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Design of an Orientation Device for Valve Bodies with Blowout Compatibility for Slug Detection

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DESIGN OF AN ORIENTATION DEVICE FOR VALVE BODIES WITH BLOWOUT COMPATIBILITY FOR SLUG DETECTION

By

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A thesis submitted to the faculty of The University of Mississippi in partial fulfillment of the requirements of the Sally McDonnel Barksdale Honors College.

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DEDICATION

This thesis is dedicated Baxter Williams Barnes, who has supported me through thick and thin and has always been a tremendous resource for engineering problem solving, and to Jason Alexander, who portrayed George Costanza on *Seinfeld*, for giving me a role model I truly admire.

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ABSTRACT

The Parker Hannifin plant in Batesville, MS, is a manufacturing plant that specializes in A/C and hydraulic tube and hose assemblies that support heavy machinery in the agriculture, logistics and construction industries. Customers place unique orders where the tube size, bending configurations, and adaptors vary widely. Some tubes require valve bodies where the customer can charge the A/C system once it has been assembled. A pierce and press machine is used to pierce the tube and place the valve body in the hole. Many tubes have asymmetric blocks at the end of the tube, with valve bodies placed at customer-requested angles relative to these blocks. The current process to achieve these angles does not allow for accurate block-to-valve orientation. Parker Hannifin requested the design of a tool which would allow block-to-valve orientation at angles with a ±2° tolerance. An orientation device that is capable of transmitting angular rotation input to the block was developed. The orientation device uses a belt drive contained within a metal housing to transmit the angular rotation input. Two hollow shafts were keyed into the pulleys used for the belt drive, with the outer shaft attached to a knob used for input and the inner shaft attached to the tooling that holds the block and tube. The inner shaft rotates the tooling with a 1:1 ratio to the input. After the pierce and press process, the piece of metal that is pierced from the tube, called a slug, must be expelled from the tube. A blowout device using pressurized air can be used to expel the slug, and a detection sensor past the orientation device ensures the slug was expelled. To ensure compatibility with the slug blowout device, the tooling and shaft have hollow inner diameters.

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1. PRODUCT DESIGN

1.1 Identification of Need

The Parker Hannifin plant in Batesville, MS is a manufacturing plant that specializes in A/C and hydraulic tube and hose assemblies that support heavy machinery in the agriculture, logistics and construction industries. Some of its largest customers are John Deere, Caterpillar, and Paccar. The floor's capacity can run several hundred different jobs through the plant daily. Each job is unique; the tube size, bending configurations, and adaptors widely vary part to part. Some tubes require valve bodies where the customer can charge the A/C system once it has been assembled. These valve bodies may also need to be at a certain angle from a defining feature on a block that is pressed onto the end of the tube. This block-to-valve body angle orientation is required in order to achieve the correct position of the valve body after completing the bending process. The valve body is applied during the pierce and press process. In the 5/8'' and 3/4" value stream, there is a horizontal pierce and press machine where valve bodies can be inserted into the wall of straight tubing at certain angles from the block at the end of the tube.

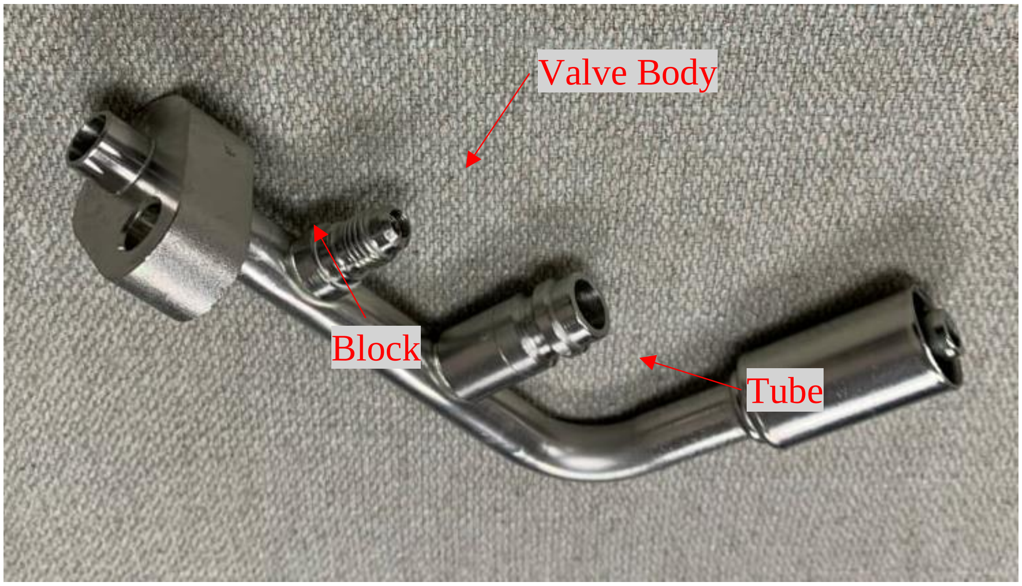


Figure 1: Tube with Block and Valve Bodies

A tool with the block profile is inserted into a sliding block that can be adjusted for the tube length. The tool can be rotated to give the block any angle of rotation, as the piercing tool for the valve body is at a fixed location. The angle of rotation is dictated by the customer print. The tool setter's job is to setup the tool and rotate it to the correct angle called out in the print. The current tools make it difficult to setup, as they are circular and provide no way of measuring at what angle the block is at relative to the piercing tool. The tool setter currently estimates the block angle, runs the setup piece and measures the block-to-valve body angle on a calibrated granite surface. The tolerance for the block-to- valve body angles are typically +/2 degrees. With such a small tolerance, setup time and scrap have become a large loss in efficiency and cost. That certain machine may also end up running over ten different jobs per day, and a lot of the time that it is running is wasted on setup. This machine needs a new way of obtaining the block-to-valve body angle more precisely in order to reduce setup times and scrap significantly.

1.*2 Background Research*

Parker Hannifin, a Fortune 250 company, is, according to their website, the "global leader in motion and control technologies." They are broken into six groups which include fluid handling, filtration, sealing and shielding, climate control, process control, and aerospace technologies. Each group is further divided into multiple divisions that are in 50 countries across 6 continents. This design will be implemented in the Hose Product Division in the Fluid Handling Group in Batesville, MS. This division services customers such as John Deere, Caterpillar, and Daimler [1]. This facility does not mass produce products but instead produces specialty products made-to-order. Product design engineers work with customers to achieve a product that meets their needs and is manufacturable.

Once product design engineers have produced a finalized CAD drawing, the part goes through the Production Part Approval Process (PPAP) to ensure the part is manufacturable, that the proper tooling is made, and that the process routing for the part works. The E-963 is the machine that Parker Hannifin has requested the orientation device be developed for. This machine was originally used for vails (end forming) and was converted to operate as a pierce and press machine. The E-963 is designed to run two diameters of tubing, five-eighths inch and three-quarters inch. It uses interchangeable grip pads (shown in figure 2) to accomplish this. The punch location is a set location on the machine. [2]

Figure 2: Pierce and Press Location (Grip Pads)

To pierce the tube, a .243" diameter punch pin (figure 3) is used.

Figure 3: Punch Pin for E-963

The punch pin leaves a hole in the tube, and the valve body is pressed in. Tools can be added to the machine to allow for various blocks and valve bodies.

Design engineers at Parker Hannifin use Geometric Dimensioning and Tolerancing (GD&T) to ensure that the parts that are ordered will all properly assemble in the final product. Tolerance defines the limits that are allowed for a specified dimension. As the tolerance becomes smaller, the precision of machines required for manufacturing increase as will the cost of manufacturing, so tolerances must be kept within a range that both allows for the part to fit in the final product and keeps manufacturing costs reasonable [3]. An example of a tolerance is shown in figure 4 below.

TOLERANCES:		X.X	± 0.1
		X.XX	± 0.06
ANGLE	± 2°	X.XXX	± 0.030

Figure 4: Tolerance for Parker Drawing

The block to valve body relationship on the print shown in figure 5 is 60 degrees from the horizontal axis.

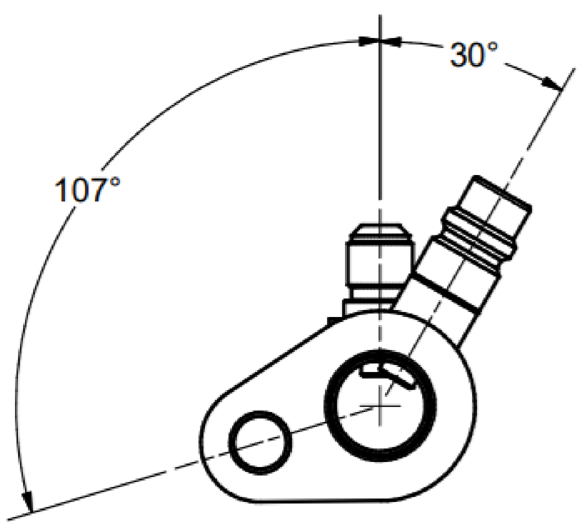


Figure 5: Valve Body to Block Relationship [2]

The print is an example of the type of print operators on the floor use during the manufacturing process. The tolerance for angle dimensions is a bilateral limit of ±2 degrees, which is a commonly used tolerance at Parker Hannifin [2]. This means that after the valve body is pressed into the tube, the range for the acceptable dimension is 58-62 degrees.

Parker Hannifin does not currently have a process to ensure that the location of the valve bodies is within the tolerance specified by the customer prior to the pierce and press process. Having a mechanism that would allow operators to orient the block and tube before the pierce and press process would greatly reduce the amount of parts that are out of tolerance, which would directly decrease the number of customer issues and scrap. An orientation device would allow the operator to set the block and tube at an angle within the acceptable tolerance for manufacturing

To transmit angular input into an angular output, several types of mechanisms can be used. Three mechanisms were under consideration for this project: spur gears, worm gears, and belt drives. Spur gears are cylindrical gears that have shafts that are parallel and coplanar and teeth that are oriented parallel to the shafts. They are the most commonly used and arguably the simplest type of gear. The gear teeth have an involute profile and mesh one at a time. Because the teeth are involute, spur gears only produce radial force, though the method of meshing produces high stresses on the gear teeth, making spur gears well suited to low-speed applications [4]. Worm gears consist of a worm and a gear with non-parallel, non-intersecting shafts oriented at 90-degree angles from each other. The worm is similar to a screw with a V-type thread, and the gear is essentially a spur gear. The worms in the worm gear may be single start or multiple start. A single-start worm advances the gear one tooth for each 360 degree turn of the worm, making worm gears ideal for large gear reductions [5]. Belt drives have traditionally been used for power transmission; however, they can also be used for transmitting angular rotation. A tensioned belt is wrapped around 2 or more pulleys, with one pulley being the input and the other being the output.

Belts can be flat or have teeth. Flat belts rely solely on friction between the flat surfaces of the belt and pulleys to turn the pulleys. Belts with teeth are called positive drive or synchronous belts and rely on the teeth of the belt engaging with grooves on the pulley. These belts are less prone to slippage because of the high frictional force between the teeth of the belt and groove of the pulley. Belt drives use readily available and relatively inexpensive components, making them ideal for use in a fast-paced manufacturing environment [6].

1.3 Goal Statement

The objective of this design process is to create a device which orients block to valve body on AC tubing at desired angles with a tolerance of ±2° while allowing for slug blowout.

1.4 Product Design Specifications

Table 1: Customer Needs

List of Customer Needs	**Importance (1 to 5, 5 being most important)**
Orients the valve body to block	5
Calibration check once per day	5
Hole for the slug to pass	3
Have a +/- 2-degree tolerance on device	4
Tool swap out required	4
Easy installation	3

Table 2: Product Design Specifications

	Parameters		
	Current Model	This Design	World Class
Purpose: The product's main goal	To orient the valve bodies to block per the customer print while leaving room for a slug detector.	To orient the valve bodies with precision per customer print while leaving room for slug detector.	To orient the valve bodies with precision and accuracy automatically per customer print in addition to a slug detector.

