based on the QFD, lists all of the measurable/testable specifications of the project, what the required test is for that specification, the risk associated with the specification, and the level of compliance necessary for the specification. The risks are either high risk (H), moderate risk (M), or low risk (L), depending on how critical the specification is and how difficult it will be to achieve. In the compliance section, an A signifies the requirement will be checked through analysis, an I signifies it will be checked by inspection, and a T signifies it will be checked by testing.

Quality Function Deployment

The Engineering Specifications for the project were written based on a Quality Function Deployment (QFD). A QFD House of Quality was completed and is attached in Appendix A. The QFD House of Quality is a useful way to organize information pertinent to the writing on engineering specifications, including customer needs and requirements.

The QFD method begins with defining the customer requirements. The customer requirements list must be formulated before continuing with the process. An initial list of the customer requirements was provided by Parker and was modified based on interaction with the project sponsor. After defining the implicit and explicit customer requirements, each requirement is weighted relative to one another based upon its value to different users. The two users for the project are the project sponsor at Parker, Eric Polcuch, and the operator or technician.

Next, background research on existing products is conducted. A list of existing solutions is compiled and ranked based on how well each existing solution meets and fulfills our customer requirements. As this is a unique project, no existing solutions exist so similar solutions for other applications were considered. At this point, additional technical research is conducted. A list of engineering specifications is written, based primarily on the customer requirements along with our understanding of the project. The QFD can be further utilized to determine decision matrix weights for each specification.

To determine the weight of each specification, the existing solutions are first ranked relative to one another based on how well they meet the engineering specifications. This step involves evaluation of the competition and assessing performance based on the engineering specifications. The next step involves evaluating the dependencies between engineering specifications. If two specifications are strongly related, both specifications may not be needed. Lastly, the final targets for each engineering specification are set based on our best current product performance. The relative weights of engineering specifications are evaluated using the information completed on the QFD and these specifications and weights can be used as the design criteria for a weighted decision matrix for project concepts.

Boundary Sketch

A boundary sketch was drawn to further define the project and all that it entails. The boundary sketch is useful for defining what subsystems are included in the project and what is and is not included within the scope of the project. Figure 18 displays the boundary sketch for this Friction Testing Machine project. The items and subsystems within the boundary are considered within the scope of the project, while anything outside of the boundary is outside the scope of the project. The boundary sketch is useful in ensuring that the specifications are complete and there are not any requirements missing or any requirements that should be removed.

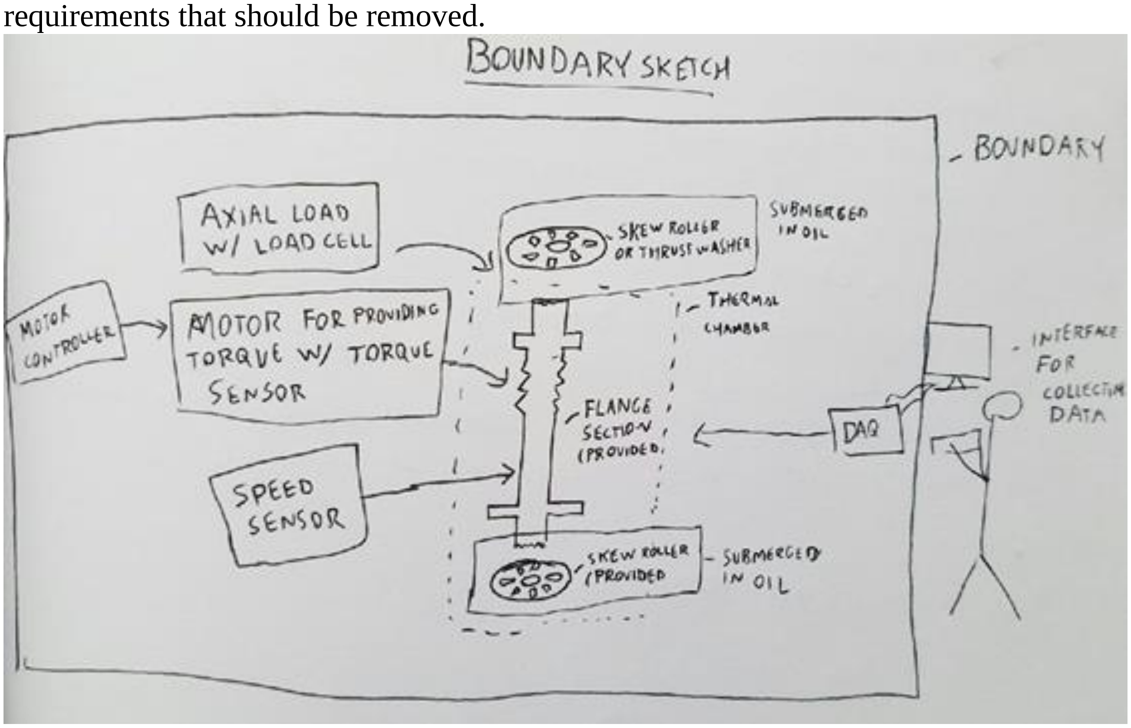


Figure 18. Boundary Sketch

DESIGN DEVELOPMENT - PRELIMINARY DESIGN

Since defining the problem, finding customer needs, conducting background research, and creating a problem statement, we have had numerous brainstorming sessions for ideation for the various functions of the machine. We performed standard brainstorming, where ideas were generated for various functions and written down on the white board. This is can be seen below in Figure 19. Another method we used was brainwriting. This approach allows for more creativity of function execution along with some initial packaging ideation. The results of our brainwriting sessions can be seen in Figure 20.

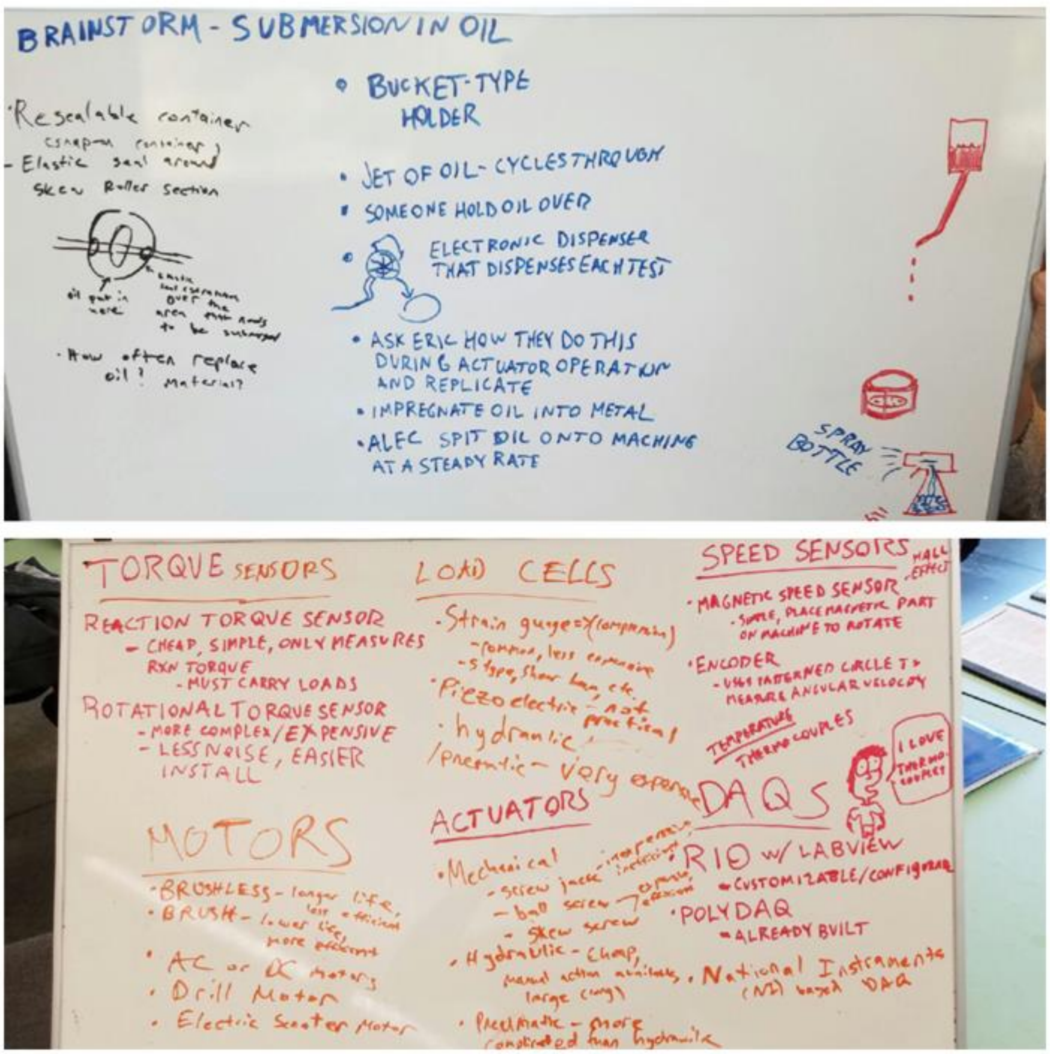


Figure 19. Brainstorming Sessions

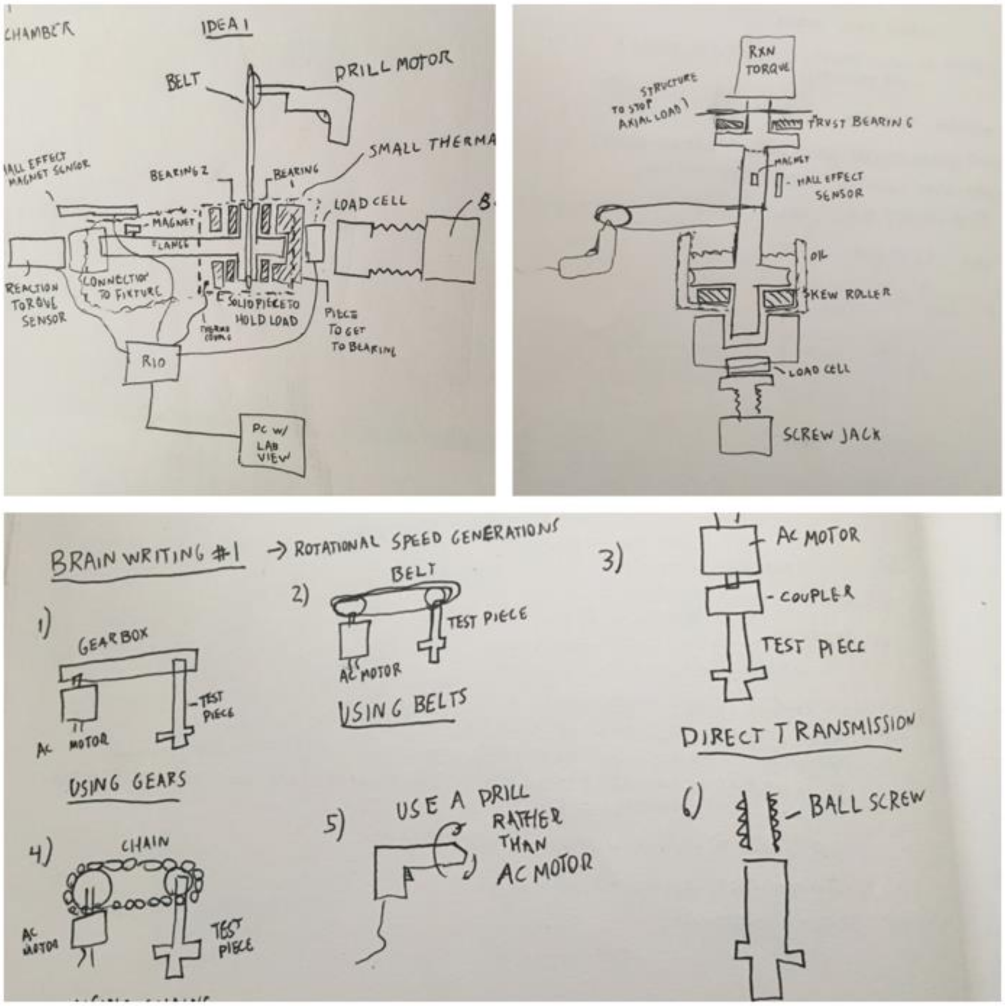


Figure 20. Brainwriting Sessions

The main functions that the brainstorming sessions focused on were: axial load generation, axial load measurement, rotational speed generation, rotational speed measurement, torque measurement, submersion in oil, supporting structure, thermal chamber, and data acquisition system. Ideas were generated in each of these categories, and then combined to look at how the different functions interact with each other. These brainstorming and brainwriting sessions helped us determine all the different components that were available for use, as well as ideas for how to submerge the bearings in oil, generate the rotational speed, and contain the axial load.

As a preliminary configuration exercise, our team constructed the rough prototype seen in Figure 21. This activity was helpful for us to see possible challenges regarding the torque sensor application, as well as the unforeseen struggle we may have containing lubricant with a horizontal configuration.

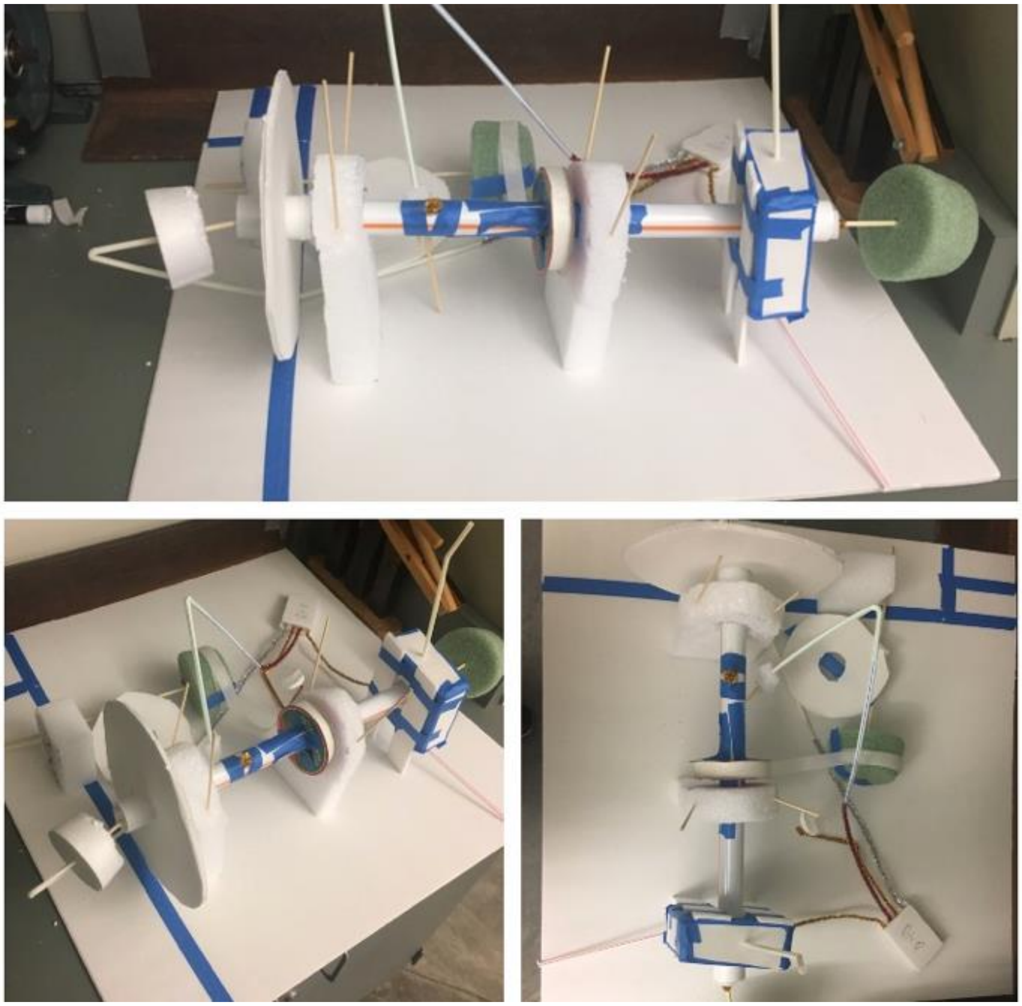


Figure 21. Ideation Build

After many ideas for the different functions were explored, these ideas were analyzed against each other in Pugh matrices (see Appendix D) in order to determine the best components for each configuration. Concepts for specific functions were evaluated via the Pugh matrices using customer requirements as criteria. Each matrix placed one concept as the datum and the remaining concepts were measured relative to it. The criteria for each concept received either a '+' meaning better, a '-' meaning worse, or an 'S' meaning about the same as the datum. These matrices allowed for the capabilities and shortcomings of each of our ideas to be explored, and later, configurations to be created and analyzed in a decision matrix.

Once the preliminary design has been reviewed and agreed upon with the project sponsor, the team can move into the detailed design phase of the project. This is the phase that the design really comes together. The sensors and loading components will be positioned, structural components will be selected and sized, wiring diagrams will be created, and other design features will be fully implemented. The purpose of this phase is to finish everything necessary in the design necessary to manufacture it completely. The cost of the overall design will also be evaluated in this phase.

Given the nature of the friction test machine, safety is a concern. Since the test fixture will be spinning, we want the test sample flange section to be covered. This will be done with the thermal chamber. Also, because there is wiring associated with this test machine, there will be considerations made to make sure the operator is safe when testing.

Concept Descriptions

Horizontal 1

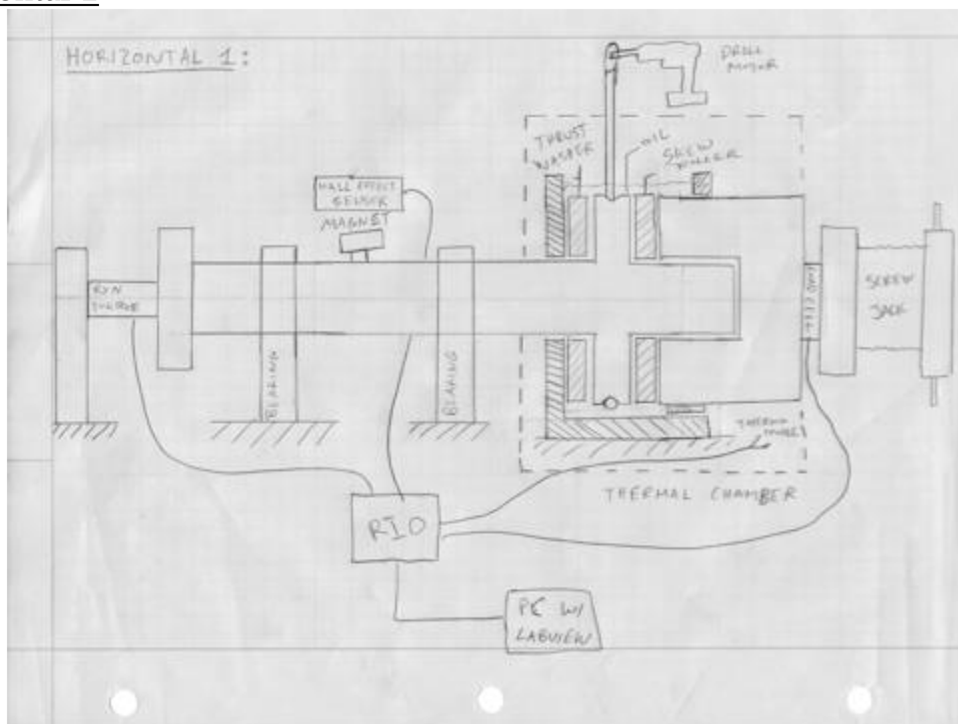


Figure 22. Horizontal 1 Configuration

The Horizontal 1 configuration was the first conceived configuration. The rotating shaft, provided by Parker, is aligned with the longitudinal axis parallel to the ground. The rotating shaft is supported with two bearings, to be provided by Parker, and mounted securely to the base. The shaft is driven via a electric drill motor with a belt. The belt is attached around the outer diameter of the flange section on the shaft side and around a disc attached to the motor, sized to provide an appropriate gear ratio. The axial load is applied by a screw jack, with the right end fixed to the structure. The screw jack can be turned to manually apply the necessary load. The load is transferred through a compression load cell to measure the load, a bracket, the skew roller plate, the flange, and the thrust washer to the oil chamber, which is connected to the structure to counteract the load. The oil chamber is a hollow cylinder, with an open top and a hole in one face to allow for the bracket to apply the axial load. A gasket is used to prevent the oil from leaking out of the chamber. The friction-torque applied to the rotating shaft by the thrust washer and skew roller plate is measured via a reaction torque sensor on the opposite side of the screw jack. The rotational speed is measured using a rotating, magnet Hall Effect sensor attached between the two supporting bearings. A plywood thermal chamber with two ducts to provide hot or cold air surrounds the test section and a thermocouple is used to measure the testing temperature. All sensors are connected to a data acquisition system (DAQ) and interfaced to a PC for recording all appropriate data.

Horizontal 2

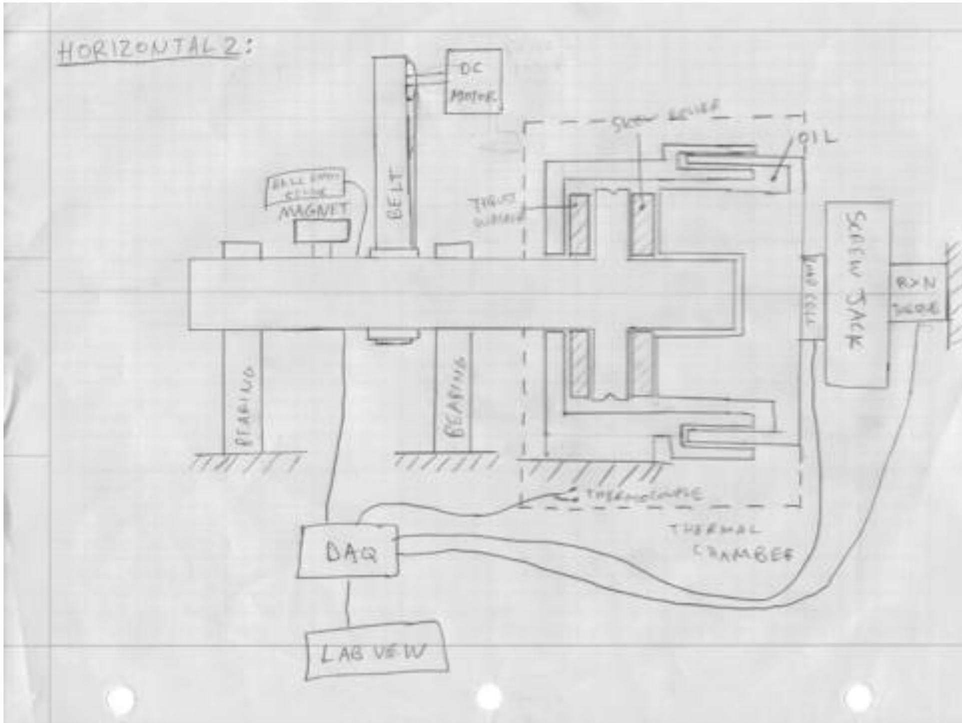


Figure 23. Horizontal 2 Configuration

The Horizontal 2 configuration is a variant of Horizontal 1. The shaft in the configuration, like Horizontal 1, is aligned with the longitudinal axis parallel to the ground. There are many similarities between the two horizontal configurations including the use of a Hall Effect sensor to measure rotational velocity, two bearings to support the shaft, a screw jack to apply axial load, a compression load cell to measure axial load, a reaction torque sensor to measure friction-torque, a thermal chamber with a thermocouple, and the use of a belt to drive the shaft. The key differences between the two configurations is the location of the drive assembly, the modified thermal chamber, and the placement of the reaction torque sensor. The shaft is driven by a DC motor with a belt attached to the existing teeth on the shaft, rather than the flange section. The oil chamber is modified to provide an enclosed area for the oil while allowing for the axial load to be applied. The chamber is a two-piece compartment with the left end fixed to the structure and the right end used to apply the axial load. The reaction torque sensor is fixed to the structure and attached to the screw jack to measure the friction-torque of the skew roller plate, independently of the thrust washer.

Vertical 1

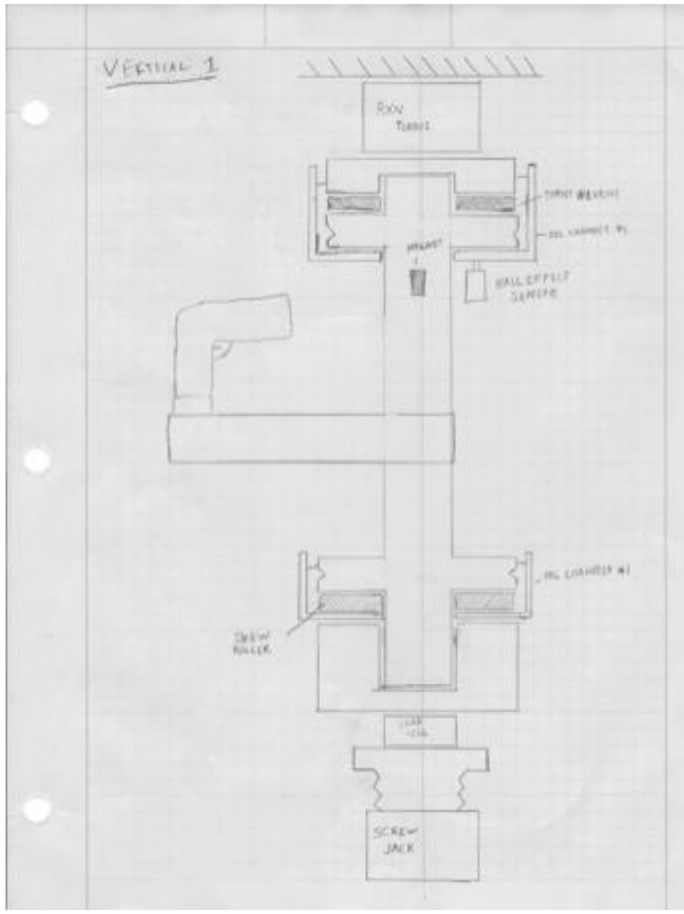


Figure 24. Vertical 1 Configuration

The Vertical 1 configuration has the flange section rotating parallel to the ground, with the flange section oriented vertically. The idea behind this configuration was to increase the distance that the compression load acted across. A second flange piece was added to the opposite side, and one bearing is on each flange piece. In this configuration, the reaction torque of a single skew roller bearing can be measured. While this might benefit is nice, the addition of a second flange poses several problems. The large load that the flange experiences could cause significant bending concerns when not opposed on the opposite side. There would also have to be two thermal chambers and two oil chambers (one for each flange section).

Vertical 2

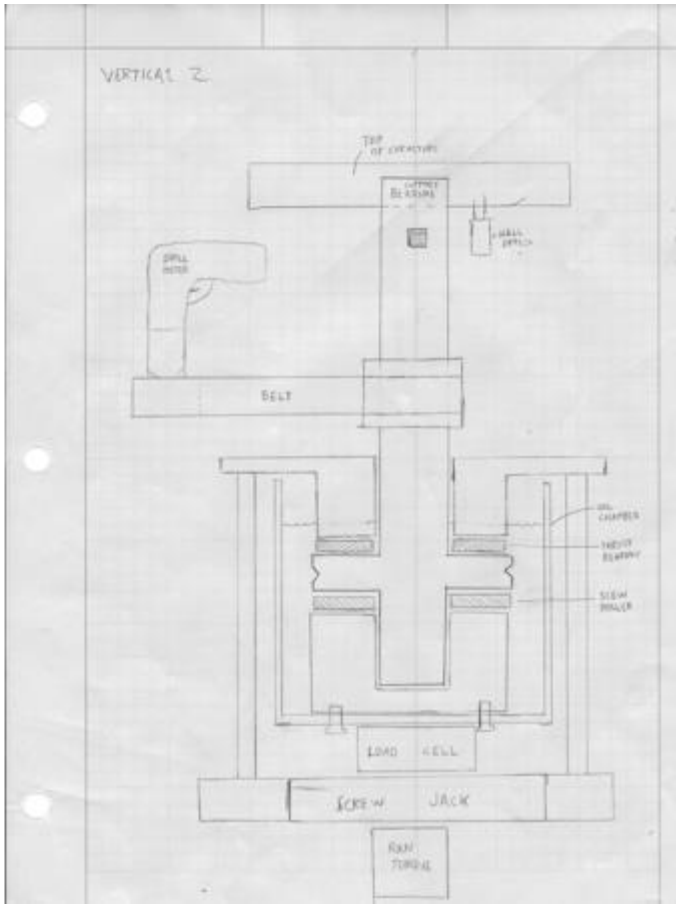


Figure 25. Vertical 2 Configuration

In the Vertical 2 configuration, the flange section rotates parallel to the ground, driven by a drill motor. This drill motor is connected to the shaft through a belt with a gear ratio that will allow for the required torque to be delivered to the test section at the necessary speed (200rpm). In this configuration, one skew roller bearing will be used, with a thrust bearing on the opposite side of the flange. In order to keep the bearings fully submerged, a simple can will be used to hold it together. The axial load will be generated through a manual screw jack and measured with a compression load cell. The load will be carried out in a loop that goes from the screw jack, into the bearings across the flange, and then through tensioning rods, will travel back into the screw jack. The reaction torque can then be measured from the screw jack, as all the frictional torque will go back into the screw jack via the load loop. This configuration simplifies the oil chamber, and provides an easy way to contain the axial load without the structure having to support the 5000 lb compression load. One challenge of this configuration is making sure that the structure takes the weight of the system above the reaction torque cell so the torque cell is not taking any axial loading. Another benefit of the vertical configuration is that it takes up less base space in the 24" x 24" maximum area, and most of the space is absorbed in the height, where the constraints are less stringent.

Vertical 3

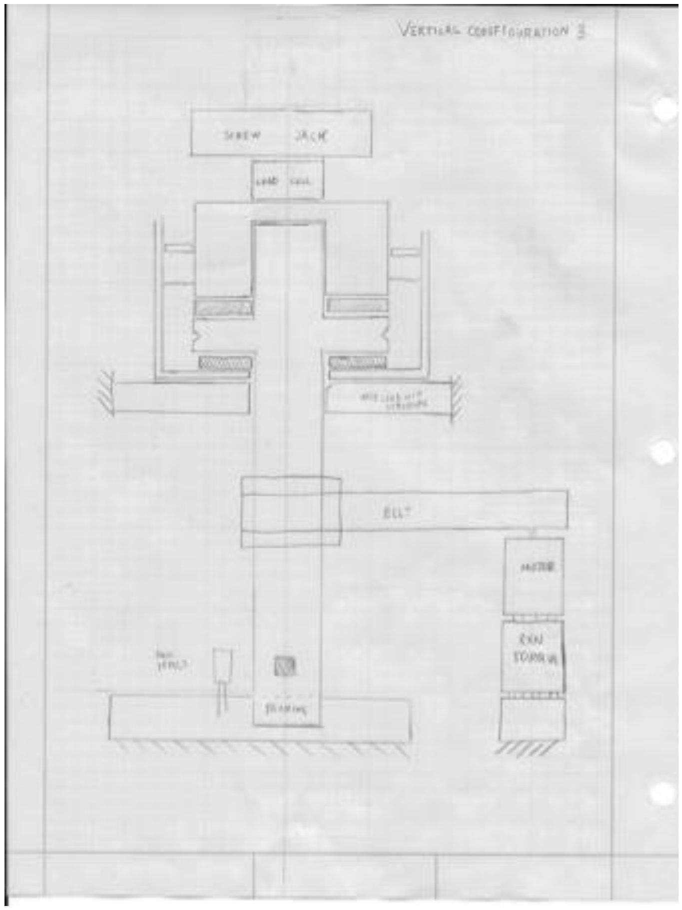


Figure 26. Vertical 3 Configuration

This configuration is very similar to Vertical 2, but the flange section is rotated 180 degrees. In this configuration, the structure is supporting the 5000 lb axial load as well as the weight of all the components of the design. This is helpful in the fact that there is not as much concern about the torque cell carrying any axial loading, as the torque cell is on top of the structure rather than on the bottom. However, the idea of the load loop that was discussed in the vertical 2 can not be applied here. This means a lot more load is now taken by the structure rather than dissipated through the components.

Configuration Evaluation

The configurations listed and described in the previous section are evaluated using a weighted design criteria. The design criteria is primarily based off the design specifications and corresponding weights that were evaluated using the QFD method (see Appendix A). Some of the weighted specifications were not used in the design criteria due to irrelevance to configuration evaluation. Additionally, other design criteria was added to the engineering specifications for use in configuration evaluation, including ease of us and manufacturing time. Ease of use allows for the inclusion of a score for the ease of operation of the testing configuration by the operator and the ability to take measurements and data quickly and efficiently. Manufacturing time is a consideration separate from the specifications since it is irrelevant to the final product, but is worthwhile for the purposed of building time constraints. The decision matrix used to evaluated the configurations relative to each other is attached in Appendix D. The configuration that received the highest overall score was the Vertical 2 configuration. Of the horizontal configurations, the Horizontal 2 configuration scored the best overall score. A Pugh matrix comparing the horizontal and vertical configurations is attached in Appendix E. The Pugh matrix results in a higher evaluation score for the vertical configuration over the horizontal configuration, further corroborating the decision matrix results.

DESIGN DEVELOPMENT - CRITICAL DESIGN

Functional Description

Figure 27, below, shows our SolidWorks model of our overall design, based on the preliminary Vertical 2 configuration. This section will discuss the aspects of the chosen design and how everything works. The major components of this design (the motor, the screw jack, the load cell, the torque sensor, and the speed sensor), will be discussed in further detail in the next section. The first thing that should be mentioned is the outer structure that contains the design. The outer structure is made out of square t-slotted framing made by 80/20. This framing is easy to assemble, relatively lightweight, and relatively inexpensive.

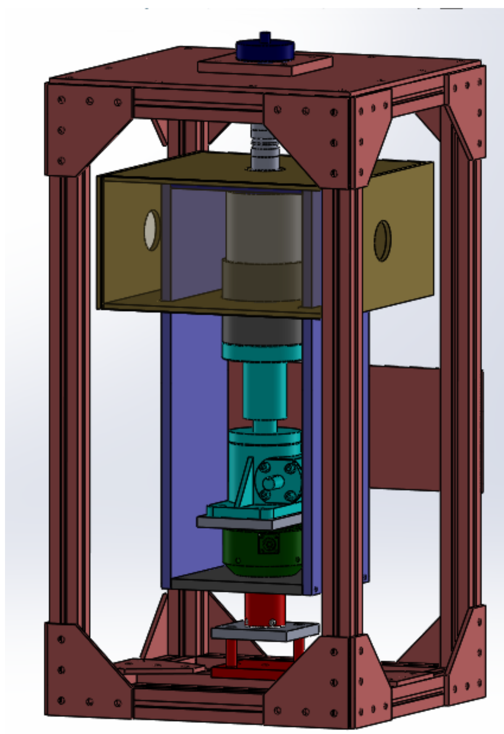


Figure 27. Overall SolidWorks Model

The top of the fixture as seen in the following figure is where the weight of the test area will be held. The top dark blue piece holds the test section provided by a ¼-20 bolt, to allow for easy assembly/disassembly. This blue piece will be supporting all the weight of the test area, and will be supported on a thrust washer to allow the test section to rotate. This dark blue piece will also connect to the drill that will be supplying the rotational power.

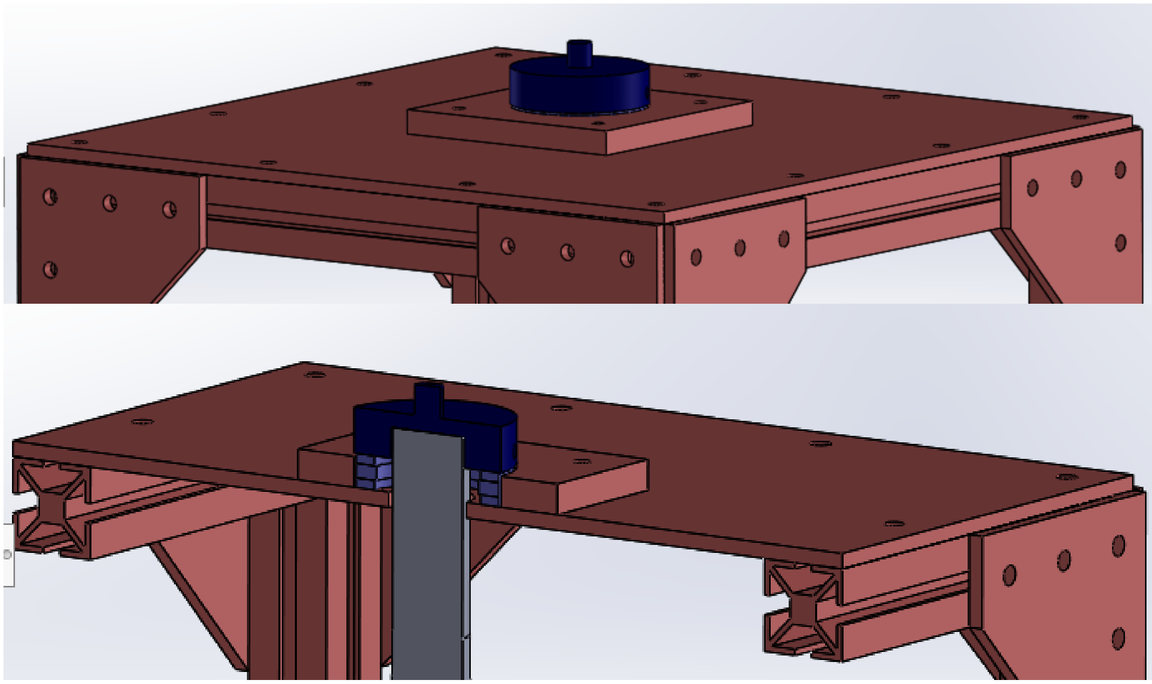


Figure 28. Top of structure

Next is the load loop. This is the area of the structure where the 5000 pound compression load is put on the flange, and the reaction torque is measured by the torque cell. The axial load is generated by a Nook Screw Jack, and runs to the test flange. The load is opposed by the top plate, creating compression on the flange section, and runs through the two long plates on the sides of the jack. This creates tension in the side plates, which connect the top and bottom plate of the load structure, creating a load loop. This load is measured by a Futek compression load cell, which is placed between the screw jack and the bottom plate of the load loop. The load loop acts a one rigid structure, from which the reaction torque cell can be used to measure the torque generated by the test flange, which is rotating from the drill connected to the test piece.

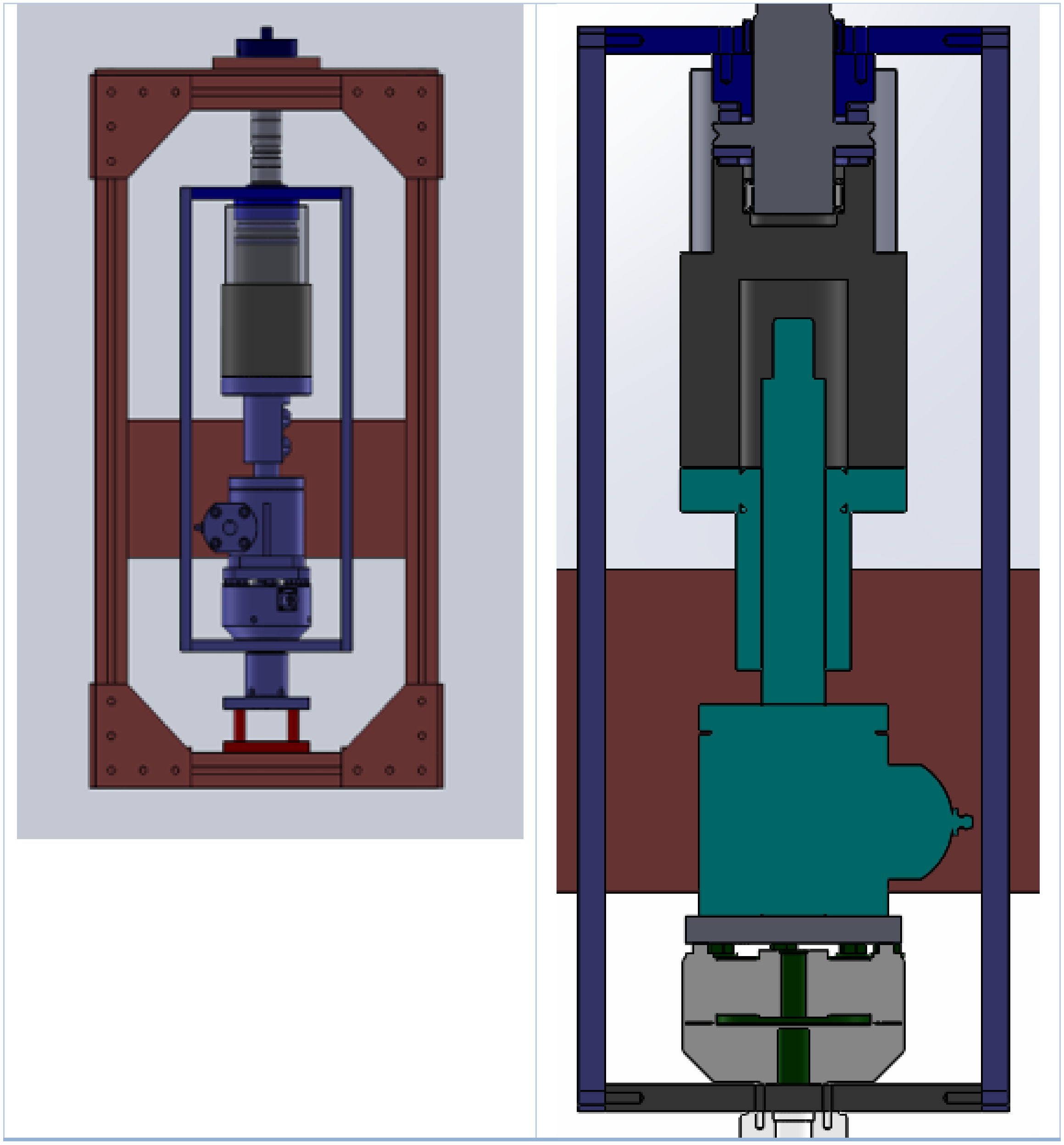


Figure 29. Load Loop

Zooming in on the test area, it can be seen how the test piece will be held in place. There are two INA bearings, seen in red, that keep the shaft aligned during testing. These bearings are press fit into the top plate and the screw jack bracket, so as to be placed as close as possible to the test area. The test area consists of a sandwich of a thrust bearing above the flange of the shaft, shown in green, a skew roller bearing below the flange on the shaft, shown in light blue, and two thrust plates forming the "bread" on either side of the bearings. These bearings are squeezed together on the flange section, which causes the friction that will be measured by the torque sensor.