Performance: How the product works overall	Can hold set orientation for given job, but no accurate way of achieving the correct orientation without testing setup piece. Tools used do not have slug detector capabilities.	Can hold orientation with visual representation of angle of orientation to reduce setup scrap. Tools are compatible with slug detector.	Can input a desired angle orientation and will automatically turn to correct angle. Slug detector capabilities built in.
Safety: How product affects safety and well-being of consumer	Safely bolted to machine, minimal pinch points. Light curtain shuts down machine if obstructed by operator or object.	Safely bolted to machine. All moving components housed for safety. Light curtain shuts down machine if obstructed by operator or object.	Safely bolted to machine. All moving components housed for safety. Light curtain shuts down machine if obstructed by operator or object.
Size: How large overall product is	7” x 3” x 3”, within safety guard constraints.	7” x 3” x 4”, within safety guard constraints.	7” x 3” x 4”, within safety guard constraints.
Weight: How much the product weighs	Made of cold rolled steel, about 5 lbs	Made of cold rolled steel, with ABS housing, about 5 lbs	Made of cold rolled steel, about 5 lbs
Life in Service: How long product lasts	24/7 for 25 years	24/7 for 25 years with little maintenance	24/7 for 25 years with maintenance
Maintenance: How much maintenance is needed after installation	May need tooling replaced if damaged	May need replacement of belt or pulley	May need regreasing of joints, calibration of automatic encoders
Shipping: How product is moved from manufacturer to consumer	By ground shipping	By ground shipping	By air/ground shipping
Packaging: How product is packaged for consumer	None	None	None
Quantity: EAU of product	1	1	1
Aesthetics: How the product looks	Boxy, minimal	Functionally minimal	Streamlined design on visible components

Materials: What the product is made of	Cold rolled steel	Cold rolled steel, aluminum housing	Heat treated steel, aluminum, and plastic
Environment: Where product is placed	5/8 Horizontal pierce and press machine at Parker Hannifin, Batesville	5/8 Horizontal pierce and press machine at Parker Hannifin, Batesville	5/8 Horizontal pierce and press machine at Parker Hannifin, Batesville

1.5 Synthesis

The main design limitation for the orientation device is the amount of open space available, particularly the available height, for the device as well as the railing that is attached to the device to fit through. Figure 6 shows the existing rail and sliding plate the device must attach to as well as the guard that blocks operators from accessing the machine from the side.

Figure 6: Guard Restriction

The guard exists because the location of the pierce and press process is a pinch point. The Occupational Health and Safety Administration (OSHA) defines a pinch point under 1910.211(d)(44) as any point where part of a person's body could be caught in [7]. Per OSHA standards, the distance from the guarding and the pinch point determines the height of the opening in the guarding. The guarding can be further expanded if other safety considerations are present such as a light curtain; however, additional safety measures are not able to be implemented to accommodate this device. Rockford Systems, the manufacturer of the pierce and press machine, provided documents indicating that the maximum height between the bottom of the guard and the table of the machine should not exceed three inches [8]. Figure 7 shows an example of the measurement device used to determine appropriate guard heights.

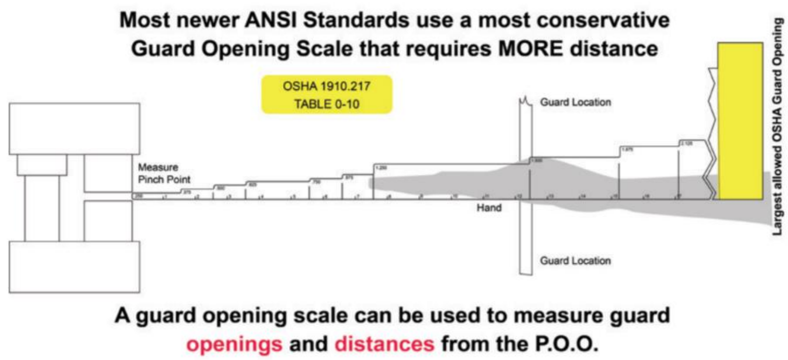


Figure 7: Example of Pinch Point Guard Height Measurement Tool [8]

The other main size constraint is the railing and sliding plate that are already attached to the machine. The orientation device must attach to this in order to accommodate the various lengths of tube that need to be processed through the machine, which limits the available width for the device.

To transmit angular rotation, three types of mechanisms to transmit that angular rotation were considered. Worm gears were the first mechanism to be considered. Fine-pitch worm gears are largely used to transmit motion rather than power, which would make them ideal for transmitting angular rotation. Using a self-locking worm gear would prevent any backdriving and keep the force from the pierce and press machine from moving the shaft from its set rotation. However, the most practical worm gear design, which would have the worm situated on top of the gear for ease of access to the operator, would not meet the height restrictions from the safety railing. Placing the worm on the sides of the gear instead of on top would be difficult for the operator to access and would be unlikely to meet the height constraints as well. In addition, achieving a 1:1 gear ratio, where the input angle would be the same as the output angle, would be incredibly difficult if not impossible to achieve with any worm gear configuration on the market. [9]

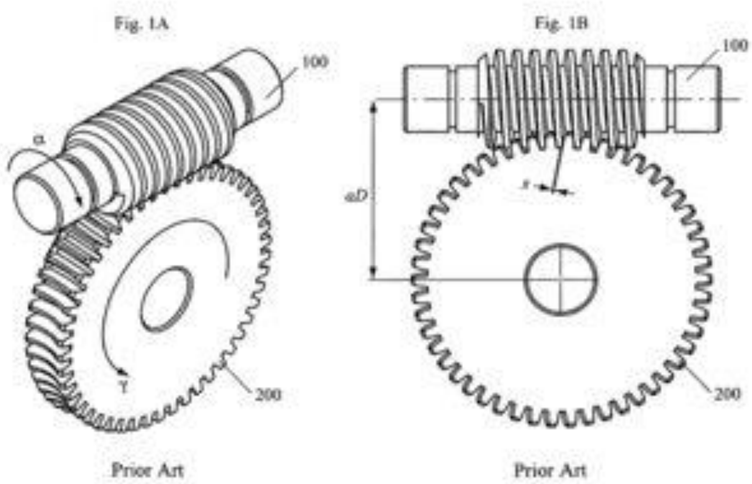


Figure 8: Worm Gear Drawing [10]

Spur gears were the next mechanism to be considered. For spur gears with the same diametral pitch, the gear ratio would be 1:1, and the input angle would equal the output angle. Spur gears could be backdriven, but the addition of a set screw would prevent the force of the pierce and press from affecting the angle that was set. When searching for commercially available gears, no gears were found that would be both small enough to meet the height requirement and have a large enough inner diameter to fit the hollow shaft that would be needed to support the tool that holds the block and tube. [9]

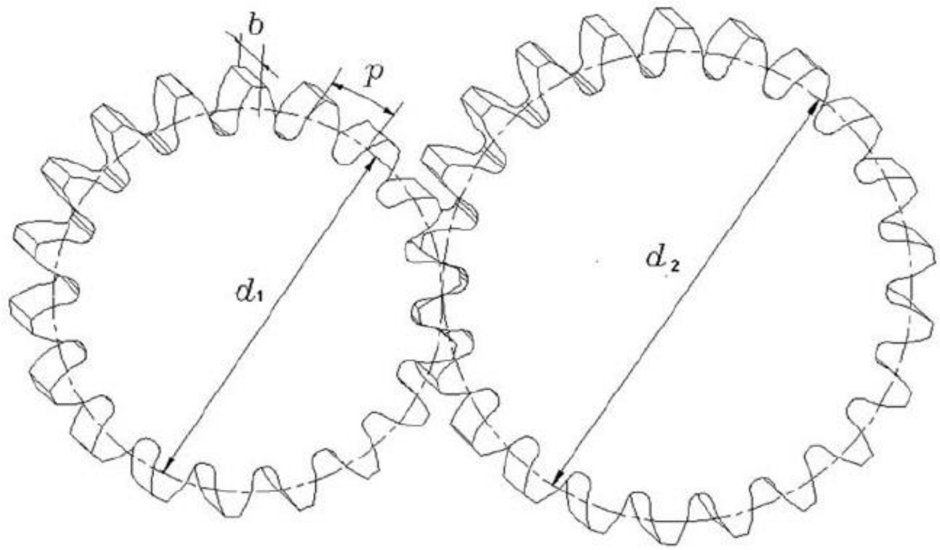


Figure 9: Spur Gear Drawing [11]

The final mechanism considered was a belt drive. Belt drives use pulleys to transmit rotation and angular velocity from an input pulley to an output pulley. Traditionally, belt drives are used in industrial power transmission applications where the speeds of the driver and driven shafts must be different or if they must be separated by long distances. A commercially available pulley was found that would not exceed the height limit with both the belt drive and housing above it, as well as having a sufficiently large inner diameter to support the shafts. If the driving pulley has the same pitch diameter as the driven pulley, the velocity ratio, and by extension the ratio of angular input to output, will be the same. Using two of the same pulleys with a belt drive would ensure a 1:1 input to output ratio. [9]

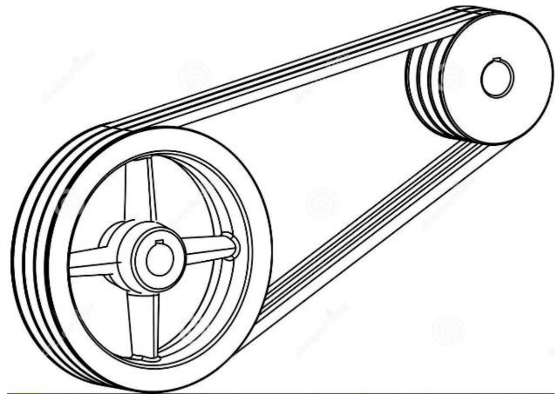


Figure 10: Belt Drive Drawing [12]

Pulleys are used for angle displacement and power transmission. The precision of those quantities is important when selecting a pulley. For the orientation device, the purpose of the pulley and belt drive is to transmit angle displacement. The material and size of the pulley, as well as the type of the pulley, were considered when making the pulley selection.

The material of the pulley is an important factor in determining its lifespan and its load capacity. Plastics are lightweight and sufficiently durable for light torque and load situations. Metals, however, generally have a longer lifespan and are more resistant to unexpectedly high loads. Although they may add to the weight of the overall design, they are more resilient and designs that utilize metal pulleys will require less maintenance. [9]

Selecting the proper size of the pulleys is crucial for the design of the orientation device. The pulleys must not exceed size constraints while maintaining the desired output torque, displacement, and velocity. The pulleys used for the orientation device have to fit within the housing of the device and avoid interfering with the movement of other components. The pulleys must also be a sufficiently large distance away from each other in order to keep proper tension on the belt. As such, their sizes must be compact while having an inner diameter large enough for the shafts to fit through. Pulleys share the same ideals as gear trains. The ratio of the sizes of the pulleys in the belt drive determines the ratio, or factor upon the output. Having the same diametral pitch of the pulleys in the belt drive will ensure at 1:1 ratio, a requirement for this design. [9]

There are several series of pulleys, and the appropriate pulley must be selected for the application. There are commercially available pulleys that are smooth for simple power transmission, that have tensioning springs, and that have grooves cut out for timing capabilities. The orientation device uses the timing belt series, as the grooves will ensure no slippage of the belt. Slippage occurs when there is a loss of friction between the belt and pulley. Having slippage occur in the orientation device is detrimental, as the device relies on an absolute zero position that has been set upon fabrication. The timing belt will ensure that absolute zero can be achieved at all times. Additionally, the series defines what type of belt will be compatible with the pulleys, as pulleys can have features such as flanges that keep the belt from translating axially on the pulley. There are also fastening systems, such as a set screw or welding flanges, that are used to secure the pulley onto its shaft. [9]

Once the series of the pulleys that are used in the design has been determined, a compatible belt must be selected. The belt must be an appropriate width and circular length. The width of the belt is restricted to the width of the pulley. The circular length of the belt is determined from the fixed length between the two pulleys and the diameter of the pulleys. The belt must be long enough to encompass both pulleys and also provide enough tension between the two pulleys in order to have enough frictional force for torque transmission. [9]

The shaft for this design was required to be hollow in order to for the slug, the piece of the tube that is removed during the pierce process by the pin, to pass through. After the slug had been pierced, a puff of compressed air blows through the tube to move the freed slug out of the tube and into a waste containment bucket. The compressed air is attached to one end of the tube while slug must pass through the end that is inserted into the orientation device. The hollow shaft design considers the minimum cross-sectional area needed to ensure the slug is easily able to pass through the shaft and tooling. A data set of about 100 individual slugs were measured for their diameter. The largest slug had a diameter of about .258 in.

The shaft and tooling should be made from a resilient material. The shaft and tools will encounter impacts from the slugs as they are blown out of the tube, from the block on the tube as it is inserted into the block profile on the tool, and from constant usage of set screws clamping down on surfaces of the tools. The shaft must also be resistant to deformation from the weight of the tooling and tube. Current designs have the block profile and shaft made from aluminum. The ductility of the aluminum has allowed the existing shaft to buckle under the pressure from the set screws and slugs. The block profile has developed many defects, making it difficult to insert the block into its profile [2]. Steel has been determined to be a much better material to construct the shaft and tooling. It is easily machinable, and resistant to deformation such as nicks, buckling, or chipping. Steel can also be heat treated to increase the hardness of the tooling and shaft surfaces and further resist damage.

The tooling design will imitate current tooling designs implemented for the other machines in the pierce and press area. The tooling can be inserted into the shaft and aligned with the use of a key. The key will ensure that the tool is properly "zeroed" every time. A set screw will then clamp down onto the tool to prevent it from moving axially along the shaft. A block profile will need to be made for each type of block that runs through the machine. These block profiles can be made externally from the insertion portion of the tool and bolted on for ease of machinability.

The housing will encase the pinch points created by the belt drive and act as a support for the shaft. The housing should be made from a lightweight yet durable material such as aluminum or sheet metal. The material selection should be sufficiently strong enough to support the weight of the mechanism while adding as little weight as possible to the structure. The housing should enclose the belt drive fully and allow the shafts to pass through. Bearings will be press fit into the housing to allow the shafts to rotate with minimal friction. The housing is secured to the single flanged sliding block by screws with nuts to lock them in place. The sides of the housing will be connected with screws, with the sides that support the shaft being made of thicker material, and the non-load-bearing pieces being made of sheet metal.

In order for this device to truly be an orientation device, there must be a way for the operator to determine the angle of the output. Because the output for this device is in a hard to reach area near a pinch point, the operator will need to be able to determine this angle from the input shaft, as well as have a convenient way to operate the device. A custom-machined knob that would be able to fit into the input shaft was essentially the only option for device operation. The knob would either be sized large enough so that the dial would fit on the knob, or small enough for the dial to fit around the knob on the housing. There are limited types of graduated dials that have a graduation range of 0°-360° with 1° precision commercially available. One type of dial could be easily adhered to the housing or to the knob, but the smallest diameter available exceeded height requirements. The second type of dial found was machined onto a thin metal cylinder and was available in sufficiently small diameters with a customizable graduation range. [13]

1.6 Analysis

There are several analyses that must be done before making a design selection. The belt length is dependent upon the pulley radius and distance between the center of the pulleys, and the pulleys must have a 1:1 input to output ratio. The output shaft must have a high enough flexural strength to support the weight of the tool that holds the block and tube and to not deform under the force of the pierce and press process. The housing itself must be able to support the weight of the shafts and anything the shaft supports as well as the weight of the bearings, and the bolts that fasten the housing on the single-flanged slide block must be able to support high shear loads in the event that there is an unexpected load placed on the housing.

Before the pulley selection is finalized, the gear ratio must be determined to ensure that the pulleys have a 1:1 input to output ratio. For both gears and pulleys, the ratio of input to output can be calculated the same way. The equation for gear and pulley ratio is given by equation 1

$$Gear\ ratio = \frac{d_1}{d_2} = \frac{\omega_1}{\omega_2} \qquad (1)$$

where d_1 is the diameter of the first gear or pulley, d_2 is the diameter of the second gear or pulley, ω_1 is the angular velocity in radians per second of the first gear, and ω_2 is the angular velocity in radians per second of the second gear. [9]

Once the pulleys are selected, the belt length must be determined. While the length between the pulleys is limited by the size of the pierce and press machine, it will be driven by available belt sizes. A belt drive diagram with labelled parameters is shown in figure 11.

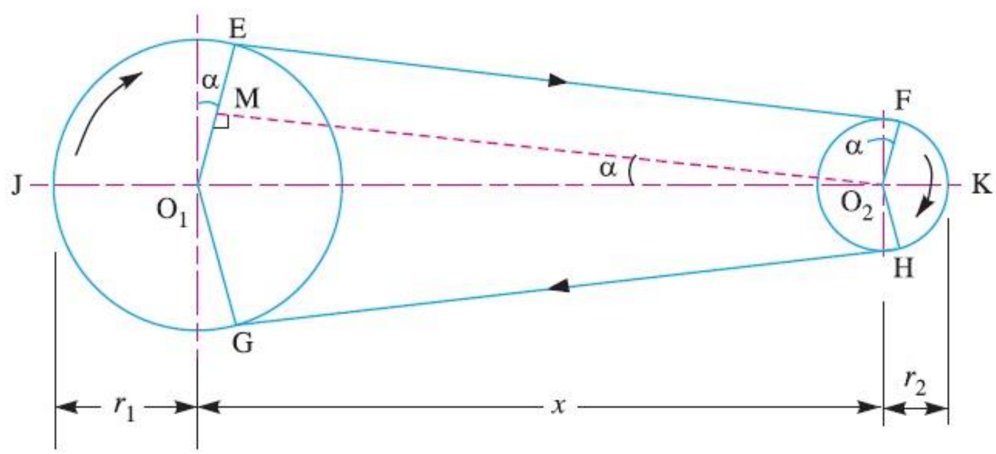


Figure 11: Belt Drive Diagram [14]

The equation for the length of the belt is given in equation 2 below

$$L = \pi(r_1 + r_2) + 2x + \frac{(r_1 - r_2)^2}{x} \tag{2}$$

where r_1 is the radius of the first pulley, r_2 is the radius of the second pulley, and x is the distance between the centers of the two pulleys. [9]

Both shafts used in the design are identical in size except for the section of the output shaft that has a keyed section to join with the handle of the tooling. The output shaft will experience significantly higher loading than the input shaft, so analysis will only need to be done to ensure that the output shaft does not fail since the input shaft would fail under the same conditions (though it does not experience them). The shaft goes completely through and extends past the housing, with each of the walls of the housing acting as a support. For the purpose of this analysis, the shaft can be simplified to an overhanging beam, with the weight of the handle, tooling, and tube simplified to a point load at the end. The free body diagram as well as the shear force and bending moment diagrams for an overhanging beam with a point load at the end are given in figure x. The reaction force at the first support is given by equation 3

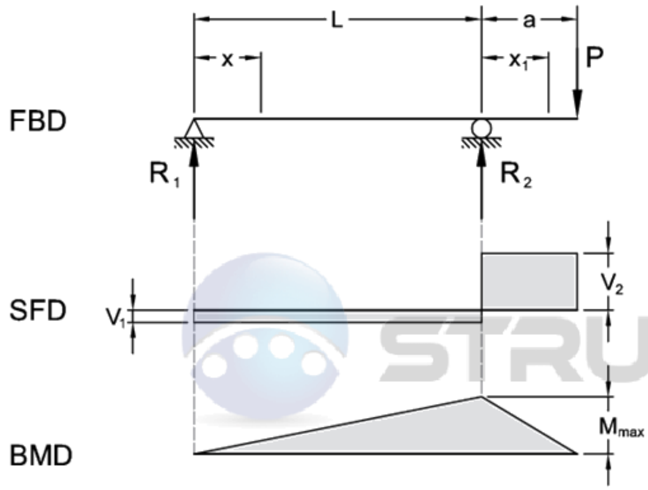
$$R_1 = \frac{P \cdot a}{L} = V_1 \tag{3}$$

where P is the point load, a is the length of the overhang, and L is the length of the beam between the supports. The reaction force at the second support is given by equation 4

$$R_2 = \frac{P}{L}(L + a) \tag{4}$$

and the maximum moment, which is located at the second support, is given by equation 5

$$M_{max} = P \cdot a \tag{5}$$



$$R_1 = V_1 \ldots\ldots = \frac{Pa}{L}$$

$$R_2 = V_1 + V_2 \ldots\ldots = \frac{P}{L}(L+a)$$

$$V_2 \ldots\ldots = P$$

$$M_{max}\ (\text{at } R_2) \ldots\ldots = Pa$$

$$M_x\ (\text{between supports}) \ldots\ldots = \frac{Pax}{L}$$

$$M_{x_1}\ (\text{for overhang}) \ldots\ldots = P(a - x_1)$$

$$\Delta_x\ (\text{between supports}) \ldots\ldots = \frac{-Pax}{6EIL}(L^2 - x^2)$$

$$\Delta_{x1}\ (\text{overhang}) \ldots\ldots = \frac{Px_1}{6EI}(2aL + 3ax_1 - x_1^2)$$

$$\Delta_{max}(\text{for overhang at } x_1 = a) \ldots\ldots = \frac{Pa^2}{3EI}(L+a)$$

$$\Delta_{max}\left(\text{between supports at } x = \frac{L}{\sqrt{3}}\right) \ldots\ldots = \frac{-PaL^2}{9\sqrt{3}EI} = 0.06415\frac{PaL^2}{EI}$$

Figure 12: Free Body, Shear Force, and Bending Moment Diagram with Equations for Overhanging Beam [15]

The shaft that supports the knob and dial will also act as an overhanging beam with the end load being the weight of the knob. The weight of the knob is significantly less than the weight of the tool and tube, and both shafts are the same size and material, so it can be assumed that if the analysis shows the shaft to be able to support the tube and tool, the shaft would also be able to support the knob.

Once the bending moments and shear stresses are calculated for the shaft, the bending stress must be calculated. The bending stress is calculated using equation 7 shown below

$$\sigma_b = \frac{My}{I} \tag{7}$$

where *M* is the maximum bending moment, *y* is the vertical distance from the neutral axis, and *I* is the moment of inertia. The moment of inertia for a cylinder is calculated using equation 8

$$I = \frac{\pi(D^4)}{64} \tag{8}$$

where *D* is the diameter. [9]

The moment of inertia for a hollow cylinder is calculated using equation 9

$$I = \frac{\pi(D^4 - d^4)}{64} \tag{9}$$

where D is the outer diameter and d is the inner diameter.

The housing that contains the belt and pulleys must be able to support the loads the output shaft will withstand. To simplify the analysis, the shaft and ball bearings in the housing are treated as a bolt, and the housing material will be evaluated for two types of failure, bearing and tearout. Bearing failure is the excessive deformation of the housing material, and the equation for bearing capacity is given by equation 9

$$R_n = 2.4dtF_u \tag{9}$$

where R_n is the nominal bearing strength, d is the diameter of the ball bearing in the hole, t is the thickness of the material, and F_u is the ultimate tensile strength of the material. If the housing suffers a tearout failure, the shaft and bearing would tear through the material above or under them. The equation for tearout failure is given by equation 10

$$R_n = 1.2l_c tF_u \tag{10}$$

where l_c is the distance between the edge of the hole and the edge of the material. [16]

The bolts that fasten the housing to the sliding plate should be able to withstand day-to-day forces that will be placed on the housing. A diagram of shear forces in a bolt and equations for stresses in the bolt and plates is shown in figure 13.