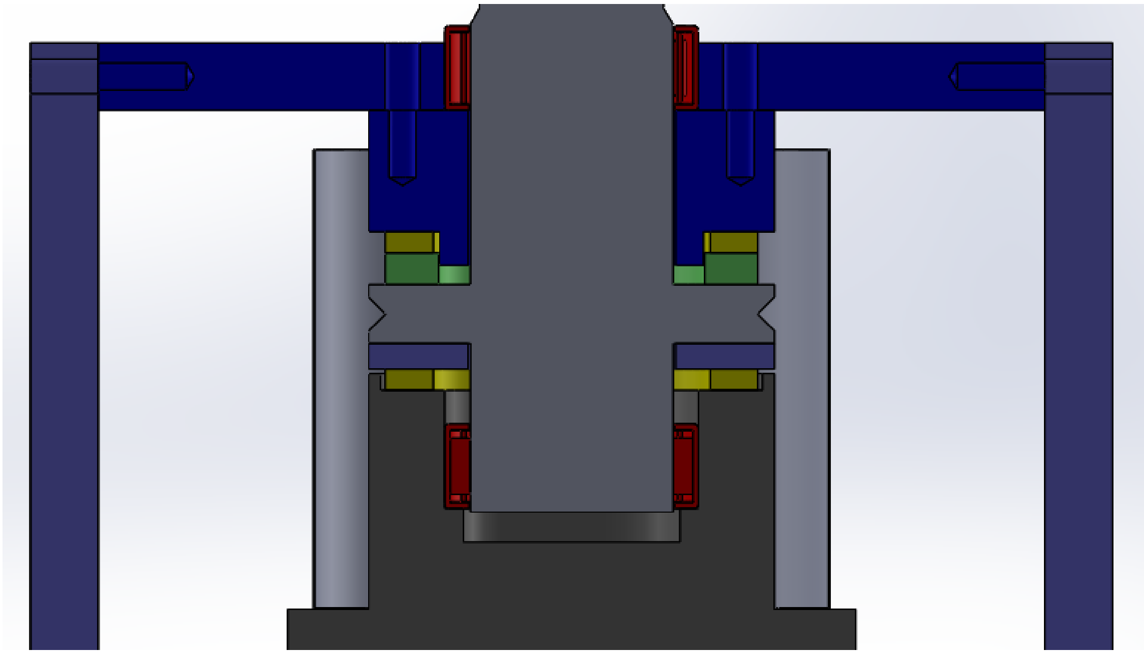


Figure 30. Test Area

Moving further down the test fixture, it can be seen how the torque will be measured. The load loop is connected directly to the torque sensor, which operates using strain gauges that measure the radial displacement of the load loop. The problem with the reaction torque sensor is that it cannot take axial load. In order to accommodate for this, the torque sensor is resting on a plate with holes for pins. These holes have a sliding fit against a plate attached to the structure that has pins press fit into it. The sensor can move up and down, sliding on the pins, but the lower half of the torque sensor is still rigidly attached to the structure.

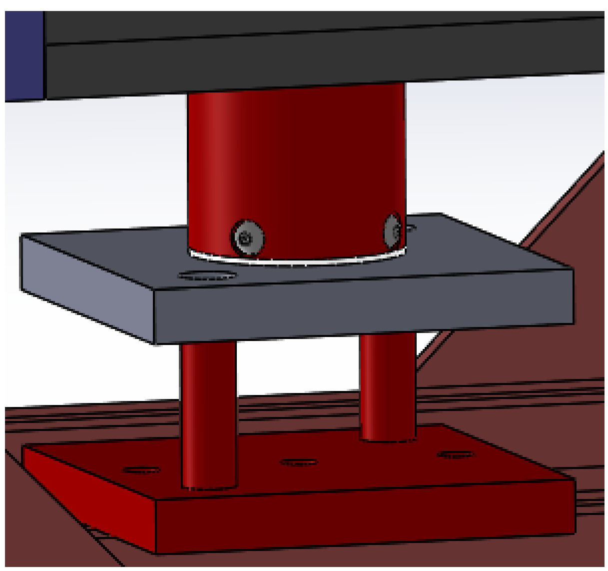


Figure 31. Torque Sensor

The last component of the mechanical design of the structure is the thermal chamber. The front plate of this thermal chamber has been made transparent for clarity. The thermal chamber is meant to allow for the introduction of hot or cold air into the test area via an air duct on the side. The thermal chamber is made of plywood, and will be nailed together, with the exception of the top piece, which will be latched on to allow for easy disassembly to access the test area. The temperature will be measured with a thermocouple.

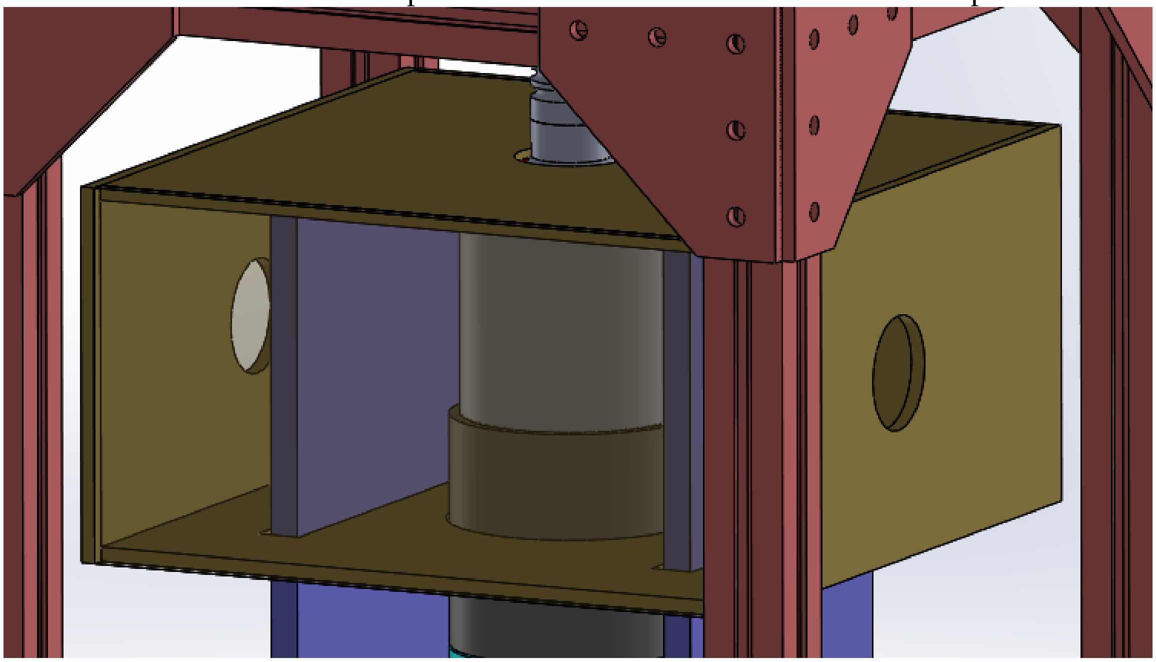


Figure 32. Thermal Chamber

Material Selection

One of the constraints of the project was to make it capable of being transported by a two-man-carry, so choosing lighter weight materials was a priority. It is for this reason that most of the test fixture is made of 6061 Aluminum. Analysis was run to make sure that the aluminum would hold the necessary loads (see the analysis section of the report), and it was found to be well within the accepted range. For the thermal chamber, plywood was chosen due to its low cost and effectiveness in containing the hot or cold air without dissipating too much heat to the sensors.

Fabrication Instructions

There are a number of custom parts to be machined for this project. The part drawings can be seen in Appendix O. Though most of the manufacturing will be simple drilling and tapping, some use of mills and lathes will be necessary. The skew roller bracket is the most complex part, seated on the screw jack and connecting to the oil container within the thermal chamber. For this part we will need to acquire a 1 ¼" drill bit, as the machine shops on campus do not have drill bits larger than an inch in diameter. We will also be using the lathe to turn down cylindrical parts to various diameters, and milling our parts that have high positional tolerances. The dowel pin alignment below the torque sensor is crucial, and to ensure the holes on the base plate and sliding plate are lined up, we will drill both simultaneously with the two plates clamped together. To allow for clearance for the sliding plate, we will drill again with a larger bit positioned over the already aligned holes. In the coming weeks, detailed manufacturing instructions for individual parts will be made.

Component Selection

<u>Axial Load Application</u>

For the axial load application, a manually driven load application is preferred due to ease of design and cost effectiveness. A manually driven axial load is acceptable for producing the amount of load necessary (5000 lbs.) and results in acceptable accuracy. The three primary contenders for the axial load application were a screw jack, a pneumatic jack, and a hydraulic jack. Other options were considered, but these three were found to be the most feasible for this system. A Pugh matrix evaluating the relative ability of each option to fulfill the design requirements and engineering specifications is attached in Appendix E.
The actuator that was found to be the most appropriate for the set-up was the screw jack. A screw jack is the simplest way to produce the axial load by converting torque to linear force. The screw jack is a purely mechanical device and is manually operable. The screw jack we have selected, the 3-BSJ-UR by Nook Industries is depicted below in Figure 33. This screw jack has a 24:1 gear ratio, which will allow fairly accurate control of the load.

Figure 33. Nook 3-BSJ-UR 24:1 Screw Jack [22]

The hydraulic jack is a good alternative to the screw jack as it is also manually operated and costs about the same. However, the hydraulic jack was determined to be inferior for this application due to the larger size and the possibility of hydraulic fluid leakage. The pneumatic jack is more expensive than the other two options and also requires compressed air to drive the load. The pneumatic jack is slightly larger than the previously mentioned alternatives and introduces safety concerns due to the use of compressed air in pressure vessels.

Drive Assembly

The configuration that was decided to be used was an electric drill motor. The other options for the motor include a single-phase AC motor, a DC brushed motor, and a DC brushless motor. The required power to be produced by the motor was found to be 1.1 hp, based on the speed and torque specifications of 200 RPM and 350 in-lb. All of the motor options are capable of producing 1.1 hp. The single-phase AC and the DC brushless motors were found to be significantly more expensive options than the electric drill motor and the DC brushed motor. The electric drill motor is slightly less expensive than the DC brushed motor. The drill motor is capable of producing variable speeds via hand operation by the operator. If automated speed control is desired for the drill or DC brushed motor, a controller will be necessary, which will increase the complexity of the design and require additional analysis. The drill motor was chosen to be the cheapest and most robust option that minimizes complexity. A belt drive was chosen rather than a gear driven arrangement because it is more cost effective and it also simplifies the design by allowing for easier placement of the motor in the configuration. A Pugh matrix evaluating the drive assembly options is attached in Appendix E.

The choice of drill is the Bosch 1034VSR corded drill. This was chosen due to its ability to produce 488 in-lb of torque at a speed of 200 RPM, which means that it can drive the shaft directly without the need for gearboxes or belts. An analysis of the motor curve can be seen in the analysis section of the report.

Figure 34. Bosch Drill Motor

Torque Sensor

For the torque cell, a reaction torque sensor was chosen over a rotary torque sensor (see background research section for explanation on the types of torque sensors). This was decided due to many considerations. One major factor was the cost; reaction cells are much simpler, and therefore much cheaper than the rotary counterpart. Another big factor is that the reaction torque cell measures only the reaction torque, which is the only measurement of interest. The reaction torque sensor that has been selected is the Futek TFF400 (see Figure 35). It can measure torque values up to 500 in-lb, which is ideal in a configuration of a skew roller bearing paired with a thrust bearing.

Figure 35. Futek TFF400 Sensor [23]

Speed Sensor

A photoelectric optical sensor was chosen to measure the rotational speed of the test shaft. There are three different types of photoelectric optical sensors: reflective, thrubeam, and retroreflective. For the purposes of this project, a reflective model was chosen. The model as shown in the next figure, has bot the emitting and receiving elements contained in one housing, Light is shot out of the housing onto a target, and then reflected back to the housing's receiving element, which sends out a voltage based on how much light is reflected back. In this project, the target would be a piece of light colored tape with dark marks along the surface. When placed around the test section, the dark marks would help determine the rotational speed of the device. The Optek OPB720A-06Z shown in Figure 37 is the chosen optical sensor, due to its simplicity and low cost (about $13).

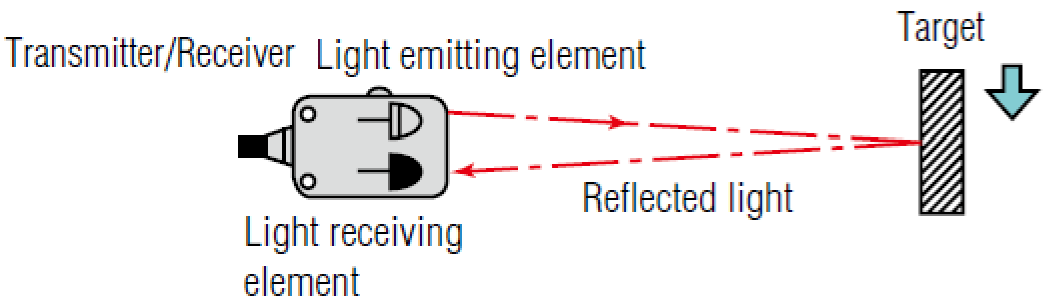


Figure 36. Depiction of how a reflective photoelectric sensor works. [25]

Figure 37. Optek OPB720A-06Z photoelectric sensor. [26]

A Hall Effect Sensor is a possible alternative to the optical sensor. The main principle is that a magnet is attached to the rotating shaft, and a non-moving sensor is placed such that the magnet will pass by it every rotation, exciting the sensor. From this excitement, the RPM of the shaft can be measured. A possible alternative design is to use a rotary encoder. A rotary encoder consists of a disk with radial etches at known angles from each other that spins with the shaft. A sensor that detects each of these etches is placed around a portion of the disk in order to determine the angular position of the disk.

Data Acquisition System

The data acquisition system we are planning to use is a National Instruments compactRIO. This module will accommodate the four channels of measurements we need to record at an ample sampling rate. This seems like the best option as it is small, programmable in LabVIEW, and adaptable to any computer with a USB. The other alternatives we eliminated were the PolyDAQ and other, larger, NI DAQ systems. The PolyDAQ is a very specific system developed at Cal Poly, and as such it is a less versatile system compared to the other two. Another drawback of the PolyDAQ is that the programming is done in python, which the team has no experience with and the sponsor has not used in the past. LabVIEW is a program that Parker has used in the past, and will have a smaller learning curve as it is similar to Simulink in MATLAB, which we have some experience with. A Pugh matrix evaluating the DAQ options is displayed in Appendix E.

Figure 38. NI USB-6000 [24]

The module in Figure 38 is a National Instruments USB-6000. This data acquisition system can sample 4 channels well over our 1kHz requirement, and plugs directly into a pc via a USB cable. The programming for this DAQ will be done in LabView, which Parker is familiar with using.

Analysis

FMEA

A failure mode effects analysis (FMEA) is included in Appendix M. The FMEA is a tool used to determine potential causes and mechanisms of failure of a design. FMEA uses a branch structure to determine the potential causes/mechanisms of failure for each potential effect of failure for each potential failure mode of each item or function. This method ensures that most or all possible causes of failure for a given design can be determined. The criticality of each potential mechanism of failure represents the priority of given failure mechanisms. The criticality is a function of the frequency or occurrence of the failure mechanism and the severity of the effect of failure for that mechanism. Failure mechanisms with a high criticality should be addressed with the highest priority. Actions can be recommended to address each failure mechanism.

Motor

The chosen motor for the design is a Bosch 1034VSR corded drill. The data sheet for this drill can be seen in Appendix G. The maximum RPM of this drill is 550 RPM at zero load, and the zero RPM max torque is 550 in-lb. From this information, as well as the fact that the motor curve is linear, a motor curve was developed, shown in Figure 39, below.

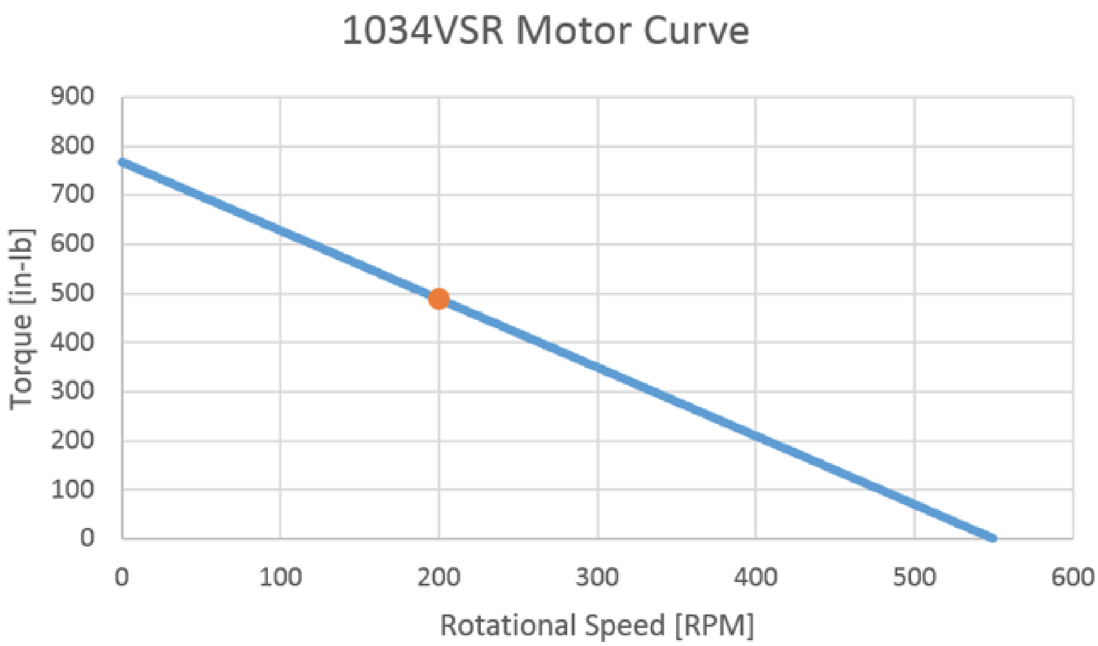


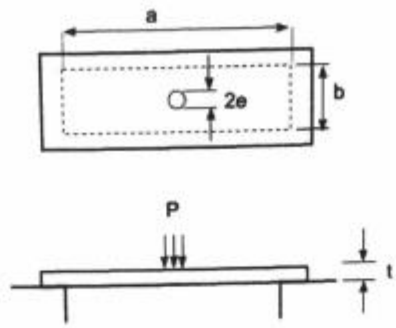
Figure 39. Motor curve for 1034VSR

The point of interest is when the operating speed of the test fixture is the greatest (200RPM), and the maximum output torque is the lowest. This point, shown as the orange dot, shows that at our maximum operating speed of 200RPM, the drill could output 488 in-lb of torque, which is well over the de. This means that the motor alone could output the necessary torque to operate the fixture without the need for a gearbox or any belt driven application. The drill is going to directly drive the test fixture, making the overall design much more simple.

Structural

In order to determine the size and material of the top plate of the structure, Roark's formulas for stress and strain were used, seen in Figure 40, below.

Rectangular Flat Plate , concentrated load at centre,edge simply supported.



If e is small then use e' as calculated below

$$e' = \left(\sqrt{1{,}6e^2 + t^2}\right) - 0{,}675t \quad \text{if } e < 0{,}5t \text{ else use } e' = e$$

$$\sigma_m = \frac{1{,}5P}{\pi t^2}\left((1+\nu)\ \ln\left(\frac{2b}{\pi e'}\right) + k_2\right) \qquad \text{At centre}$$

$$y_m = k_1\ \frac{Pb^2}{Et^3} \qquad \text{At centre}$$

	a/b								
	1,0	1,1	1,2	1,4	1,6	1,8	2,0	3,0	4 ->
k_1	0,127	0,138	0,148	0,162	0,17	0,177	0,180	0,185	0,185
k_2	0,435	0,565	0,650	0,789	0,875	0,927	0,958	1,000	0,000

Figure 40. Roark's formula for chosen load case

The analysis discussed can be found in Appendix H. The load case was a simply supported rectangular flat plate with a concentrated load at the center. The overall weight of the load loop was estimated to be 62 pounds, and a factor of safety of about 1.6 was added to make the load seen by the plate 100 pounds. The chosen material of the plate was 6061 Aluminum (yield strength 39E3 psi), since it is lightweight and cheap compared to steel or more exotic metals. First, a ½" plate was analyzed, and the max stress was found to be 378.44 psi, with a max deflection of 0.002601 inches which is well below the limits of the material. Since ½" is rather thick, a ¼" plate was then analyzed. The max shear stress of this thickness was found to be 1431.48 psi, with a max deflection of 0.020808 inches. This means that the safety factor of using a ¼" 6061 aluminum plate is 27.2445, which is plenty of reason to say that the plate will work on this structure.

Bolt Shear

The two areas analyzed for shear strength on this machine are the dowel pins at the bottom of the structure attached to the torque sensor, and the grade 8, ¼ - 20 bolts attaching the tension plates to the load loop.

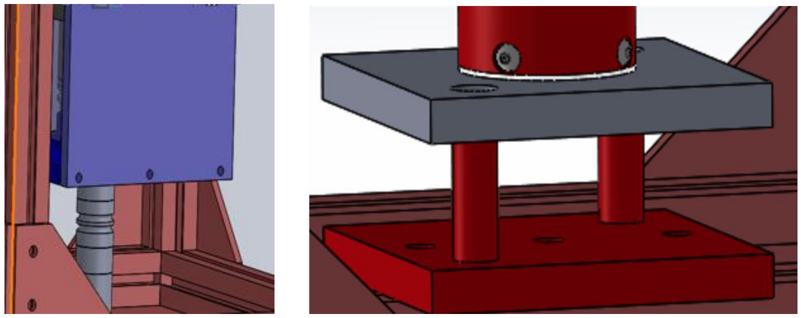


Figure 41. Shear Analysis

Using Shigley's Mechanical Design, [27] the maximum shear force for each bolt and dowel pin could be found. For the bolts in shear attaching the tension plate to the load loop, the resultant shear force for each bolt was calculated from the tensile and torsional forces applied. Using the resultant force and cross sectional area of the bolt, the maximum shear stress in each bolt could be found. The methodology of this calculation can be seen in Appendix I. For ease of operator assembly and disassembly, less bolts are advantageous, so a bolt diameter of ¼" was selected. As seen in Appendix I, one ¼"-20, grade 8 bolt has a single shear strength of 4417 lbs. Most data on bolts includes tensile yield strength, but several sources had shear yield strength values for various bolt sizes. The resultant shear force on each tension plate is approximately 2500 lbs, and 6 bolts per plate will be included for an added factor of safety.

The shear analysis on the two dowel pins is much more simple. Using the same equations from Shigley's, the resultant force on each pin could be calculated. These pins are facing very minimal shear force compared to their single shear strength. A 1/8" steel alloy dowel pin has over 1800lbs shear strength, and the shear force being applied during operation of the machine will be about 200lbs as seen in Appendix I. For our model, 2, ½" diameter dowel pins will be implemented to allow axial motion of the torque sensor while constricting torsional motion to ensure accurate measurements.

Bending

Along with shear calculations on the tension plates, critical bending stress in each plate was an area of interest. Again, using Shigley's Mechanical Design book [27], the max stress in each plate was calculated using the method shown in Appendix I. Using the applied moment, the area moment of inertia, and the perpendicular distance to the neutral axis, the critical bending stress was found to be about 1200psi. Given the plates are 6061 Aluminum with a yield stress of 35,000psi, plate bending is not of concern for our machine.

Thermal

An initial thermal analysis was conducted with our initial design. The purpose of the thermal analysis was to determine if the temperature sensitive measuring devices, including the load cell and the torque sensor, would be unaffected by the hot or cold temperature of the thermal chamber. The main concern for in developing the thermal analysis was the close proximity of the load cell to the thermal chamber in the initial design. The thermal chamber was modeled as flat plates with a constant internal temperature at the high and low extremes of our given temperature range, 160 F and –65 F, respectively. The plywood plate enclosing the chamber was assumed to have two modes of heat transfer: conduction and natural convection. The results of the analysis yielded surface temperatures of 132 F and –37 F for the hot and cold conditions, respectively. With these initial results, the effects of the temperature gradients on the load cell were concerning, so the load cell was repositioned below the screw jack. The new position of the load cell is far enough away from the thermal chamber for it to be unaffected by temperature conditions. Thermal analysis was performed using Excel, shown in Appendix J.

Cost

The estimated cost of purchased parts is presented in the bill of materials (BOM) in Appendix K. The BOM includes all of the parts modeled in the solid model, including all manufactured parts, purchased parts, fasteners, etc. The BOM only includes pricing information for purchased parts, directly from the supplier. The BOM separates subassemblies and the respective parts in each subassembly, along with respective part numbers. The part numbering system is shown in the BOM. The total cost estimate for purchased parts is $3334.62. Spec sheets for included parts are shown in Appendix G and drawings are included in Appendix O.

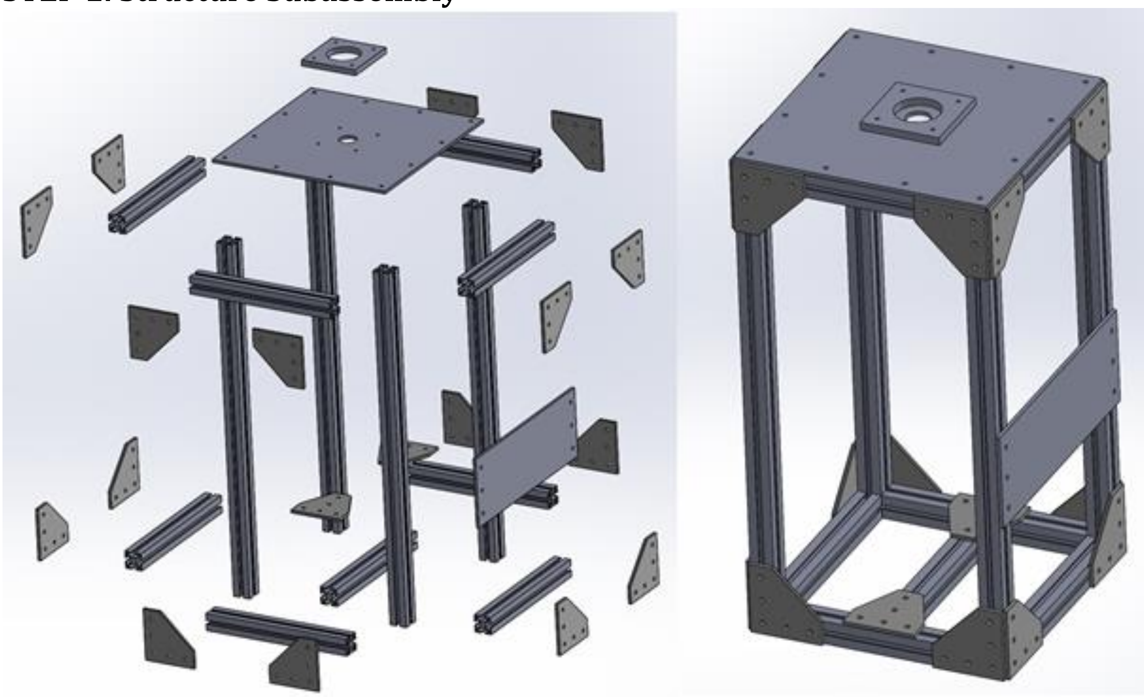
The source budget, shown in Appendix L, includes the total cost estimation, from both purchased and manufactured parts. The total cost is estimated to be $4,133.15, which is well under the $5,000 budget. For this project, our sponsor, Parker, sent Cal Poly $2,000 to use to purchase materials and parts for use on the project to ease and hasten the procurement process. We propose for Parker to purchase the two high-ticket items, namely the screw jack and torque sensor, and for Cal Poly to purchase the remaining parts. The estimated cost breakdown for this proposition is $2,595.00 and $1,538.15 for Parker and Cal Poly, respectively. We are below budget for the planned purchased parts using the $2,000 Cal Poly budget.

Design Verification

A design verification plan (DVP) is included in Appendix N. The test plan includes planned tests to be completed on the build system to verify that it meets the design specifications. Estimated dates for testing is included.

Assembly Instructions

STEP 1: Structure Subassembly



Structure Exploded View Structure Subassembly

Figure 42. Assembly of Structure Subassembly

The first step in assembling the Friction Test Machine (FTM) is to assemble the structure. The structure will be assembled with bolts fastened to brackets. Depicted in the figure above. After assembling the 80/20 aluminum structure, the side panel is fastened to two vertical members with six bolts, three on each side. Next, the top plate is bolted to the smaller support plate using four bolts in a square hole-pattern. The top plate is then bolted to the top of the 80/20 structure on all sides.

STEP 2: Pinned Bracket Subassembly

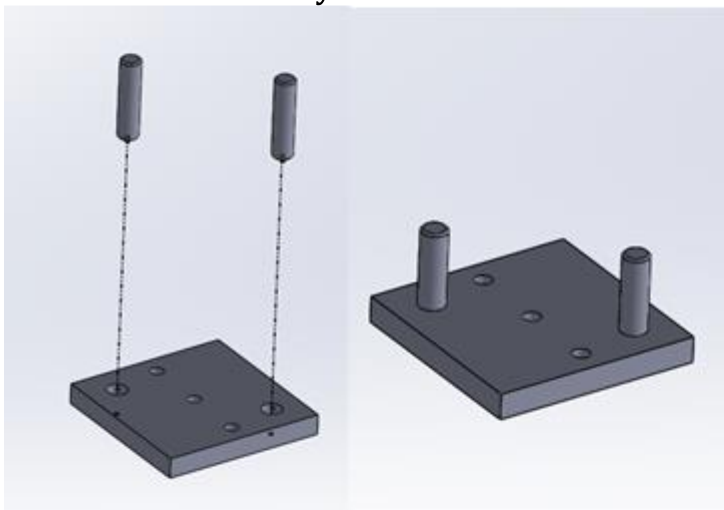


Figure 43. Assembly of Pinned Bracket

The next step in assembly is to press fit the two pins into the pinned bracket subassembly. The pins will be press fit flush with the bottom of the plate using an arbor press. The pinned bracket subassembly is then assembled with the structure using three bolts to attach it to the 80/20 structure, as shown below.

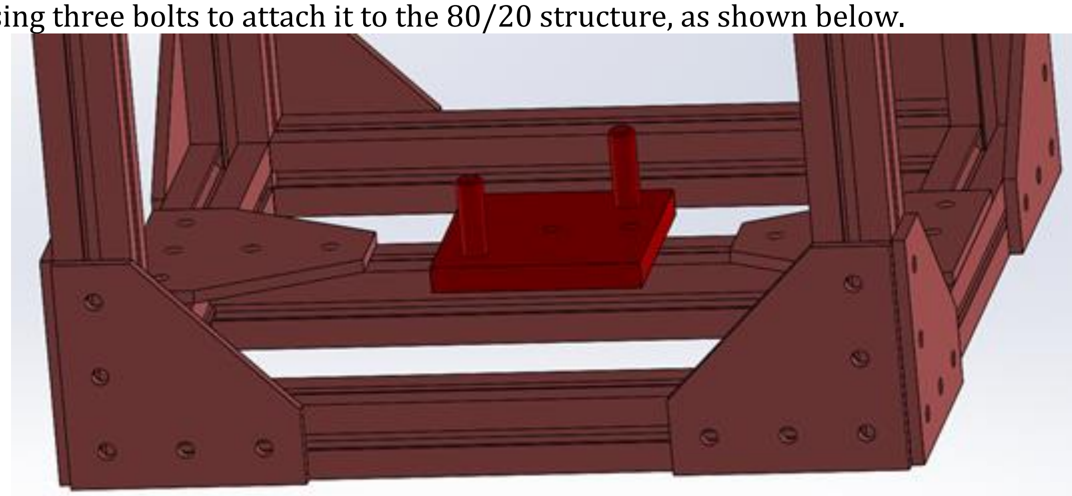


Figure 44. Assembly of Pinned Bracket with Structure

STEP 3: Oil Bracket Subassembly

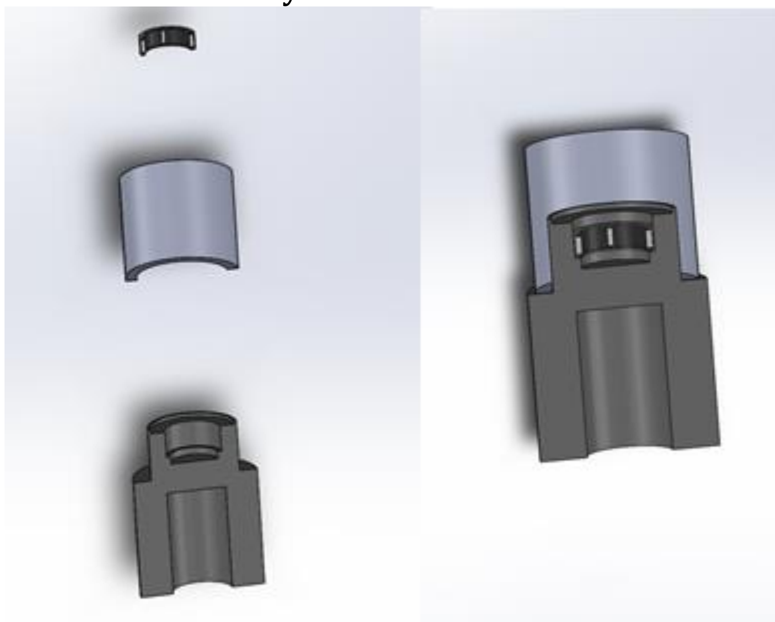


Figure 45. Oil Bracket Subassembly

Step 3 involves the assembly of the oil bracket. The oil bracket consists of the bracket (dark grey) pictured above with the tin oil container (light grey) and a press-fit bearing (dark grey). The bearing will be lightly pressed into the bracket with an arbor press. Next, the inner ring of the oil container is epoxied to the bracket to create a seal to prevent oil leakage.

STEP 4: Load Loop Bottom Subassembly

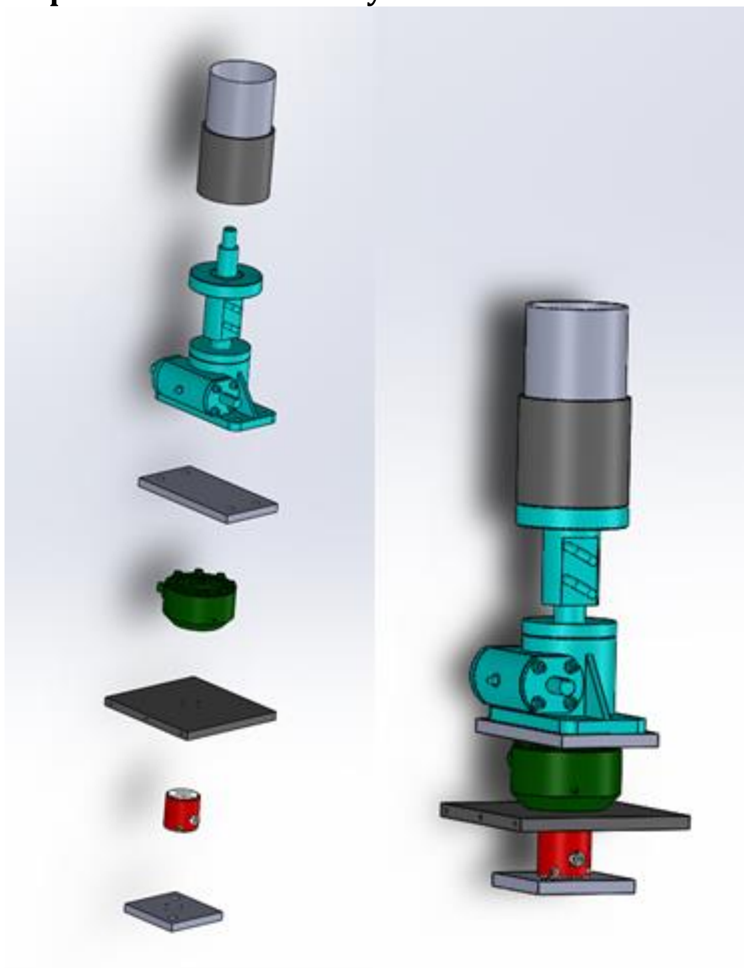


Figure 46. Load Loop Bottom Subassembly

After assembling the oil bracket, the bottom section of the load loop can be assembled. Starting from the top, the oil bracket is fastened to the screw jack using four screws. The bottom of the screw jack is fastened to the interface plate using four bolts. The interface plate is attached to the load cell using a central screw. The load cell is attached to the dark grey bottom plate again with a center screw. The torque sensor, in red, is attached to its two mating plates using four screws on each side in a square hole-pattern.

STEP 5: Load Loop Top Subassembly

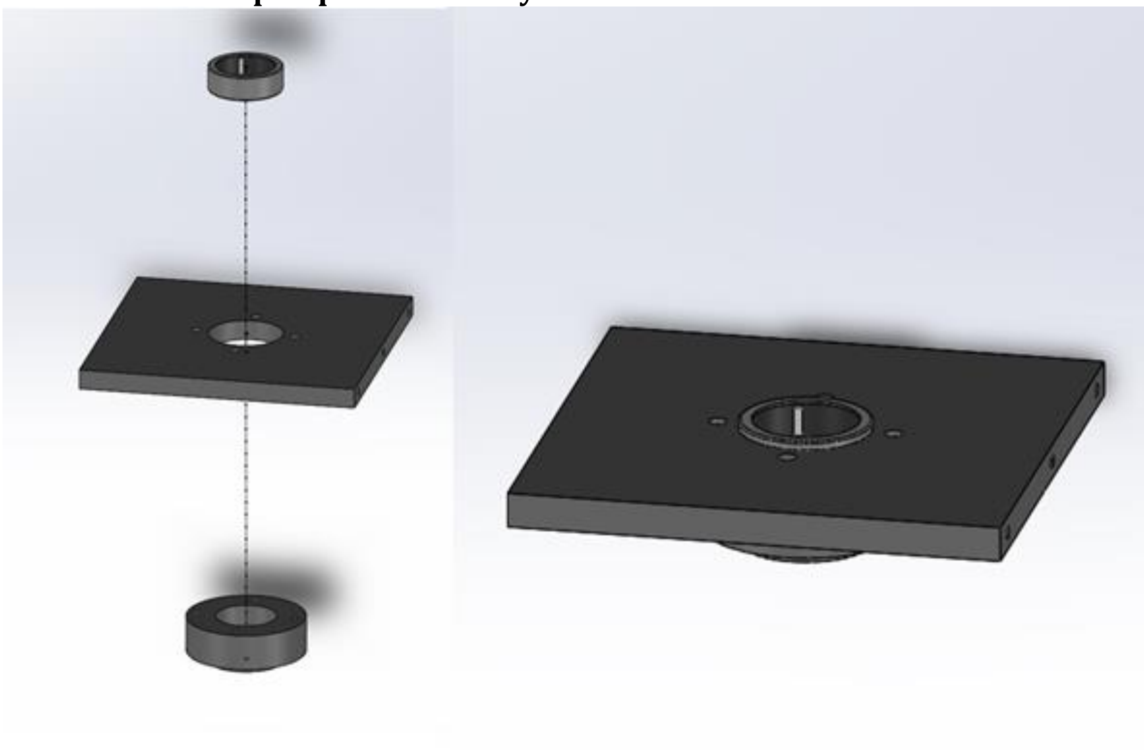


Figure 47. Load Loop Top Subassembly

The top structure of the lop loop is assembled in step 5. The plate is screwed into four tapped holes on the cylindrical bracket to secure them to one another. The bearing (top) is then press fit into the subassembly using an arbor press.

STEP 6: Load Loop Assembly

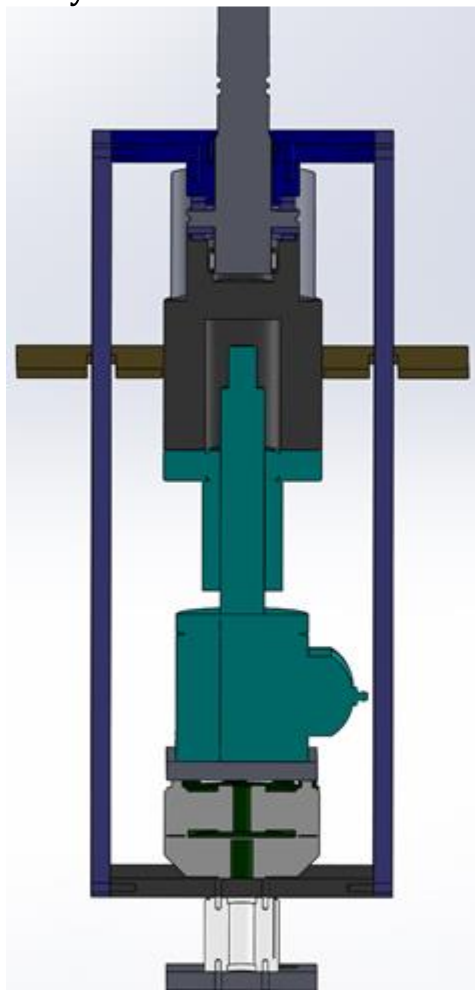


Figure 48. Load Loop Assembly

The two assemblies of the load loop are then assembled to create the final load loop assembly. The load loop bottom subassembly will be supported by the bottom dark grey plate with two wood blocks. The thrust plate and skew roller plate (light blue, bottom side) will be positioned into the dark grey oil bracket. The shaft (light grey) will be placed into the bearing and on top of the skew roller plate. The bottom piece of the thermal chamber (brown) will be slide down before adding the load loop top subassembly. The thrust washer and thrust plate (light blue, top side) are placed on the top side of the shaft's flange. The load loop top assembly is slide down the shaft and placed atop the thrust plate. Next, the blue tension plates are slid into the slots in the bottom thermal chamber piece and fastened to the top and bottom plates of the load loop t complete the assembly.

STEP 7: Thermal Chamber Assembly

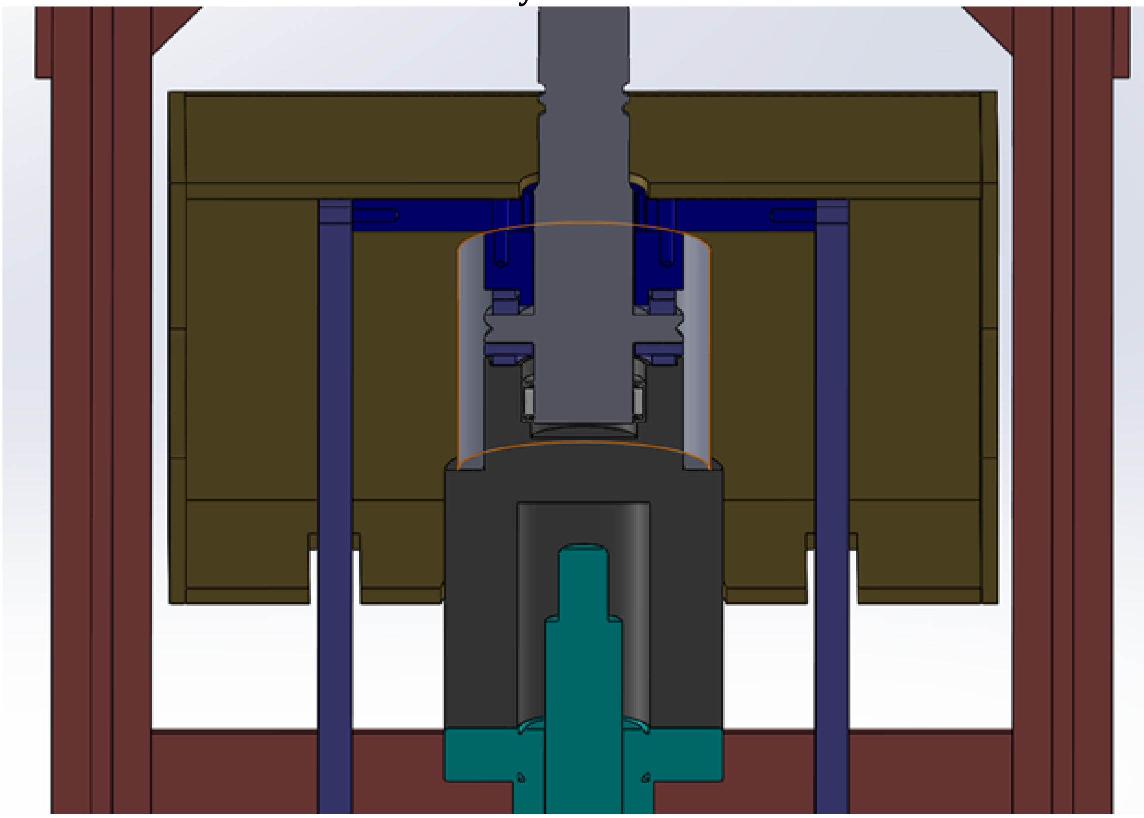


Figure 49. Thermal Chamber Assembly

Step 7 will consist of the completion of the thermal chamber around the load loop. The top of the thermal chamber will be slid over the top of the shaft and the side panels will be nailed to the bottom piece and connected to the top piece via removable clamps.