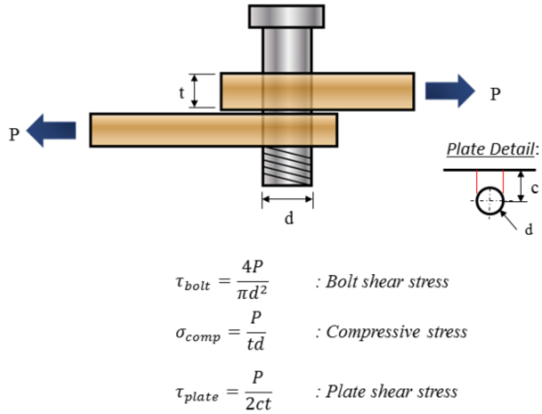


Figure 13: Shear Stress at Bolted Joints [17]

Though failure of the bolts is unlikely, the bolts would fail in shear before in compression, so only the analysis of the shear stress on the bolt must be done. The equation for shear stress in a bolt is given by equation 11

$$\tau_{bolt} = \frac{4P}{\pi d^2} \tag{11}$$

where P is the force on the plates and d is the diameter of the bolt.

1.7 Selection

To aid in the design selection, a selection matrix was used. The cost, safety, performance, and reliability are given a weighting factor that reflect their importance in the design selection. The total of the weighting factors is one; each factor is effectively a percentage. Each of the three designs that were considered were given a score out of ten, with one being the worst and ten being the best, for each criterion and the score was multiplied by the weighting factor and totaled to give a composite score. The design with the highest rank was selected.

Table 3: Selection Matrix

	Cost		Safety		Performance		Reliability		Rank
Weighting Factor	.15		.35		.30		.20		1
Design 1: Worm Gear	4	0.6	1	.35	7	2.1	8	1.6	4.65
Design 2: Spur Gear	8	1.2	2	.7	7	2.1	8	1.6	5.6
Design 3: Belt and Pulleys	8	1.2	9	3.15	9	2.7	7	1.4	8.45

The first design consisted of a worm gear, which would enable the output shaft to be turned to a desired angle of rotation from the top of the orientation device. As the operator turns the worm, the gear connected to the shaft that holds the tool body would rotate to the desired angle. Several issues arose with this design. Worm gears typically create a significant gear reduction. The operator could have to turn the worm an excessively large number of times in order to achieve the desired angle. Without the 1:1 gear ratio, the worm mechanism would not be compatible with a dual 1-180 angle dial, which is required for the operators to determine the desired angle. Also, the selection of commercially available worm gears was limited and no hollow worm gears with a large enough inner diameter for slug blowout compatibility could be found. The 90-degree change in direction of the power transmission also did not fit within the size constraints for safety reasons.

The second design used a parallel shaft spur gear system with two spur gears of the same diametral pitch. The operator would turn the knob on the input shaft, and the output shaft, which would be attached to the tool that holds the block and tube, would rotate the same angular distance, but in the opposite direction. Because the gears would rotate in the opposite direction of one other, the operator would set the angle from the side of the orientation device opposite of the pierce and press machine. This would give the operator sufficient space to use the device, though the operator would not be able to view the output while turning the knob. While the parallel spur gears would be cost effective and reliable, this design does not fit the size constraints of the guarding for the machine, and the guarding would need to be partially removed to allow for this design to be implemented, creating a safety hazard.

The third design uses pulleys and a belt drive to transmit angular rotation. Two timing belt pulleys of equal diameter are placed sufficiently far apart to create tension on the belt. Because the pulleys are of equal diameter, a 1:1 input angle to output angle can be achieved. This design is the only design out of the three to not exceed the height requirement from the guard. All other designs would have required too much of the guard to be removed to operate safely. Additionally, this design is approximately as reliable as the others, has a sufficiently low cost, and has the desired performance.

1.8 Detailed Design

The exploded and assembled views for the final design of the orientation device are shown in figures 14 and 15.

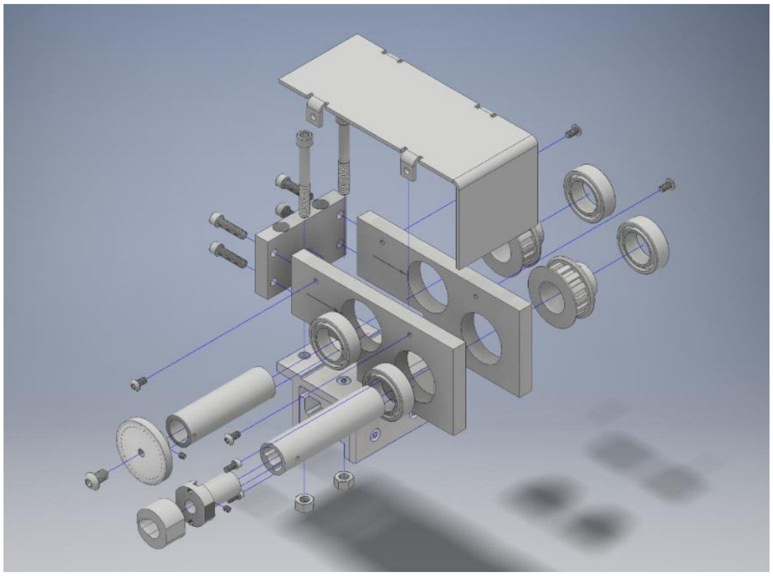


Figure 14: Overall Design View, Exploded

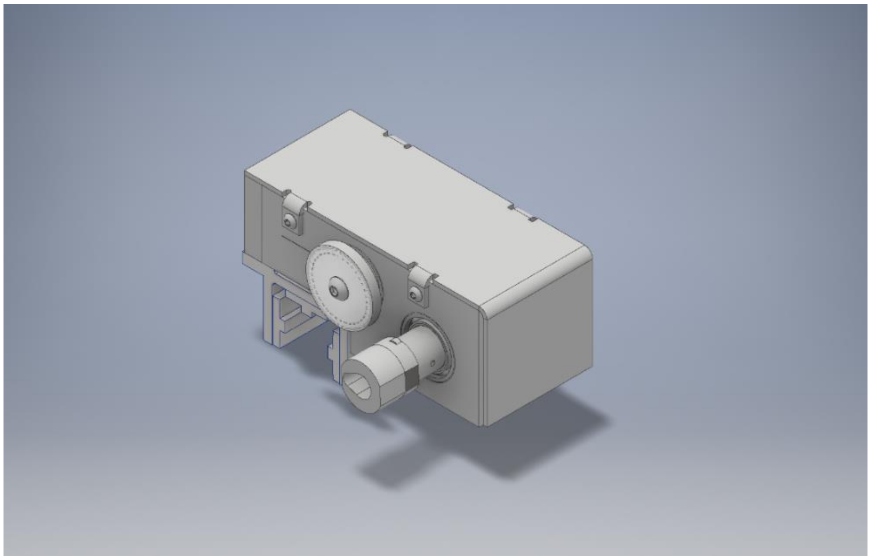


Figure 15: Overall Design View, Assembled

The orientation device consists of two pulleys, a belt, two shafts (one keyed), four ball bearings, a handle for the tool body, a knob for turning the input shaft, a graduated dial plate, a metal housing, and several screws, nuts, and bolts to fasten parts together appropriately. The pulleys and belt are fully contained within the housing to prevent access to the pinch points. The ball bearings are installed on the sides of the housing that are parallel to the pulleys. The keyed shaft goes through the bearing and pulley that are placed lower within the housing. The handle for the tool body is inserted into the keyed shaft. The non-keyed shaft goes through the bearing and pulley situated higher within the housing. The input shaft knob is press fit into the non-keyed shaft, and a set screw is placed through the shaft and knob handle to keep the knob in place. The graduated dial is bolted onto the knob handle face. The sides of the housing parallel to the shaft are the same material and thickness as the housing piece that connects the housing to the slider plate. A piece of sheet metal is used to complete the remainder of the housing, and all the housing pieces are fixed together and to the slider plate using screws, nuts, and bolts.

An L-series pulley from McMaster-Carr (Part # 6495K14) was selected for use in the belt drive. This pulley was the only available timing belt pulley that met the size constraints of the design. The pulley has a maximum belt width of 0.5 inch and has flanges on both sides of the grooves that prevent the belt from slipping off the pulley. The pitch diameter of the pulley is 1.671 inches and the pitch of the groove is .375 inch [18]. The pulley has a hollow inner diameter which would allow a shaft to be press fit through the pulley; however, no commercially available pulleys had a large enough inner diameter to fit the shaft selection, so a tool maker at the Parker Hannifin plant will bore out the diameter to 1 inch. The pulley has a hole bored on the side of the hub for the use of a set screw to ensure that the pulley does not slip on the shaft. Figure 16 shows the pulley with designations dimension titles.

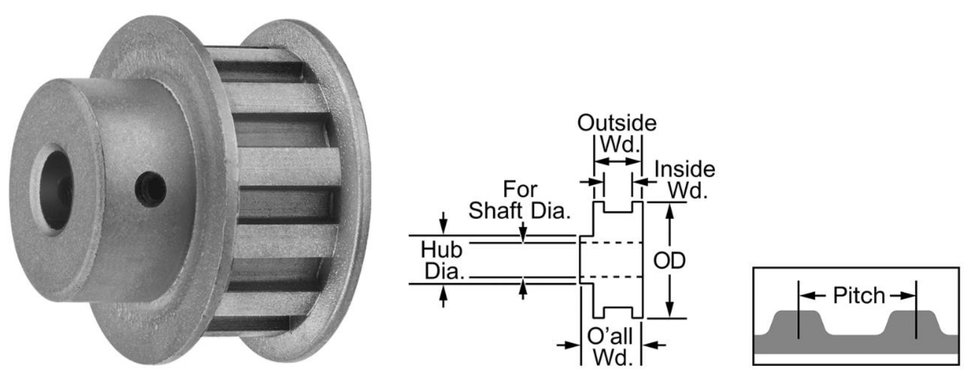


Figure 16: L Series Timing Belt Pully [18]

Because this design requires a 1:1 gear ratio, both of the pulleys used in the belt drive are the same and therefore have the same diameter. Plugging the diameters into equation 1 proves that the gear ratio is 1:1.

$$Gear\ ratio = \frac{d_1}{d_2} = \frac{1.671\ in}{1.671\ in} = 1$$

An L-series timing belt manufactured by GPR Industrial (Part #98L050) was selected for this design. The teeth on the belt create additional tension, ensuring that no slipping occurs while the shafts are being rotated. This keeps the orientation angle of the two pulleys calibrated. The pitch of the belt is 0.375 inch and the width of the belt is 0.5 inch, which match the groove pitch and width of the pulleys, respectively [19]. Figure 17 shows a side view of the belt with dimensions.

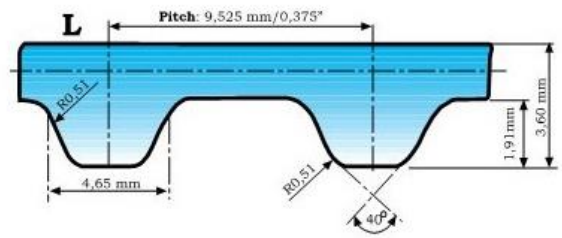


Figure 17: Belt Dimensions [19]

The pitch length of this belt is 9.75 inches with 26 teeth. Using equation 2 from the analysis section, rearranged to solve for distance x, gives the required distance between the two pullies, which is 2.25 inches.

$$L = \pi(r_1 + r_2) + 2x + \frac{(r_1 - r_2)^2}{x}; r_1 = r_2, so\ x = \frac{L - \pi\ (r_1 + r_2)}{2} = \frac{9.75\ in - \pi\ (0.8355\ in + 0.8355\ in)}{2} = 2.25\ inches$$

Two shafts were designed for this device. Both shafts will be machined from 1018 steel with a yield strength of 53,700 psi (pounds per square inch) and have an outer diameter of 1 inch and an inner diameter of 0.76 inch. A key was added to the design of the output shaft to connect the tool body to the shaft. The key will be machined within the inner diameter of the shaft by an outsourced tooling specialist with a wire EDM machine. This key will be able to mate with a keyhole designed on the tool body. The key has a 0.462-inch inner diameter and is completed by mating the key built within the shaft to the keyhole on the tool body. The remaining length of the tool shaft has an inner diameter of 0.75 inch. The outer diameter of the shaft is 1inch, which is sufficiently larger than the inner diameter to maintain reasonable wall thickness. The 1-inch outer diameter also eases the bearing selection, as there are many commercially available roller bearings with an inner diameter of 1 inch. Refer to Figure A10 in the Appendix for designation of diameters. Figure 18 shows the tooling shaft with key.

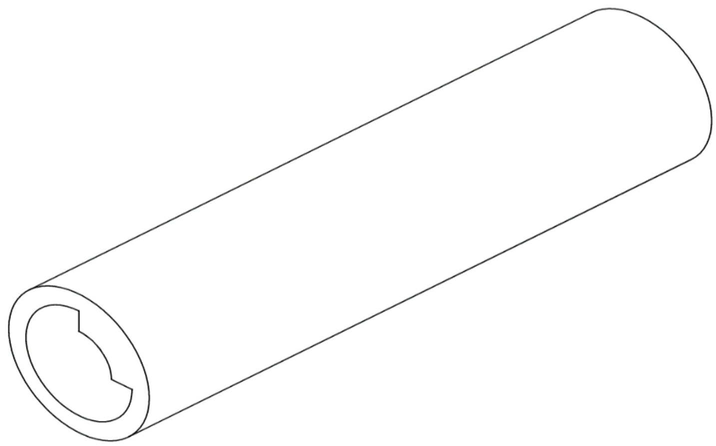


Figure 18: Shaft with Key

The input shaft was created to accommodate the graduated dial. The outer diameter and inner diameter are the same as the tooling shaft, 1.00 inch and 0.75 inch respectively. The inner and outer diameters of the two shafts are identical so that two identical pulleys could be used in the belt drive to ensure a 1:1 ratio. Having the same dimensions also ensured that the two shafts could be made of the same stock material. The length of the input shaft is shorter, however, at 3.4” long, compared to the tooling shaft, at 4.00” long. This was done to keep the dial that will attach to this shaft close to the indicator notch. An 8-36 threaded hole is bored into the side wall for a set screw. This will allow the set screw to clamp down onto the dial once it is inserted. Figure 19 is the drawing of the dial shaft.

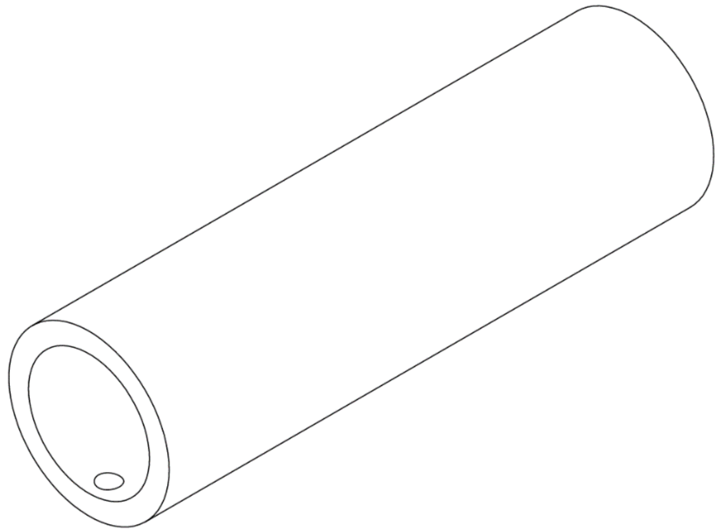


Figure 19: Shaft without Key

To ensure the shaft can withstand expected day-to-day loads, the maximum weight that can be applied on the shaft without yielding was calculated. The shaft was simplified to an overhanging beam with a point load at the end for structural calculations. It can be assumed that an overhanging beam with a large point load at the end would have the maximum bending moment at the support nearest to the load, which in this case would be the side of the housing that faces towards the machine. The first 3.08 inches of the shaft is covered with the remaining 0.92 inches of the shaft protruding past the housing. The keyed section of the shaft begins at 2.75 inches and extends until the end of the shaft. This means that the maximum bending moment will occur at the keyed portion of the shaft. The shaft will only bear loads when the tool handle is placed into the shaft, so the moment of inertia must be calculated for the shaft and the tool handle, which when joined form a cylinder (non-hollow). Equation 8 is used for this calculation.

$$I = \frac{\pi(D^4)}{64} = \frac{\pi(1\ in^4)}{64} = 0.0491\ in^4$$

A non-maximum moment will also occur at the first support, where the cross-section of the shaft is a hollow cylinder, and moment of inertia must be calculated for that area as well, using equation 9.

$$I = \frac{\pi(D^4 - d^4)}{64} = I = \frac{\pi(1\ in^4 - 0.76 in^4)}{64} = 0.0327\ in^4$$

The maximum allowable moment for the keyed section of the shaft with the tool handle inserted is determined using equation 7. The distance from the neutral axis is half the diameter as the neutral axis occurs at the center of the shaft.

$$\sigma_b = \frac{My}{I}; M_{allow} = \frac{\sigma_y I}{y} = \frac{(57{,}300\ psi)(0.0491\ in^4)}{0.5\ in} = 5273\ lb \cdot in$$

The maximum allowable moment for the non-keyed, hollow section of the shaft is determined using equation 7 as well, with the same distance from the neutral axis.

$$\sigma_b = \frac{My}{I}; M_{allow} = \frac{\sigma_y I}{y} = \frac{(57{,}300\ psi)(0.0327\ in^4)}{0.5\ in} = 3512\ lb \cdot in$$

The hollow section of the shaft has the smaller maximum allowable moment, so the maximum allowable load is determined using the maximum allowable moment for the hollow section. Equation 5 is used to determine the maximum allowable load. The distance *a* between the load and the second support is the length of the shaft plus the thickness of the tooling minus the length of the shaft between the supports.

$$M_{max} = P \cdot a;\ P = \frac{M_{max}}{a} = \frac{3512\ lb \cdot in}{4in + 0.4in - 3.08in} = 2660\ lb$$

The maximum allowable load for the shaft far exceeds even the most extreme loading cases the shaft would experience. A shear force and moment diagram, shown in figure 20, shows the locations of the maximum shear stress and bending moment.

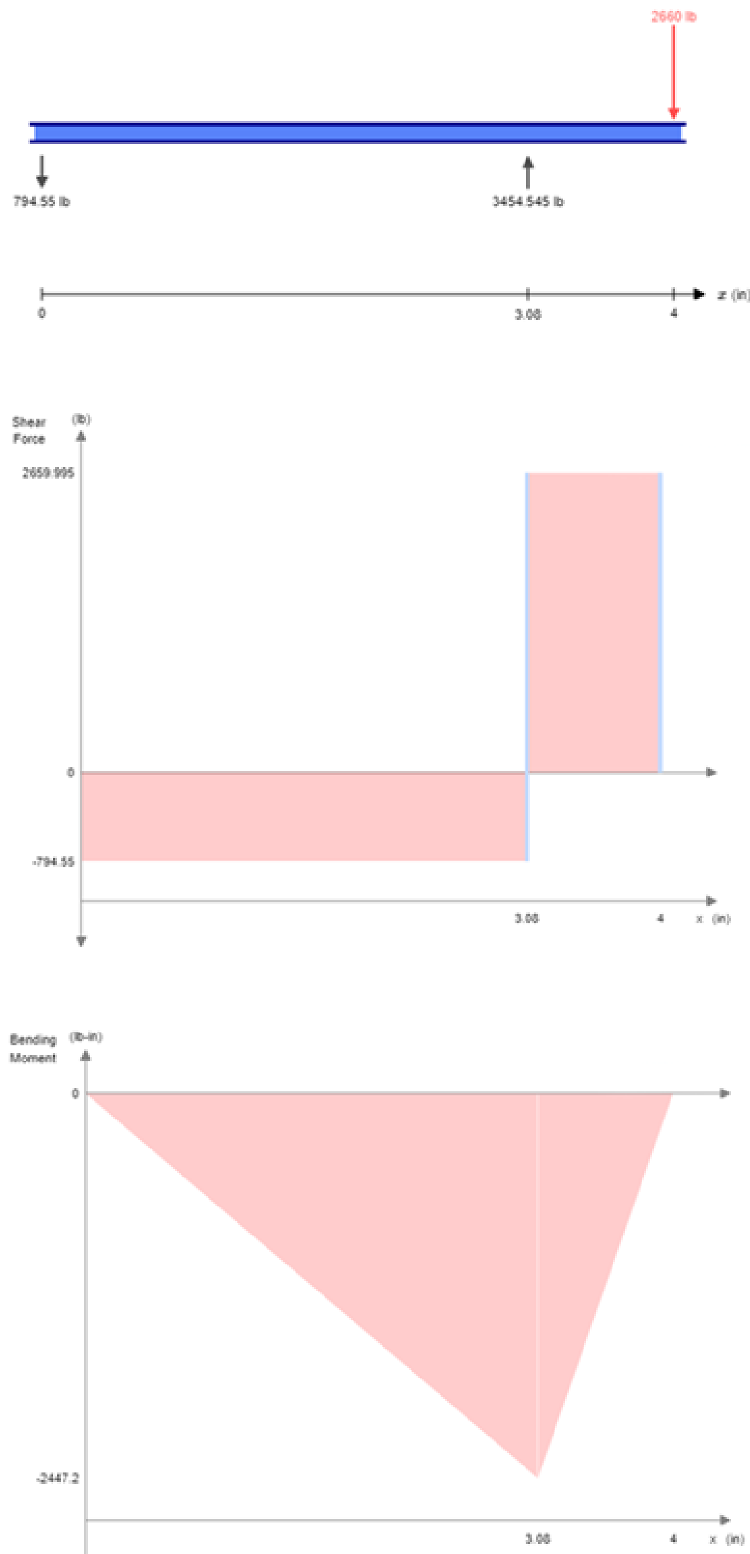


Figure 20: Shear Force and Moment Diagrams for Overhanging Beam with End Load

The tooling for this design needs to be zeroed at a certain position in relation to a defining feature on the block profile, so the existing tooling was redesigned to work with the orientation device. The keyed design will help achieve a constant zero reference every time the tool is utilized. The key will match the profile on the shaft, ensuring the tool is inserted into the shaft the same every time. The tooling consists of two components: the universal insertion key, and the block profile. Figure 21 is the design for the universal portion of the tool body.

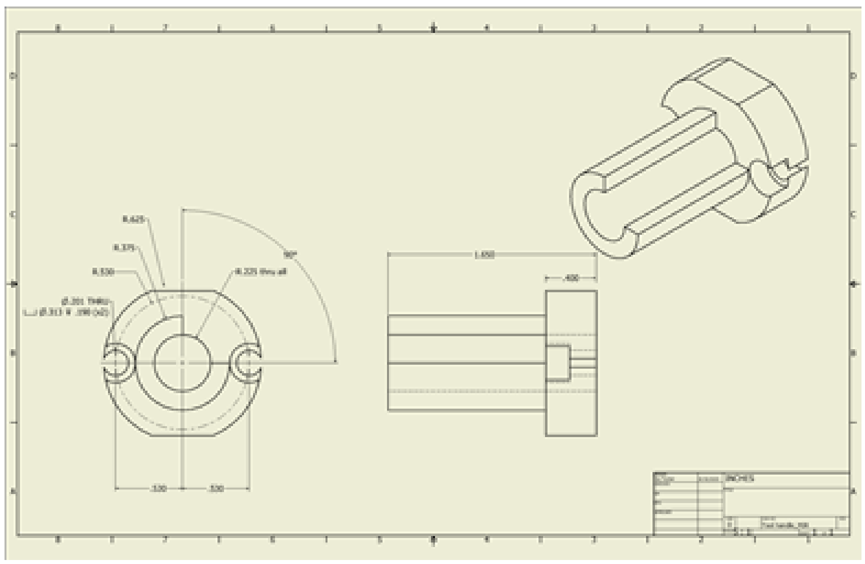


Figure 21: Universal portion of tool body part

This component is considered to be a "blank", as it can be universally used for various block profiles. A 0.450-inch diameter cut is bored through the entire body for slug blowout compatibility. This portion features the keyhole that will be inserted into the shaft. The keyhole is cut out of the handle due to ease of machinability. This cut can easily be made on a milling machine, which is readily available at the facility. Since the shaft would only need to be manufactured once, the keyhole was decided to be put on the tool body. A set screw will be used to clamp down onto the tool body to prevent it from moving axially within the shaft. Figure 22 shows an example of a block profile.