STEP 8: Structure and Load Loop Assembly

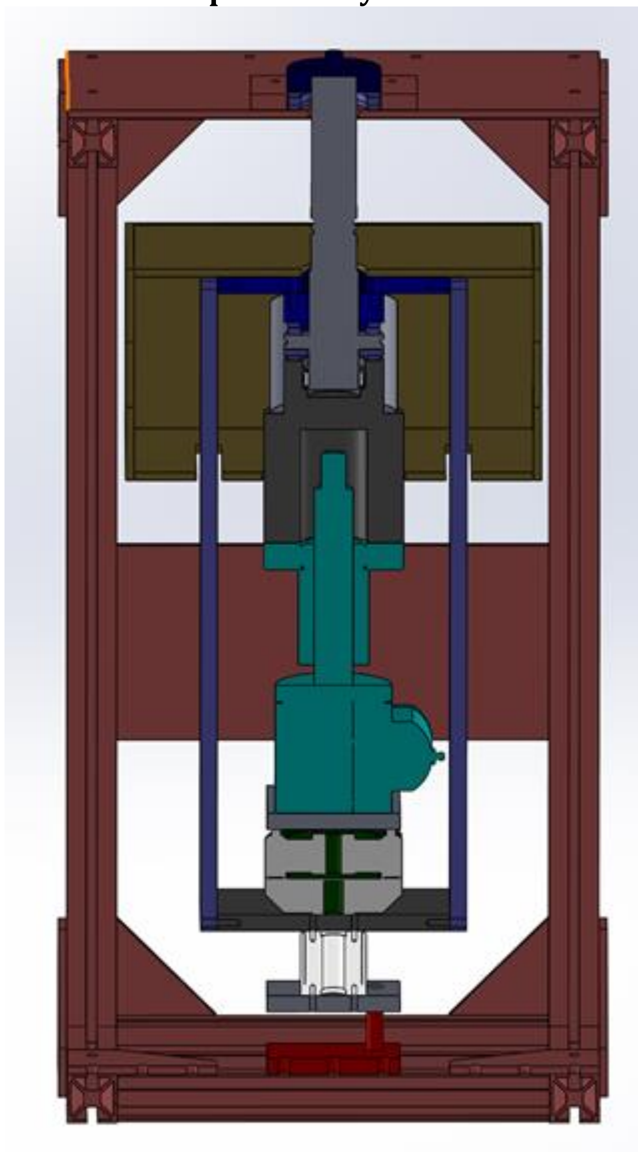


Figure 50. Load Loop and Structure Assembly

Step 8 involves the placement of the previous load loop assembly into the structure subassembly. The load loop assembly with be inserted through the side of the structure and the top of the light grey shaft will be placed through the hole in the top plate of the structure, where the light blue thrust plate and bearing reside. The dark blue piece at the top will be connected to the shaft via a through bolt. The sliding grey plate at the bottom of the load loop will be aligned with the red pinned bracket subassembly before allowing the weight of the load loop to be supported by the top.

STEP 9: Motor Mounting

Figure 51. Bosch Drill Motor

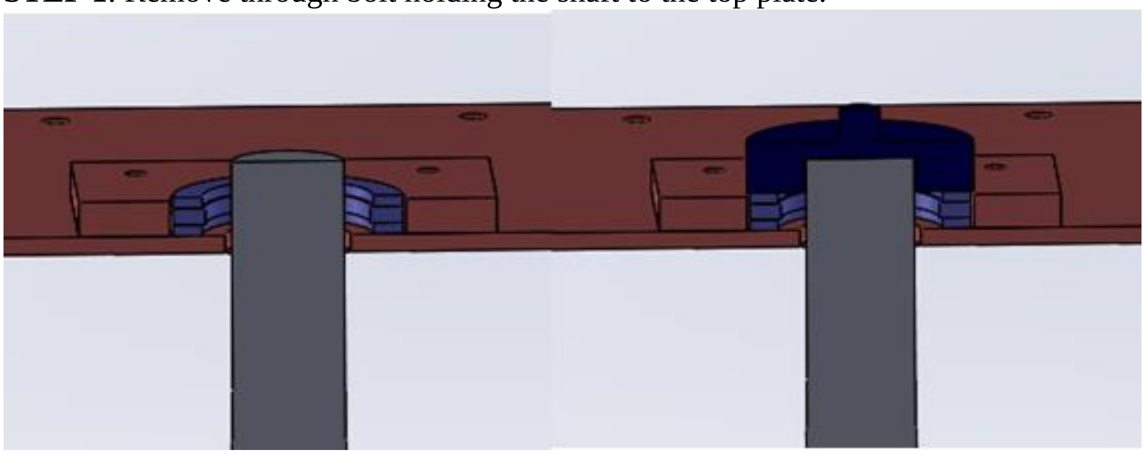
The assembly of the FTM is concluded with the mounting of the motor, which will directly drive the shaft via the blue piece at the top of the assembly. The motor mounting assembly instructions will be included when the motor mounting design is complete after the procurement of the motor.

Maintenance Considerations

Use of the FTM will require disassembly and reassembly of the machine several times to collect data with various configurations and conditions. The process of disassembling and reassembling the FTM involves several steps due to the nature of the design. In order to best accommodate for the user, instructions for the disassembly and reassembly of the FTM have been written, shown below.

Disassembly

STEP 1: Remove through bolt holding the shaft to the top plate.



Before After

Figure 52. Shaft Bolt Removal

Following the removal of the shaft, the load loop will be supported from the bottom via brackets, to ensure the torque sensor does not receive any axial load.

STEP 2: Remove top plate of the thermal chamber.

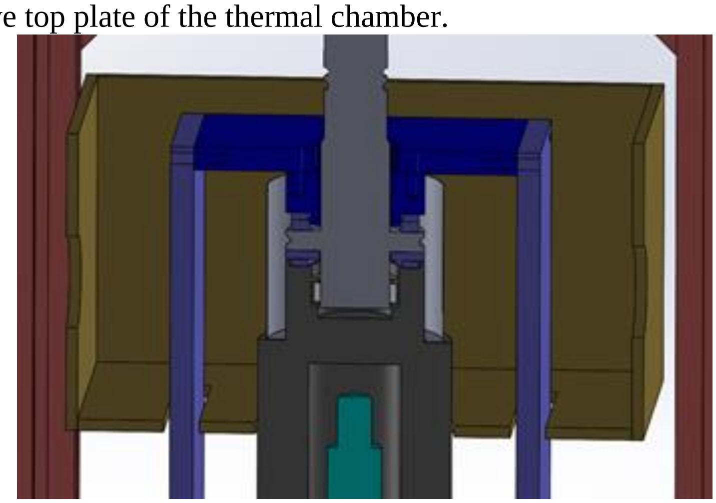


Figure 53. Thermal Chamber Top Plate Removed

After removing the top plywood panel of the thermal chamber by unlatching the latches and sliding the panel off from the top, the resto of the thermal chamber will slide down and rest on supports at the bottom. This allows for room to see and work with the test section.

STEP 3: Drain oil.

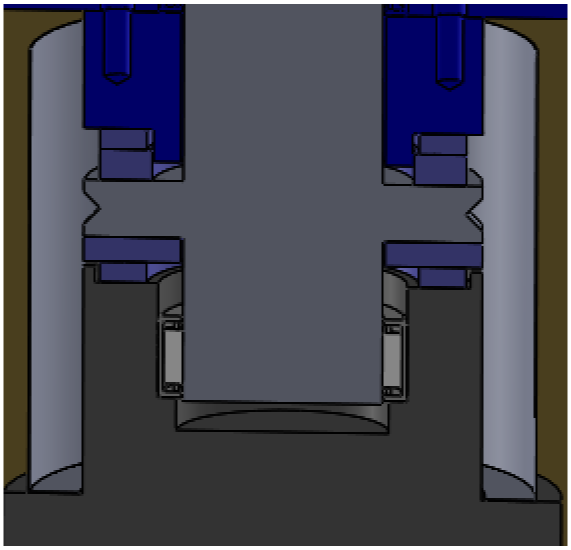


Figure 54. Oil Chamber

The oil will be drained via a plug in the bottom right hand side of the grey oil chamber, pictured above. This will allow for the removal of the oil from the test section before the shaft is removed.

STEP 4: Unscrew side tension plates from the top plate.

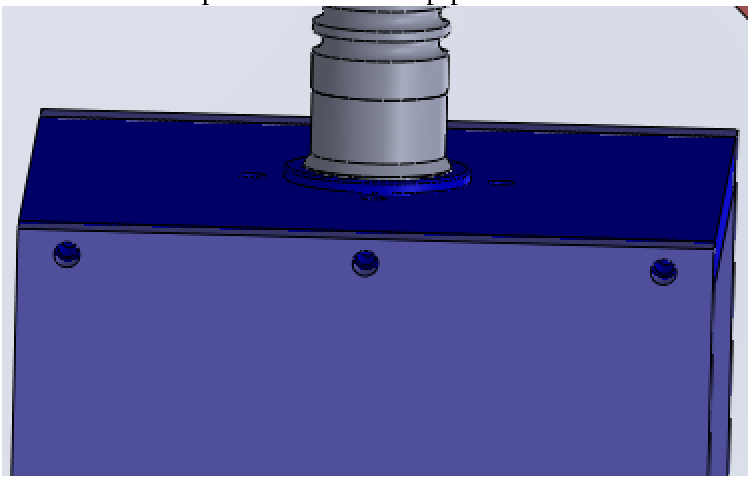


Figure 55. Tension Plate Removal

The tension plates on either side of the load loop will be unfastened from the top plate to allow for removal of the test section.

STEP 5: Remove shaft (grey) and top plate (blue) from the oil chamber.

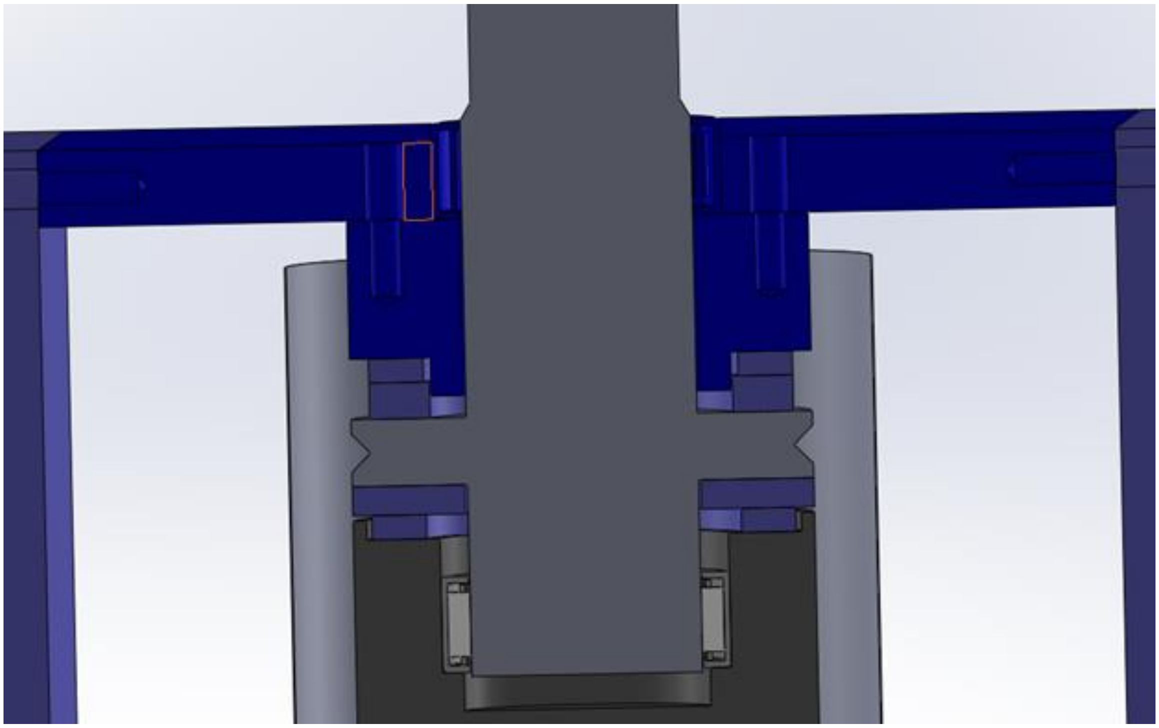


Figure 56. Remove Shaft from Oil Chamber

The shaft (grey, center) will be removed from the top along with the thrust bearing and thrust plate (light blue, above flange) and the load loop top assembly (dark blue). At this point the thrust bearing may be removed if necessary. The skew roller plate and thrust plate (light blue, below flange) will remain in the oil chamber and must be removed by hand. The oil chamber may be cleaned at this time if a different oil is to be used. The skew roller and thrust bearing may be interchanged for varying test conditions if necessary. This concludes the disassembly of the FTM for the operator.

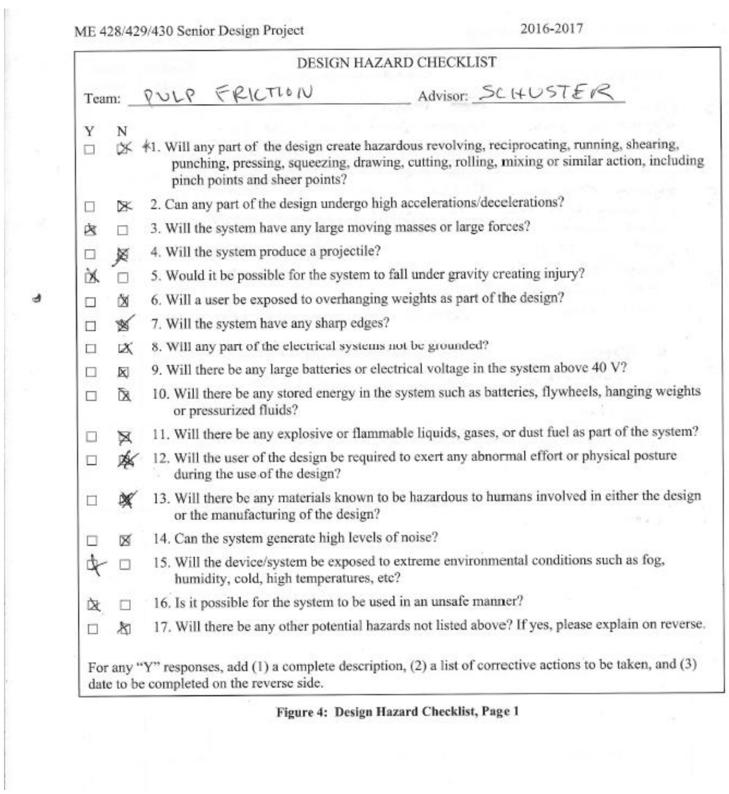
Reassembly

The reassembly of the FTM for testing is largely the same as the disassembly, in the reverse order. The primary difference is that the plug in the oil chamber must be inserted and the oil will be added to the chamber via a plastic tube after the shaft has been placed in the chamber.

Safety Checklist

A safety checklist was created (see Figure 57) in order for the design to take into account all dangerous aspects of the project and minimize all safety risks. From this list, four hazards were specifically called to attention. The first hazard comes from the 5000 lb axial load being applied by the screw jack. In order to handle this load and prevent overloading, the structure will be designed to take that load and a limit switch or mechanical stop will be added to prevent overloading. The next hazard is tipping hazard due to having the motor mounted up high on the structure. Next is a hazard to the sensors due to the thermal chamber. In order to fix this, the design will include insulating the sensors from the heat/cold air of the thermal chamber.

The test machine design shown in this document has many safety considerations. The user safety is our priority as an operator will be using this device. The 5000lb load on the skew rollers is contained within a load loop inside of the structure. The components of the load loop have been analyzed and designed with a high factor of safety to prevent failure. The mounting of the drill on the top of the structure allows the operator to easily access it, and flats along the attachment will ensure a tight grip of the motor to the rotating piece. The sensors have been placed away from the thermal chamber to improve accuracy of test results and functionality. Lastly, in order to account for possible operator hazards, we will make an operator manual that emphasizes how to use the test machine safely.



ME 428/429/430 Senior Design Project 2016-2017

DESIGN HAZARD CHECKLIST

Team: PULP FRICTION Advisor: SCHUSTER

Y	N	
☐	☒	1. Will any part of the design create hazardous revolving, reciprocating, running, shearing, punching, pressing, squeezing, drawing, cutting, rolling, mixing or similar action, including pinch points and sheer points?
☐	☒	2. Can any part of the design undergo high accelerations/decelerations?
☒	☐	3. Will the system have any large moving masses or large forces?
☐	☒	4. Will the system produce a projectile?
☒	☐	5. Would it be possible for the system to fall under gravity creating injury?
☐	☒	6. Will a user be exposed to overhanging weights as part of the design?
☐	☒	7. Will the system have any sharp edges?
☐	☒	8. Will any part of the electrical systems not be grounded?
☐	☒	9. Will there be any large batteries or electrical voltage in the system above 40 V?
☐	☒	10. Will there be any stored energy in the system such as batteries, flywheels, hanging weights or pressurized fluids?
☐	☒	11. Will there be any explosive or flammable liquids, gases, or dust fuel as part of the system?
☐	☒	12. Will the user of the design be required to exert any abnormal effort or physical posture during the use of the design?
☐	☒	13. Will there be any materials known to be hazardous to humans involved in either the design or the manufacturing of the design?
☐	☒	14. Can the system generate high levels of noise?
☒	☐	15. Will the device/system be exposed to extreme environmental conditions such as fog, humidity, cold, high temperatures, etc?
☒	☐	16. Is it possible for the system to be used in an unsafe manner?
☐	☒	17. Will there be any other potential hazards not listed above? If yes, please explain on reverse.

For any "Y" responses, add (1) a complete description, (2) a list of corrective actions to be taken, and (3) date to be completed on the reverse side.

Figure 4: Design Hazard Checklist, Page 1

	Description of Hazard	Planned Corrective Action	Planned Date	Actual Date
3	5000 lb force applied by jack to system	~~Add~~ Design structure to support 5000 lb and add limit switch to prevent user from manually over-loading		
5	system may be top heavy with heavy motor mounted in vertical config.	Design, Analyze, test structure to prevent failure under its own weight		
*15	Exposed to hot/cold temp inside ~~the~~ test area	Design thermal chamber and conduct analysis to verify sufficient insulation		
16	System may be overloaded or otherwise operated unsafely	Write operating guidelines and instructions for technician		

Figure 57. Hazard Safety Checklist

MANAGEMENT PLAN

Roles within the team will be designated a lead who is responsible for the completion of that subsystem. For organizational roles, Megan will be the communications officer, Alec will be the team treasurer, and Brandon will be the secretary and recorder. As communications lead, Megan will be the main point of communication with the sponsor, facilitating meetings with the sponsor and group. As treasurer, Alec will maintain the team's travel and materials budget. Brandon will maintain the information repository and enforce our meetings stay according to agenda.

For specific design duties we have also assigned leaders to each main area. Megan will focus on the base fixture design, the DAQ system, and controlling the rotational speed. Alec is in charge of axial loading selection and implementation, along with determining the motor we will use. Brandon's duties include torque application and measurement, the wiring of the machine, and the design of the thermal chamber. Brandon will be the CAD lead, Megan will be manufacturing lead, and Alec will be our chief editor.

Appendix F contains our Gantt chart, outlining our schedule for the rest of the project. Our tasks and subtasks are listed for the remainder of the year along with their durations. Some of the main focuses include the programming of the data acquisition system and the manufacturing of custom parts. These are items that will take time and extra learning for our team, so we want to start early in preparation for iterations.

Manufacturing

Following the approval of our design after CDR we procured components and manufactured custom parts. Many of our parts were simple and required only a drill press and chop saw, while some took more time, strategy, specialized equipment and assistance.

Mechanical

Immediately after CDR we first began purchasing the raw material for machining. We purchased all of the aluminum plates from Metals Depot and all of the cylindrical stock from McMaster-Carr, as it was most cost effective. Because all of the senior project groups would be also utilizing the Cal Poly machine shops in spring quarter, we wanted to get a jump on starting and being ahead of schedule should we have any trouble getting on machines, mistakes or alignment issues.

We initially did the plates with easier features that only required a drill press and worked our way to the cylindrical parts, which required a lathe. For mating parts where alignment is crucial we match drilled. To cut parts to length we used a chop saw and a horizontal band saw. The horizontal band saw worked great for our cylindrical parts, and gave a clean cut that we could face on a lathe for a nice surface finish. We did run into an issue cutting and drilling the test shaft with the chop saw and drill press, because we didn't realize the shaft is hardened steel. Our last piece to be manufactured was the thermal chamber. Initially we didn't know how we would saw the ply wood to achieve the large circles for the mating ducts, but were able to use hole saws which left a nice clean and perfect circular cut.

Most of the manufacturing process went seamlessly, but the few challenges we faced are documented in the following section. The cylindrical part over the screw jack, the cylindrical part on top of the assembly that the drill grasps on to, and all of the plates with holes larger than an inch were machined by a friend at Dow-Key Microwave on CNC mills. These parts required more equipment and machining experience than we have access to.

Issues Encountered

There were several issue we ran into during our manufacturing process. The main problems we discovered were related to the following parts:

1. The provided test section shaft
2. The load cell
3. General hole alignment difficulties on the drill press
4. Equipment and skill restrictions

1 - The provided test section shaft

2 - The Load Cell

There were two issues that we ran into regarding the load cell. The first one related to the bolts that center the load cell on the assembly. The only 5/8-18 bolts we were able to locate were grade 8 bolts that had large bolt heads. This was a problem because the bolt heads stuck out on either side, which got in the way of the torque cell and the screw jack. In order to solve this, we used an angle grinder to cut the bolt heads off, and then epoxied one of the bolts into the plate the load cell connects to. The second issue was that the bolts that are on the load cell were slightly taller than the smaller active end of the load cell. Our first option to solve this was to create a spacer out of some scrap aluminum. We had trouble getting the threaded hole completely perpendicular to the spacer, and it ultimately failed. We then realized that using a few washers would accomplish the same goal, and ultimately went with using two 5/8 washers.

3- General hole alignment difficulties on the drill press

One general problem we had to overcome was aligning holes using the drill press. To alleviate this problem, we match drilled as many holes as possible to their counterparts to ensure proper alignment. For several of the mating plates where hole alignment is paramount, match drilling was performed. The plates that mate with the alignment pins, and the two side plates on the load loop that mate to the top and bottom plates of the loop, are match drilled to ensure successful assembly. As indicated on each side plate, both have a specific side and direction. These two plates were more difficult in particular because of the orientation at 90° from the mating top and bottom plates. Mid-machining we adjusted by drilling and tapping two of the three holes and bolting the plates together before drilling the third hole. This ensured a successful alignment.

4- Equipment restrictions

Two of the cylindrical parts we machined on a lathe in our machine shop, while the large cylindrical part over the screw jack, and the part the drill clamps on to, were outsourced to be made at a machine shop with CNC mills. The Cal Poly shops do not have CNC machines accessible to students unfamiliar with programming and running the machines. Also, all holes greater than an inch were machined off site because of drill bit size restrictions available to us on campus.

Electrical

In order to measure the reaction torque, axial load, temperature, and speed, we needed to figure out how to connect our sensors to a data acquisition system as well as power them all up. We decided to accomplish this by using a power supply that would output enough voltage for all the sensors and amplifiers to operate properly. This one power supply was directed to each of the sensors via terminal blocks. Originally, we purchased a 12V power supply, but when we were given the torque and load cell amplifiers required at least 14 volts. We looked into our power supply and found that it can be adjusted to output 14 volts. All of our terminal blocks, amplifiers, and power supply had DIN rail mounting, which made it easy to keep it compact.

One thing we realized was the complexity of getting a thermocouple reading on the data acquisition system we had, so we made the decision not to capture temperature on the DAQ. We used three channels of our DAQ, Torque, Load, and speed.

In order to ensure safety due to our power supply having exposed leads, we needed to add an electrical box. This box encapsulated all of the supplies on our DIN rail, with punch outs to connect our wires to the sensors. The electrical box was mounted to a side plate on the structure, so that all our electrical components are connected to our main structure.

Software

Software was created using LabVIEW to read outputs of the sensors, namely the optical speed sensor, reaction-torque sensor, and the load cell. The temperature will be measured by a type-E thermocouple, which is not currently integrated into the data acquisition system (DAS) or corresponding software. After consulting with Cal Poly professors, the integration of the thermocouple with the DAS was determined to be a complicated task. As the temperature will be at steady state during the testing, thousands of temperature measurements during the test are unnecessary. It should be acceptable to record one constant temperature value for the entire test run. The software was written using LabVIEW 2015 in conjunction with NI-DAQmx. NI-DAQmx is a driver provided by National Instruments (NI) with the purchase of the USB-6000 DAQ. NI-DAQmx allows for the collection of data from the DAS to a LabVIEW block diagram code.

The software implements a DAQmx task to acquire data from the specified three channels for the optical speed sensor, torque sensor, and load cell. The software currently defines the load cell as channel 0, the torque sensor as channel 1, and the speed sensor as channel 2 in the DAQ. The channel corresponding to each sensor can be modified by editing the DAQmx task code in the LabVIEW block diagram interface. The LabVIEW code outputs 1-D waveform data for the multi-channel task. The data is plotted on the front panel for the multi-channel task. The data is plotted on a waveform graph on separate plots. From here, the data can be exported to a number of different formats, including text files and excel files. The software is currently set-up to take data for 10 seconds at a 1 kHz frequency. This can be modified in the DAQmx task code. The code currently only outputs the electrical signal, or voltage over time, produced by the sensors. This allows us to check for noise in the system and for inconsistencies. The scaling/calibration for the sensors has not been applied yet, but it can be applied in the DAQmx task by adding a scaling factor/polynomial to the analog voltage task for the load/torque sensors.

Testing

Planned Test Procedures

Below is a description of all the tests that we planned on running for this project.

5000 lb Load Test:

The purpose of this test is to demonstrate the ability of the structure to support a 5000 lb axial load without yielding or breaking.

Procedure:

1. Assemble and set-up the FTM per Assembly Instructions and Set-Up Instructions provided in the Operator's Manual.
2. Turn on power supply by turning on the power strip while it is plugged into an outlet.
3. Plug computer into DAQ and open Labview code to give real-time read outs for the sensor values, namely the load cell force reading.
4. Zero the load cell for the static weight of the system that is resting on the load cell.
5. Torque the screw Jack handle by hand slowly. Step up to a 5000 lb load in 1000 lb intervals and pause for about 60 seconds per interval.
6. Verify that the system is functional by inspection after each interval.
7. Disassemble the device per instructions in the Operator's Manual and inspect critical load carrying parts and fasteners for damage.
8. Publish results.

Torque Cell + Speed Sensor Repeatability Tests

The purpose of the torque cell and speed sensor repeatability tests is to demonstrate the ability of the FTM to take statistically repeatable data within a reasonable confidence interval. The data for speed and torque measurements can be taken simultaneously in the same run using the DAQ with Labview.

1. Set up and assemble the FTM as shown in the Operator's Manual.
2. Review Operation Instructions provided in the Operator's Manual before running the test.
3. Grease the two thrust bearings and the skew roller bearing in preparation for the test. The test section will not be fully submerged in oil for this test. Thus, it is important to run these tests at a relatively low speed/axial load to prevent damage to the system.
4. Load the test section with a small arbitrary load value (50 or 100 lb). Record the value and verify the value is static during the course of the test via the DAQ.
5. Run the system 5 times (see operation instructions) to obtain torque, speed, and load values for each run. Save the data.
6. Run the system 5 more times at two additional arbitrary load values and the motor power. Record the torque, speed, and load data.
7. Run the system 5 times at two different motor power values (clamp the drill trigger harder or softer) at the first arbitrary load value.
8. Perform statistical uncertainty analysis on the acquired data to determine the standard deviation and to verify repeatability.

Two Man Lift/Weight

The purpose of the test is to verify that the FTM can be carried and moved by a team of two persons reasonably.

1. Assembly FTM per Operator's Manual instructions.
2. Carry the structure a distance of 50 feet between two people with negligible fatigue and at a walking pace.
3. Weigh structure and include weight in product specifications.
4. Measure final dimensions of the structure to include in the product specifications.

Friction Calibration Test

The purpose of the friction calibration test is to determine the quantity of friction introduced to the torque sensor by the two thrust bearings in the system, the two bearings in the system, and any other friction introduced by non-perfect alignment of the system or other factors. This will allow us to accurately measure the friction of only the skew roller by calibrating the other influences out.

1. Assemble and set-up the FTM per the Operator's Manual.
2. Replace the skew roller bearing with a thrust bearing. Grease the bearings. Do not run the system dry.
3. Run the system 5 times (per operating instructions) at the same arbitrary load and motor power performed in the torque cell + speed sensor reliability tests. Save the data for analysis.
4. Run the system 5 times at the two other arbitrary loads performed in the performed in the reliability test.
5. Perform analysis comparing the thrust bearing and skew roller results to determine the torque introduced to the sensor by the system and the bearings. Predict the friction created by the skew roller.
6. Publish methodology and results and provide an estimate of the total system friction (it may be a function of load and speed) and the thrust bearing friction (will be a function of load and speed).

Tests Completed

The first test that we ran was the two man lift test. Alec and Brandon were successfully able to carry the assembly with relative ease for the 50 feet with no fatigue. The structure was not too heavy for a single person, although the dimensions make it difficult for a single person to carry. Overall, the structure is reasonably transported and operated by two persons.

Next, testing of the mechanical system was performed. After the system was assembled, we verified the mechanical operation. The system was run at a minimal applied load, which was applied by torqueing the screw jack by hand without a handle. The skew roller bearings and thrust bearings were coated with a general purpose grease to lubricate the system and to prevent damage while running. Once the set-up for the test was completed, the motor was run by pushing on the trigger, thus driving the system. We started by driving the system at low speed and slowly ramped up to 200 RPM. The system was operable up to and past 200 RPM with the small applied load. There were no discernable problems with the system and everything appeared to be aligned and rotating properly. The system has yet to be run at a significant load (above 50 lbs) so it remains to be seen whether any problems will arise when driving the system at a high load.

After verifying the functionality of the mechanical system, the system was tested with the sensors wired and connected to the DAQ to verify the ability of the device to read measurements for torque, speed, and load. The Labview program created is able to read the electrical signals created by each sensors, as read from the USB-600 DAQ. The calibration of the sensors, converting the electrical signal, measured as volts over the time interval, to the respective measured outputs (load, torque, and speed) has not been completed. For the initial data acquisition test runs, there was an excess amount of noise in the signals and there appeared to be interference between the electrical signals in different channels of the DAQ. The figure below shows initial test data obtain from the sensors including the noise and interference.

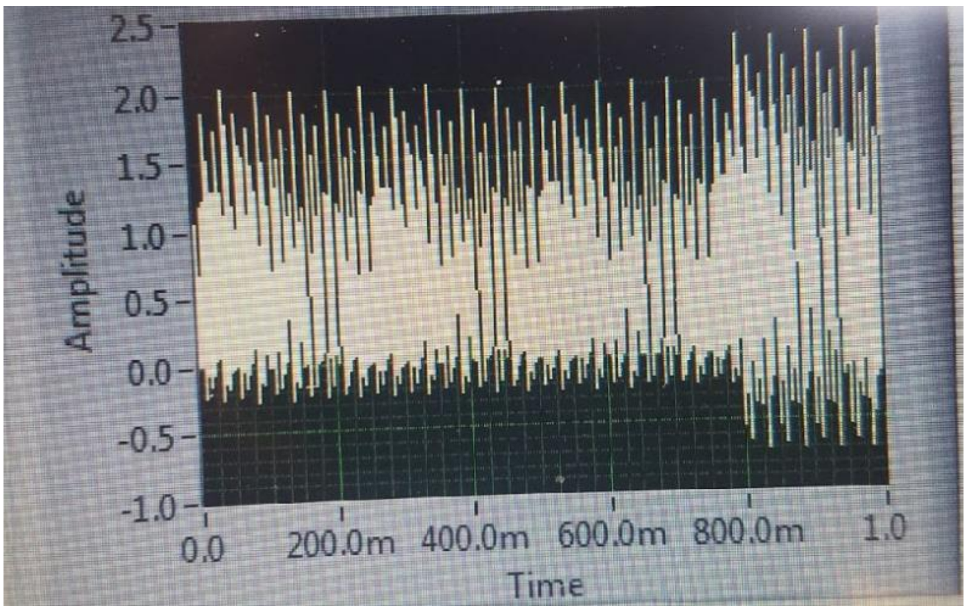
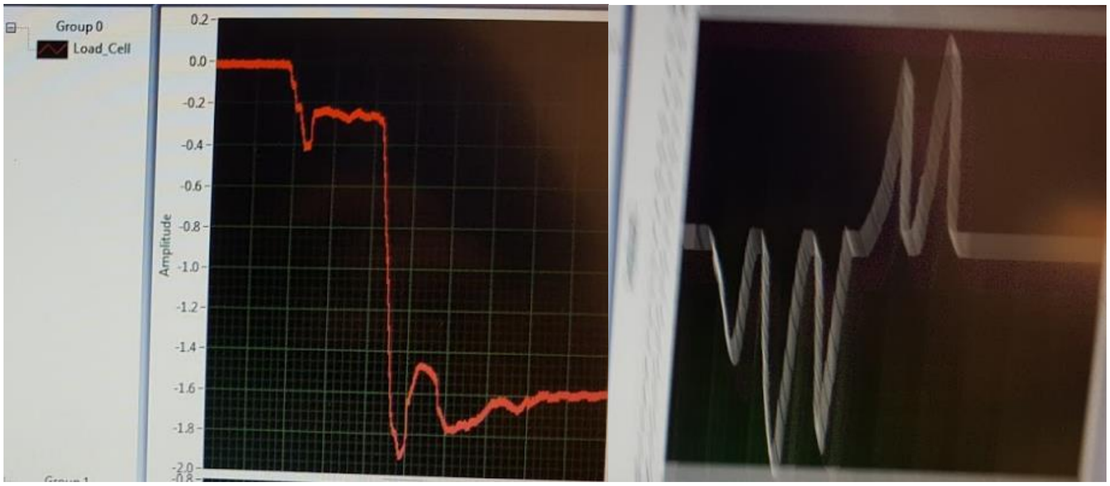


Figure 58. Torque Data Measurement with Excess Noise

After examining the electrical and DAQ systems and consulting with personnel at Parker, we determined a fundamental issue with our data collection set-up. A majority of the wires were shielded to prevent corruption of the electric signals, but the data acquisition system was not grounded. We rewired the system and included the ground for the DAQ to receive better results. We tested each sensor individually by applying a torque, load, or speed independently of one another. The system was then tested with a test run and the data appeared to be significantly smoother and free of the random noise we saw before. The figure below shows the less-noisy data for the torque and load sensors.



Load Cell Measurement Torque Measurement

Figure 59. Load and Torque Signals due to Applied Loads

We were unable to complete the rest of the planned testing, including the 5000 lb load test, sensor repeatability test, and the friction calibration test. In order to perform these tests, it is necessary to calibrate the sensors to receive accurate data outputs. These tests will be important in verifying the precision and accuracy of the instrumentation and results.

Improvements

There are a number of different things that can be done to improve the system that the team was unable to complete in time. For the mechanical system, there are a few things that can be done better. The interface between the load cell and the screw jack is not rotationally fixed, so the screw jack is capable of rotating within the load loop. Ideally, this would be fixed by threading the interface plate between the screw jack and the load cell. At the tested load values, the screw jack does not rotate due to the friction between the plate and the load cell. However, if the torque created by the system becomes large than the friction between the two surface, the screw jack may slip and rotate. The figure below shows the screw jack/load cell interface plate.

Additional mechanical improvements include insulation of the gaps in the thermal chamber, a drain mechanism to more easily remove and refill the oil chamber, and the inclusion of a handle to properly load the system. The thermal chamber has large gaps in it so that it is able to slide on to the system without interfering with the torque measurement. These gaps may be insulted with a thermal tape. Additionally, the thermal chamber is simply resting on the system and is not fixed in place, so it is capable of sliding around. This can be fixed by locking the chamber to the top of the load loop or by making it smaller so it fits securely around the load loop. The oil chamber is currently difficult to fill and refill. A mechanism to facilitate the oil removal and filling would improve the system. Additionally, the oil chamber has not been verified to be leak-free with oil in it. A handle for the screw jack will be provided by Parker in order to properly load the system. The device is currently difficult to disassemble and reassemble, which is inherent in the design. The assembly may be facilitated with stands to more easily hold the load loop while installing it.

Improvements to the software and electrical system may be necessary to obtain good data. The software needs to be calibrated to output the torque, load, and speed values in the respective desired units, rather than in volts. An additional improvement would be to incorporate the thermocouple into the data acquisition system. Additionally shielding may be provided for the electrical system to prevent noise and the mechanical system may need to be grounded to get better measurements.

The data reliability and calibration tests on the device remain to be completed. Data should be taken to verify the statistical and measurement uncertainty created by the device. Additionally, a test should be run to calibrate the friction created by the system so only the torque measured by the skew roller bearings is acquired. Verification of the ability of the system to operate under the maximum applied load still needs to be completed.

Statement of Disclaimer

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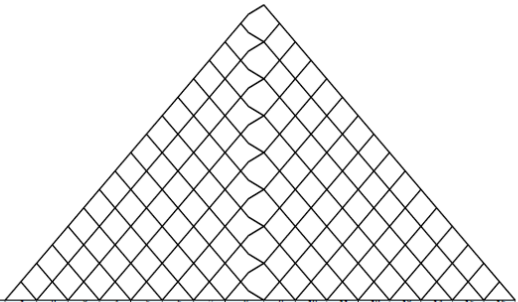
APPENDIX A – Quality Function Deployment House of Quality

QFD: House of Quality
Project:
Revision:
Date:

Correlations	
Positive	+
Negative	–
No Correlation	

Relationships	
Strong	●
Moderate	○
Weak	▽

Direction of Improvement	
Maximize	▲
Target	◇
Minimize	▼

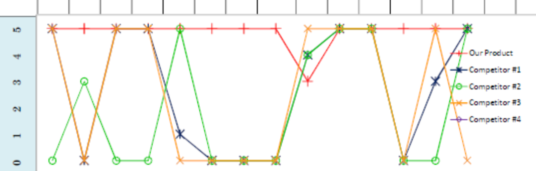


Row #	Weight Chart	Relative Weight	Eric Polouch	Technician			Maximum Relationship	WHAT: Customer Requirements (explicit & implicit) / HOW: Engineering Specifications	1 Size	2 Axial Load	3 Generate and measure a torque	4 Rotational Speed at Max Torque	5 Data Acquisition System	6 submerge the skew roller section under oil	7 Thermal Chamber	8 Thrust Washer (if applicable)	9 Cost	10 Weight	11 Power	12 flange sample and scew roller bearings	13 Geometry	14 Stability	15	16	Our Product	DC motor dynamometer	Ametek FT1 Series Friction Tester Machine	Rotational Torque Sensor
1		6%	2	9			9	Bench top or vertical floor standing configuration	○											▽	●	●			5	5	5	0
2		6%	3	9			9	24 x 24 inch base maximum	●									○		▽	○				5	5	0	0
3		7%	9	4			9	The fixture shall be able to generate an axial load across the skew rollers of 5000 lb.		●							○								5	0	4	0
4		7%	9	4			9	The fixture shall be able to generate a torque of 600 in-lb per bearing			●						○								5	5	0	5
5		7%	9	4			9	The fixture shall be able to generate the torque in item 5 above at a speed of 0 to 200 rpm				●					▽								5	0	0	0
6		6%	4	8			9	A data acquisition system shall be provided to capture the sensor data and output this data to Text, ASCI or other digital format for post processing					●				▽								5	0	0	0
7		7%	9	5			9	The Fixture shall be designed to be able to submerge the skew roller section under oil						●											5	0	0	0
8		5%	5	5			9	A box shall be provided around the skew assembly to allow the introduction of hot and / or cold air							●		▽								5	0	0	0
9		3%	5	1			9	Thrust Washer (if applicable) shall be High Carbon Steel, through hardened to HRc 58 min. Ground to meet finish, flatness and parallelism								●									5	0	0	0
10		1%	1	1			9	Cost goal < $5000									●								3	5	3	5
11		8%	7	9			9	Fixture must fit through standard 30 inch doorway	●											▽	●				5	0	0	0
12		7%	4	9			9	Fixture must be capable of 2-man carry (not including data acquisition system)	○									●			●				5	5	2	0
13		8%	8	7			9	Fixture must be able to run on standard 115 Vac 20 amp max wall power											●						5	5	0	5
14		11%	10	10			9	Must use provided flange sample piece and scew roller bearings												●					5	0	0	0
15		11%	10	10			9	Must collect accurate data		●	●	●	●	●	●							●			5	5	5	5
16		0%																										

Column #	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
HOW MUCH: Target	24" X 24" X 80"	5000 Lb	600 in-lb per bearing	0 to 200 RPM	1Khz minimum sample rate at 4 channels	Bearings 100% submerged	Temperature between -65 deg F to 160 deg F	through hardened to HRc 58 min. Ground to meet finish, flatness and parallelism	<$5000	150lb max	standard 115 Vac 20 amp max wall power	are the flange sample and scew bearings in the design	places to grip	Is tester stable during testing or is there the possibility of movement		
Max Relationship	9	9	9	9	9	9	9	9	9	9	9	9	9	9		
Technical Importance Rating	170.5	156.3	156.3	156.3	151.6	161.1	142.1	28.4211	68.95	80.53	71.05	115.3	208.4	146.8		
Relative Weight	9%	9%	9%	9%	8%	9%	8%	2%	4%	4%	4%	6%	11%	8%		
Weight Chart																

Current Product Assessment - Engineering Specifications

	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Our Product	5	5	5	5	5	5	5	5	3	5	5	5	5	5		
Competitor #1: *DC Motor Dynamometer*	5	0	5	5	1	0	0	0	4	5	5	0	3	5		
Competitor #2: *Ametek FT1 Series Friction Tester Machine*	0	3	0	0	5	0	0	0	4	5	5	0	0	5		
Competitor #3: *Rotational Torque Sensor*	5	0	5	5	0	0	0	0	5	5	5	0	5	0		



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

Template Revision: 0.
Christopher Hettles

APPENDIX B – Engineering Specifications

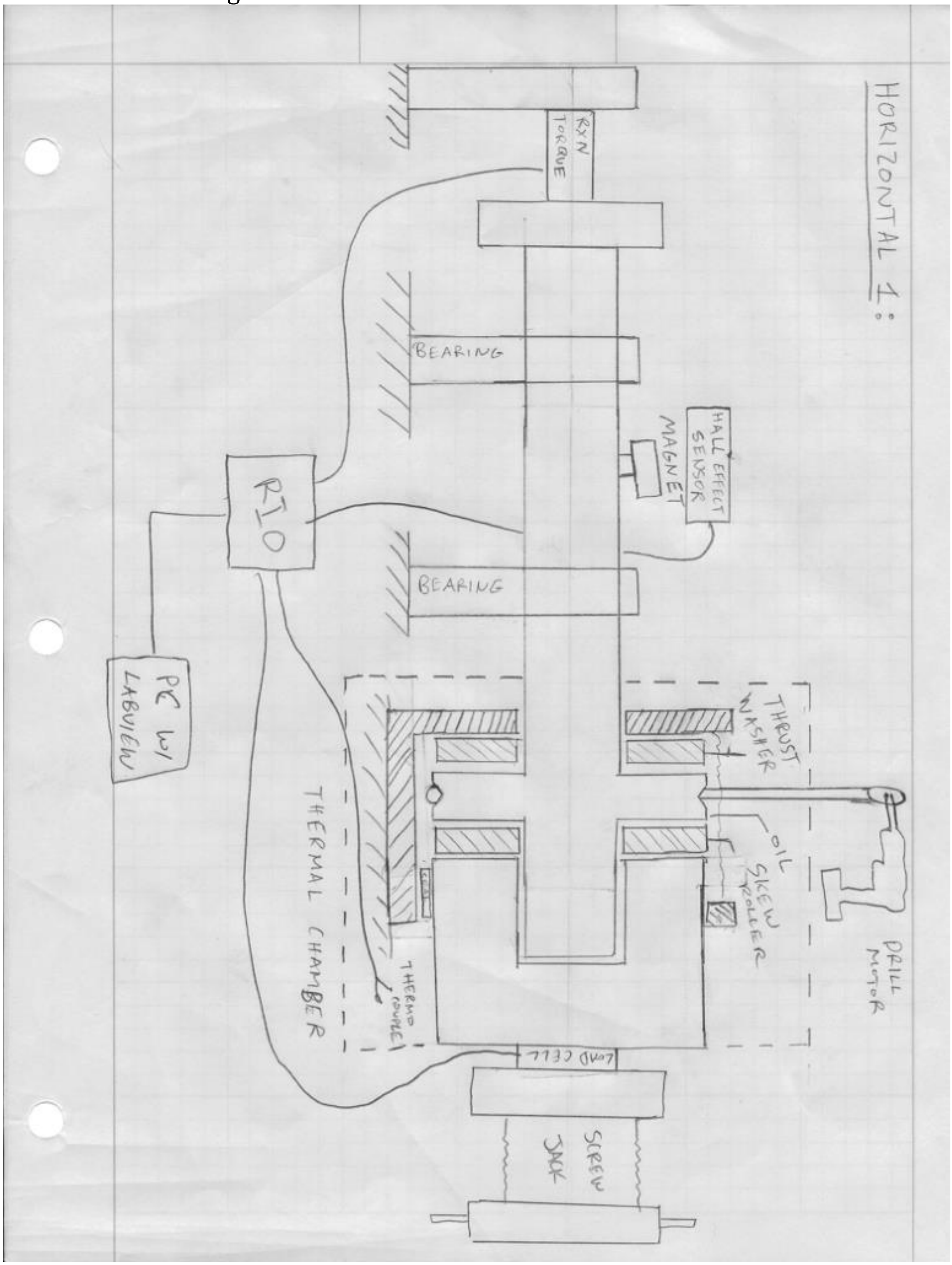
Table 1: Engineering Specifications Table

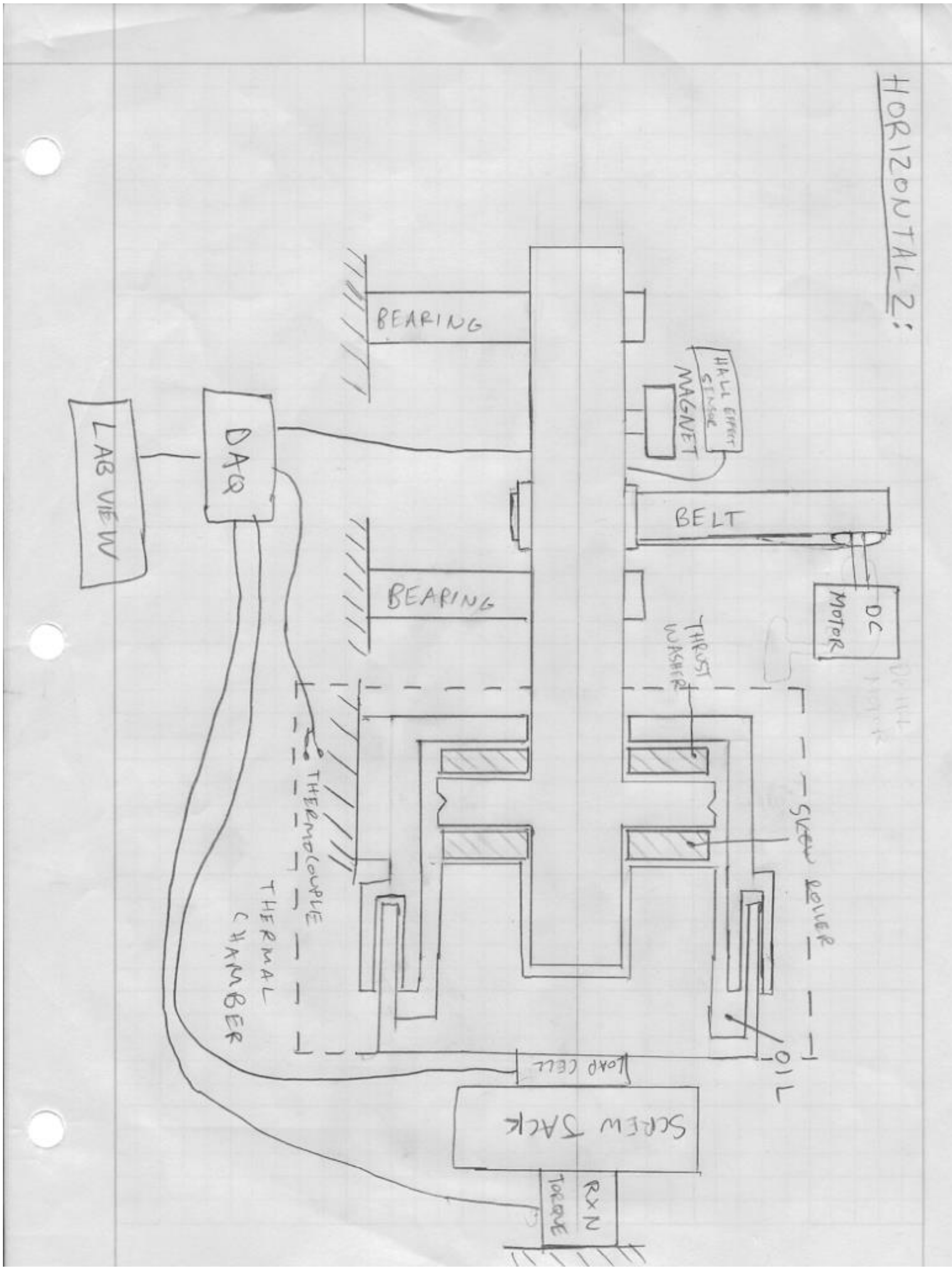
Spec. #	**Parameter Description**	**Requirement or Target (units)**	**Tolerance**	**Risk**	**Compliance**
1	Weight	150lb	Max	L	A, T
2	Size	24in x 24in x 80in	Max	L	A, T
3	Geometry	Places to grip		L	A
4	Axial Load	5000lb	Min	M	A, T
5	Apply/Measure Torque	600 in.lb	Min	M	A, T
6	Rotational Speed	0 to 200 rpm		M	A, T
7	DAQ	1KHz sample rate. 4 channels	Min	H	A, T
8	Thermal Chamber	-65°F to 160°F		M	A, T
9	Power	115 VAC 20 Amp	Max	M	A, T
10	Cost	$5000	Max	M	A
11	Submersible	Bearings 100% submerged		M	A, T
12	Flange sample	Parker unit used		L	I
13	Skew roller bearings	Parker bearings used		L	I
14	Stability	Stability during test		L	A, T

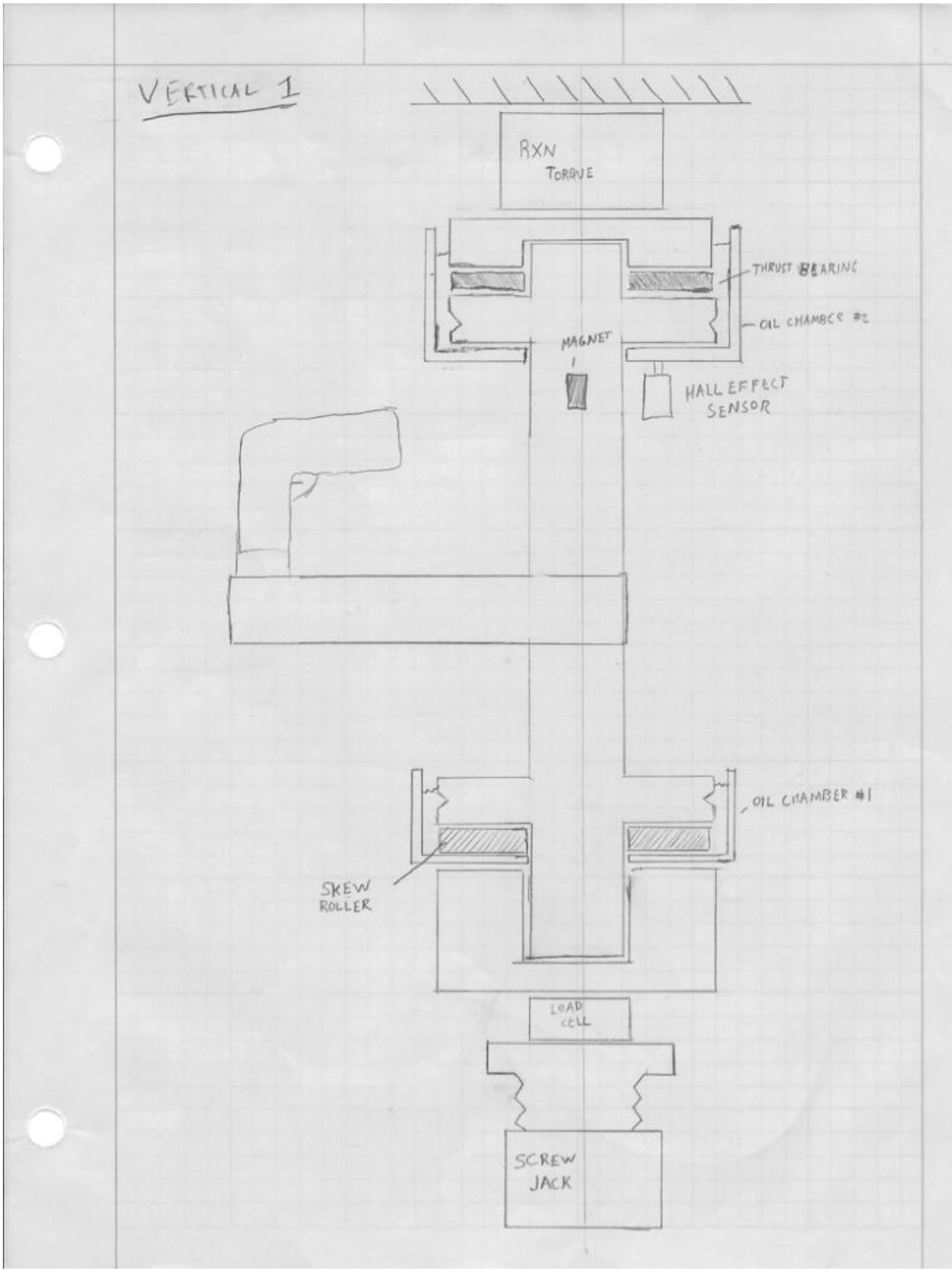
High (H)
Moderate (M)
Low(L)

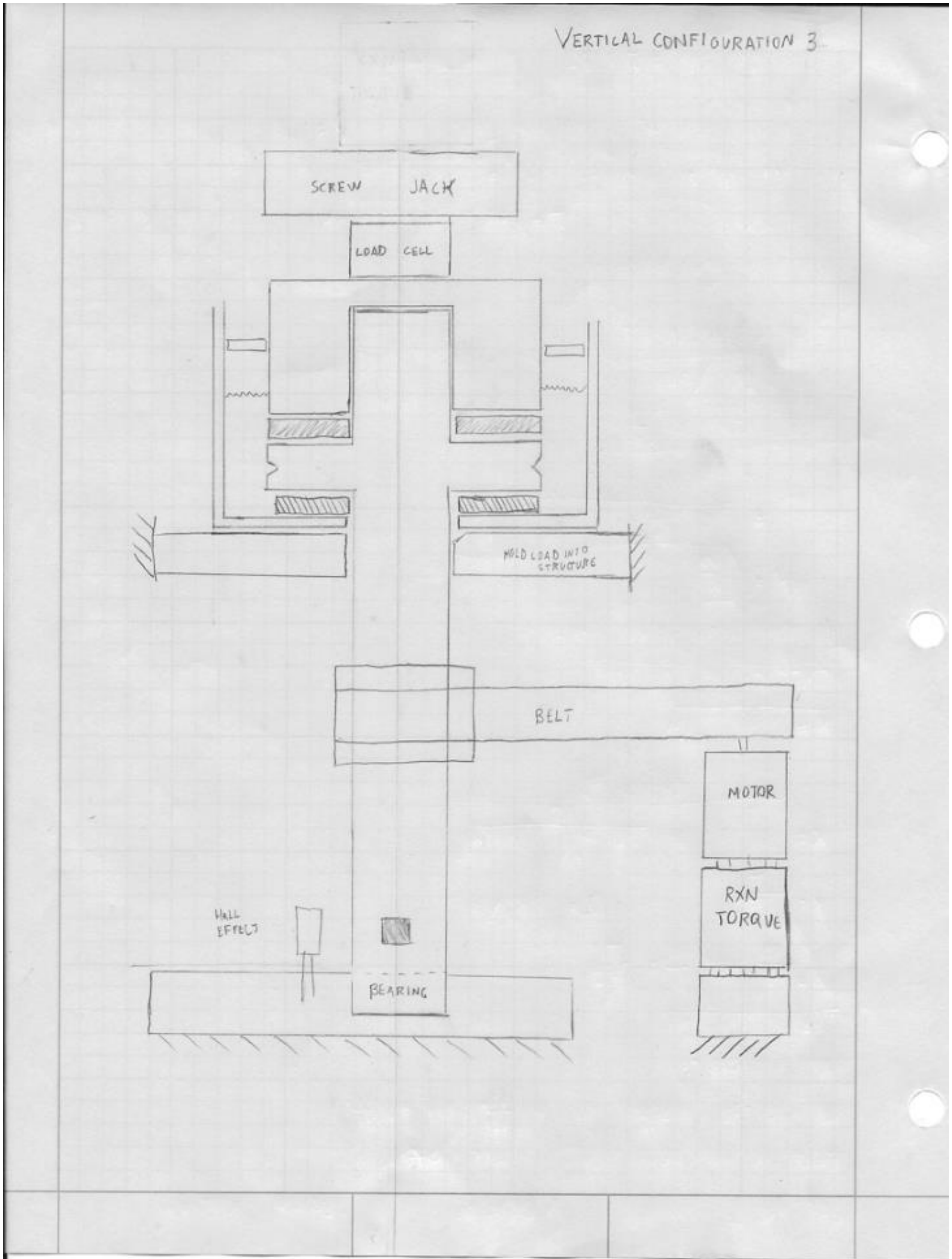
Analysis (A)
Test (T)
Similarity to Existing Designs (S)
Inspection (I)

APPENDIX C – Configurations




HORIZONTAL 2:
BEARING
HALL EFFECT SENSOR
MAGNET
BELT
DC MOTOR
BEARING
THRUST WASHER
LAB VIEW
DAQ
THERMOCOUPLE
THERMAL CHAMBER
SKEW ROLLER
OIL
LOAD CELL
SCREW JACK
RXN TORQUE


VERTICAL 1
RXN TORQUE
THRUST BEARING
OIL CHAMBER #2
MAGNET
HALL EFFECT SENSOR
OIL CHAMBER #1
SKEW ROLLER
LOAD CELL
SCREW JACK


VERTICAL CONFIGURATION 3
SCREW JACK
LOAD CELL
HOLD LOAD INTO STRUCTURE
BELT
MOTOR
RXN TORQUE
HALL EFFECT
BEARING

APPENDIX D – Configuration Decision Matrix

DECISION MATRIX							
		24" X 24" Base Max	48" Height Max	5000 Lb Generation	Abililty to measure reaction torque 600 in-lb per bearing	0 to 200 RPM	Bearings 100% submerged
	Weight	10.79%	10.79%	9.89%	11.38%	9.89%	10.19%
Horizontal 1	Rating	6	9	8	6	8	5
	Wgt Rtg	0.647	0.971	0.791	0.683	0.791	0.509
Horizontal 2	Rating	6	9	8	9	8	5
	Wgt Rtg	0.647	0.971	0.791	1.025	0.791	0.509
Vertical 1	Rating	9	7	8	6	8	7
	Wgt Rtg	0.971	0.755	0.791	0.683	0.791	0.713
Vertical 2	Rating	9	7	8	8	8	7
	Wgt Rtg	0.971	0.755	0.791	0.911	0.791	0.713
Vertical 3	Rating	9	7	8	8	8	7
	Wgt Rtg	0.971	0.755	0.791	0.911	0.791	0.713

		Cost <$5000	150lb max Total Weight	Manufacturing time	Ease of Use	Stability of Tester	Total
	Weight	4.36%	5.09%	6.32%	12.02%	9.29%	100.00%
Horizontal 1	Rating	6	7	6.5	5	8	-
	Wgt Rtg	0.262	0.357	0.411	0.601	0.743	6.77
Horizontal 2	Rating	6	7	6.5	6	8	-
	Wgt Rtg	0.262	0.357	0.411	0.721	0.743	**7.23**
Vertical 1	Rating	5	6	6.5	7	6	
	Wgt Rtg	0.218	0.306	0.411	0.841	0.557	7.04
Vertical 2	Rating	5	6	7	8	6	-
	Wgt Rtg	0.218	0.306	0.443	0.961	0.557	**7.42**
Vertical 3	Rating	5	6	7	8	5	-
	Wgt Rtg	0.218	0.306	0.443	0.961	0.464	7.32

APPENDIX E – Component Pugh Matrices

Horizontal vs. Vertical Configuration Pugh Matrix

Criteria	Vertical Configuration	Horizontal Configuration
Bench top or vertical floor standing configuration	D	S
24 x 24 inch base maximum		-
The fixture shall be able to generate an axial load across the skew rollers of 5000 lb.	A	S
The fixture shall be able to generate a torque of 600 in-lb per bearing		S
The fixture shall be able to generate the torque in item 5 above at a speed of 0 to 200 rpm	T	S
A data acquisition system shall be provided to capture the sensor data and output this data to Text, ASCI or other digital format for post processing		S
The Fixture shall be designed to be able to submerge the skew roller section under oil	U	-
A box shall be provided around the skew assembly to allow the introduction of hot and / or cold air		S
Thrust Washer (if applicable) shall be High Carbon Steel, through hardened to HRc 58 min. Ground to meet finish, flatness and parallelism	M	S
Cost goal <$5000		S
Fixture must fit through standard 30 inch doorway		+
Fixture must be capable of 2-man carry (not including data acquisition system)		S
Fixture must be able to run on standard 115 Vac 20 amp max wall power		S
Must use provided flange sample piece and scew roller bearings		S
Must collect accurate data		S
∑+	-	1
∑-	-	2
∑S	-	12

Axial Load Pugh Matrix

Concept	Screw jack	Hydraulic	Pneumatic	Electromechanical
Criteria	1	2	3	4
Safety	+	S	D	S

Ease of use	-	-	A	-
5000lbs	S	S	T	S
size	+	S	U	S
cost	+	+	M	+
Σ-	1	1	0	1
Σ+	3	1	0	1
ΣS	1	3	0	3

Drive Assembly Pugh Matrix

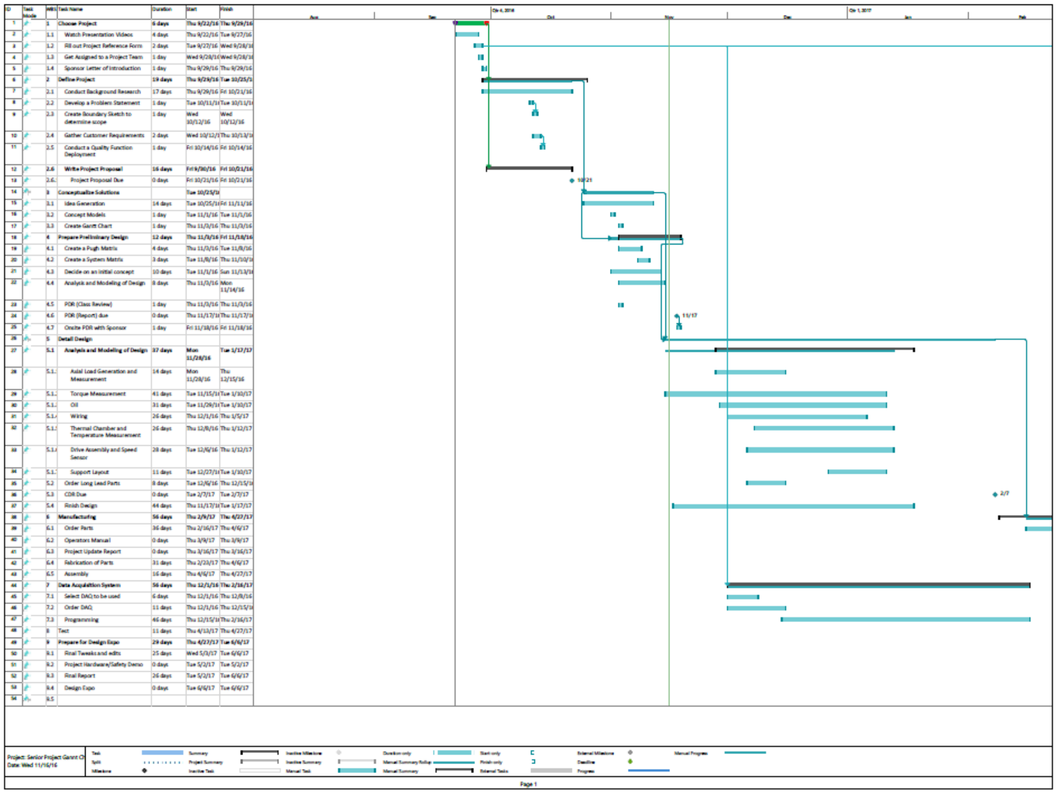
Criteria	Belt Driven	Directly Gear Driven	DC Motor Mounted to Flange
Bench top or vertical floor standing configuration	S		S
24 x 24 inch base maximum	S	D	+
The fixture shall be able to generate an axial load across the skew rollers of 5000 lb.	S		S
The fixture shall be able to generate a torque of 600 in-lb per bearing	+	A	-
The fixture shall be able to generate the torque in item 5 above at a speed of 0 to 200 rpm	S		-
A data acquisition system shall be provided to capture the sensor data and output this data to Text, ASCI or other digital format for post processing	S	T	S
The Fixture shall be designed to be able to submerge the skew roller section under oil	S		S
A box shall be provided around the skew assembly to allow the introduction of hot and / or cold air	S	U	+
Thrust Washer (if applicable) shall be High Carbon Steel, through hardened to HRc 58 min. Ground to meet finish, flatness and parallelism	S		S
Cost goal <$5000	S	M	+
Fixture must fit through standard 30 inch doorway	-		+
Fixture must be capable of 2-man carry (not including data acquisition system)	S		S
Fixture must be able to run on standard 115 Vac 20 amp max wall power	S		S
Must use provided flange sample piece and scew roller bearings	S		S
Must collect accurate data	S		S
∑+	1	-	4
∑-	1	-	2
∑S	13	-	9

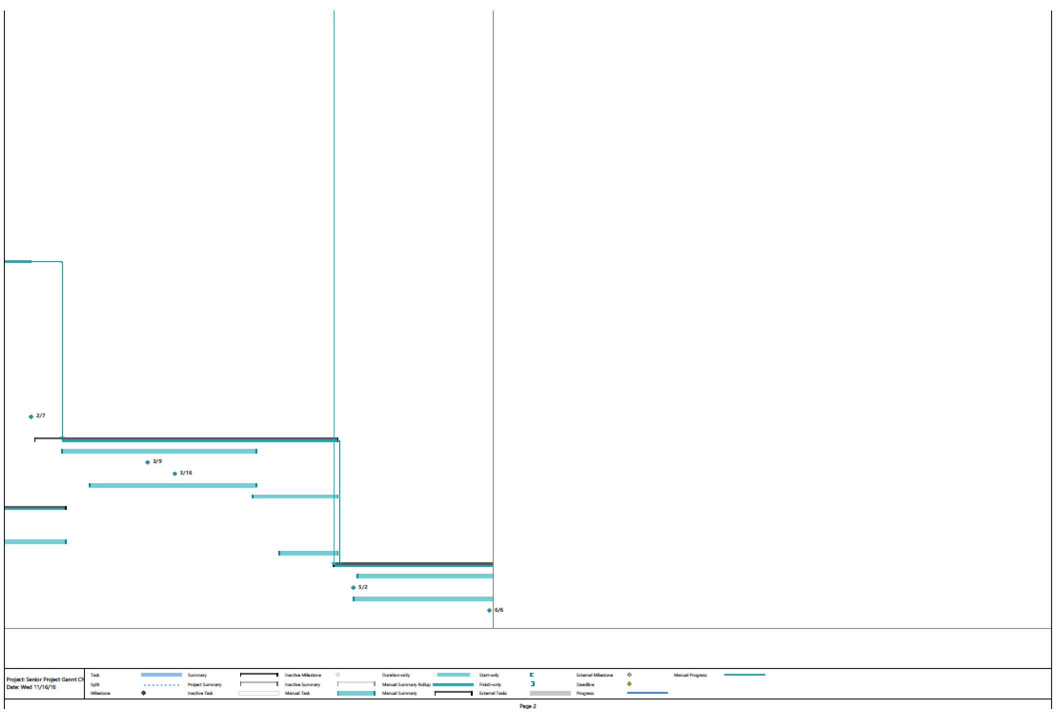
Data Acquisition System

Concept	**NIcompactRIO**	**PolyDAQ**	**NIcompactDAQ**
Criteria	1	2	3
Customizable	+	S	D

Ease of programming	S	-	A
Download data CSV	S	S	T
Recording frequency	+	+	U
Record 4 channels	S	S	M
size	+	S	
Σ-	0	1	0
Σ+	3	1	0
ΣS	3	4	0

APPENDIX F - Gantt Chart





APPENDIX G - Spec Sheets

SHARE IT

PRODUCT DESCRIPTION

Description

This drill is a 1/2 in. pistol grip heavy duty high amperage pro drill. With an 8 Amp motor and 1/2 in. keyed chuck. Extra long 8 ft rubber cord with 3-prong grounded plug. Variable speed, soft grip trigger, with reversing switch for greater control. Two finger trigger design and lock-on button for ease of operation.

Includes

- Auxiliary Handle -- 2 602 025 094
- Chuck Key -- 2 610 348 635

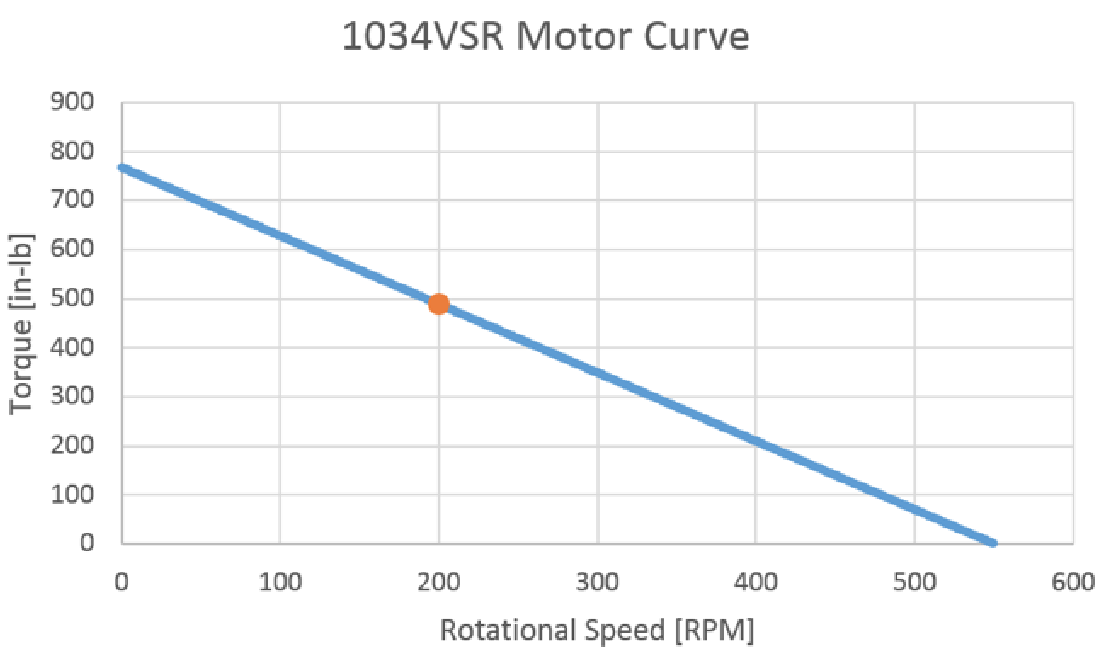
TECHNICAL DATA

Specifications

Attributes	
Amperage	8.0
Chuck Design	Keyed
Chuck Size	1/2 in
Length	11.2"
Max. Hole Diameter in Steel-Twist Drill Bit	1/2"
No Load RPM	0-550
Rating	120 V
Size	Spindle - 1/2" - 20"
Torque (in. lbs.)	767
Voltage	120V
Weight	5.3lb
Width	2.5"
Includes	Auxiliary Handle -- 2 602 025 094, Chuck Key -- 2 610 348 635

Benefits

- 8.0 Amp all ball bearing motor, 0 to 550 RPM
- Heavy duty, keyed chuck
- Variable speed, soft grip trigger, with reversing switch — for greater control
- Two finger trigger design and lock-on button — for ease of operation
- Ergonomic handle with soft grip — for increased comfort
- Precision cut steel gears
- Magnesium front housing — for increased durability
- Extra long 8 Ft. rubber cord with 3-prong grounded plug
- 360° auxiliary handle with optional depth gauge



National Instruments USB 6000 DAQ Module

DEVICE SPECIFICATIONS

NI USB-6000

Low-Cost Data Acquisition: 10 kS/s, 8 AI, 4 DIO Bus-Powered USB

The following specifications are typical at 25 °C, unless otherwise noted. For more information about the NI USB-6000, refer to the *NI USB-6000 User Guide* available from *ni.com/manuals*.

Analog Input

Number of analog inputs	8, single-ended
Input resolution	12 bits
Maximum sample rate (aggregate), system-dependent	10 kS/s
Converter type	Successive approximation
AI FIFO	2,047 samples
Timing resolution	125 ns (8 MHz timebase)
Timing accuracy	100 ppm of actual sample rate
Input range	±10 V
Working voltage	±10 V
Input impedance	>1 MΩ
Overvoltage protection	±30 V
Trigger sources	Software, PFI 1
System noise[1]	10 mVrms
Absolute accuracy at full scale, single-ended	
Typical at 25 °C	26 mV
Maximum over temperature	135 mV

Note Absolute accuracy at full scale on the analog input channels is determined using the following assumptions: *Number of readings* = 100, *Coverage factor* = 3 σ.

[1] System noise measured at maximum sample rate.

Digital I/O

Number of digital I/O	4
Function	
P0.0/PFI 0	Static digital I/O or counter source
P0.1/PFI 1	Static digital I/O or AI Start Trigger
P0.2	Static digital I/O
P0.3	Static digital I/O
Direction control	Each channel individually programmable as input or output
Output driver type	Each channel individually programmable as open collector or active drive
Absolute maximum voltage range	0 V to 5 V with respect to D GND
Pull-down resistor	47.5 kΩ to D GND
Power-on state	Input

Digital Input

Input voltage range (powered on)	0 V to 5 V
Input voltage range (powered off)	0 V to 3.3 V
Input voltage protection	±20 V, for up to 24 hours

⚠ **Caution** Do not leave a voltage above 3.3 V connected on any DIO line when the device is powered off. This may lead to long term reliability issues.

Minimum V_{IH}	2.4 V
Maximum V_{IL}	0.8 V
Maximum input leakage current	
At 3.3 V	0.8 mA
At 5 V	4.5 mA

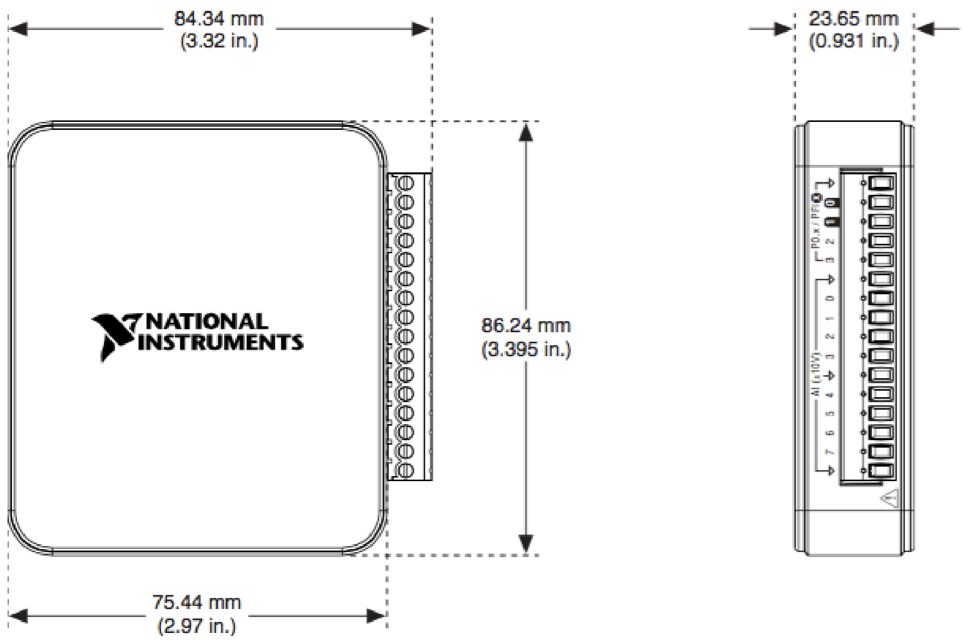
Digital Output (Active Drive)

Maximum V_{OL} (4 mA)	0.8 V
Maximum V_{OL} (1 mA)	0.2 V
Minimum V_{OH} (4 mA)	2.2 V
Minimum V_{OH} (1 mA)	2.9 V

Physical Characteristics

Dimensions	
Without screw terminal connector plug	75.44 mm × 86.24 mm × 23.65 mm (2.97 in. × 3.40 in. × 0.93 in.)
With screw terminal connector plug	84.34 mm × 86.24 mm × 23.65 mm (3.32 in. × 3.40 in. × 0.93 in.)

Figure 1. NI USB-6000 Dimensions



Weight	
Without screw terminal connector plug	73 g (2.58 oz)
With screw terminal connector plug	84 g (3 oz)
I/O connector	USB Micro-B receptacle, (1) 16-position screw terminal plug
Screw-terminal wiring	1.31 mm^2 to 0.08 mm^2 (16 AWG to 28 AWG)
Torque for screw terminals	0.22 to 0.25 N · m (2.0 to 2.2 lb. · in.)

If you need to clean the module, wipe it with a dry towel.

Safety Voltages

Connect only voltages that are within these limits.

Channel-to-GND	±30 V max, Measurement Category I

Measurement Category I is for measurements performed on circuits not directly connected to the electrical distribution system referred to as MAINS voltage. MAINS is a hazardous live electrical supply system that powers equipment. This category is for measurements of voltages from specially protected secondary circuits. Such voltage measurements include signal levels, special equipment, limited-energy parts of equipment, circuits powered by regulated low-voltage sources, and electronics.

Caution Do not use this module for connection to signals or for measurements within Measurement Categories II, III, or IV.

Environmental

Operating temperature (IEC 60068-2-1 and IEC 60068-2-2)	0 °C to 40 °C
Storage temperature (IEC 60068-2-1 and IEC 60068-2-2)	-40 °C to 85 °C
Operating humidity (IEC 60068-2-56)	5% to 90% RH, noncondensing
Storage humidity (IEC 60068-2-56)	5% to 95% RH, noncondensing
Pollution Degree (IEC 60664)	2
Maximum altitude	2,000 m

Indoor use only.

Safety

This product is designed to meet the requirements of the following electrical equipment safety standards for measurement, control, and laboratory use:

- IEC 61010-1, EN 61010-1
- UL 61010-1, CSA 61010-1

Note For UL and other safety certifications, refer to the product label or the *Online Product Certification* section.

Electromagnetic Compatibility

This product meets the requirements of the following EMC standards for electrical equipment for measurement, control, and laboratory use:

- EN 61326-1 (IEC 61326-1): Class A emissions; Basic immunity
- EN 55011 (CISPR 11): Group 1, Class A emissions
- EN 55022 (CISPR 22): Class A emissions
- EN 55024 (CISPR 24): Immunity
- AS/NZS CISPR 11: Group 1, Class A emissions
- AS/NZS CISPR 22: Class A emissions
- FCC 47 CFR Part 15B: Class A emissions
- ICES-001: Class A emissions

Note In the United States (per FCC 47 CFR), Class A equipment is intended for use in commercial, light-industrial, and heavy-industrial locations. In Europe, Canada, Australia and New Zealand (per CISPR 11) Class A equipment is intended for use only in heavy-industrial locations.

Note Group 1 equipment (per CISPR 11) is any industrial, scientific, or medical equipment that does not intentionally generate radio frequency energy for the treatment of material or inspection/analysis purposes.

Note For EMC declarations and certifications, and additional information, refer to the *Online Product Certification* section.

CE Compliance CE

This product meets the essential requirements of applicable European Directives, as follows:

- 2014/35/EU; Low-Voltage Directive (safety)
- 2014/30/EU; Electromagnetic Compatibility Directive (EMC)

Online Product Certification

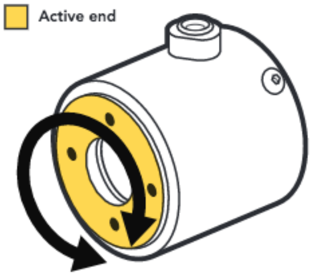
Refer to the product Declaration of Conformity (DoC) for additional regulatory compliance information. To obtain product certifications and the DoC for this product, visit *ni.com/certification*, search by model number or product line, and click the appropriate link in the Certification column.

Futek Model TF400 Torque Sensor

FEATURES

- Easily integrates into OEM applications
- Designed for Torque auditing
- Aluminum construction
- Built-in overload protection on lower ranges
- Strain gauge based

Active end



– Output (CCW)
+ Output (CW)

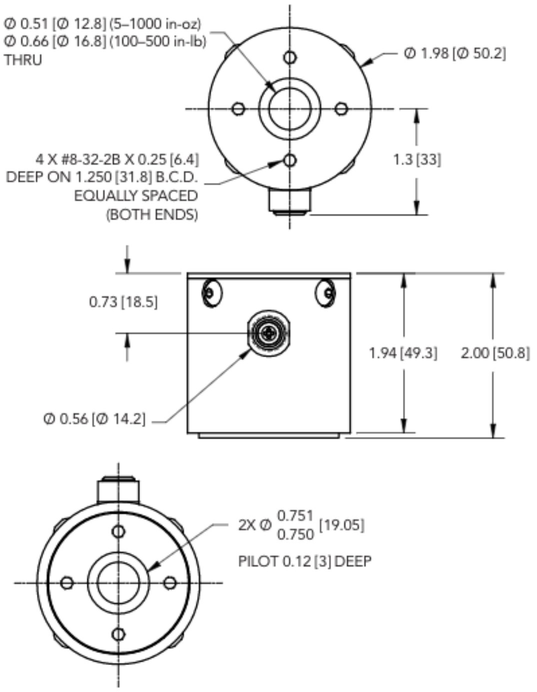
SPECIFICATIONS	
PERFORMANCE	
Nonlinearity	±0.2% of RO
Hysteresis	±0.2% of RO
Nonrepeatability	±0.05% of RO
ELECTRICAL	
Rated Output (RO)	1 mV/V nom (5 in-oz) 2 mV/V nom (10 in-oz to 500 in-lb)
Excitation (VDC or VAC)	18 max
Bridge Resistance	350 Ohm nom (5 to 1000 in-oz) 700 Ohm nom (100 to 500 in-lb)
Connection	4 Pin LEMO® Receptacle (EGG. OB. 304 CLL)
Wiring/Connector Code	CC4
MECHANICAL	
Weight (approximate)	9 oz [250 g]
Safe Overload	300% (5 to 400 in-oz) of RO 150% (1000 in-oz to 500 in-lb) of RO
Material	Aluminum
IP Rating	IP40
TEMPERATURE	
Operating Temperature	–60 to 200°F (–50 to 93°C)
Compensated Temperature	60 to 160°F (15 to 72°C)
Temperature Shift Zero	±0.002% of RO/°F (0.0036% of RO/°C)
Temperature Shift Span	±0.002% of Load/°F (0.0036% of Load/°C)
CALIBRATION	
Calibration Test Excitation	10 VDC
Calibration (standard)	5-pt CW
Calibration (available)	5-pt CW & CCW
Shunt Calibration Value	60.4 kOhm (10 to 1000 in-oz) 100 kOhm (5 in-oz, 100 to 500 in-lb)

Sensor Solution Source
Load · Torque · Pressure · Multi-Axis · Calibration · Instruments · Software

www.futek.com

ANSI Z540-1 ISO 17025 ISO 9001 ISO 13485 U.S. Manufacturer

DIMENSIONS inches [mm]



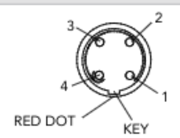
CAPACITIES

ITEM #	in-oz	Nm	Torsional Stiffness in-oz/rad
FSH02587	5*	0.04	325
FSH02588	10*	0.07	650
FSH02590	50*	0.35	3500
FSH02592	160*	1.1	11000
FSH02593	400*	2.8	30000
FSH02594	1000	7.1	71000
FSH02595	100 in-lb	11	77000 in-lb/rad
FSH02596	200 in-lb	22	95000 in-lb/rad
FSH02597	500 in-lb	56	199000 in-lb/rad

* WITH OVERLOAD PROTECTION
FOR HIGHER CAPACITIES REFER TO MODELS TFF600–750 AND TDF600–675

CONNECTOR CODE (CC4)

PIN	WIRING CODE
1	+ EXCITATION/RED
4	– EXCITATION/BLACK
2	+ SIGNAL/GREEN
3	– SIGNAL/WHITE



Drawing Number: FI1251-F

10 Thomas, Irvine, CA 92618 USA
Tel: (949) 465-0900
Fax: (949) 465-0905

www.futek.com

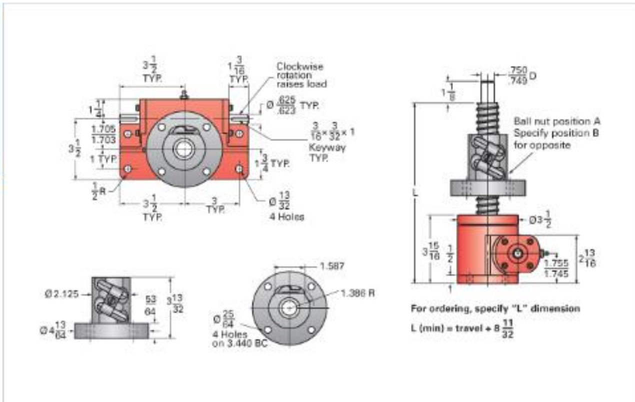
Nook ActionJac 3-BSJ-UR 24:1

NOOK

Product Specification Sheet

ActionJac 3-BSJ-UR 24:1

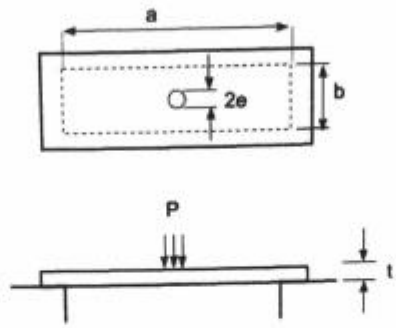
Call 800-321-7800 or visit us online at www.nookindustries.com to configure and order your ActionJac 3-BSJ-UR 24:1 today!