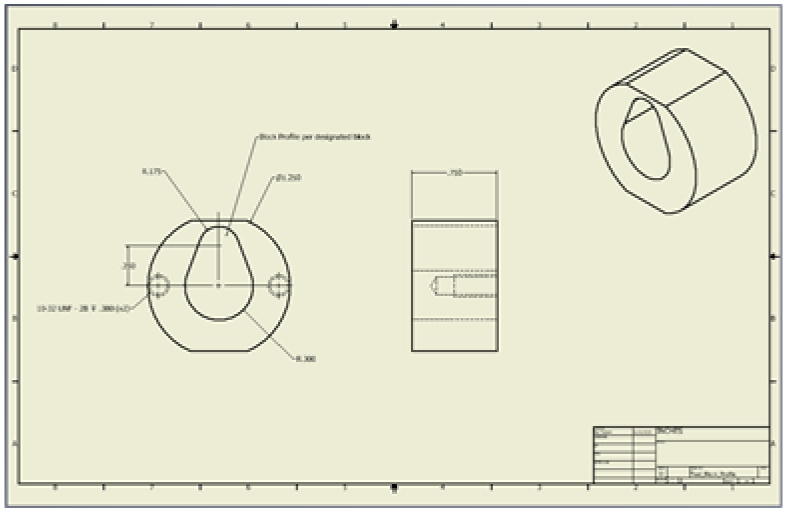


Figure 22: Block Profile

The profiles will be determined from the specified block design. The machinability of the blocks is much simpler as the profile can be cut through the entire thickness. The parts are fastened together using 10-32 socket head cap screw bolts. Two flats were designated to be milled on the curve of the tools as a surface to place an angle finder in the need to confirm the angle orientation.

The housing will be made of out of Aluminum 6061 and sheet metal. Two walls made of aluminum will have ball bearings press fit into them to support the shafts. One of these walls is shown in figure 23.

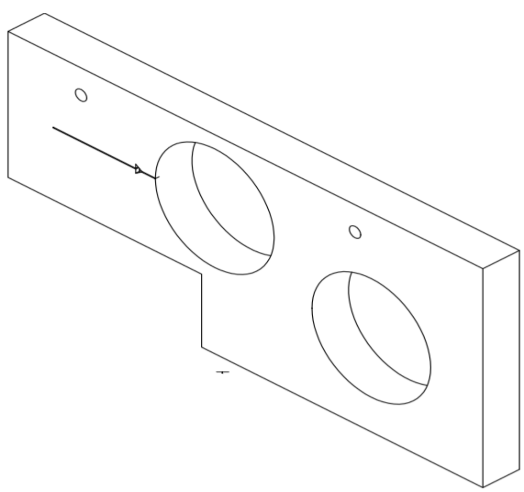


Figure 23: Bearing Wall

The dimensions of this wall can be seen in figure A7 in the appendix. The width of the wall is 0.5 inch, which will allow bearings that are 0.5-inch thick to fit into the wall. To ensure that the wall can support the weight of the shafts and any loads the shafts will bear, bearing and tearout calculations were done. The maximum bearing capacity is the maximum load the wall can take before extensive deformation and is calculated using equation 9.

$$R_n = 2.4dtF_u = 2.4 \cdot 1.625in \cdot 0.5\ in\ \cdot 45000\ psi = 87750lb$$

To calculate the maximum applied load on the shaft before bearing failure, the forces at the reactions were solved and the bearing load was set as the reaction force. Equation 3 is used to find the maximum applied load at the end of the shaft to not exceed the bearing capacity for the first support.

$$R_1 = \frac{P \cdot a}{L}; P = \frac{R \cdot L}{a} = \frac{87550\ lb \cdot 3.08\ in}{1.32\ in} = 204750\ lb$$

Equation 4 is used to find the maximum applied load at the end of the shaft to not exceed the bearing capacity at the second support.

$$R_2 = \frac{P}{L}(L + a); P = \frac{R \cdot L}{(L + a)} = \frac{87750\ lb \cdot 3.08\ in}{(3.08\ in + 1.32\ in)} = 61425\ lb$$

The maximum end load that can be applied to the shaft is lower for the second support, so the maximum load for bearing capacity is 61425 lbs.

The maximum tearout capacity is the maximum load the wall can take before the shaft tears out of the wall and is calculated using equation 10. Like the bearing capacity, the tearout capacity must be calculated for both supports. The calculations for the tearout capacity at the first support are done first.

$$R_n = 1.2 l_c t F_u = 1.2 \cdot 0.9\ in \cdot 0.5\ in \cdot 45000\ psi = 24300\ lb;\ R_1 = \frac{P \cdot a}{L}; P = \frac{R \cdot L}{a}$$

$$= \frac{24300\ lb \cdot 3.08\ in}{1.32\ in} = 56700\ lb$$

Next, the tearout capacity calculations at the second support are done.

$$R_n = 1.2 l_c t F_u = 1.2 \cdot 0.476\ in \cdot 0.5\ in \cdot 45000\ psi = 12852\ lb;\ R_1 = \frac{P}{L}(L + a); P = \frac{R \cdot L}{(L+a)} = \frac{12852\ lb \cdot 3.08\ in}{(3.08\ in + 1.32\ in)} = 8996\ lb$$

The second support has the lower tearout capacity, so the maximum load that can be applied to the end of the shaft without causing a tearout failure is 8996 lb.

These two walls are connected to the third wall, also made from 0.5-inch-thick aluminum, by ¼-20 socket head screws. This wall will be directly connected to the single flanged sliding block by two 5/16-18 socket head screws. The dimensions of this wall can be seen in figure A8 in the appendix. A drawing is shown in figure 22 below.

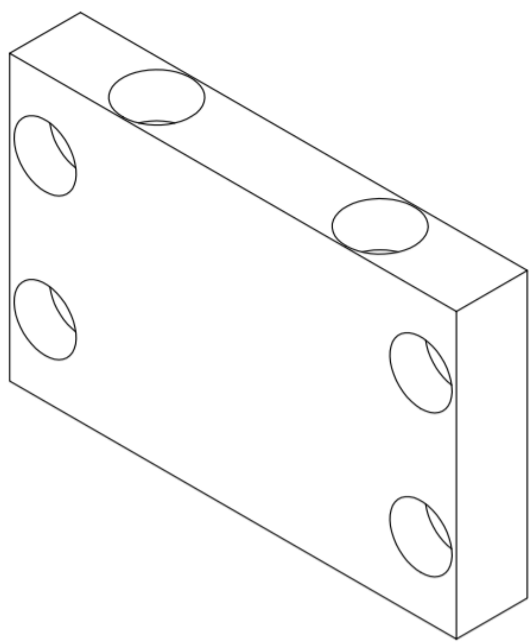


Figure 24: Sliding Block Connecting Wall

The final piece of the housing is made from sheet metal. A drawing of the sheet metal housing section is shown in figure 25 while the dimensions can be viewed in figure A9 in the appendix. This wall encloses the orientation device so that no pinch points are exposed. It will be connected to the two walls that hold the bearings with 10-32 button head screws.

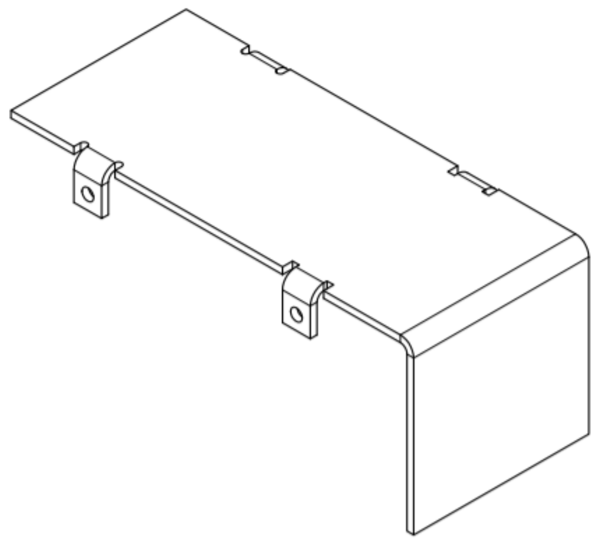


Figure 25: Housing Cover

Several types of bolts are used in this design. For assembling the housing, two 5/16-18 X 1.5-inch socket head cap screws (SHCS) were placed through the supporting wall that will connect to the slide block. Bolt shear calculations were done on these bolts to determine the maximum load the bolts can withstand, using equation 11.

$$\tau_{bolt} = \frac{4P}{\pi d^2} = \frac{4(3264\ lb)}{\pi(0.313\ in)^2} = 1448\ psi$$

The maximum shear load for a 5/16-18 x 1.5-inch SHCS is 3264 lbs [20]. The nominal diameter of the screw is 0.313 inches. The maximum allowable shear stress for the screws attaching the housing to the slide block is 1,448 psi, which is well above what the bolts would experience even during extreme use.

Since the holes through the wall and slide block are both clearance and counter bore, two 5/16-18 lock nuts are fastened onto the bolt, clamping the wall and slide block together. This size was chosen because there were existing holes of this size on the slide block per manufacturer's design. Next, four 1/4-20 X 1-inch SHCS were used to fasten the bearing housing walls to the supporting wall. Finally, four 10-32 X 3/8-inch button head cap screws (BHCS) were used to fasten the housing cover to the bearing housing walls. Button head was an aesthetic decision, as well as a safety decision. The button head prevents the operator's hands/gloves from getting caught or bashed upon the harsh edge. For the tooling to be inserted into the orientation device, two 8-32 X 3/8-inch were used to bolt the tool body handle to the desired block profile. In order to mount the graduated dial plate onto the knob, a 5/16-18 X ½-inch BHCS was screwed into the center of knob and dial to clamp them together. All screws are fabricated with black oxide coated stainless steel to prevent rusting over time. The head type for all the screws have a hex profile, as allen keys are the universal tool used on all fasteners. All drawings of specified fasteners used are in the appendix.

An acetal plastic ball bearing with 316 stainless steel balls manufactured by McMaster-Carr (Part #6455K43), shown in figure 26, was selected for use in the housing of the device. A plastic bearing was chosen because it is lightweight and would minimize the weight of the overall device. The bearing chosen fits a shaft diameter of 1 inch with a maximum unilateral tolerance of .004 inches. The shaft will be press fit into the bearing and the bearing is press fit into the housing. The outer diameter of the bearing is 1.625 inches with a maximum unilateral tolerance of -.004 inches [21]. The width is 0.5 inches which matches the width of the housing wall. Four of these are used in the design; each shaft is press fit into two bearings with one bearing used for each side of the housing. This bearing does not require lubrication so it will not need regular maintenance. Since this will be manually turned at a low angular velocity within one revolution, there is no need to consider the maximum speed and typical bearing life calculations would not be meaningful. The radial load capacity is 50 and 75 pounds for static and dynamic loading, respectively.