Details	
Capacity [Ton]	3
Gear Ratio	24:1
Turns of Worm per Inch Travel	58.1
Torque to Raise 1 lb. [in-lb]	0.0070
Lifting Screw Diameter [in]	1.17
Screw Lead [in]	0.413
Root Diameter [in]	0.870
BackDrive Holding Torque [ft-lb]	2
Tare Drag Torque [in-lb]	6
Performance Specifications	
Maximum Allowable Input [hp]	0.50
Maximum Input Torque [in-lb]	42
Maximum Worm Speed at Rated Load [rpm]	750
Maximum Load at 1750 RPM [lb]	2572
Starting Torque	1.5 x Running Torque
Weight	
Base 0 Travel [lb]	18.5
Per Inch of Travel [lb]	0.60
Grease [lb]	0.5

APPENDIX H – Structural Analysis

Rectangular Flat Plate , concentrated load at centre,edge simply supported.



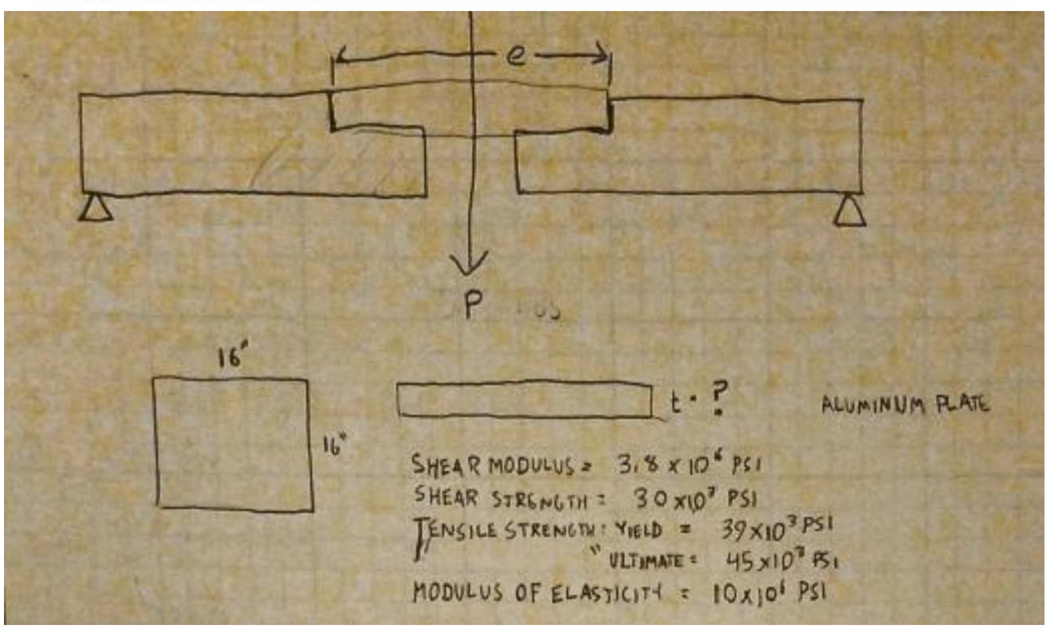
If e is small then use e' as calculated below

$$e' = \left(\sqrt{1{,}6e^2 + t^2}\right) - 0{,}675t \quad \text{if } e < 0{,}5t \text{ else use } e' = e$$

$$\sigma_m = \frac{1{,}5P}{\pi t^2}\left((1 + v)\ \ln\left(\frac{2b}{\pi e}\right) + k_2\right) \qquad \text{At centre}$$

$$y_m = k_1\ \frac{Pb^2}{Et^3} \qquad \text{At centre}$$

	a/b								
	1,0	1,1	1,2	1,4	1,6	1,8	2,0	3,0	4 ->
k_1	0,127	0,138	0,148	0,162	0,17	0,177	0,180	0,185	0,185
k_2	0,435	0,565	0,650	0,789	0,875	0,927	0,958	1,000	0,000



1/2"	ALUMINUM PLATE	

$e = 2.756\text{ IN}$ $\quad a = 16\text{ IN}$ $\quad K_1 = 0.127$

$P = 100\text{ LB}$ $\quad b = 16\text{ IN}$ $\quad K_2 = 0.435$

$t = \frac{1}{2}"$ $\quad E = 10\times10^6\text{ PSI}$ $\quad \nu = 0.33$

$P = (P_{STRUCTURE} + P_{SCREW\,JACK})\,K_{FS}$ $\quad P_{STRUCTURE} = 42\text{ LB}$ $\quad P_{SCREW\,JACK} = 20\text{ LB}$

$P = (42\text{ LB} + 20\text{ LB})(1.6129)$ $\quad K_{F.S.} = 1.6129$

$\underline{P = 100\text{ LB}}$

$$e' = \left(\sqrt{1.6e^2 + t^2}\right) - 0.675t$$

$$e' = \left(\sqrt{1.6(2.756\text{ IN})^2 + (0.5\text{ IN})^2} - .675(0.5\text{ IN})\right)$$

$$\underline{e' = 3.18427\text{ IN}}$$

$$\sigma_m = \frac{1.5P}{\pi t^2}\left((1+\nu)\ln\left(\frac{2b}{\pi e'}\right) + K_2\right)$$

$$\sigma_m = \frac{(1.5)(100\text{ LB})}{\pi(0.5\text{ IN})^2}\left((1+0.33)\ln\left(\frac{2\cdot 16\text{ IN}}{\pi\cdot 3.18427\text{ IN}}\right) + 0.435\right)$$

$$\boxed{\sigma_m = 378.44\text{ PSI}}$$

$$y_m = k_1\frac{Pb^2}{Et^3}$$

$$y_m = 0.127\cdot\frac{(100\text{ LB})(16\text{ IN})^2}{10\times10^6\text{ PSI}\cdot(.5\text{ IN})^3}$$

$$\boxed{y_m = 0.002601\text{ IN}}$$

1/4"	ALUMINUM PLATE	

$e = 2.756\,in$ $\quad a = 16\,in$ $\quad K_1 = 0.127$

$P = 100\,LB$ $\quad b = 16\,in$ $\quad K_2 = 0.435$

$t = 1/4"$ $\quad E = 10\times10^6\,psi$ $\quad \nu = 0.33$

$P = (P_{STRUCTURE} + P_{SCREW\,JACK}) \cdot K_{FS}$ $\qquad P_{STRUCTURE} = 42\,LB$ $\qquad K_{FS} = 1.6129$

$P = (42\,LB + 20\,LB) \cdot 1.6129$ $\qquad P_{SCREW\,JACK} = 20\,LB$

$\underline{P = 100\,LB}$

$$e' = \left(\sqrt{1.6e^2 + t^2}\right) - 0.675t$$

$$e' = \sqrt{1.6\,(2.756\,in)^2 + (0.25\,in)^2} - 0.675 \cdot (0.25\,in)$$

$$\underline{e' = 3.45286\,in}$$

$$\sigma_m = \frac{1.5P}{\pi t^2}\left((1+\nu)\ln\left(\frac{2b}{\pi e'}\right) + K_2\right)$$

$$\sigma_m = \frac{(1.5)(100\,LB)}{\pi\,(0.25\,in)^2}\left((1+0.33)\ln\left(\frac{2\cdot 16\,in}{\pi\cdot 3.45286\,in}\right) + 0.435\right)$$

$$\boxed{\sigma_m = 1431.48\ PSI}$$

$$y_m = K_1\,\frac{Pb^2}{Et^3}$$

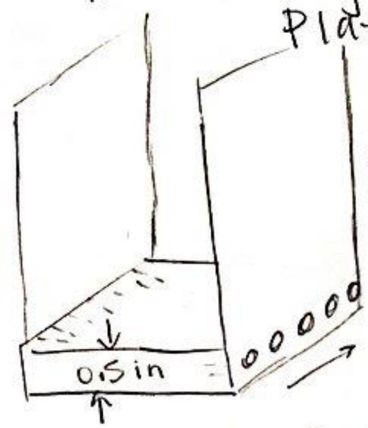
$$y_m = 0.127\,\frac{(100\,LB)(16\,in)^2}{(10\times10^6\,psi)(0.25\,in)^3}$$

$$\boxed{y_m = 0.020808\ in}$$

APPENDIX I - Shear and Stress Analysis

1) Tension Plate Shear Analysis

1/28/17 Bolt Analysis For Tension Plates



$V = \frac{M}{r} = \frac{400 \text{ in}\cdot\text{lb}}{4 \text{ in}} = 100 \text{ lb}$

Shigley's Ch 8 Section 12
Bolted and Riveted Joints Loaded in Shear:

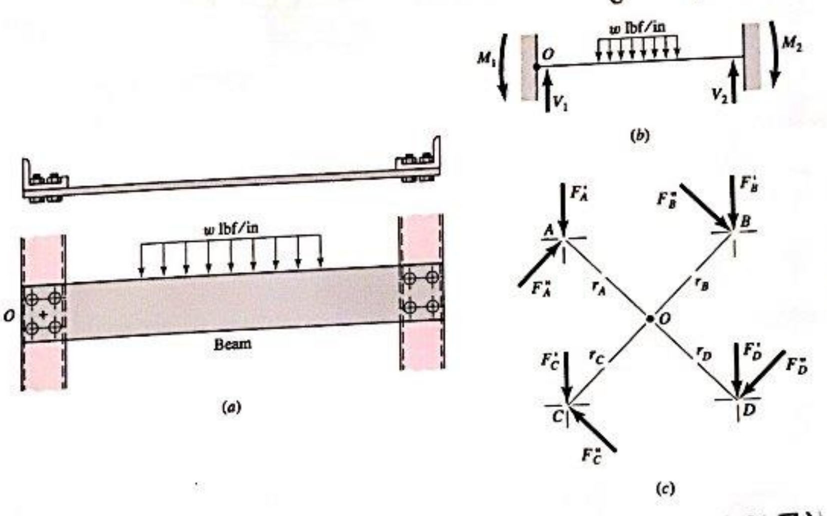
Bending Moment: $M = \frac{Ft}{2}$ — F: shear force, t: thickness of connected parts

Bending stress: $\sigma = \frac{M}{I/c}$ — I/c: section modulus for weakest member

Pure shear failure: $\tau = \frac{F}{A}$ — A: cross-sectional area of all rivets in the group

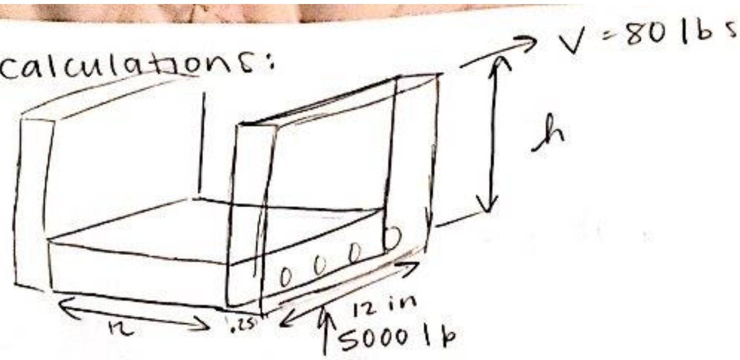
Figure 8-27

(*a*) Beam bolted at both ends with distributed load; (*b*) free-body diagram of beam; (*c*) enlarged view of bolt group centered at *O* showing primary and secondary resultant shear forces.



(63)

Calculations:



$T = F \times r$

$500 \text{ in} \cdot \text{lb} = V\,(6.25")$

$V = 80 \text{ lbs}$

$M = V h$

Primary shear load per bolt:

- $F' = \frac{V}{n} + \frac{5000 \text{ lb}}{n}$

Theoretically all bolts should take the same shear force since they are linearly arranged in the horizontal.

Secondary shear force:

- $F'' = \frac{M}{n}$

— Shear force resultant of F' and F"

The thread length of inch-series bolts, where d is the nominal diameter, is

$$L_T = \begin{cases} 2d + \frac{1}{4} \text{ in} & L \leq 6 \text{ in} \\ 2d + \frac{1}{2} \text{ in} & L > 6 \text{ in} \end{cases}$$

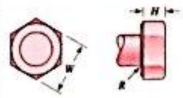
and for metric bolts is

$$L_T = \begin{cases} 2d + 6 & L \leq 125 \quad d \leq 48 \\ 2d + 12 & 125 < L \leq 200 \\ 2d + 25 & L > 200 \end{cases}$$

64

Table A-29

Dimensions of Square and Hexagonal Bolts



CDR

Design Analysis: Bolt shear

Nominal Size, in	Square W	Square H	Regular Hexagonal W	Regular Hexagonal H	Regular Hexagonal R_{min}	Heavy Hexagonal W	Heavy Hexagonal H	Heavy Hexagonal R_{min}	Structural Hexagonal W	Structural Hexagonal H	Structural Hexagonal R_{min}
$\frac{1}{4}$	$\frac{3}{8}$	$\frac{11}{64}$	$\frac{7}{16}$	$\frac{11}{64}$	0.01						
$\frac{5}{16}$	$\frac{1}{2}$	$\frac{13}{64}$	$\frac{1}{2}$	$\frac{7}{32}$	0.01						
$\frac{3}{8}$	$\frac{9}{16}$	$\frac{1}{4}$	$\frac{9}{16}$	$\frac{1}{4}$	0.01						
$\frac{7}{16}$	$\frac{5}{8}$	$\frac{19}{64}$	$\frac{5}{8}$	$\frac{19}{64}$	0.01						
$\frac{1}{2}$	$\frac{3}{4}$	$\frac{21}{64}$	$\frac{3}{4}$	$\frac{11}{32}$	0.01	$\frac{7}{8}$	$\frac{11}{32}$	0.01	$\frac{7}{8}$	$\frac{5}{16}$	0.009
$\frac{5}{8}$	$\frac{15}{16}$	$\frac{27}{64}$	$\frac{15}{16}$	$\frac{27}{64}$	0.02	$1\frac{1}{16}$	$\frac{27}{64}$	0.02	$1\frac{1}{16}$	$\frac{25}{64}$	0.021
$\frac{3}{4}$	$1\frac{1}{8}$	$\frac{1}{2}$	$1\frac{1}{8}$	$\frac{1}{2}$	0.02	$1\frac{1}{4}$	$\frac{1}{2}$	0.02	$1\frac{1}{4}$	$\frac{15}{32}$	0.021
1	$1\frac{1}{2}$	$\frac{21}{32}$	$1\frac{1}{2}$	$\frac{43}{64}$	0.03	$1\frac{5}{8}$	$\frac{43}{64}$	0.03	$1\frac{5}{8}$	$\frac{39}{64}$	0.062
$1\frac{1}{8}$	$1\frac{11}{16}$	$\frac{3}{4}$	$1\frac{11}{16}$	$\frac{3}{4}$	0.03	$1\frac{13}{16}$	$\frac{3}{4}$	0.03	$1\frac{13}{16}$	$\frac{11}{16}$	0.062
$1\frac{1}{4}$	$1\frac{7}{8}$	$\frac{27}{32}$	$1\frac{7}{8}$	$\frac{27}{32}$	0.03	2	$\frac{27}{32}$	0.03	2	$\frac{25}{32}$	0.062
$1\frac{3}{8}$	$2\frac{1}{16}$	$\frac{29}{32}$	$2\frac{1}{16}$	$\frac{29}{32}$	0.03	$2\frac{3}{16}$	$\frac{29}{32}$	0.03	$2\frac{3}{16}$	$\frac{27}{32}$	0.062
$1\frac{1}{2}$	$2\frac{1}{4}$	1	$2\frac{1}{4}$	1	0.03	$2\frac{3}{8}$	1	0.03	$2\frac{3}{8}$	$\frac{15}{16}$	0.062
Nominal Size, mm											
M5	8	3.58	8	3.58	0.2						
M6			10	4.38	0.3						
M8			13	5.68	0.4						
M10			16	6.85	0.4						
M12			18	7.95	0.6	21	7.95	0.6			
M14			21	9.25	0.6	24	9.25	0.6			
M16			24	10.75	0.6	27	10.75	0.6	27	10.75	0.6
M20			30	13.40	0.8	34	13.40	0.8	34	13.40	0.8
M24			36	15.90	0.8	41	15.90	0.8	41	15.90	1.0
M30			46	19.75	1.0	50	19.75	1.0	50	19.75	1.2
M36			55	23.55	1.0	60	23.55	1.0	60	23.55	1.5

So,

$$L_T = 2d + \frac{1}{4} \text{ in}$$

$$L_T = 2\left(\frac{1}{4}''\right) + \frac{1}{4}$$

$$L_T = \frac{3}{4}''$$

$$H_{nut} = \frac{11}{64} \text{ in}$$

washer $\approx$ 2mm = 0.08 in

Bolt length = 1 in ?

65

Table A–31

Dimensions of Hexagonal Nuts

Nominal Size, in	Width W	Height H Regular Hexagonal	Height H Thick or Slotted	Height H JAM
$\frac{1}{4}$	$\frac{7}{16}$	$\frac{7}{32}$	$\frac{9}{32}$	$\frac{5}{32}$
$\frac{5}{16}$	$\frac{1}{2}$	$\frac{17}{64}$	$\frac{21}{64}$	$\frac{3}{16}$
$\frac{3}{8}$	$\frac{9}{16}$	$\frac{21}{64}$	$\frac{13}{32}$	$\frac{7}{32}$
$\frac{7}{16}$	$\frac{11}{16}$	$\frac{3}{8}$	$\frac{29}{64}$	$\frac{1}{4}$
$\frac{1}{2}$	$\frac{3}{4}$	$\frac{7}{16}$	$\frac{9}{16}$	$\frac{5}{16}$
$\frac{9}{16}$	$\frac{7}{8}$	$\frac{31}{64}$	$\frac{39}{64}$	$\frac{5}{16}$
$\frac{5}{8}$	$\frac{15}{16}$	$\frac{35}{64}$	$\frac{23}{32}$	$\frac{3}{8}$
$\frac{3}{4}$	$1\frac{1}{8}$	$\frac{41}{64}$	$\frac{13}{16}$	$\frac{27}{64}$
$\frac{7}{8}$	$1\frac{5}{16}$	$\frac{3}{4}$	$\frac{29}{32}$	$\frac{31}{64}$
1	$1\frac{1}{2}$	$\frac{55}{64}$	1	$\frac{35}{64}$
$1\frac{1}{8}$	$1\frac{11}{16}$	$\frac{31}{32}$	$1\frac{5}{32}$	$\frac{39}{64}$
$1\frac{1}{4}$	$1\frac{7}{8}$	$1\frac{1}{16}$	$1\frac{1}{4}$	$\frac{23}{32}$
$1\frac{3}{8}$	$2\frac{1}{16}$	$1\frac{11}{64}$	$1\frac{3}{8}$	$\frac{25}{32}$
$1\frac{1}{2}$	$2\frac{1}{4}$	$1\frac{9}{32}$	$1\frac{1}{2}$	$\frac{27}{32}$
Nominal Size, mm				
M5	8	4.7	5.1	2.7
M6	10	5.2	5.7	3.2
M8	13	6.8	7.5	4.0
M10	16	8.4	9.3	5.0
M12	18	10.8	12.0	6.0
M14	21	12.8	14.1	7.0
M16	24	14.8	16.4	8.0
M20	30	18.0	20.3	10.0
M24	36	21.5	23.9	12.0
M30	46	25.6	28.6	15.0
M36	55	31.0	34.7	18.0

$$\tau_{shear} = \frac{F}{A_S}$$

$$\tau_{shear} = \frac{F' + F''}{\frac{\pi (d_m)^2}{4}}$$

Bending stress:

$$M = V(h)$$

$$I = I_{bar}$$

$$I = \frac{t_{plate} W_{plate}^3}{12}$$

66

$$\sigma = \frac{Mc}{I} = \left[\left(\frac{(\text{lb}\cdot\text{in})(\text{in})}{\text{in}^4}\right)\right] = \text{psi}$$

c: perpendicular distance to neutral axis

$c = \frac{w}{2}$ in our case

SINGLE SHEAR CALCULATIONS, MIN. LBS

NOMINAL BOLT DIAMETER	BODY SHEAR AREA, SQ IN	GR 2	A307A & B	GR 5 / A325	GR 8 / A490
1/4"	0.04908	2,179.2	1,766.9	3,533.8	4,417.2
5/16"	0.07669	3,405.0	2,760.8	5,521.7	6,902.1
3/8"	0.11044	4,903.5	3,975.8	7,951.7	9,939.6
7/16"	0.15033	6,674.7	5,411.9	10,823.8	13,529.7
1/2"	0.19634	8,717.5	7,068.2	14,136.5	17,670.6
9/16"	0.24850	11,033.4	8,946.0	17,892.0	22,365.0
5/8"	0.30679	13,621.5	11,044.4	22,088.9	27,611.1
3/4"	0.44178	19,615.0	15,904.1	31,808.2	39,760.2
7/8"	0.60132	21,647.5	21,647.5	43,295.0	54,118.8
1"	0.78539	28,274.0	28,274.0	56,548.1	70,685.1
1 1/8"	0.99401	35,784.4	35,784.4	62,622.6	89,460.9
1 1/4"	1.22718	44,178.5	44,178.5	77,312.3	110,446.2

DEFINITIONS:
- > Ultimate Tensile Strength, UTS - PSI - Lbs/Square Inch
- > Ultimate Shear Strength, USS - PSI USS = .6 X UTS
- > Body Shear Area, BSA - Square Inches
- > Single Shear Strength, SSS - Lbs SSS = USS X BSA

Shear Force on each plate	5001	lbs
bolts used	1/4 20	
max shear load/grade 8 bolt	4417	lbs
n (# of bolts)	Shear Force/Bolt (lb)	Shear Stress/Bolt (psi)
1	5001	101879.6215
2	2500.5	50939.81073
3	1667	33959.87382
4	1250.25	25469.90537
5	1000.2	20375.92429
6	833.5	16979.93691
7	714.4285714	14554.23164
8	625.125	12734.95268
9	555.6666667	11319.95794
10	500.1	10187.96215
Bending stress in plate		
M	2300	lb.in
I_bar	7.145833333	in^4
Critical Bending Sress (sigma)	1126.530612	psi

2) Dowel Pin Shear Analysis

1/31/17 Dowel Pin Shear Analysis

Max torque 400 in·lb

$$V = \frac{M}{r}$$

$$V = \frac{400 \text{ in}\cdot\text{lb}}{2 \text{ in}}$$

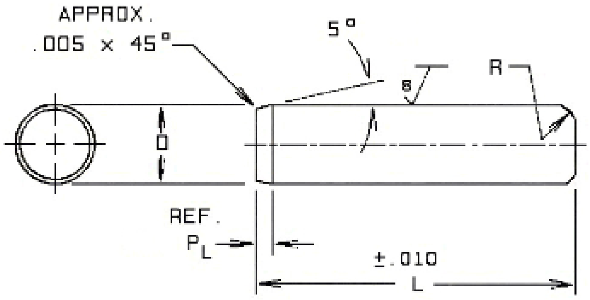
$$V = 200 \text{ lb}$$

1/8" dowel pin max shear strength:
1840 lb. (alloy steel)

CDR

Design Analysis: Dowel pins

Inch Series Dowel Pins



Nominal Diameter	Diameter D Standard Pin Max.	Standard Pin Min.	Oversize Pin Max.	Oversize Pin Min.	PL Point Length Ref.	R Top Radius Basic
1/8	0.1253	0.1251	0.1261	0.1259	0.045	0.031
3/16	0.1878	0.1876	0.1886	0.1884	0.060	0.046
1/4	0.2503	0.2501	0.2511	0.2509	0.070	0.062
5/16	0.3128	0.3126	0.3136	0.3134	0.070	0.062
3/8	0.3753	0.3751	0.3761	0.3759	0.080	0.078
7/16	0.4378	0.4376	0.4386	0.4384	0.090	0.093
1/2	0.5003	0.5001	0.5011	0.5009	0.090	0.109
5/8	0.6253	0.6251	0.6261	0.6259	0.090	0.125
3/4	0.7503	0.7501	0.7511	0.7509	0.120	0.125
7/8	0.8753	0.8751	0.8761	0.8759	0.120	0.125
1	1.0003	1.0001	1.0011	1.0009	0.120	0.125

Mechanical Properties

Nominal Diameter	Single Shear Strength Pounds	Double Shear Strength Pounds	Recommended Hole Size Standard Pins Max.	Min.
1/8	1840	3680	0.1250	0.1245
3/16	4150	8300	0.1875	0.1870
1/4	7360	14720	0.2500	0.2495
5/16	11500	23000	0.3125	0.3120
3/8	16550	33100	0.3750	0.3745
7/16	22550	45100	0.4375	0.4370
1/2	29450	58900	0.5000	0.4995
5/8	46000	92000	0.6250	0.6245
3/4	66200	132400	0.7500	0.7495
7/8	90200	180400	0.8750	0.8745
1	117800	235600	1.0000	0.9995

Applicable Standard: ASME B18.8.2
Alloy Steel; Core Hardness Rc 47-58
Case Hardness Rc 60 min.

APPENDIX J: THERMAL ANALYSIS

Thermal Calculations			
Dimensions			
t	0.0127	m	
W	0.3048	m	
B	0.3048	m	
Area	0.09290304	m^2	
Perimeter	1.2192	m	
L	0.0762	m	
Properties of wood			
	cold	hot	
k_wood	0.12	0.12	W/m-K
Properties of air			
	cold	hot	units
k_air	0.01599	0.01599	W/m-K
visc	0.00001343	0.00001343	m^2/s
alpha	0.00001887	0.00001887	m^2/s
Tchamber	219.261	344.261	K
Tambient	293	293	K
Tsurface	**234.684827**	**328.395975**	**K**
g	9.81	9.81	m/s^2
beta	0.00379014	0.00321856	1/K
Ra	3785497.52	1951198.474	-
hbar	2.49912511	4.235089709	W/(m^2-K)
q_1	13.5394055	-13.9266997	
q_2	13.5394023	-13.9266422	
Tsurface	**-36.967311**	**131.7127549**	**F**

APPENDIX K - Bill of Materials

						Numbering System - ABCDE				
			Indented Bill of Material (BOM)			Lvl	0	1	$ 2.00	$ 3.00
			Friction Test Machine (FTM)			Letter(s)	A	BC	D	E
Assembly Level	**Part Number**		**Description**			**Vendor**	**Part No.**	**Qty**	**Cost**	**Ttl Cost**
		Lvl0	Lvl1	Lvl2	Lvl3					
0	10000	FTM Assy								$ -
1	10100		Structure							$ -
2	10110			31 inch 80/20		80/20	1515-ULS	4	$ 14.04	$ 56.16
2	10120			13 inch 80/20		80/20	1515-ULS	9	$ 7.02	$ 63.18
2	10130			80/20 T-Bracket		80/20	47065T271	2	$ 7.80	$ 15.60
2	10140			80/20 90-Bracket		80/20	47065T279	16	$ 7.10	$ 113.60
2	10150			Side Plate		-	-	1	$ -	$ -
2	10160			Top Plate		-	-	1	$ -	$ -
2	10170			Bearing Plate		-	-	1	$ -	$ -
2	-			80/20 fasteners		80/20	47065T97	108	$ 0.68	$ 73.44
2	-			1/4-20, 1" Hex Screw		McMaster	92240A542	4	$ 0.11	$ 0.42
2	-			1/4-20 Hex Nut		McMaster	91845A029	4	$ 0.04	$ 0.15
2	-			1/4 Washer		McMaster	92141A029	8	$ 0.03	$ 0.27
1	10200		Thermal Chamber							$ -
2	10210			Duct Side Plate		-	-	2	$ -	$ -
2	10220			Side Plate		-	-	2	$ -	$ -
2	10230			Top Plate		-	-	1	$ -	$ -
2	10240			Btm Plate		-	-	1	$ -	$ -
2	10250			Surface Latch		Penn Elcom	7538	4	$ 9.09	$ 36.36
1	10300		Load Loop BTM							$ -
2	10310			Oil Bracket						$ -
3	INA SCE2410				Shaft Bearing	Amazon	INA SCE2410	1	$ 11.05	$ 11.05
3	10311				Oil Container	-	-	1	$ -	$ -
3	10312				Skew Roller Bracket	-	-	1	$ -	$ -
2	10320			Screw Jack		NOOK	3-BSJ-UR 24:1	1	$ 1,645.00	$1,645.00
2	10330			Load Cell Plate		-	-	1	$ -	$ -
2	10340			Load Cell		Futek	LCF455	1	$ -	$ -
2	10350			Btm Bracket		-	-	1	$ -	$ -
2	10360			Torque Cell		Futek	FSH02597	1	$ 950.00	$ 950.00
2	10370			Sliding Plate Top		-	-	1	$ -	$ -
2	-			3/8-24, 1.5" Hex Screw		McMaster	92198A357	8	$ 0.61	$ 4.90
2	-			3/8-24 Hex Nut		McMaster	91845A125	4	$ 0.09	$ 0.35
2	-			3/8 Washer		McMaster	92141A031	12	$ 0.05	$ 0.60
2	-			8-32, .75" Hex Screw		McMaster	92314A197	8	$ 0.15	$ 1.22
2	-			#8 Washer		McMaster	92141A009	8	$ 0.02	$ 0.16
1	10400		Pinned Bracket							$ -
2	10410			Sliding Plate BTM		-	-	1	$ -	$ -
2	10420			0.5", 2" Dowel Pin		McMaster	97395A776	2	$ 8.17	$ 16.34
1	10500		Load Loop TOP							$ -
2	INA SCE2410			Shaft Bearing		Amazon	INA SCE2410	1	$ 11.05	$ 11.05
2	10510			Load Loop Bracket (a)		-	-	1	$ -	$ -
2	10520			Load Loop Bracket (b)		-	-	1	$ -	$ -
2	-			1/4-20, 1" Hex Screw		McMaster	92240A542	4	$ 0.11	$ 0.42
2	-			1/4 Washer		McMaster	92141A029	4	$ 0.03	$ 0.13
1	10600		Shaft			-	-	1	$ -	$ -
1	10700		Skew Roller Plate			-	-	1	$ -	$ -
1	10800		Thrust Washer			Timken	KB1110TVPB	2	$ 22.00	$ 44.00
1	10900		Thrust Plate			Amazon	Koyo GS.81110	4	$ 17.84	$ 71.36
1	11000		Top Attachment			-	-	1	$ -	$ -
1	11100		Tension Plate			-	-	2	$ -	$ -
1	11200		1034VSR Drill			Home Depot	1000388953	1	$ 211.00	$ 211.00
1	-		1/4-20, 1" Hex Screw			McMaster	92240A542	8	$ 0.11	$ 0.85
1	-		1/4 Washer			McMaster	92141A029	10	$ 0.03	$ 0.34
1	-		1/4-20, 3.5" Hex Screw			McMaster	92198A556	1	$ 0.34	$ 0.34
1	-		1/4-20 Hex Nut			McMaster	91845A029	4	$ 0.04	$ 0.15
1	-		Epoxy			McMaster	7605A11	1	$ 6.17	$ 6.17
1	11300		NI USB 6000			National Instruments	782602-01	1	$ 150.00	$ 150.00
	Total Parts							**266**		**$3,334.62**

APPENDIX L - Source Budget

Source Budget							
Part No.	**Size**	**Material**	**Vendor**	**Qty**	**Cost**	**Ttl Cost**	**Part(s) Used On**
P314T6	1/4"x6"x16"	6061 Aluminum	Metals Depot	1	$ 31.96	$ 31.96	Side Plate
P314T6	1/4"x16"x16"	6061 Aluminum	Metals Depot	1	$ 141.76	$ 141.76	Top Plate
P312T6	1/2"x5"x5"	6061 Aluminum	Metals Depot	1	$ 36.28	$ 36.28	Bearing Plate
Plywood	4'x4'	Plywood	Home Depot	2	$ 8.99	$ 17.98	Thermal Chamber Sides
-	-	Plywood	Home Depot	2	$ -	$ -	-
-	-	Plywood	Home Depot	2	$ -	$ -	-
-	-	Plywood	Home Depot	1	$ -	$ -	-
-	-	Plywood	Home Depot	1	$ -	$ -	-
-	d = 4", h~3"	Metal	-	1	$ -	$ -	Oil Container
8974K61	d = 4-1/4", L = 1'	6061 Aluminum	McMaster	1	$ 93.35	$ 93.35	Skew Roller Bracket
P312T6	1/2"x4"x7.5"	6061 Aluminum	Metals Depot	1	$ -	$ -	Load Cell Plate
P312T6	1/2"x7"x7"	6061 Aluminum	Metals Depot	1	$ 47.11	$ 47.11	Btm Bracket
P312T6	1/2"x4"x4"	6061 Aluminum	Metals Depot	1	$ 32.22	$ 32.22	Sliding Plate Top
P312T6	1/2"x4"x4"	6061 Aluminum	Metals Depot	1	$ 32.22	$ 32.22	Sliding Plate Bottom
9056K44	OD=3", ID=1.5",L=.5'	6061 Aluminum	McMaster	1	$ 49.56	$ 49.56	Load Loop Bracket (a)
P312T6	1/2"x7"x7"	6061 Aluminum	Metals Depot	1	$ 47.11	$ 47.11	Load Loop Bracket (b)
-	-	-	Parker	1	$ -	$ -	Shaft
-	-	-	Parker	1	$ -	$ -	Skew Roller Plate
8974K55	d = 2-7/8", L = 6"	6061 Aluminum	McMaster	1	$ 31.72	$ 31.72	Top Attachment
P314T6	1/4"x7"x21"	6061 Aluminum	Metals Depot	2	$ 43.63	$ 87.26	Tension Plate
1515-ULS	-	-	80/20	4	$ 14.04	$ 56.16	31 inch 80/20
1515-ULS	-	-	80/20	9	$ 7.02	$ 63.18	13 inch 80/20
47065T271	-	-	80/20	2	$ 7.80	$ 15.60	80/20 T-Bracket
47065T279	-	-	80/20	16	$ 7.10	$ 113.60	80/20 90-Bracket
47065T97	-	-	80/20	108	$ 0.68	$ 73.44	80/20 fasteners
92240A542	-	-	McMaster	4	$ 0.11	$ 0.42	1/4-20, 1" Hex Screw
91845A029	-	-	McMaster	4	$ 0.04	$ 0.15	1/4-20 Hex Nut
92141A029	-	-	McMaster	8	$ 0.03	$ 0.27	1/4 Washer
7538	-	-	Penn Elcom	4	$ 9.09	$ 36.36	Surface Latch
INA SCE2410	-	-	Amazon	1	$ 11.05	$ 11.05	Shaft Bearing
3-BSJ-UR 24:1	-	-	NOOK	1	$ 1,645.00	$ 1,645.00	Screw Jack
LCF455	-	-	Futek	1	$ -	$ -	Load Cell
LCF455	-	-	Futek	1	$ -	$ -	Load Cell
FSH02597	-	-	Futek	1	$ 950.00	$ 950.00	Torque Cell
92198A357	-	-	McMaster	8	$ 0.61	$ 4.90	3/8-24, 1.5" Hex Screw
91845A125	-	-	McMaster	4	$ 0.09	$ 0.35	3/8-24 Hex Nut
92141A031	-	-	McMaster	12	$ 0.05	$ 0.60	3/8 Washer
92314A197	-	-	McMaster	8	$ 0.15	$ 1.22	8-32, .75" Hex Screw
92141A009	-	-	McMaster	8	$ 0.02	$ 0.16	#8 Washer
97395A776	-	-	McMaster	2	$ 8.17	$ 16.34	0.5", 2" Dowel Pin
INA SCE2410	-	-	Amazon	1	$ 11.05	$ 11.05	Shaft Bearing
92240A542	-	-	McMaster	4	$ 0.11	$ 0.42	1/4-20, 1" Hex Screw
92141A029	-	-	McMaster	4	$ 0.03	$ 0.13	1/4 Washer
KB1110TVPB	-	-	Timken	2	$ 22.00	$ 44.00	Thrust Washer
Koyo GS.81110	-	-	Amazon	4	$ 17.84	$ 71.36	Thrust Plate
1000388953	-	-	Home Depot	1	$ 211.00	$ 211.00	1034VSR Drill
92240A542	-	-	McMaster	8	$ 0.11	$ 0.85	1/4-20, 1" Hex Screw
92141A029	-	-	McMaster	10	$ 0.03	$ 0.34	1/4 Washer
92198A556	-	-	McMaster	1	$ 0.34	$ 0.34	1/4-20, 3.5" Hex Screw
91845A029	-	-	McMaster	4	$ 0.04	$ 0.15	1/4-20 Hex Nut
7605A11	-	-	McMaster	1	$ 6.17	$ 6.17	Epoxy
782602-01	-	-	NI	1	$ 150.00	$ 150.00	DAQ module
Number of Parts				**244**	**TOTAL**	**$ 4,133.15**	
					Cal Poly	**$ 1,538.15**	
					Parker	**$ 2,595.00**	

APPENDIX M – Failure Mode Analysis

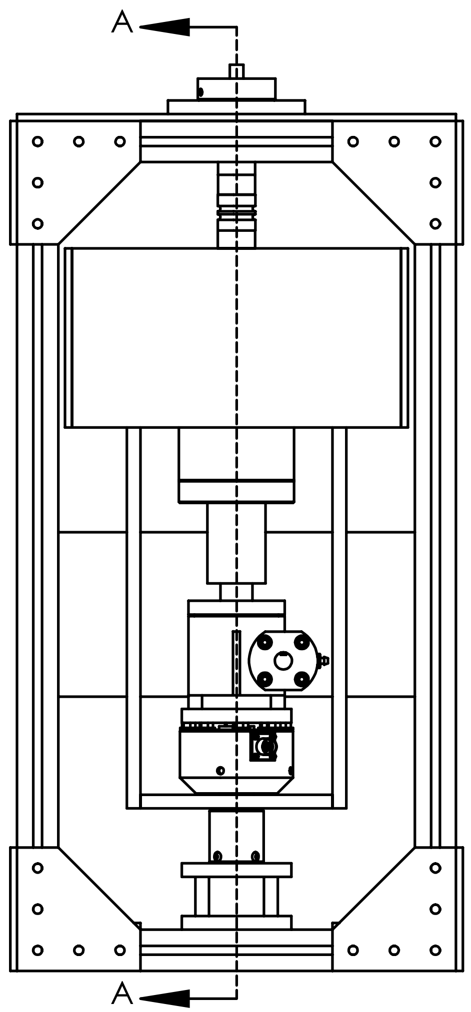
Item / Function	Potential Failure Mode	Potential Effect(s) of Failure	Severity	Potential Cause(s) / Mechanism(s) of Failure	Occurence	Criticality	Recommended Action(s)	Responsibility & Target Completion
Generate Axial Load	Apply too much load	Damage to system	7	User error of screw jack	6	42	Have specified number of turns to apply specified 5000 lb load.	Brandon Killian - May 2016
				Screw jack breaks	3	21		
		Innaccurate Test data	5	User error of screw jack	6	30	Have specified number of turns to apply specified 5000 lb load.	Brandon Killian - May 2016
				Screw jack breaks	3	15		
	Apply too little load	Measurement of torque will not be accurate to the load case specified	5	User error of screw jack	6	30	Have specified number of turns to apply specified 5000 lb load.	Brandon Killian - May 2016
				Screw jack breaks	3	15		
Measure Torque	No Measurement	Test cannot be performed	6	Problem with reaction torque sensor	3	18	Make sure sensor is placed within specified temp ranges and doesn't experience too much axial load that	Brandon Killian - February 2016
	Measures the wrong torque	Data is flawed	5	Poor location of sensor	1	5	Place torque sensor in place where torsion isn't effected	Brandon Killian - February 2016
				Temperature damages to sensor	3	15		
	Torque is induced with the measurement	Data is flawed	5	Device is too stiff, reacts to torsion	1	5	Place torque sensor in place where torsion isn't effected	Brandon Killian - February 2016
	Measurement is unreliable	Data is flawed	5	Device gives different numbers when test is run multiple times	3	15	Ensure calibration is correct before taking data	Brandon Killian - May 2016
				Device is not properly calibrated	3	15		
Generate rotational speed to drive the shaft	No Speed is generated	Test will not be performed	6	Motor is broken	1	6	Ensure machine is connected to power.	Megan Hardisty - May 2016
				Motor is not powered	1	6		
	No Speed is transmitted to the shaft	Test will not be performed	6	Belt/gear system breaks	1	6	Check belt connection before starting test.	Megan Hardisty - May 2016
	The speed is not steady/reliable	Data is flawed	5	motor cannot be accurately controlled	6	30	*this will be human error from the drill	-
Measure Speed	No Measurement	Test cannot be performed	6	Sensor is not powered	1	6	Ensure machine is connected to power.	Megan Hardisty - May 2016
				Sensor is damaged and cannot turn on	1	6		
	Measurement is unreliable	Data is flawed	5	Sensor does not have enough marks around the shaft to be accurate at low speeds	1	5	Make sure optical speed sensor has appropriate components and is calibrated before delivering to customer.	Brandon Killian - May 2016
				sensor is not properly calibrated	3	15		

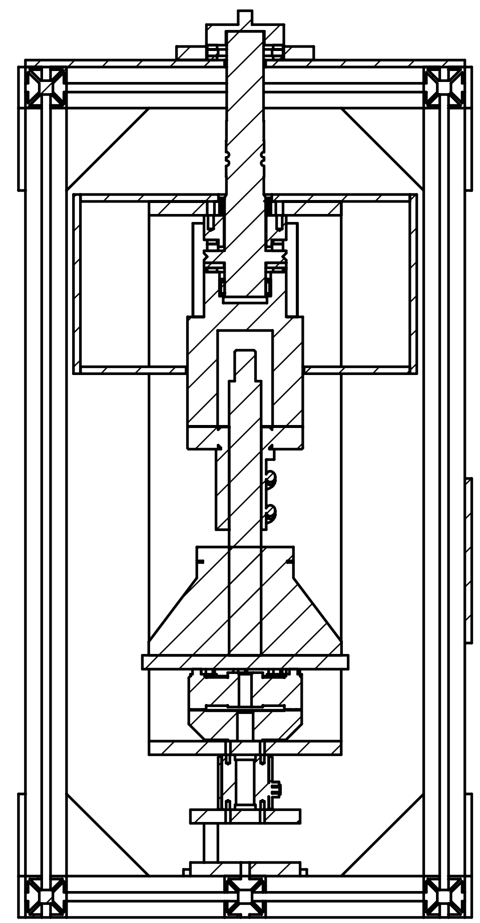
Item / Function	Potential Failure Mode	Potential Effect(s) of Failure	Severity	Potential Cause(s) / Mechanism(s) of Failure	Occurence	Criticality	Recommended Action(s)	Responsibility & Target Completion
Measure Load	Measures the wrong Load	Data is flawed	5	Load cell takes the weight of the structure as well as the axial load	1	5	Place load cell where it won't take the weight of the structure.	Alec Makowski - May 2016
	Measurement is unreliable	Data is flawed	5	Load cell is not properly calibrated	3	15	Calibrate correctly before providing machine to customer.	Alec Makowski - May 2016
Collect Data	No data collection	Test cannot be verified	6	DAQ is not programmed correctly	1	6	Check data collection with DAQ before providing machine to customer.	Megan Hardisty - May 2016
	Data is unreliable	Data is flawed	5	sensors are innacurate	1	5	Check data collection with DAQ before providing machine to customer.	Megan Hardisty - May 2016
				DAQ is not programmed correctly	1	5		
Measure Temperature	No Measurement	Test cannot be performed	6	Thermocouple is not connected	3	18	Check thermocouple placement before test.	Brandon Killian - May 2016
	Measures the wrong Temperature	Data is flawed	5	Thermocouple is not placed correctly	8	40	Check thermocouple placement before test.	Brandon Killian - May 2016
	Measurement is unreliable	Data is flawed	5	Themocouple is damaged	6	30	Verify DAQ collections before providing machine to customer.	Megan Hardisty - May 2016
Submerge roller bearings/thrust bearings in oil	Leak	Bearings get damaged	7	container is cracked	3	21	Check lubricant container before starting each test. Make container out of strong material	Alec Makowski - May 2016
				container isn't sealed correctly	3	21		
	Splash into other components	Components are damaged	7	container has no lid	1	7	Check lid is tightly on container before running test.	Alec Makowski - May 2016
	Innadequate lubrication	Bearings get damaged	7	not enough oil is in the chamber	3	21	Have a mark for correct oil level and check before each test.	Alec Makowski - May 2016
Contain hot/cold air	Sensors damaged	Broken sensors	7	Condensation on the shaft	3	21	Oil shaft to eliminate frost problem. Make sure the sensor temperature ranges are within our chamber fluctuation temps.	Alec Makowski - February 2016
				Temperature leakage onto the sensors	3	21		
				sensors are in the thermal chamber	3	21		
	Temperature is not accurate/not the desired temperature	Data is flawed	5	chamber is not well insulated	1	5	Check data collection with DAQ before providing machine to customer.	Megan Hardisty - May 2016
				Thermocouple is innaccurate	3	15		
				Flow is unreliable	1	5		
Support the components	Components shift location after testing	Data becomes innaccurate	5	Structure does not hold down components well	3	15	More bolts or epoxy permanent parts of structure. Add more components to make robust.	Alec Makowski - May 2016
				Structure is not bolted together well	3	15		
		Sensors get damaged	7	Structure does not hold down components well	3	21	More bolts or epoxy permanent parts of structure. Add more components to make robust.	Alec Makowski - May 2016
				Structure is not bolted together well	3	21		