Figure 26: Acetal Plastic 316 Stainless Steel Ball Bearing (6455K43) [21]

Set screws are used in this design in order to fasten parts together and prevent any unwanted axial movement. Four small 8-36 X 3/16-inch set screws were each used to set the dial, the tool body, and the two pulleys to prevent any axial motion along the shafts. Failure to keep all components aligned could cause the belt to snap and damage the device. The set screws are also vital for the pulleys and dial to ensure that they do not rotate about the shaft. The material of the set screws is black oxide coated stainless steel, which is the same material used for the other screws. The hex profile was selected for the general use of Allen hex keys used at the plant. The set screw is shown in Figure 27.

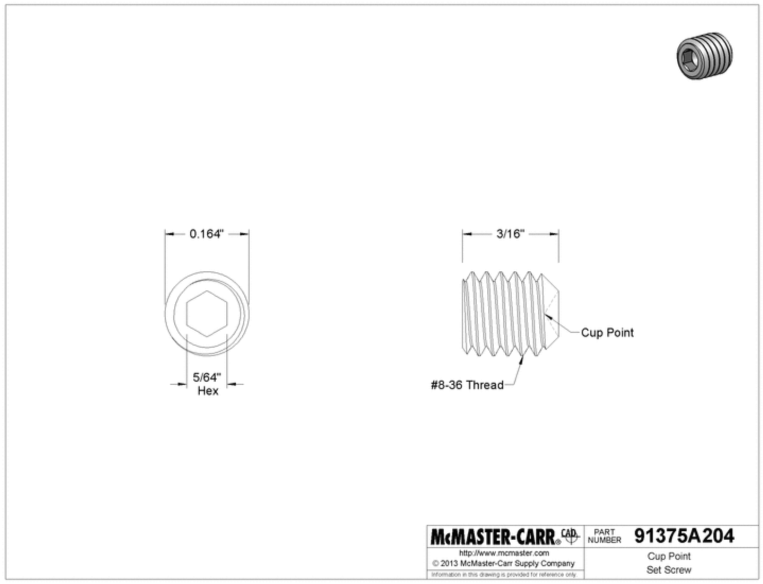


Figure 27: 8-36 X 3/16" Set screw

The knob, shown in figure 28, is press fit into the shaft and a set screw keeps the knob in place. The knob is a cylindrical disk machined from aluminum with a diameter of 2 inches and a thickness of 0.25 inch. The knob handle, which is press fit into the shaft and machined from the same piece of aluminum, has a 0.753-inch diameter. The knob handle has a 0.177-inch diameter hole bored into the side for the set screw, and a hole sized for a 5/16-18 screw to fasten the dial plate onto the knob.

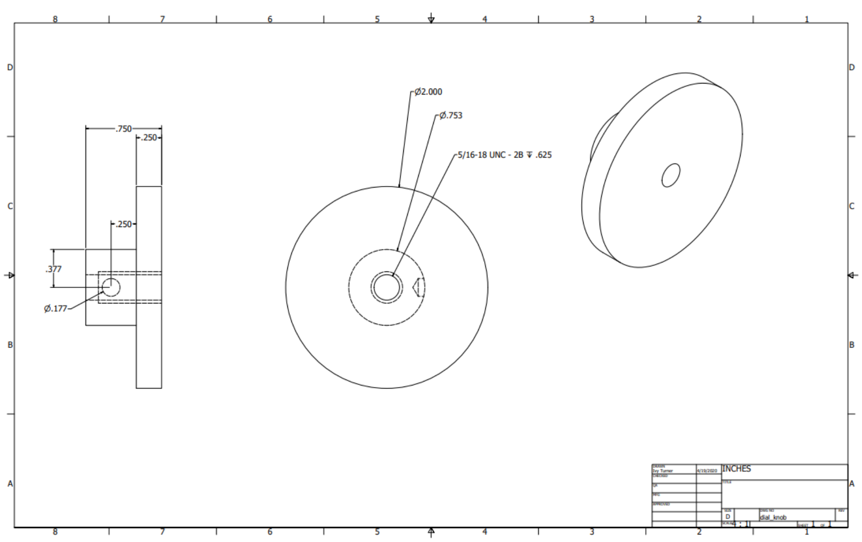


Figure 28: Drawing of Dial Knob

A graduated engraved flat disk manufactured by Berg is used as the dial for the orientation device. A drawing of the A9-2 engraved flat disk is shown in figure 29 below.

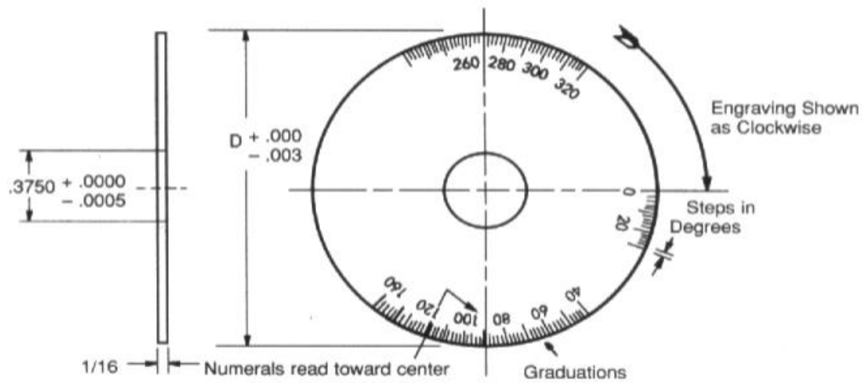


Figure 29: Drawing of Dial Plate [13]

The plate is made of black anodized aluminum, has an outer diameter of 2 inches, a bore diameter of 3/8 inch, and a thickness of 1/16 inch. The graduation range is from 0° to 360°. Because the operators on the floor who will be using this device read angles from 0°-180°, the dial plate is custom-ordered to go from 0°-180° on both the upper section and lower section. A 5/16-18 screw is used to fasten the dial plate to the knob. A notch is engraved on the housing at the 0° mark to allow operators to determine the angle of rotation.

1.9 Design Summary/Conclusion

The orientation device for valve bodies was designed to be both precise and easy to use on the manufacturing floor. Operators can easily adjust the tool handle to an angle within the tolerance of the part without encountering any hazards due to pinch points. The overall design consists of two L-series pulleys and an L-series timing belt supported on two shafts enclosed in a housing. The two shafts are 1-inch steel tubes with an inner diameter of 0.76 inch. The input shaft has a knob press fit into the shaft with a graduated dial plate bolted onto the face. A notch is engraved at the 0° point to allow for the determination of the input angle. The output shaft has a key machined into part of the inner diameter that mates with the keyhole in the tooling handle. Housing walls made of aluminum support the shafts, which are fit through ball bearings that are press fit into the housing. A piece of sheet metal encloses the remainder of the housing. A variety of screws, bolts, and nuts are used to fasten the housing to the slide block that allows the orientation device to move to accommodate various tube lengths, and to fasten the sections of the housing together. This design is largely composed of commercially available, inexpensive parts, with the rest of the parts easily fabricated by machinists at Parker Hannifin. When this device is implemented, it will greatly reduce the amount of scrapped parts caused by valve body placement being out of tolerance and save the company money.

2. FUTURE WORK

Future work for this design includes manufacturing and integrating the slug blowout device. Due to impacts from COVID-19, the orientation device could not be manufactured and implemented in the plant as originally scheduled. Once the design is returned to Parker Hannifin, parts will be ordered and machined as necessary for manufacturing. Additionally, the device that blows compressed air through the tube to expel the slug is not currently functioning. In the future, the slug blowout mechanism will be modified to allow full functioning for this pierce and press machine and will be integrated with the orientation device, which has considerations for the slug blowout built in.