APPENDIX N – Design Verification Plan

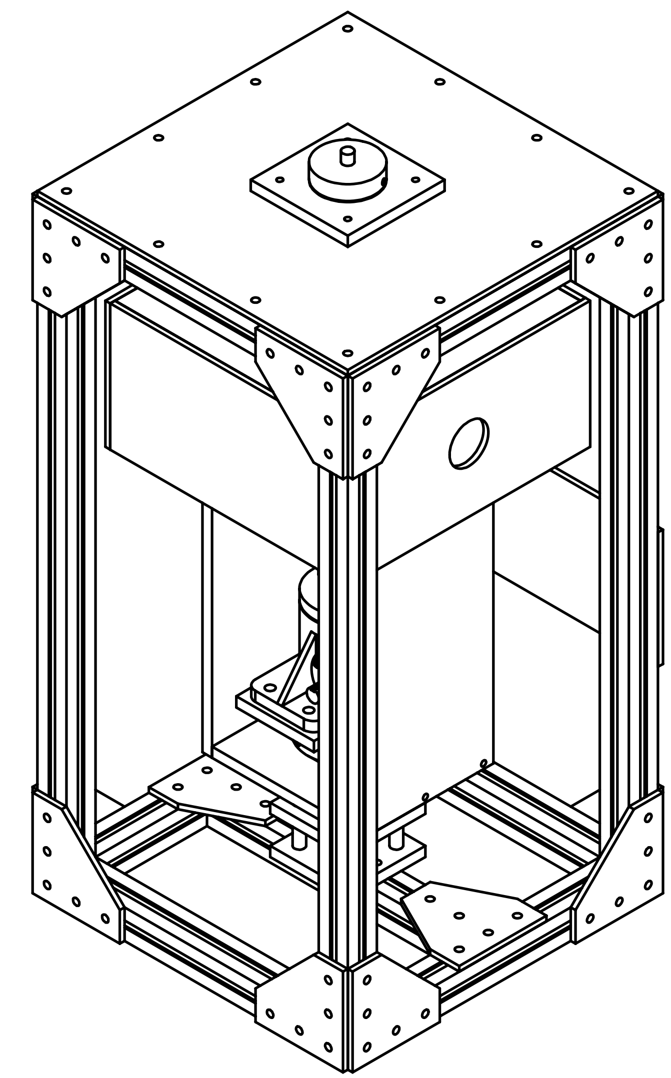
			ME428 DVP&R Format						
Report Date			Sponsor						Componen
TEST PLAN									
Item No	Specification or Clause Reference	Test Description	Acceptance Criteria	Test Responsibility	Test Stage	SAMPLES		TIMING	
						Quantity	Type	Start date	Finish date
1	1	Measure weight with scale	<150 lb	Brandon	Final			4/13/2017	4/13/2017
2	2	Measure dimensions with tape measure	<24x24x48 in	Megan	Final			4/13/2017	4/13/2017
3	3	Test participants can grip and carry structure through doorway, sample of adults with a doorway	95% can grip structure	Megan	Final			4/13/2017	4/18/2017
4	4	Measure max axial load with compression load cell	>5000 lb	Alec	Component			3/18/2017	3//22/2017
5	5	Measure max torque at 200 RPM with torque sensor	>350 in-lb	Brandon	Component			3/18/2017	3//22/2017
6	6	Measure max rotational speed with optical speed sensor	>200 RPM	Brandon	Component			3/18/2017	3//22/2017
7	7	4 channel DAQ sample rate	>1kHz	Megan	Component			3/18/2017	3//22/2027
8	8	Measure S.S. temp range, type E thermocouple	-65 to 160 F	Alec	Final			4/13/2017	4/18/2017
9	9	Works off Standard Wall Outlet in test area with wall outlet	Works	Brandon	Rough			4/13/2017	4/15/2017
10	10	Cost	<$5000	Alec	Final			4/13/2017	4/20/2017
11	11	Visual inspection of oil leakage during test, requires test area with wall outlet	no oil leakage	Alec	Component			4/13/2017	4/22/2017
12	14	Visual inspection of stability during test	Components move as designed	Megan	Final			4/13/2017	4/18/2017

APPENDIX O - Drawings





SECTION A-A
SCALE 1 : 7



ITEM NO.	PART NUMBER	SW-Title(Title)	QTY.
1	10100	STRUCTURE ASSY	1
2	10200	THERMAL CHAMBER	1
3	10300	LOAD LOOP BTM	1
4	10400	PINNED BRACKET	1
5	10500	LOAD LOOP TOP	1
6	10600	SHAFT	1
7	10700	SKEW ROLLER PLATE	1
8	10800	THRUST WASHER	2
9	10900	THRUST PLATE	4
10	11000	TOP ATTACHMENT	1
11	11100	TENSION PLATE	2
12	-	1/4-20, 1" HEX SCREW	8
13	-	1/4" WASHER	10
14	-	1/4-20, 3.5" HEX SCREW	1
15	-	1/4-20 HEX NUT	4

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 1/100
ANGULAR: MACH ± 1° BEND ± 2°
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005

INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5

MATERIAL

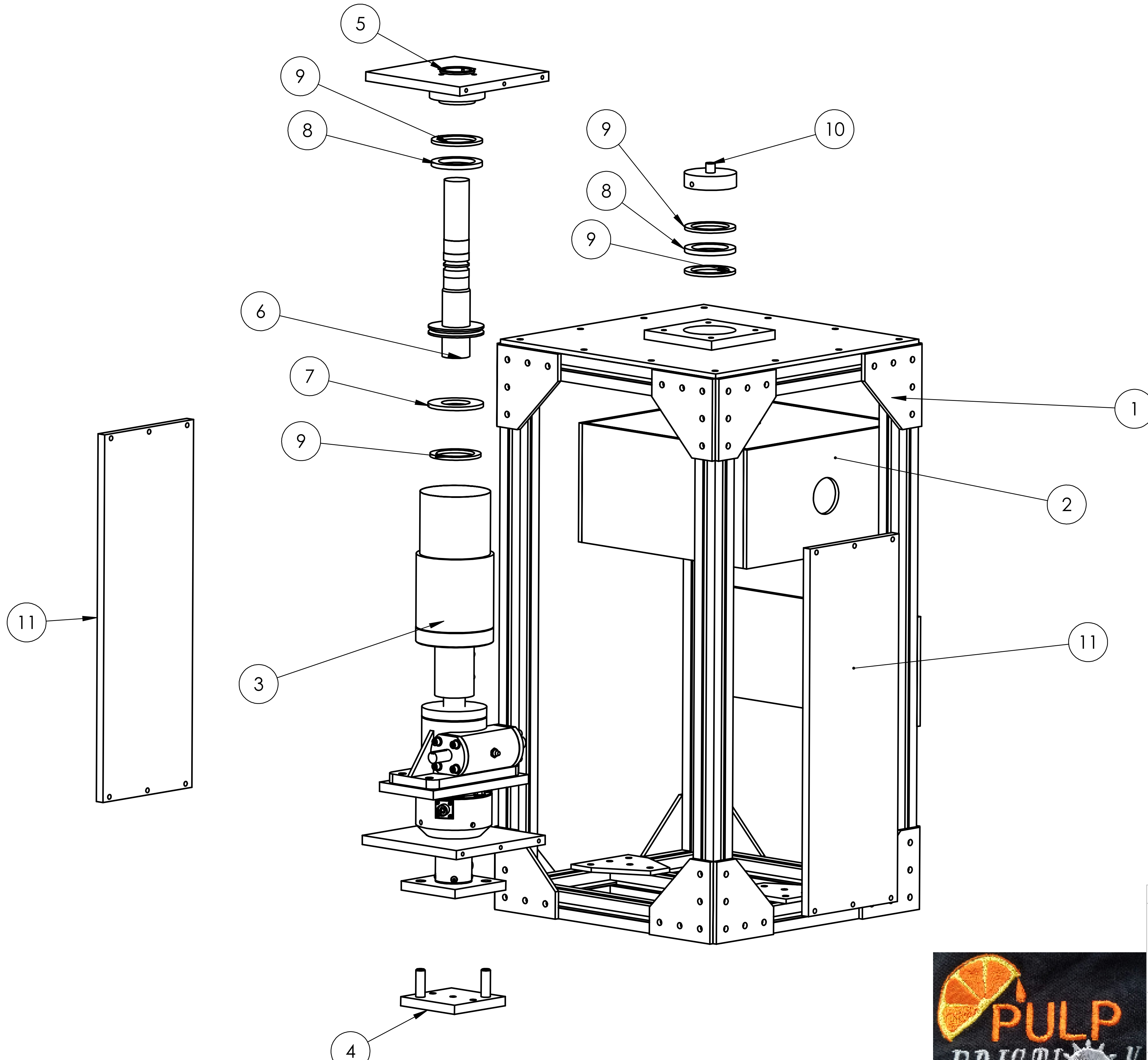
FINISH

NEXT ASB. NO.:

TITLE:
FRICTION TEST MACHINE

SIZE: B
DWG. NO.: 10000
REV

SCALE: 1:20 WEIGHT: SHEET 1 OF 2



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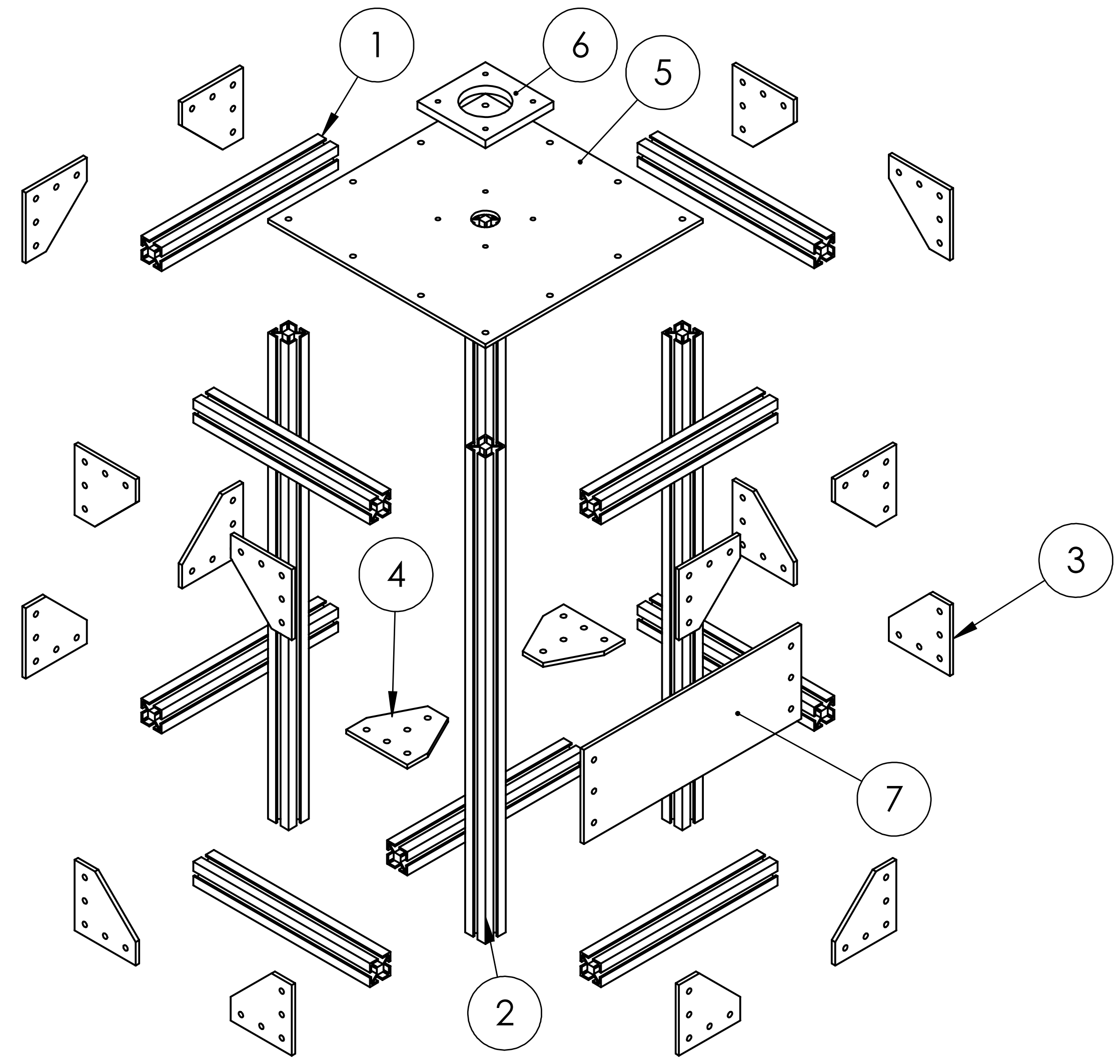
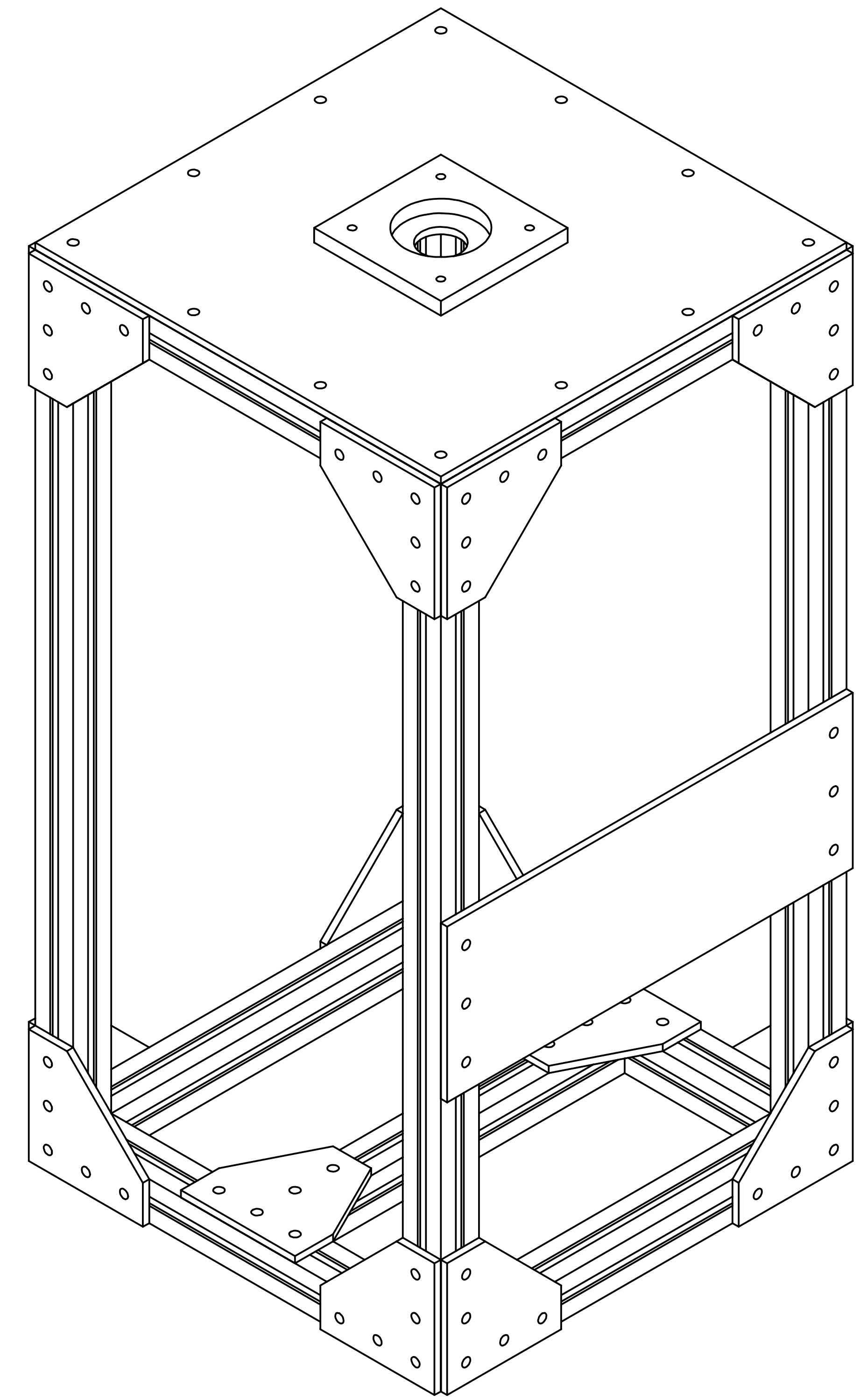
UNLESS OTHERWISE SPECIFIED:	NEXT ASB. NO.:
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ±1/100 ANGULAR: MACH ±1° BEND ±2° TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.005	TITLE: FRICTION TEST MACHINE
INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5	
MATERIAL	SIZE B DWG. NO. 10000 REV
FINISH	SCALE: 1:20 WEIGHT: SHEET 2 OF 2





ITEM NO.	PART NUMBER	TITLE	QTY.
1	10120	13 IN 80/20	9
2	10110	31 IN 80/20	4
3	10140	80/20 90 DEG BRACKET	16
4	10130	80/20 T-BRACKET	2
5	10160	TOP PLATE	1
6	10170	BEARING PLATE	1
7	10150	SIDE PLATE	1
8	-	80/20 FASTENERS	108
9	-	1/4-20, 1" HEX SCREW	4
10	-	1/4-20 HEX NUT	4
11	-	1/4" WASHER	8
12	10180	OPTEK OPB720A-06Z SENSOR	1

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±1/100
ANGULAR: MACH ±1° BEND ±2°
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC
TOLERANCING PER: ASME Y14.5

MATERIAL

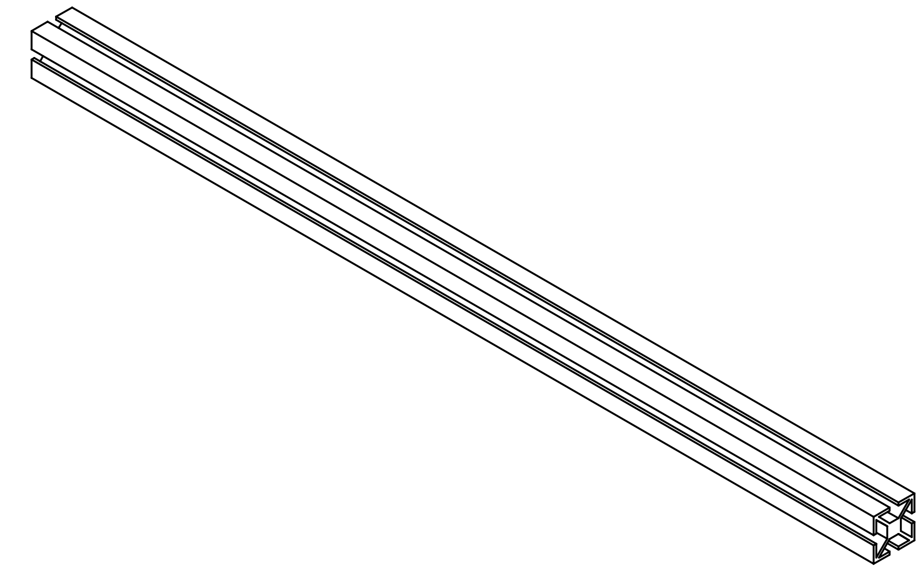
FINISH

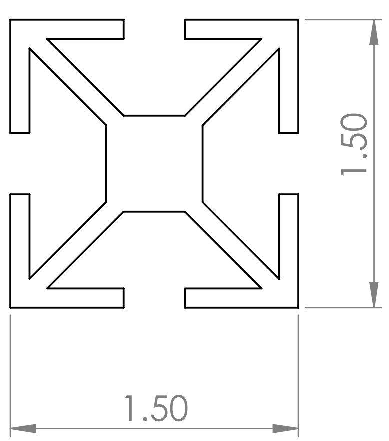
NEXT ASB. NO.: 10000

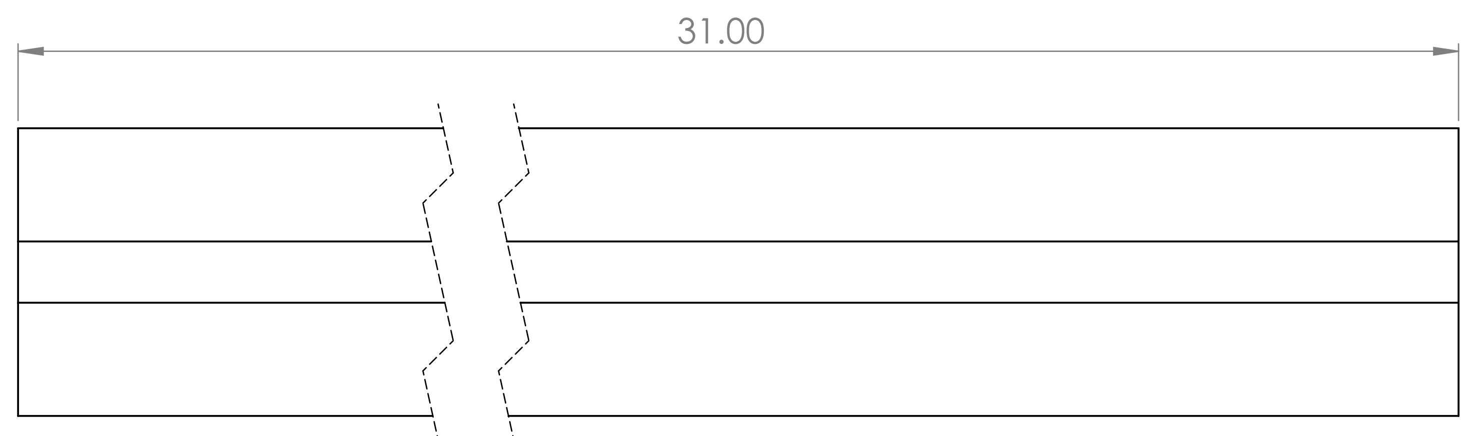
TITLE:
STRUCTURE

SIZE B | DWG. NO. 10100 | REV

SCALE: 1:20 | WEIGHT: | SHEET 1 OF 1







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UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±1/100
ANGULAR: MACH ±1° BEND ±2°
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC
TOLERANCING PER: ASME Y14.5

MATERIAL
6061 ALUMINUM

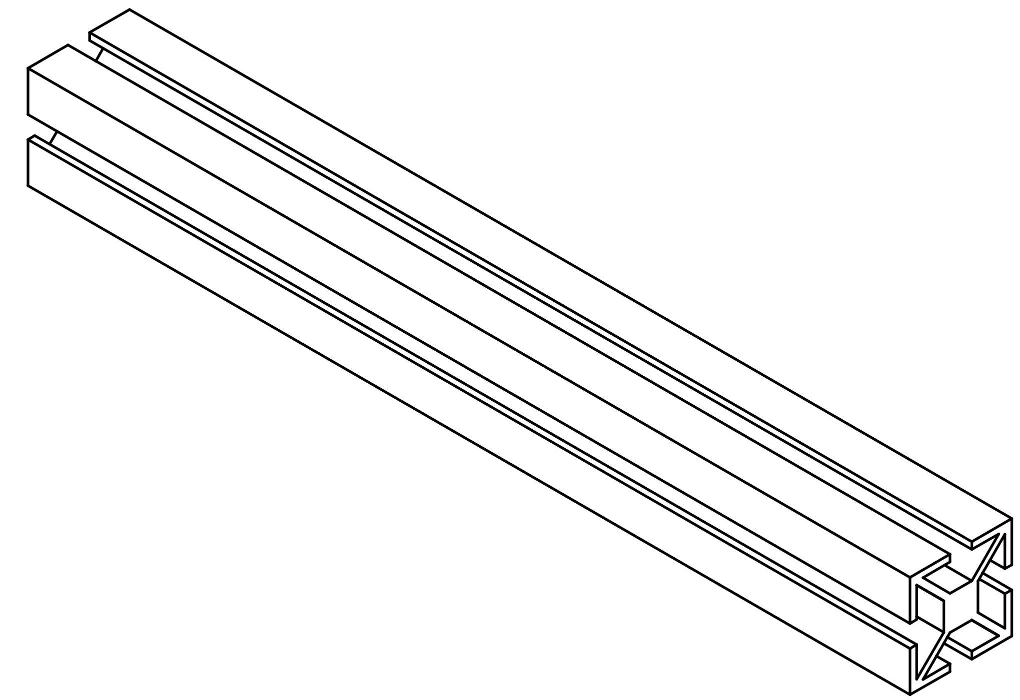
FINISH
32

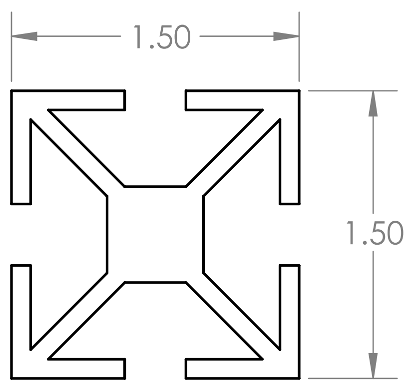
NEXT ASB. NO.: 10100

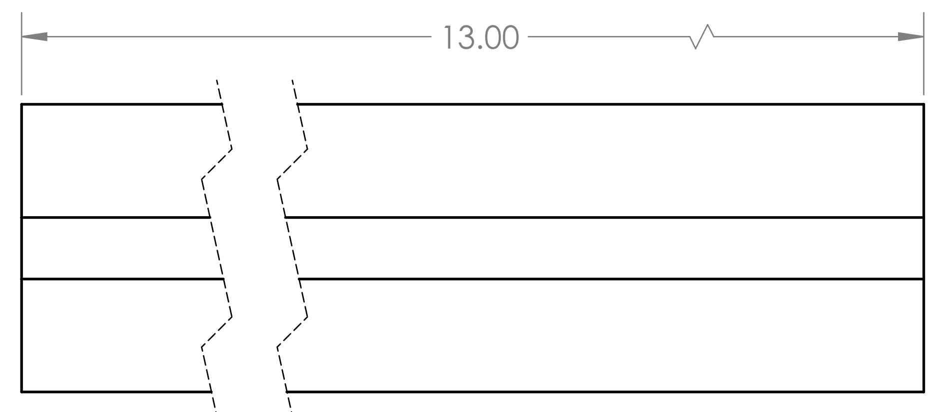
TITLE:
31 IN 80/20

SIZE B | DWG. NO. 10110 | REV

SCALE: 1:10 | WEIGHT: | SHEET 1 OF 1







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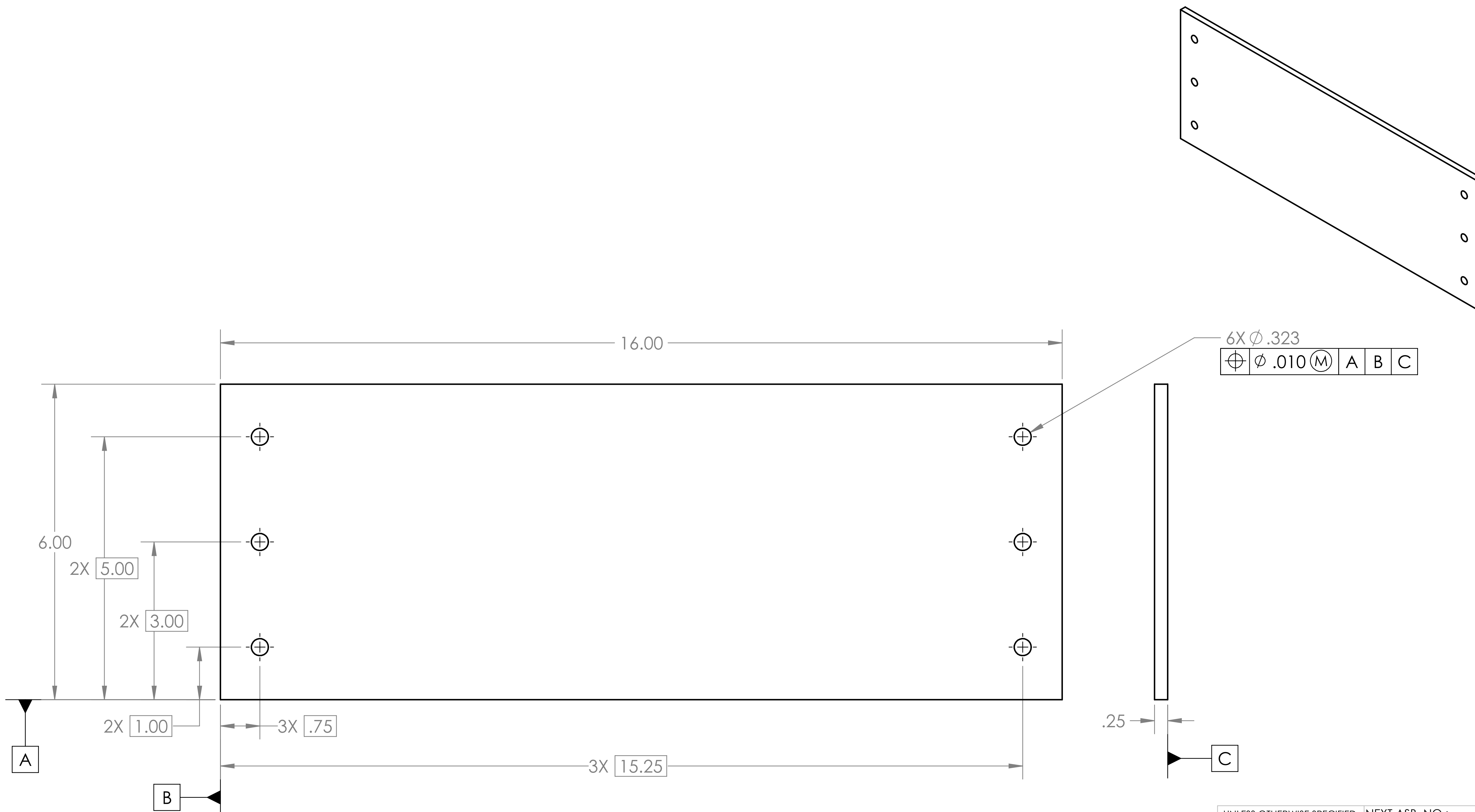
UNLESS OTHERWISE SPECIFIED:	NEXT ASB. NO.: 10100	
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ±1/100 ANGULAR: MACH ±1° BEND ±2° TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.005	TITLE: 13 IN 80/20	
INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5		
MATERIAL 6061 ALUMINUM	SIZE B	DWG. NO. 10120 REV
FINISH 32	SCALE: 1:5 WEIGHT:	SHEET 1 OF 1

PULP FRICTION



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UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 1/100
ANGULAR: MACH ± 1° BEND ± 2°
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005

INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5

MATERIAL: 6061 ALUMINUM

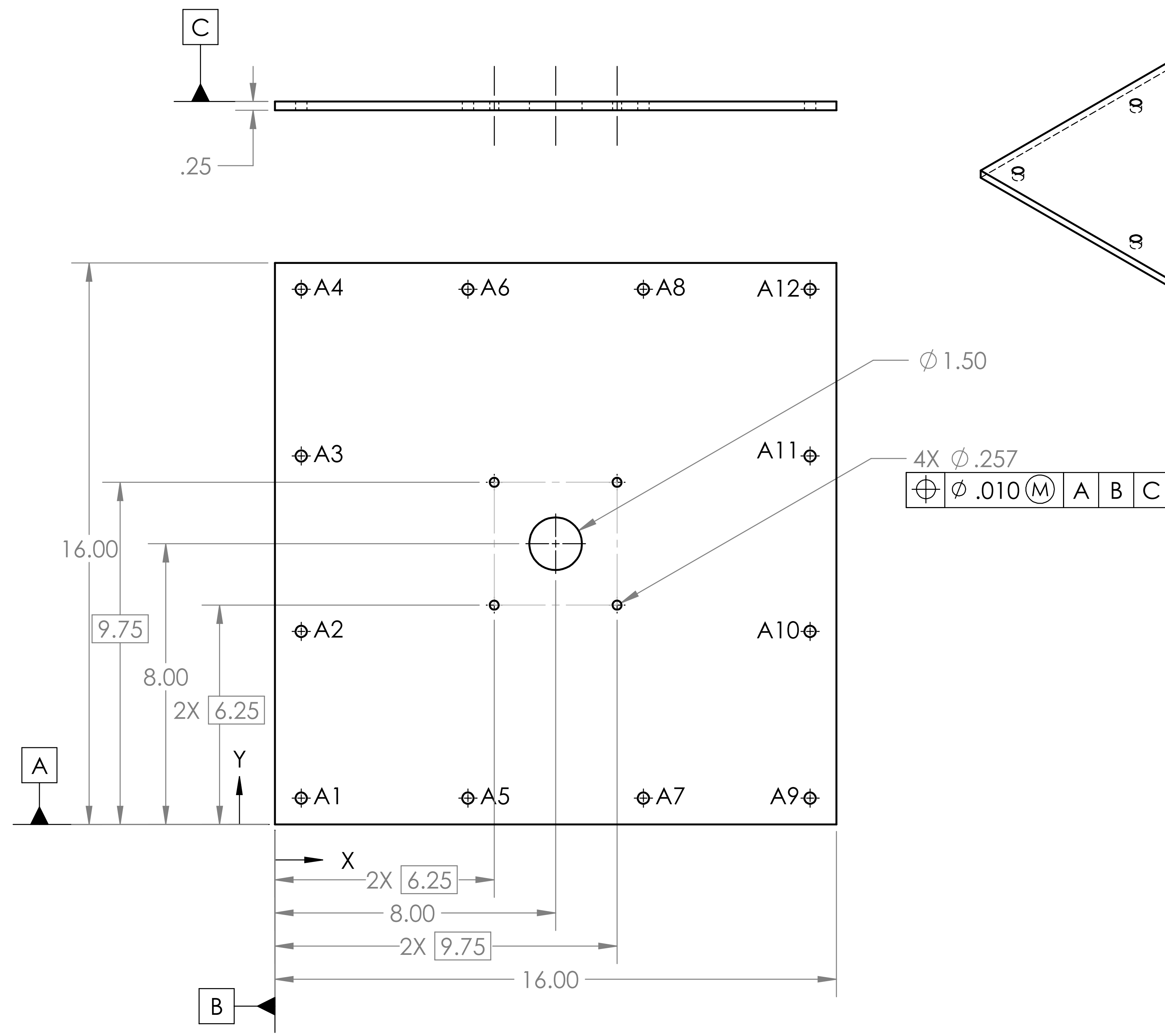
FINISH: 32

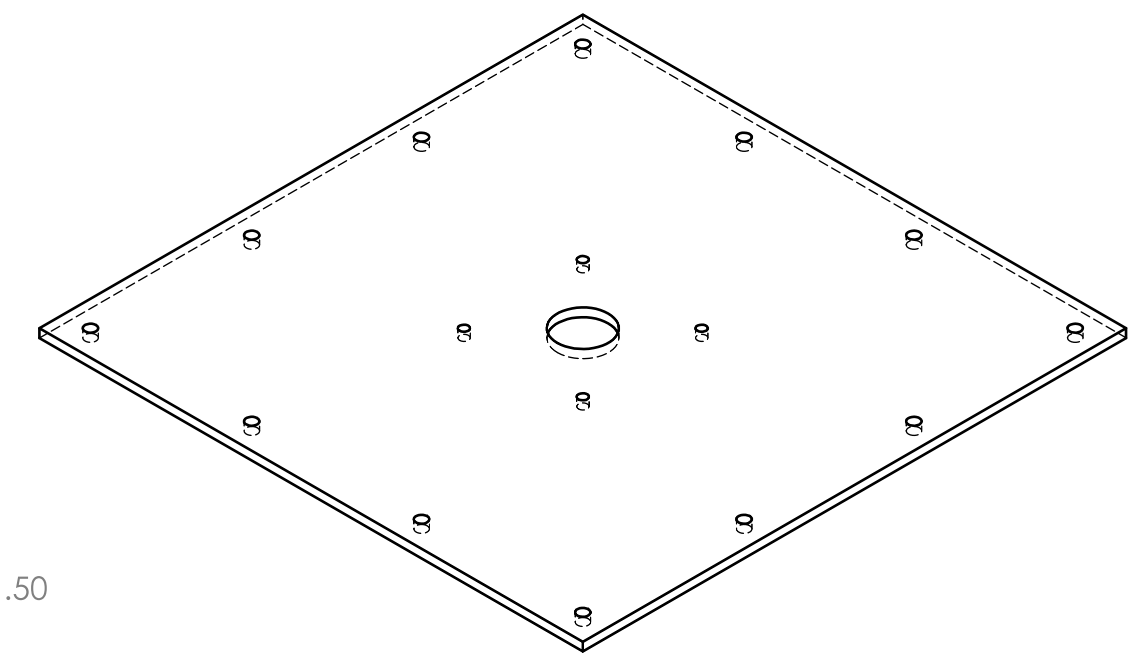
NEXT ASB. NO.: 10100

TITLE: SIDE PLATE

SIZE: B | DWG. NO.: 10150 | REV

SCALE: 1:2 | WEIGHT: | SHEET 1 OF 1





TAG	X LOC	Y LOC	SIZE
A1	.75	.75	∅ .323 THRU
A2	.75	5.50	∅ .323 THRU
A3	.75	10.50	∅ .323 THRU
A4	.75	15.25	∅ .323 THRU
A5	5.50	.75	∅ .323 THRU
A6	5.50	15.25	∅ .323 THRU
A7	10.50	.75	∅ .323 THRU
A8	10.50	15.25	∅ .323 THRU
A9	15.25	.75	∅ .323 THRU
A10	15.25	5.50	∅ .323 THRU
A11	15.25	10.50	∅ .323 THRU
A12	15.25	15.25	∅ .323 THRU

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 1/100
ANGULAR: MACH ± 1° BEND ± 2°
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005

INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5

MATERIAL 6061 ALUMINUM

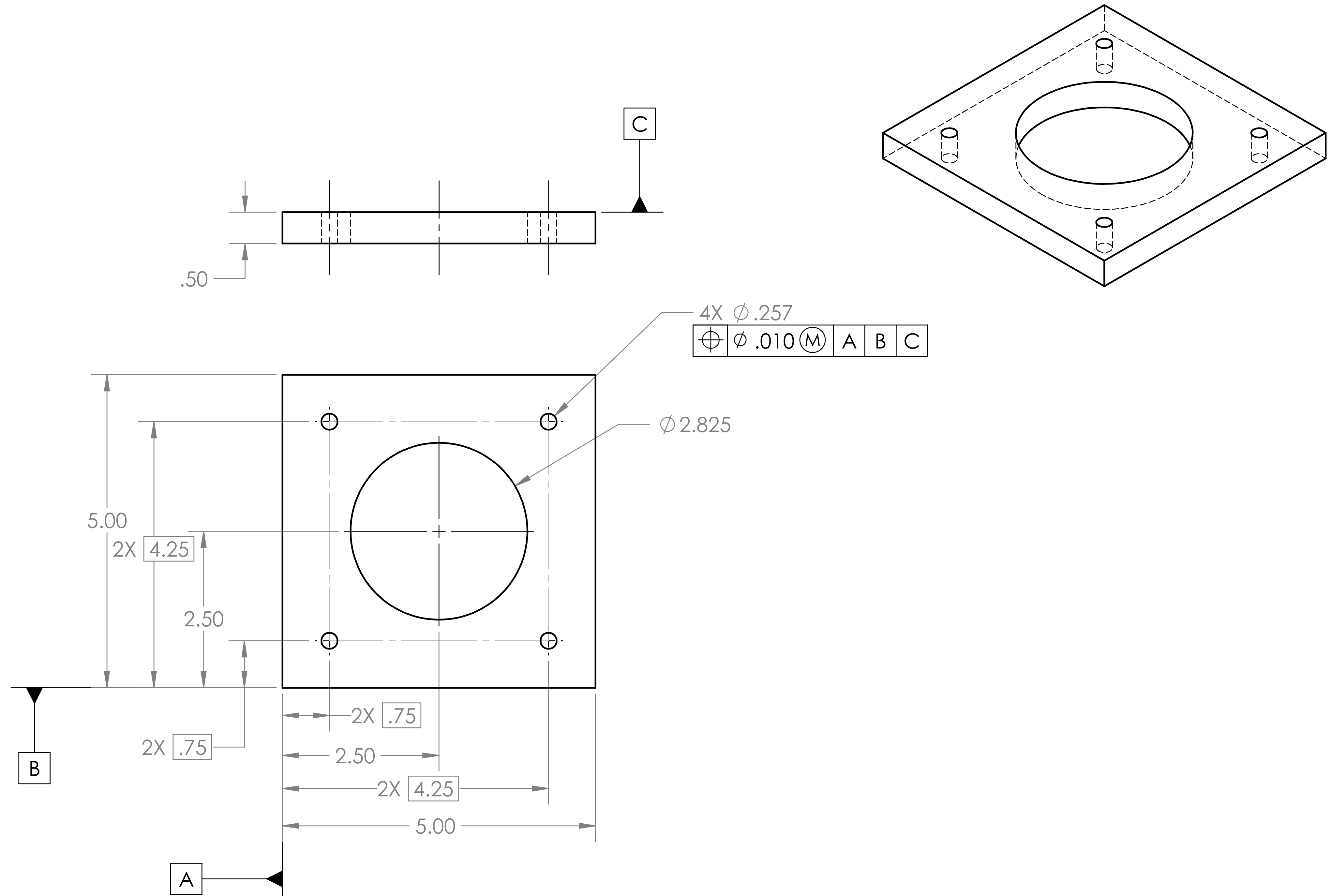
FINISH 32

NEXT ASB. NO.: 10100

TITLE: TOP PLATE

SIZE B | DWG. NO. 10160 | REV

SCALE: 1:5 | WEIGHT: | SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED:	NEXT ASB. NO.: 10100
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ±1/100 ANGULAR: MACH ±1° BEND ±2° TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.005	TITLE: BEARING PLATE
INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5	
MATERIAL 6061 ALUMINUM	SIZE B · DWG. NO. 10170 · REV
FINISH 32	SCALE: 1:2 · WEIGHT: · SHEET 1 OF 1

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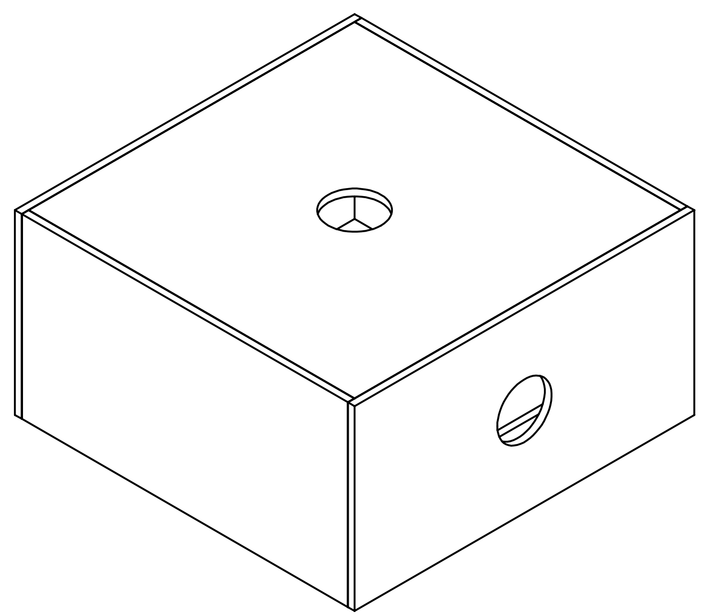
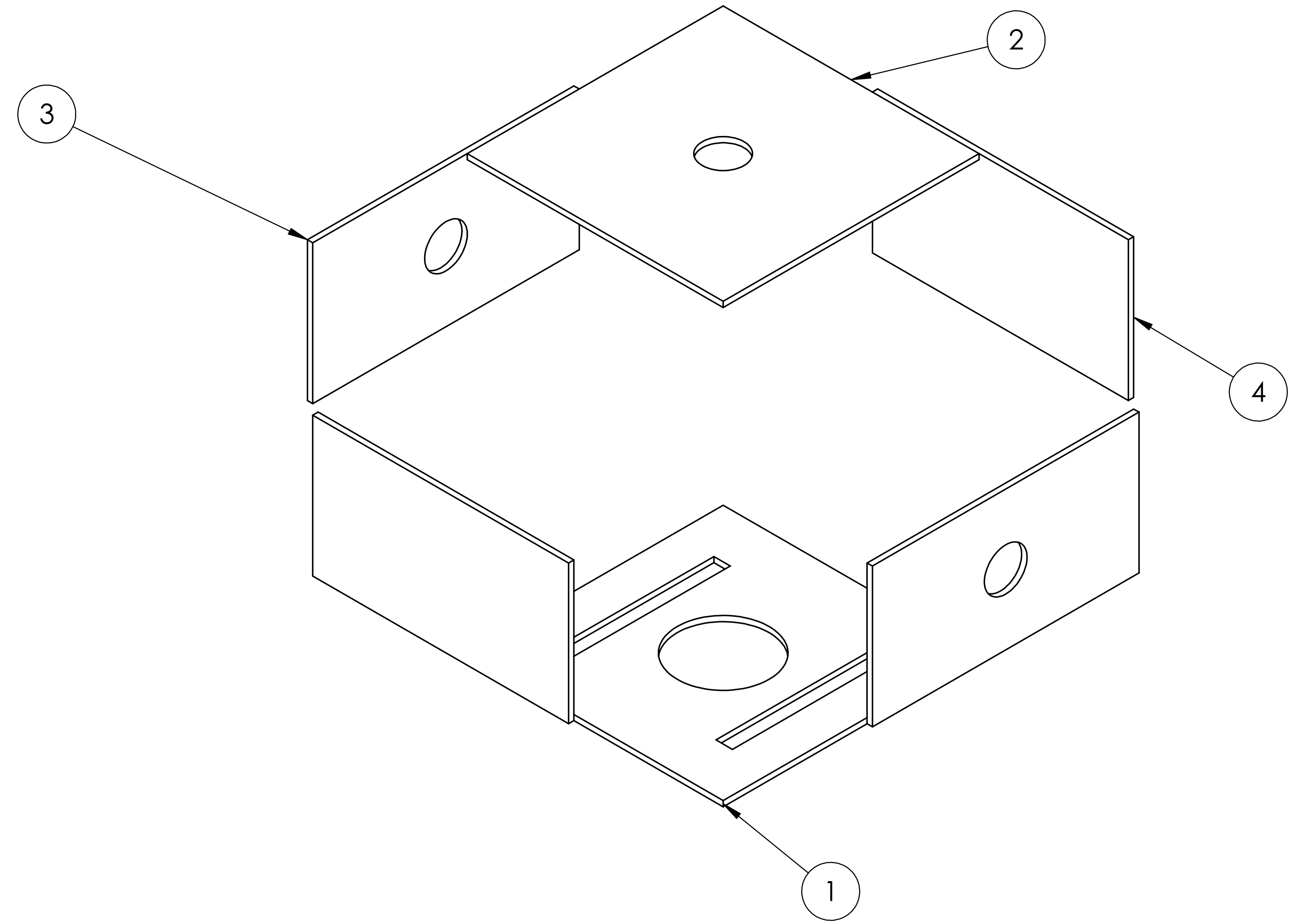
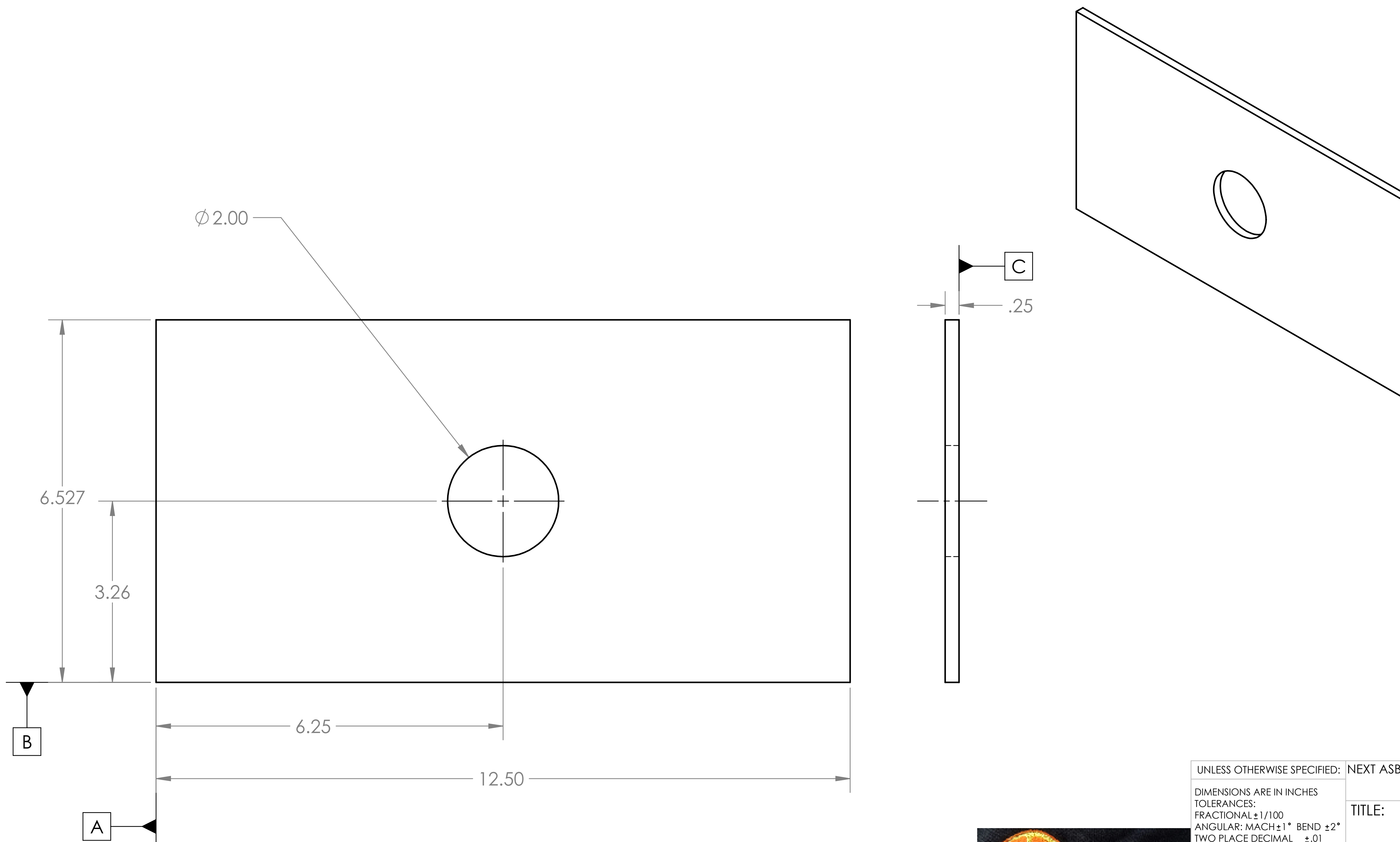




ITEM NO.	PART NUMBER	TITLE	QTY.
1	10240	BTM PLATE	1
2	10230	TOP PLATE	1
3	10210	SIDE PLATE	2
4	10220	SIDE PLATE	2

UNLESS OTHERWISE SPECIFIED:	NEXT ASB. NO.: 10000	
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ±1/100 ANGULAR: MACH ±1° BEND ±2° TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.005	TITLE: THERMAL CHAMBER	
INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5		
MATERIAL	SIZE B	DWG. NO. 10200	REV
FINISH	SCALE: 1:10 WEIGHT:	SHEET 1 OF 1



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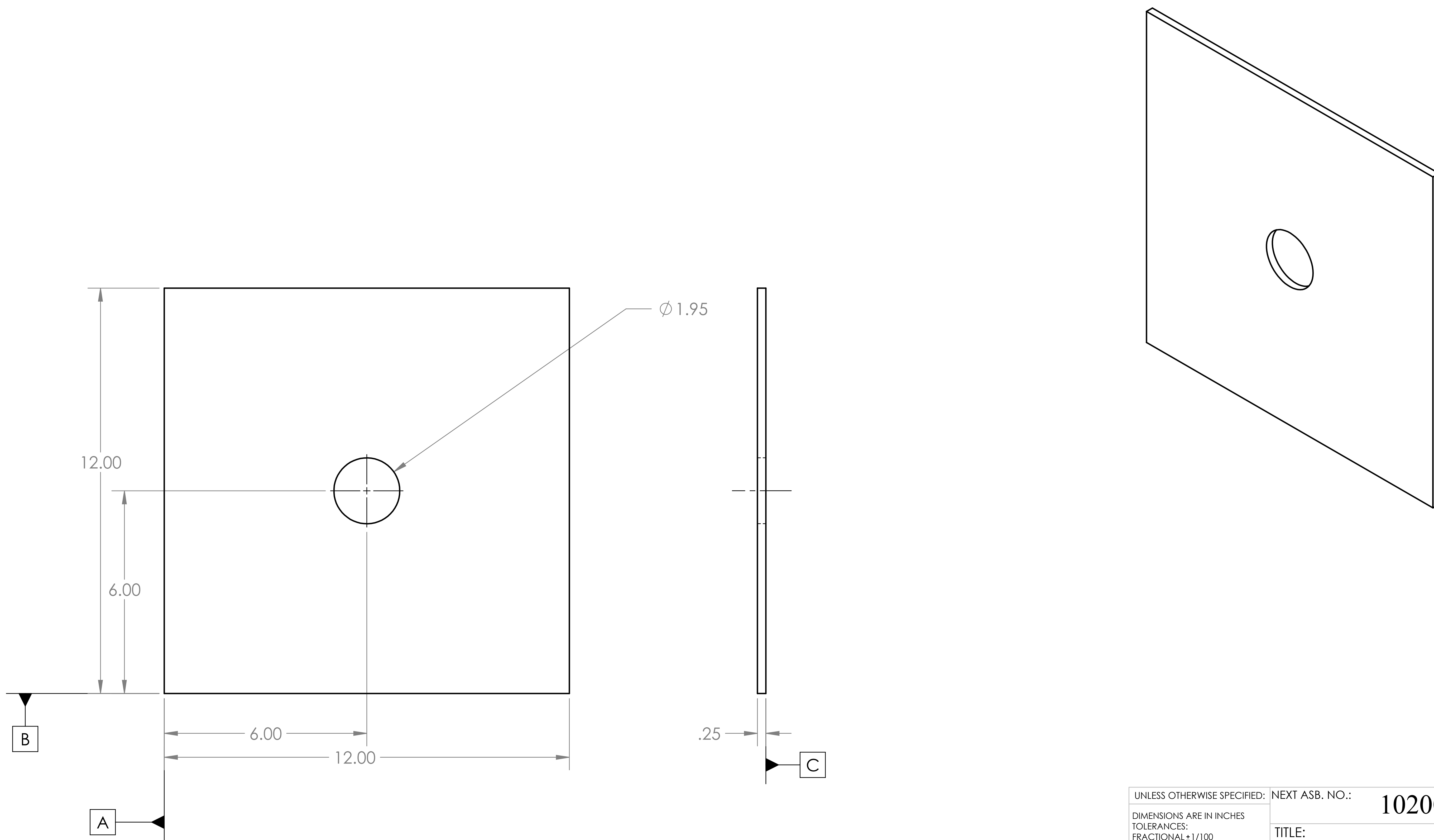
UNLESS OTHERWISE SPECIFIED:	NEXT ASB. NO.: 10200
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ±1/100 ANGULAR: MACH ±1° BEND ±2° TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.005	TITLE: SIDE PLATE
INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5	
MATERIAL PLYWOOD	SIZE B · DWG. NO. 10210 · REV
FINISH	SCALE: 1:2 · WEIGHT: · SHEET 1 OF 1

UNLESS OTHERWISE SPECIFIED:	NEXT ASB. NO.: 10220	
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ±1/100 ANGULAR: MACH ±1° BEND ±2° TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.005	TITLE: SIDE PLATE	
INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5		
MATERIAL PLYWOOD	SIZE B	DWG. NO. 10220 · REV
FINISH	SCALE: 1:2 · WEIGHT:	SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED:	NEXT ASB. NO.: 10200
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL±1/100 ANGULAR: MACH±1° BEND ±2° TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.005	TITLE: TOP PLATE
INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5	
MATERIAL PLYWOOD	SIZE B DWG. NO. 10230 REV
FINISH	SCALE: 1:5 WEIGHT: SHEET 1 OF 1

PULP FRICTION

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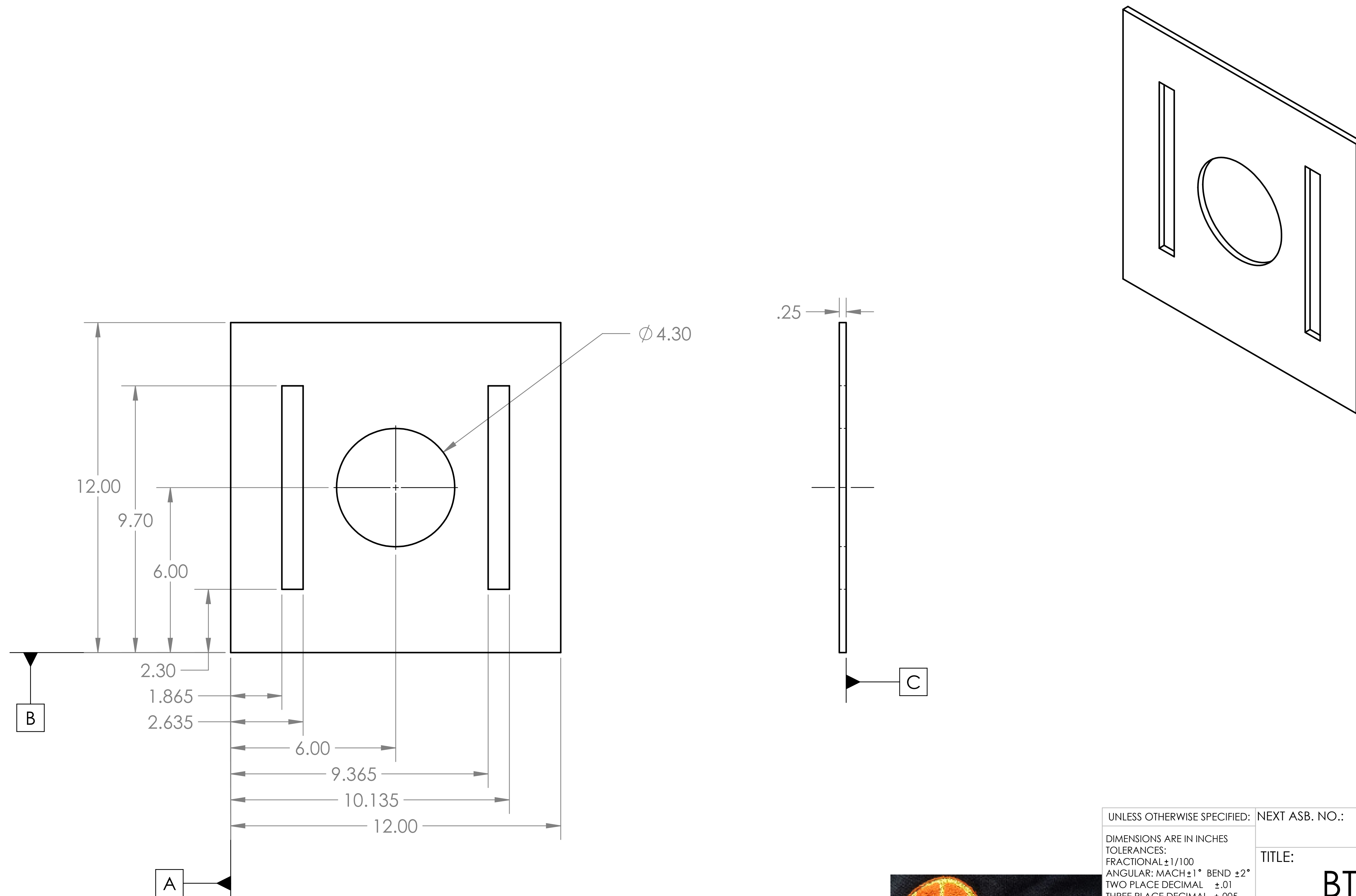

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UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±1/100
ANGULAR: MACH ±1° BEND ±2°
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5

MATERIAL: PLYWOOD

FINISH

NEXT ASB. NO.: 10240

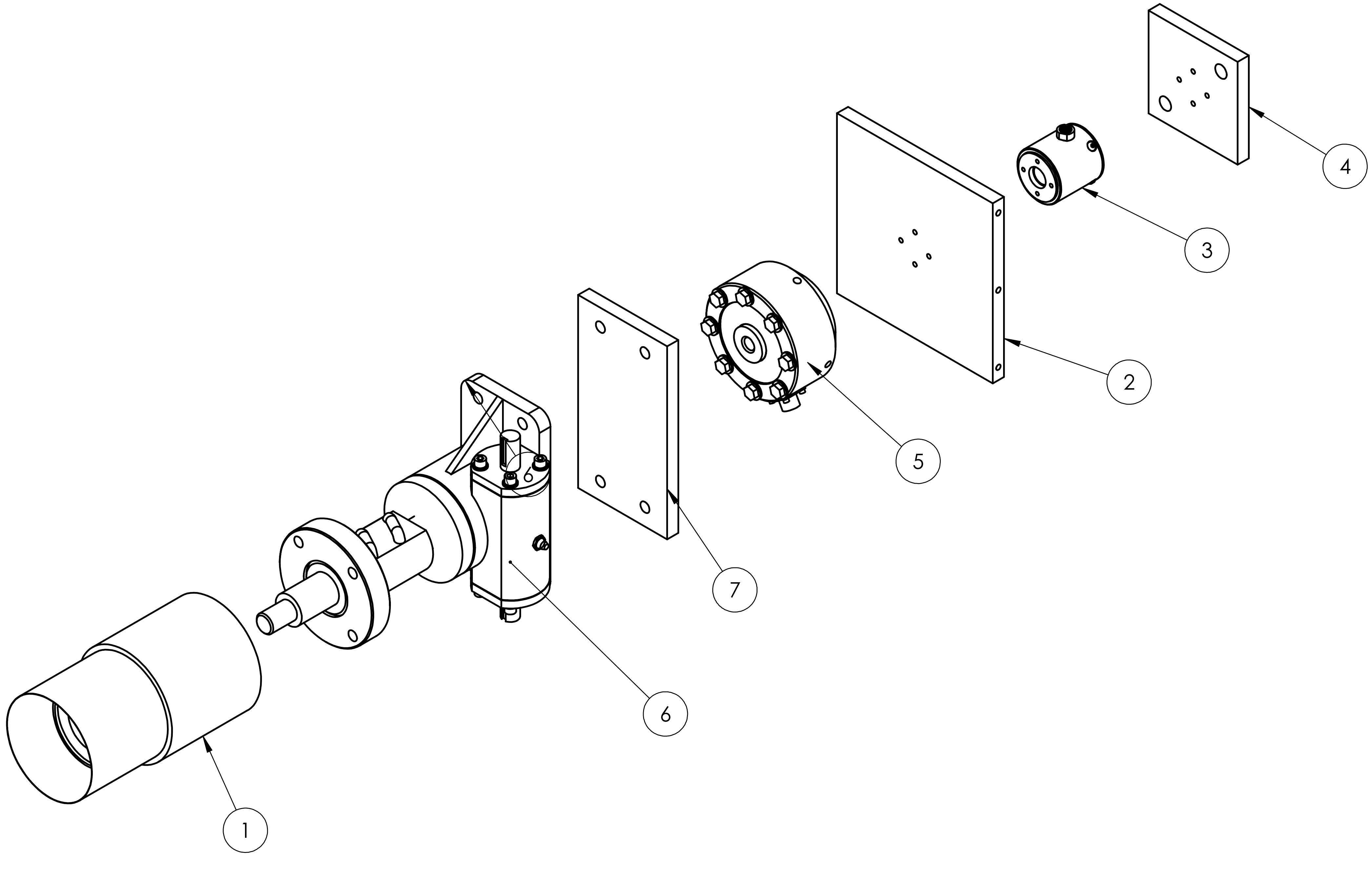
TITLE: BTM PLATE

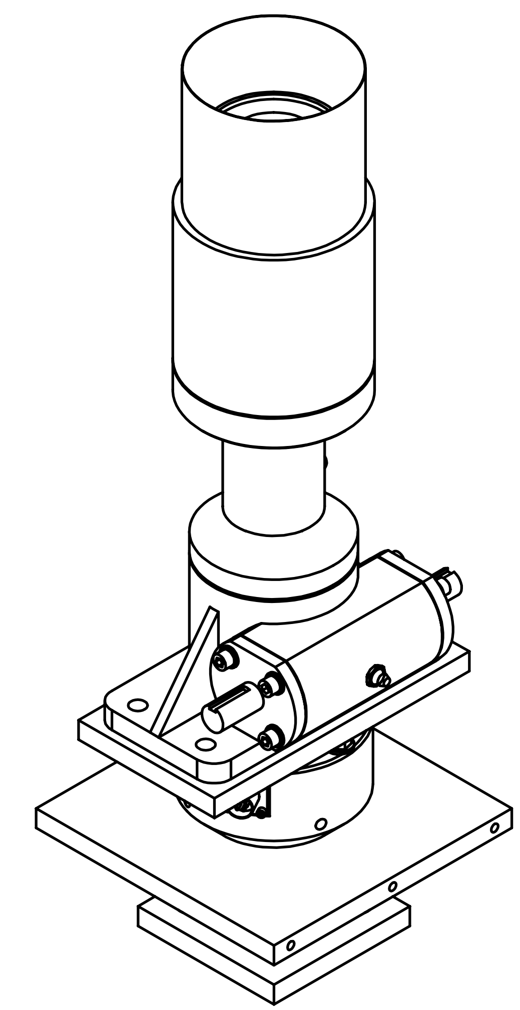
SIZE: B
DWG. NO. 10240
REV

SCALE: 1:5 WEIGHT: SHEET 1 OF 1

4 3 2 1

B





B

A

ITEM NO.	PART NUMBER	TITLE	QTY.
1	10310	OIL BRACKET	1
2	10350	BTM BRACKET	1
3	10360	TORQUE CELL	1
4	10370	SLIDING PLATE TOP	1
5	10340	LOAD CELL	1
6	10320	SCREW JACK	1
7	10330	LOAD CELL PLATE	1
8	-	3/8-24 1.5" HEX SCREW	8
9	-	3/8-24 HEX NUT	4
10	-	3/8" WASHER	12
11	-	8-32 0.75" HEX SCREW	8
12	-	#8 WASHER	8

PULP FRICTION

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±1/100
ANGULAR: MACH ±1° BEND ±2°
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5

MATERIAL

FINISH

NEXT ASB. NO.: 10000

TITLE:

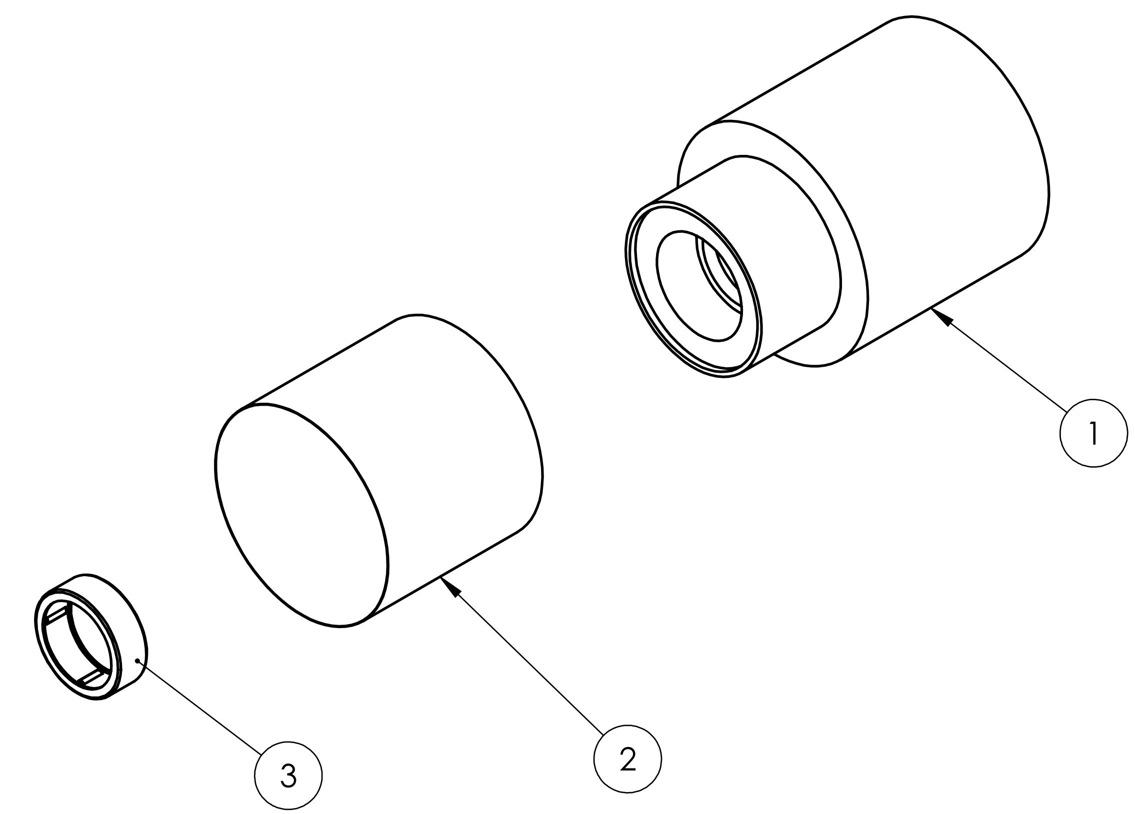
LOAD LOOP BTM

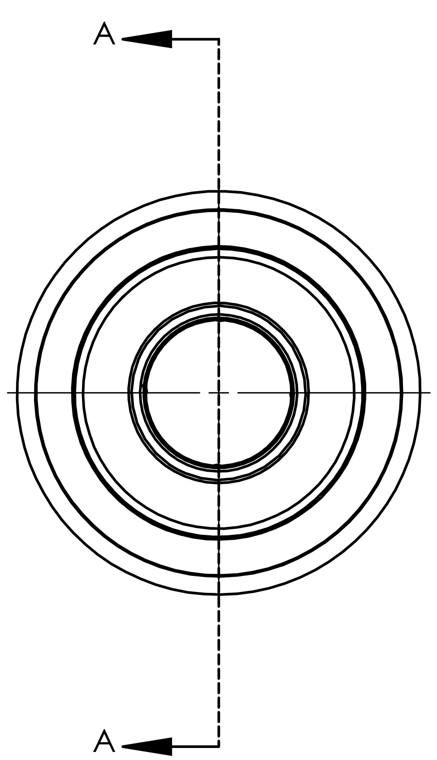
SIZE B | DWG. NO. 10300 | REV

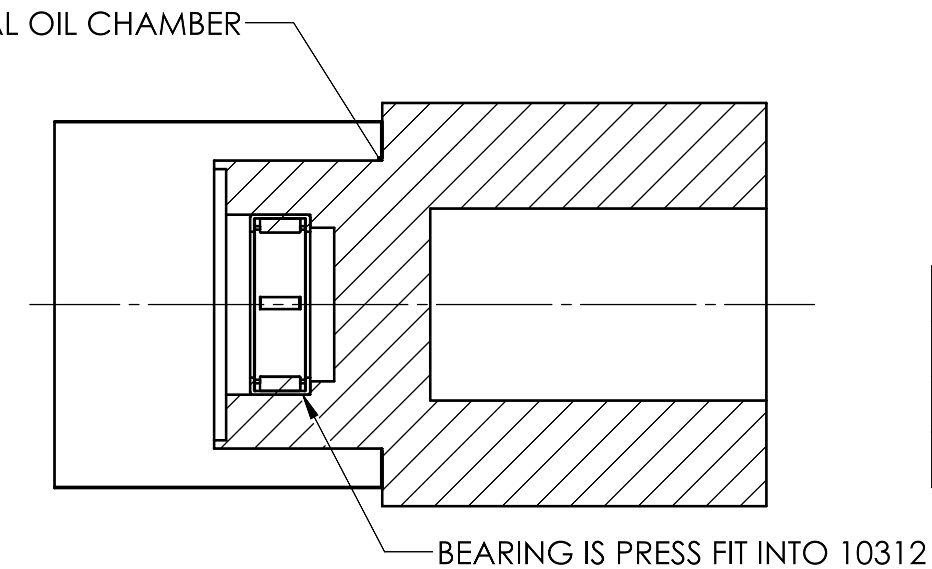
SCALE: 1:10 | WEIGHT: | SHEET 1 OF 1

A

4 3 2 1







SECTION A-A
SCALE 1 : 2

ITEM NO.	PART NUMBER	SW-Title(Title)	QTY.
1	10312	SKEW ROLLER BRACKET	1
2	10311	OIL CONTAINER	1
3	INA SCE2410	SHAFT BEARING	1

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±1/100
ANGULAR: MACH ±1° BEND ±2°
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC
TOLERANCING PER: ASME Y14.5

MATERIAL

FINISH

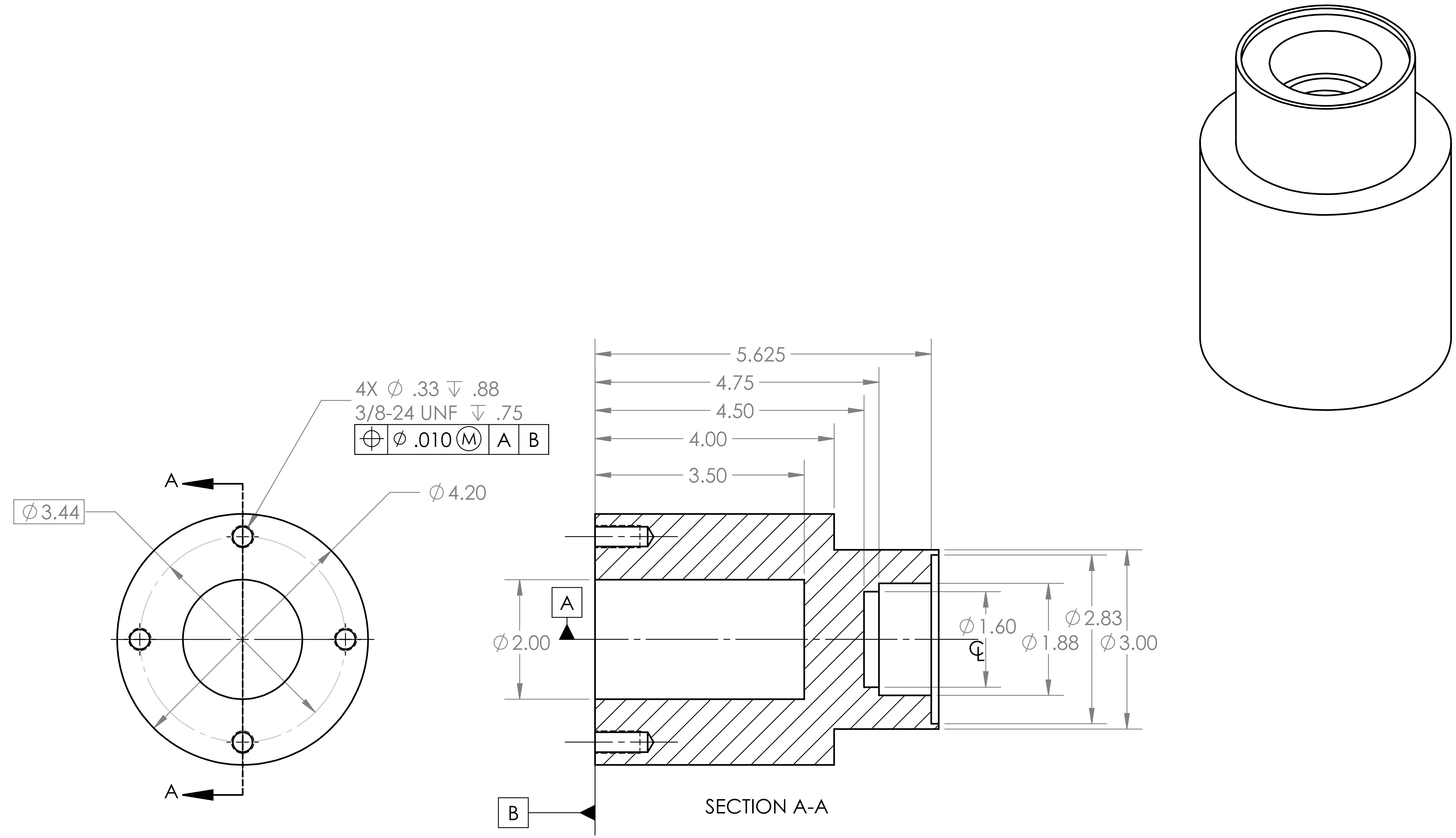
PULP FRICTION

NEXT ASB. NO.: 10300

TITLE:
OIL BRACKET

SIZE **B** DWG. NO. 10310 REV

SCALE: 1:5 WEIGHT: SHEET 1 OF 1



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UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±1/100
ANGULAR: MACH ±1° BEND ±2°
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5

MATERIAL
6061 ALUMINUM

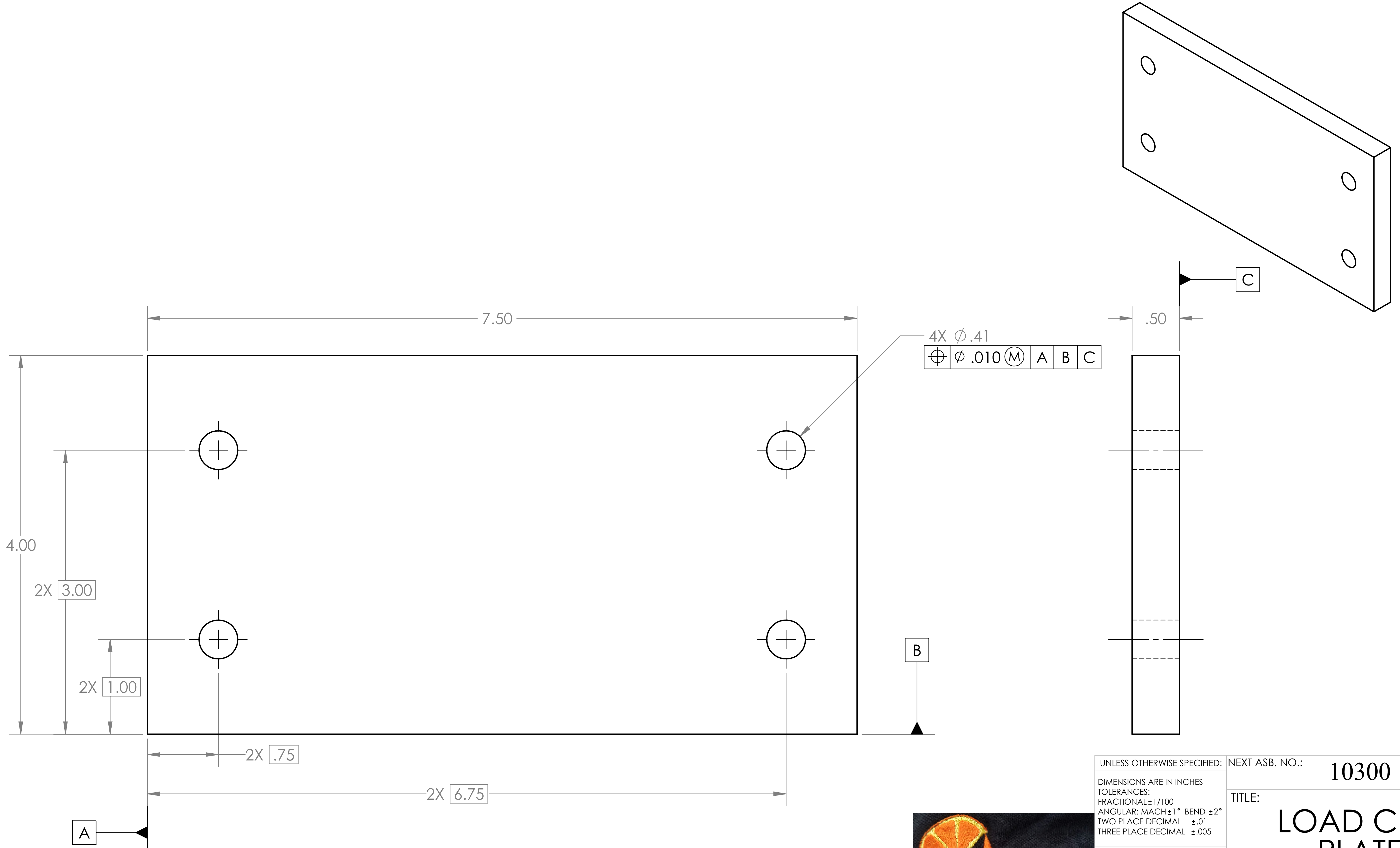
FINISH
32

NEXT ASB. NO.: 10310

TITLE:
OIL BRACKET

SIZE **B** DWG. NO. 10312 REV

SCALE: 1:2 WEIGHT: SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±1/100
ANGULAR: MACH ±1° BEND ±2°
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5

MATERIAL: 6061 ALUMINUM

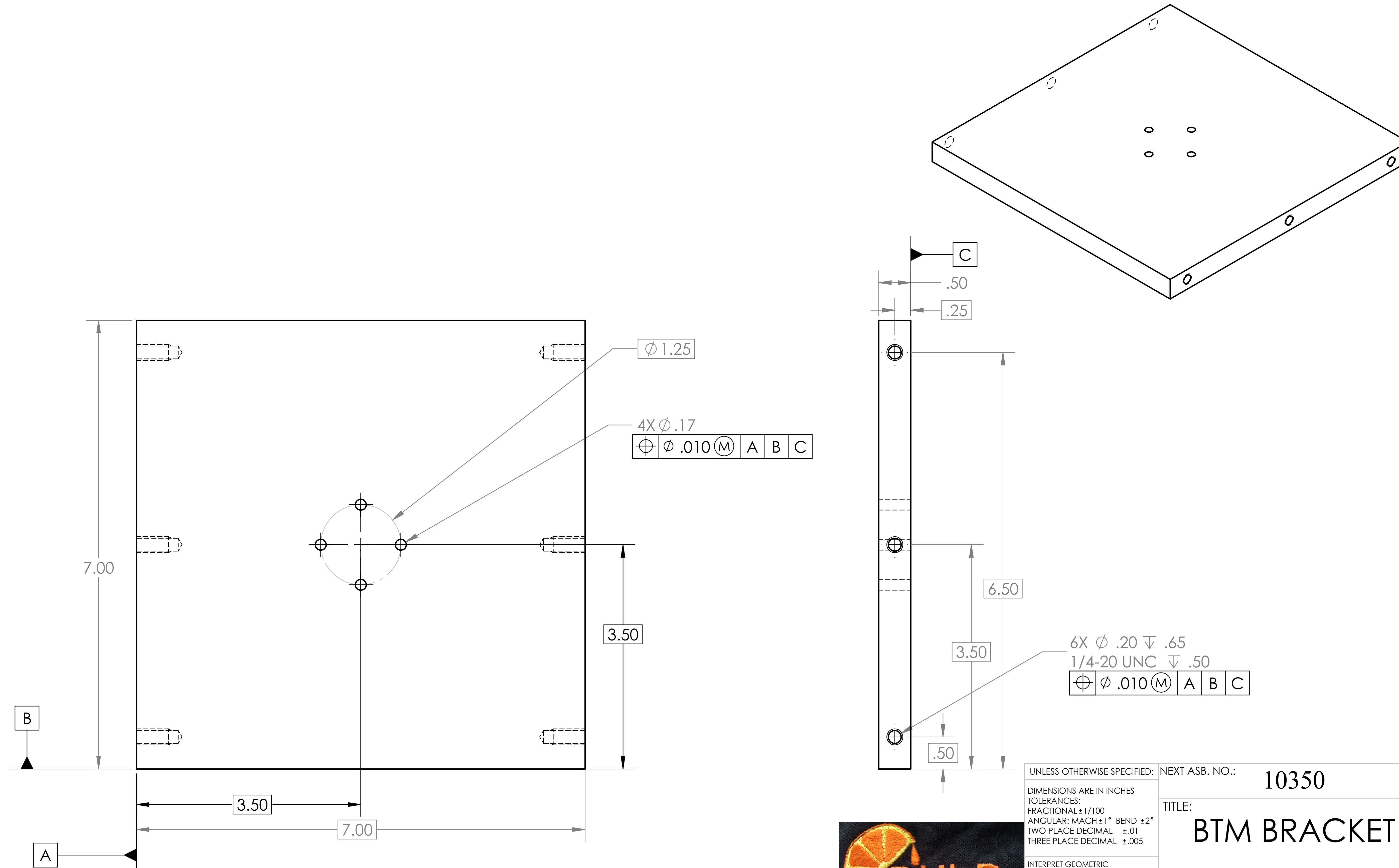
FINISH: 32

NEXT ASB. NO.: 10300

TITLE: LOAD CELL PLATE

SIZE: B | DWG. NO.: 10330 | REV:

SCALE: 1:1 | WEIGHT: | SHEET 1 OF 1



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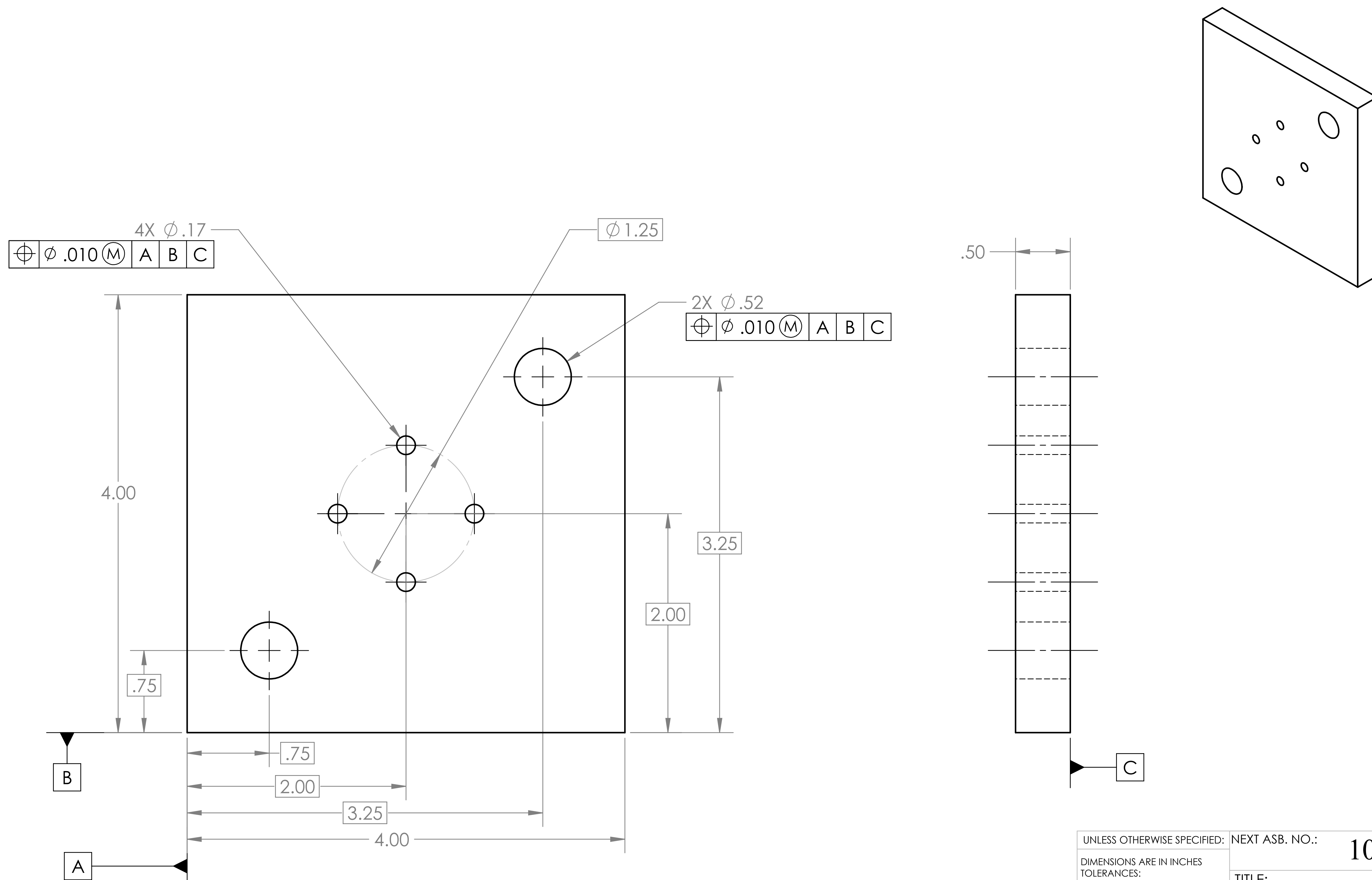
UNLESS OTHERWISE SPECIFIED:	NEXT ASB. NO.: 10350	
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ±1/100 ANGULAR: MACH ±1° BEND ±2° TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.005	TITLE: BTM BRACKET	
INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5		
MATERIAL 6061 ALUMINUM	SIZE B	DWG. NO. 10350 / REV
FINISH 32	SCALE: 1:2 WEIGHT:	SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±1/100
ANGULAR: MACH ±1° BEND ±2°
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5

MATERIAL: 6061 ALUMINUM

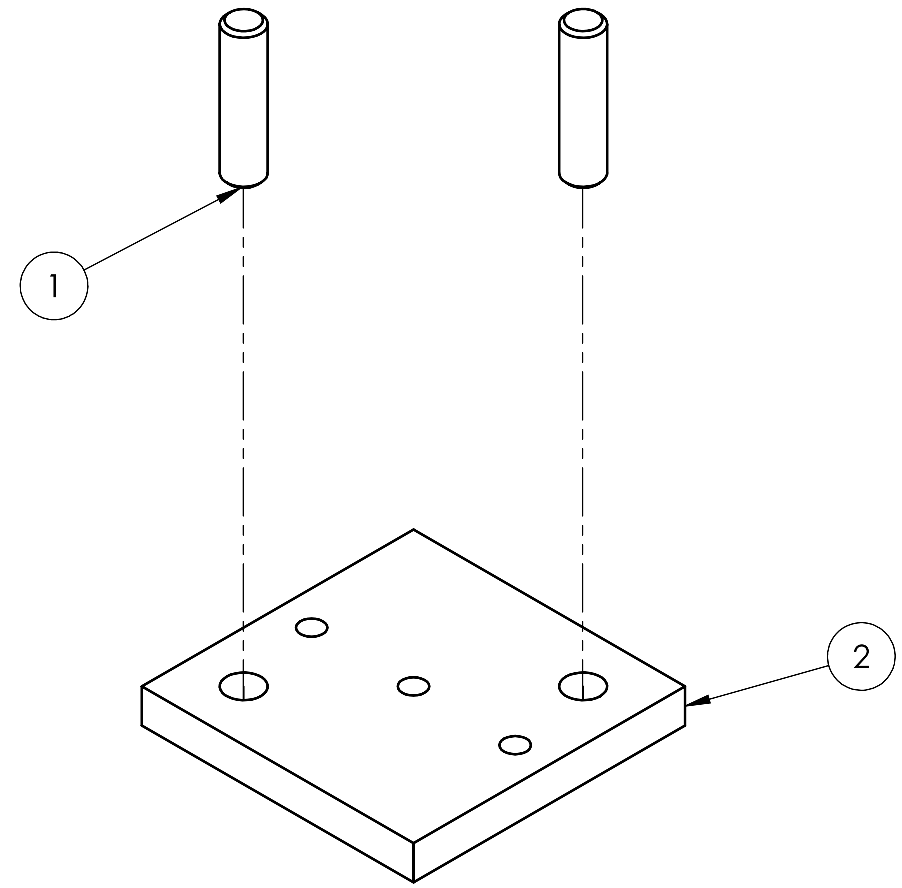
FINISH: 32

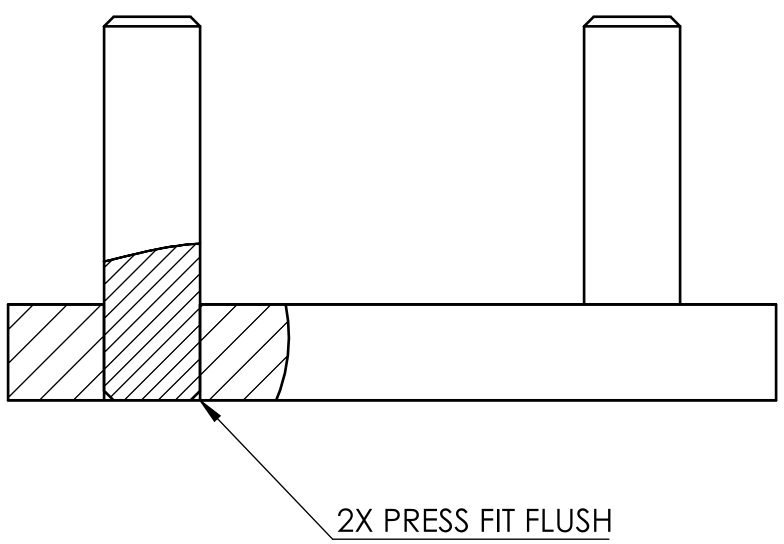
NEXT ASB. NO.: 10300

TITLE: SLIDING PLATE TOP

SIZE: B | DWG. NO. 10370 | REV

SCALE: 1:1 | WEIGHT: | SHEET 1 OF 1





ITEM NO.	PART NUMBER	TITLE	QTY.
1	10420	0.5", 2" DOWEL PIN	2
2	10410	SLIDING PLATE BTM	1

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±1/100
ANGULAR: MACH ±1° BEND ±2°
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC
TOLERANCING PER: ASME Y14.5

MATERIAL

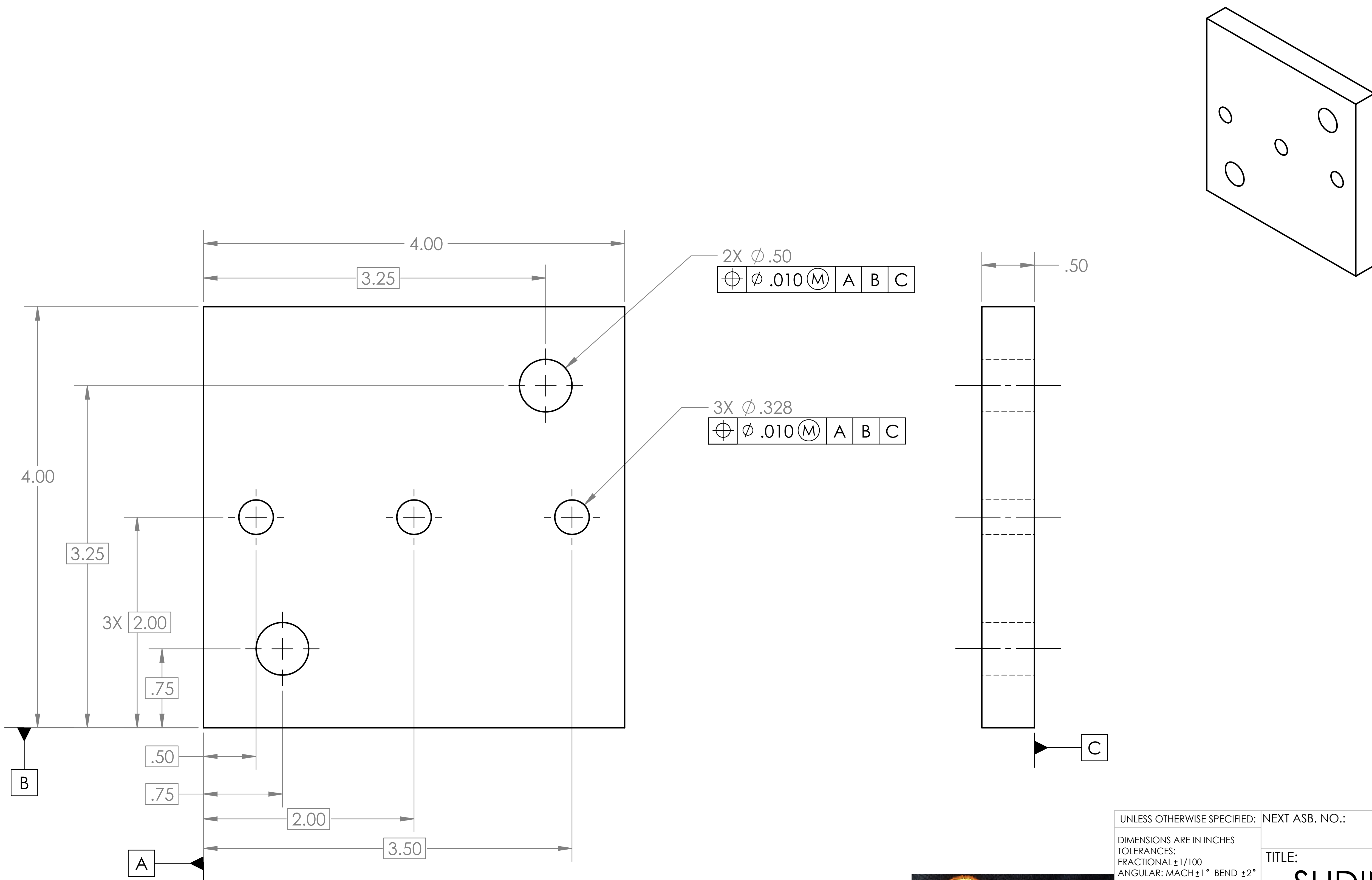
FINISH

NEXT ASB. NO.: 10000

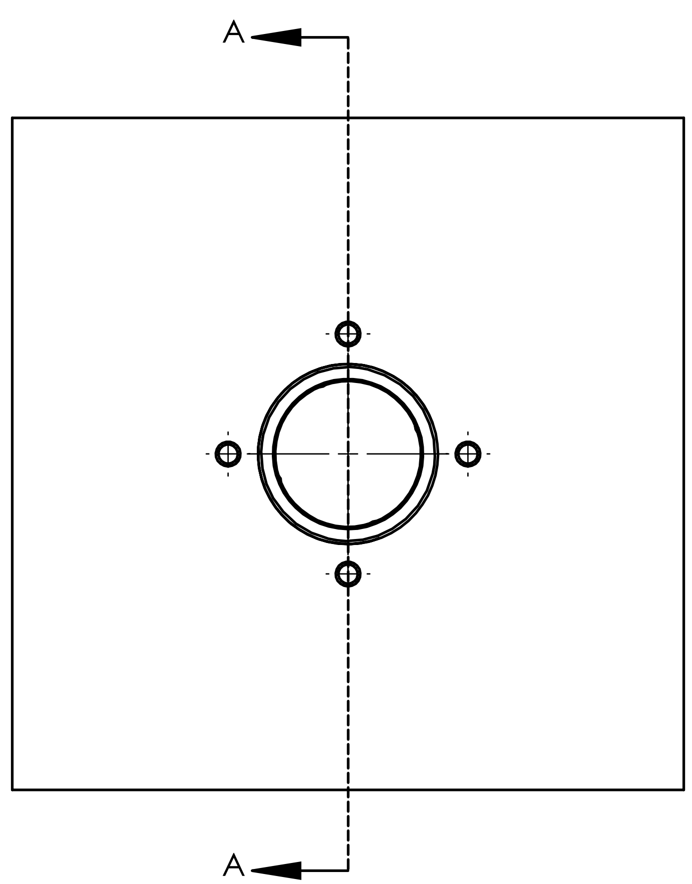
TITLE:
PINNED BRACKET

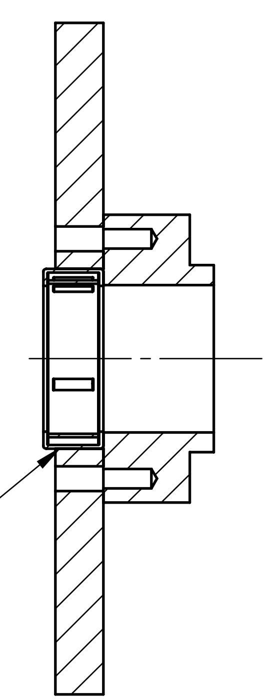
SIZE **B** DWG. NO. 10400 REV

SCALE: 1:1 WEIGHT: SHEET 1 OF 1



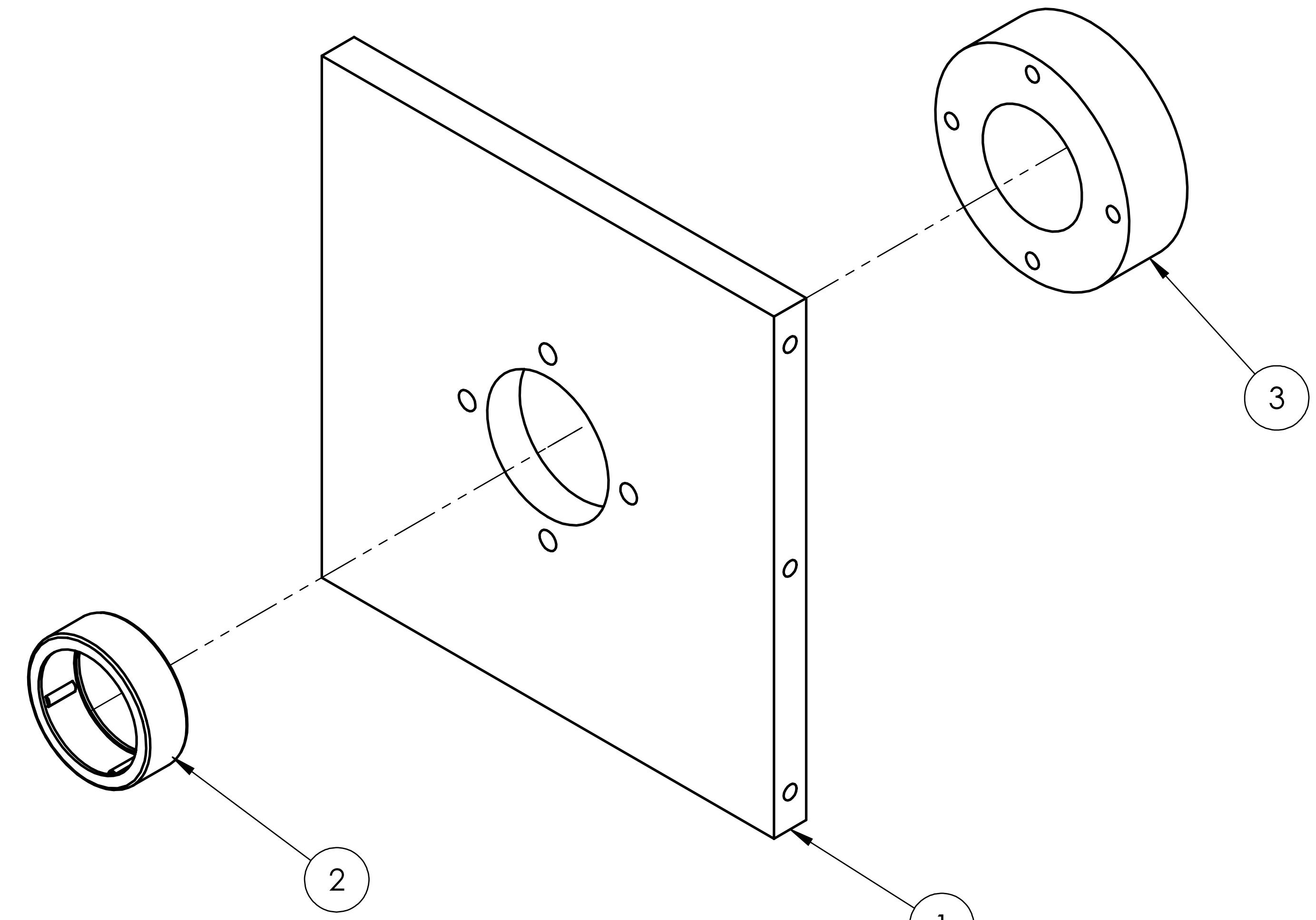
UNLESS OTHERWISE SPECIFIED:	NEXT ASB. NO.: 10400
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL±1/100 ANGULAR: MACH±1° BEND ±2° TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.005	TITLE: SLIDING PLATE BTM
INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5	
MATERIAL 6061 ALUMINUM	SIZE B DWG. NO. 10410 REV
FINISH 32	SCALE: 1:1 WEIGHT: SHEET 1 OF 1





PRESS FIT BEARING

SECTION A-A
SCALE 1 : 2



ITEM NO.	PART NUMBER	TITLE	QTY.
1	10520	LOAD LOOP BRACKET (B)	1
2	INA SCE2410	SHAFT BEARING	1
3	10510	LOAD LOOP BRACKET (A)	1
4	-	1/4-20, 1" HEX SCREW	4
5	-	1/4" WASHER	4

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±1/100
ANGULAR: MACH ±1° BEND ±2°
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC
TOLERANCING PER: ASME Y14.5

MATERIAL

FINISH

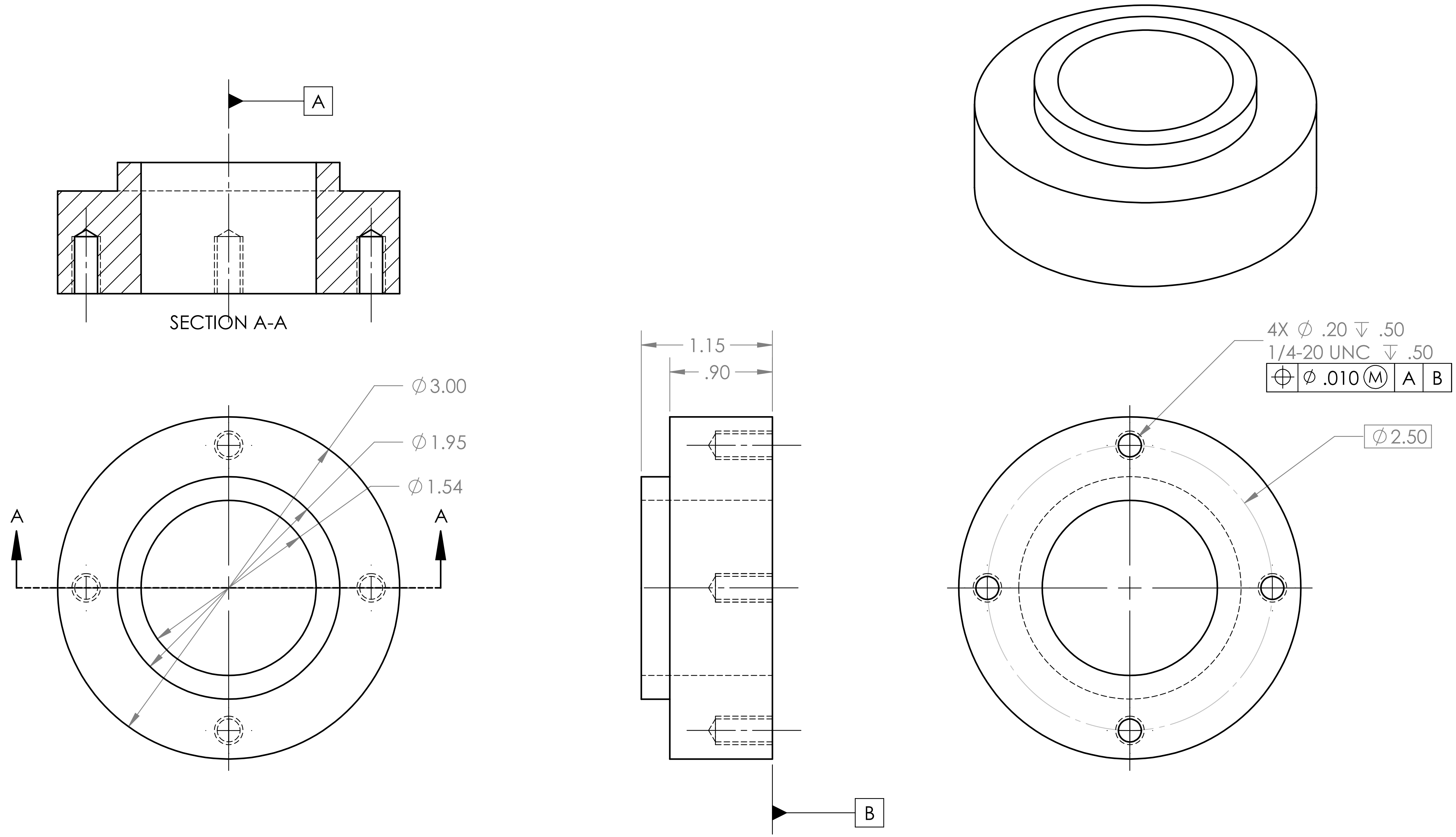
NEXT ASB. NO.: 10000

TITLE:
LOAD LOOP TOP

SIZE B | DWG. NO. 10500 | REV

SCALE: 1:5 | WEIGHT: | SHEET 1 OF 1

PULP FRICTION



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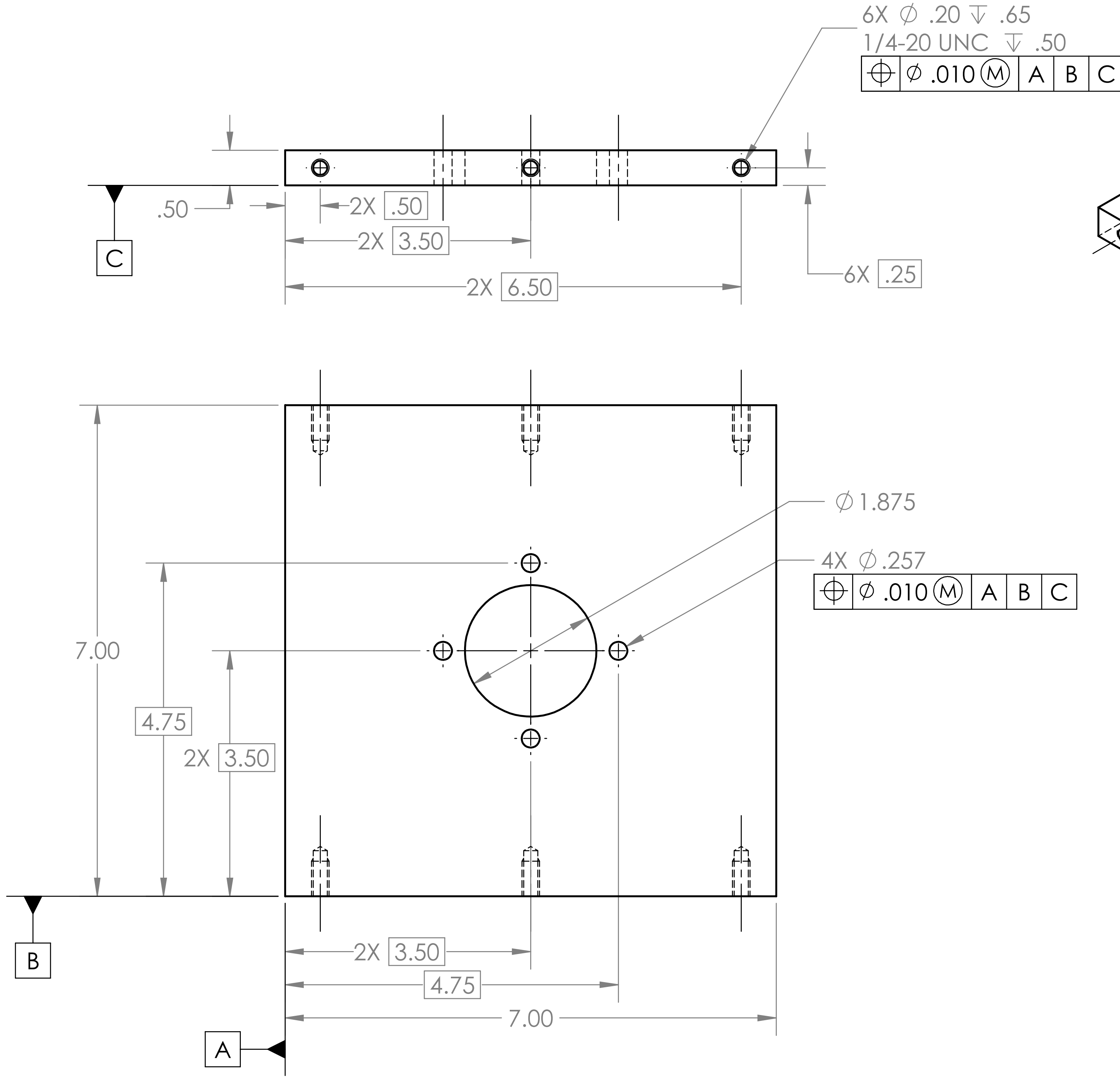
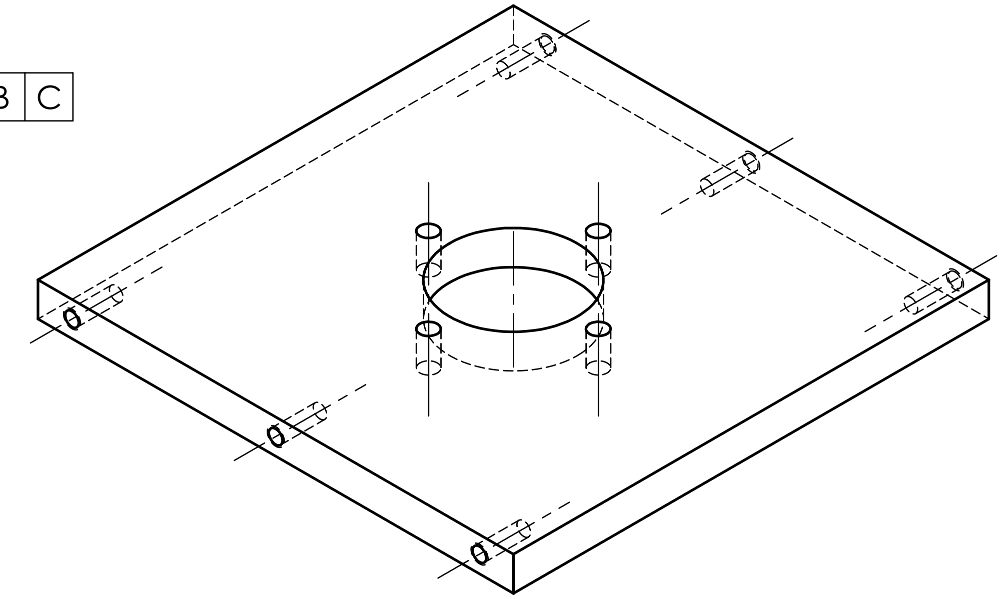
UNLESS OTHERWISE SPECIFIED:	NEXT ASB. NO.: 10500	
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ±1/100 ANGULAR: MACH ±1° BEND ±2° TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.005	TITLE: LOAD LOOP BRACKET (A)	
INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5		
MATERIAL 6061 ALUMINUM	SIZE B	DWG. NO. 10510 · REV
FINISH 32	SCALE: 1:1 · WEIGHT:	SHEET 1 OF 1





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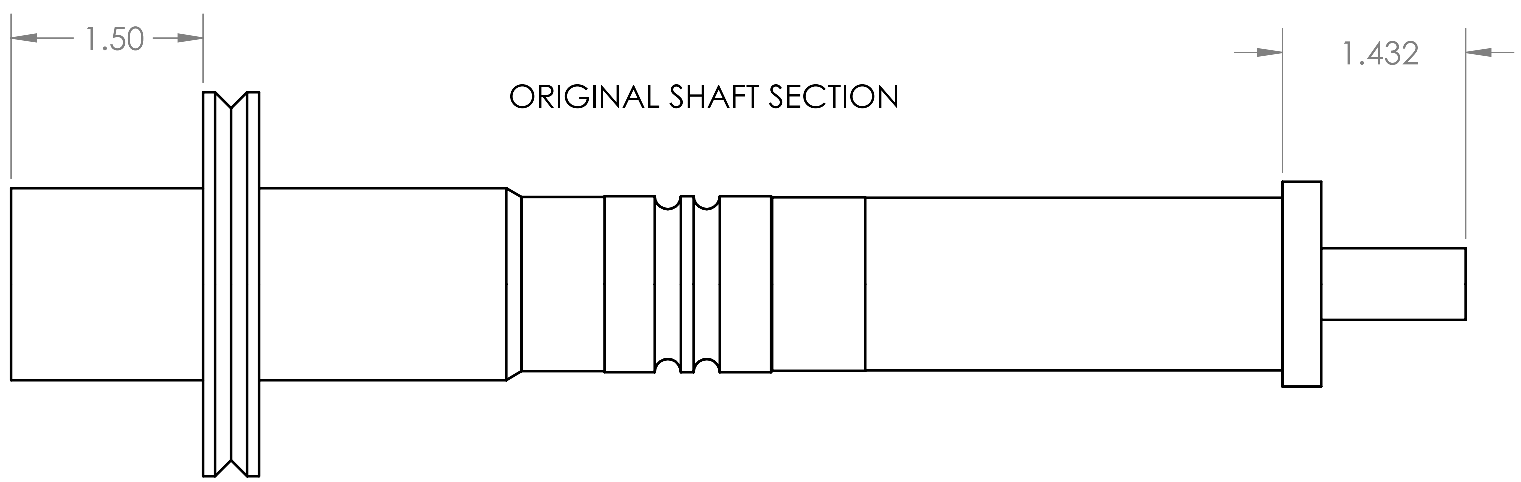
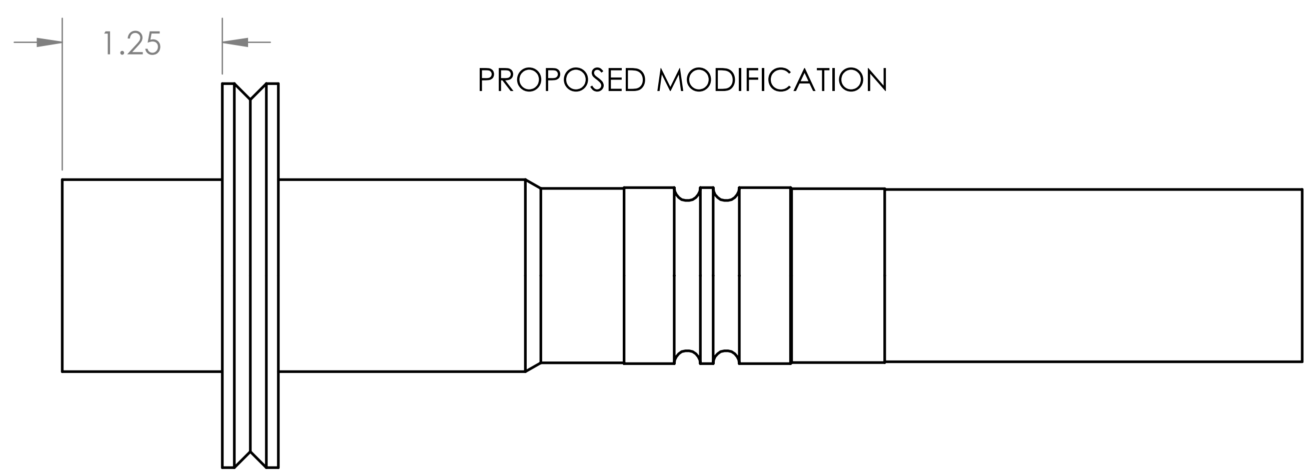
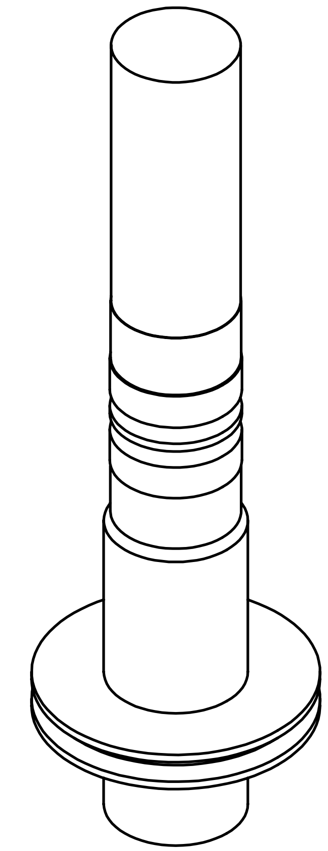
UNLESS OTHERWISE SPECIFIED:	NEXT ASB. NO.: 10500
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ±1/100 ANGULAR: MACH ±1° BEND ±2° TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.005	TITLE: LOAD LOOP BRACKET (B)
INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5	
MATERIAL 6061 ALUMINUM	SIZE B DWG. NO. 10520 REV
FINISH 32	SCALE: 1:2 WEIGHT: SHEET 1 OF 1



ORIGINAL SHAFT SECTION

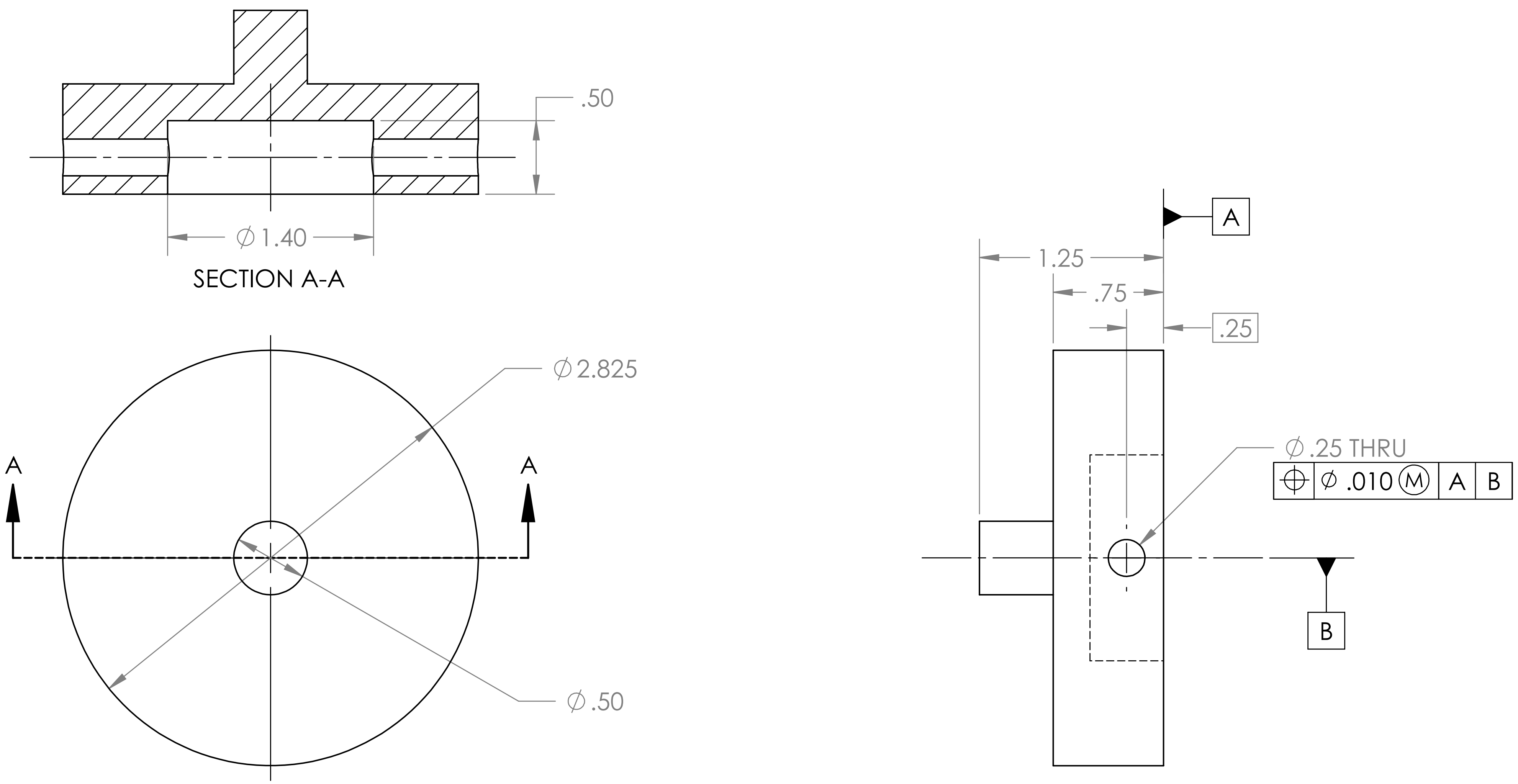


PROPOSED MODIFICATION

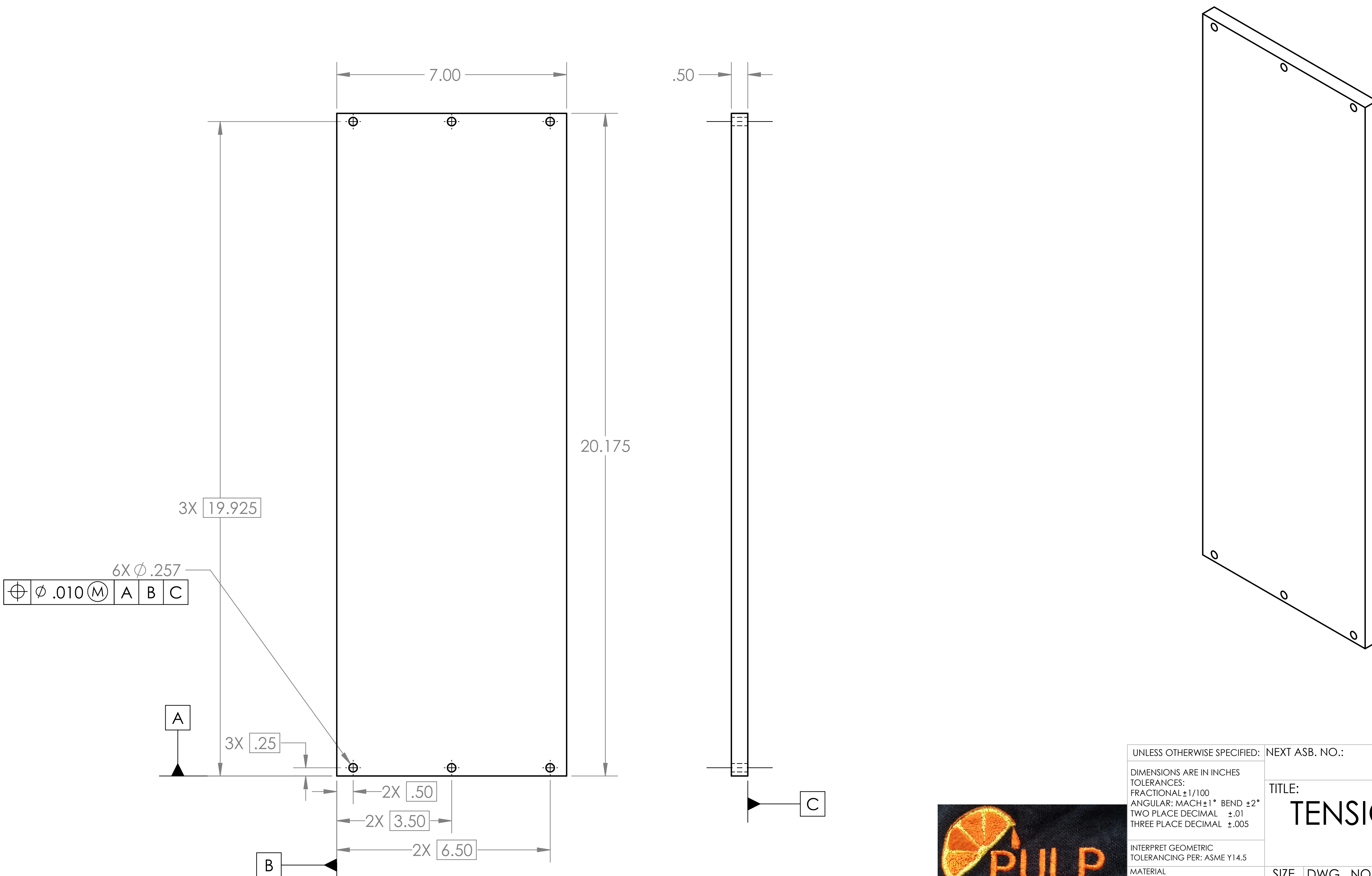


UNLESS OTHERWISE SPECIFIED:	NEXT ASB. NO.: 10600	
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ±1/100 ANGULAR: MACH ±1° BEND ±2° TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.005	TITLE: SHAFT	
INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5		
MATERIAL 6061 ALUMINUM	SIZE B / DWG. NO. 10600	REV
FINISH	SCALE: 1:2 / WEIGHT:	SHEET 1 OF 1

PULP FRICTION



UNLESS OTHERWISE SPECIFIED:	10000
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ±1/100 ANGULAR: MACH ±1° BEND ±2° TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.005	TITLE: TOP ATTACHMENT
INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5	
MATERIAL 6061 ALUMINUM	SIZE B · DWG. NO. 11000 · REV
FINISH 32	SCALE: 1:1 · WEIGHT: · SHEET 1 OF 1



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UNLESS OTHERWISE SPECIFIED:	NEXT ASB. NO.: 10000	
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ±1/100 ANGULAR: MACH ±1° BEND ±2° TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.005	TITLE: TENSION PLATE	
INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5		
MATERIAL 6061 ALUMINUM	SIZE B	DWG. NO. 11100
FINISH 32	REV	
	SCALE: 1:5	WEIGHT:
	SHEET 1 OF 1	