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4. APPENDIX

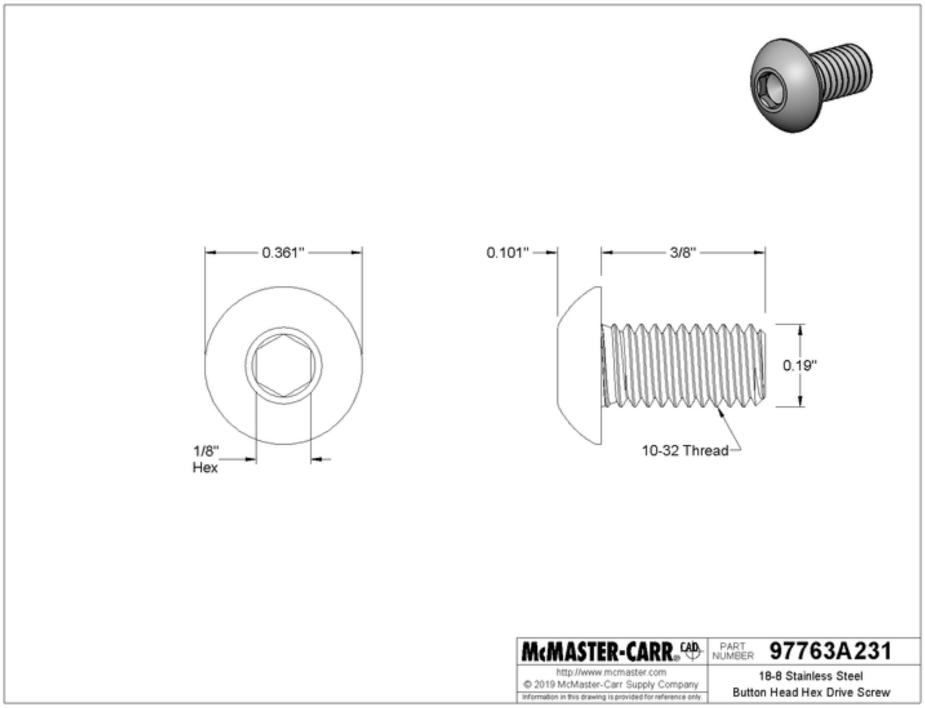


Figure A1: 10-32 X 3/8 BHCS

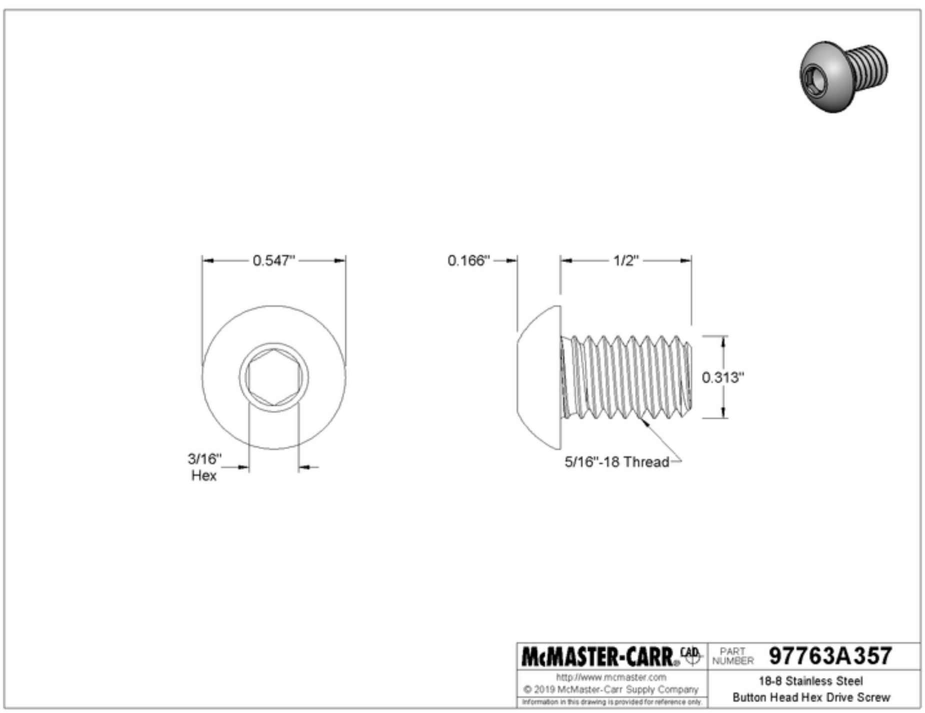


Figure A2: 5/16-18 X ½" BHCS

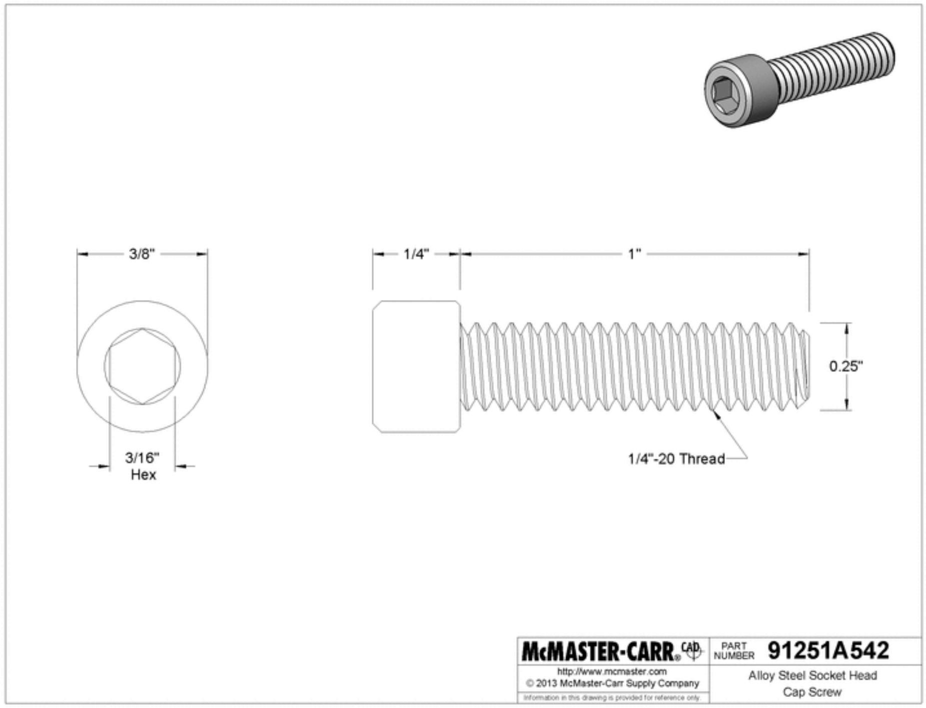


Figure A3: ¼-20 X 1” SHCS

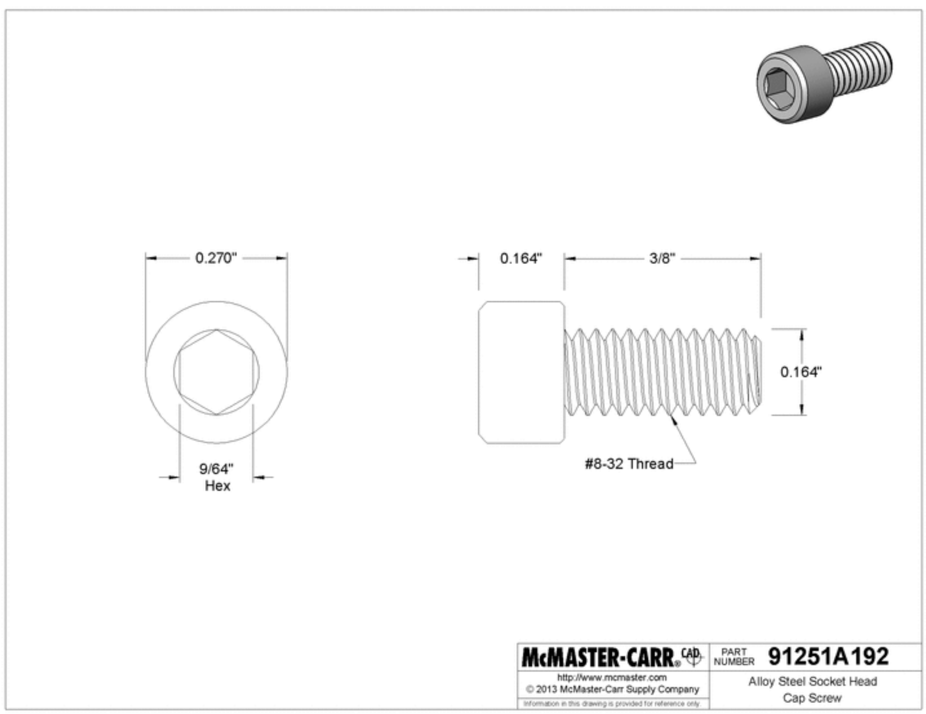


Figure A4: 8-32 X 3/8” SHCS

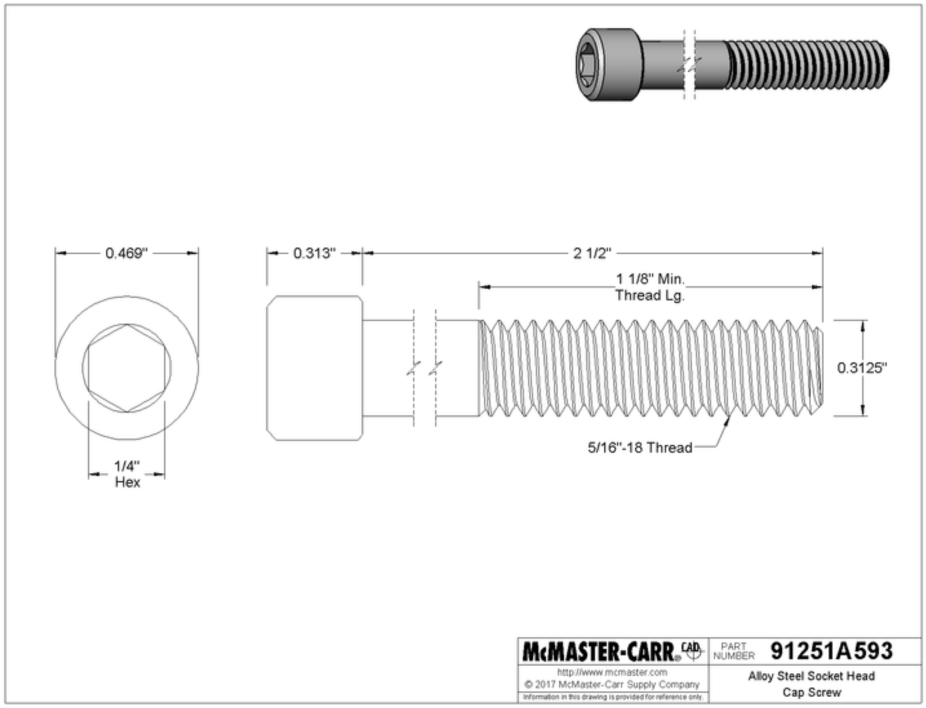


Figure A5: 5/16-18 X 2-½" SHCS

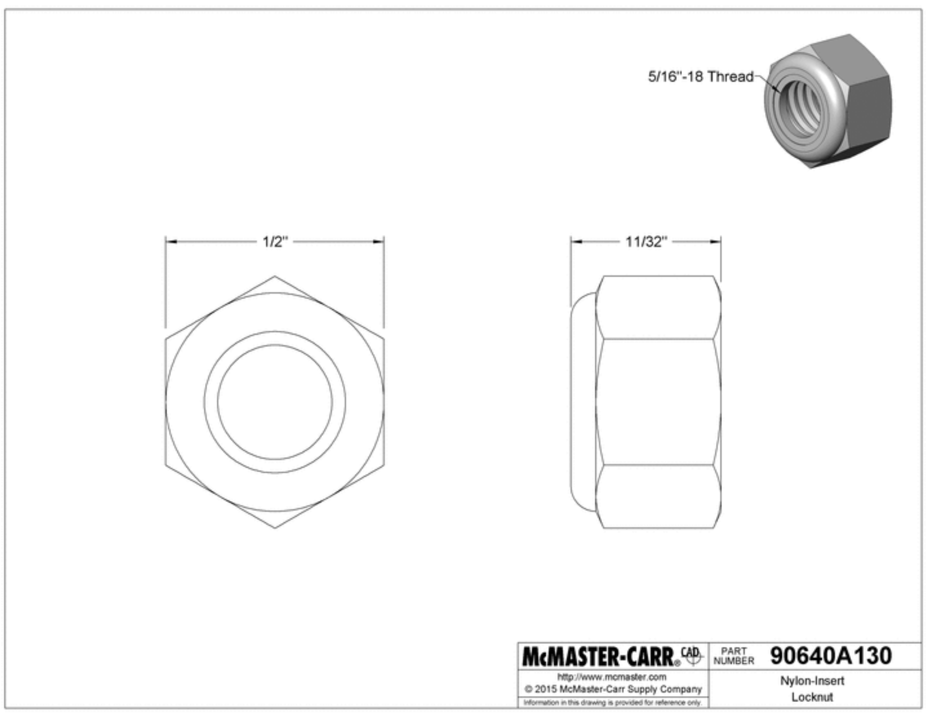


Figure A6: 5/16-18 Nylon Locknut

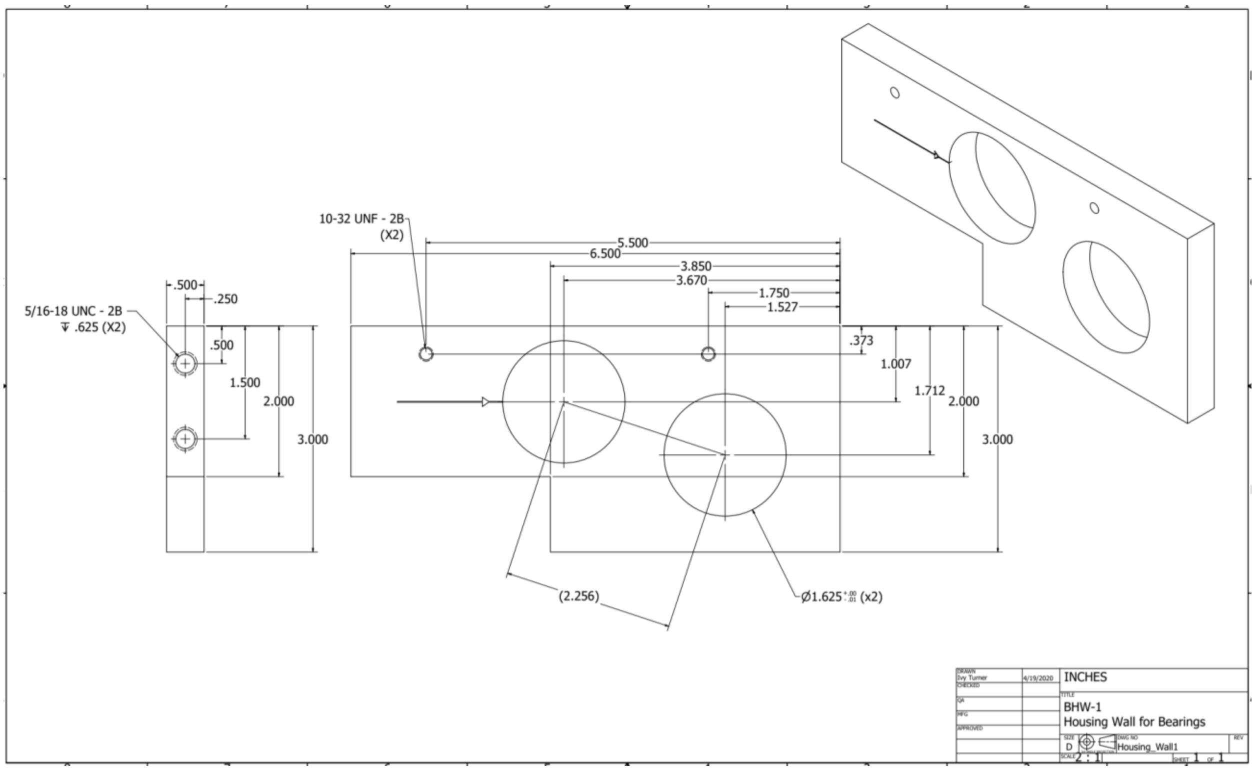


Figure A7: Bearing Wall

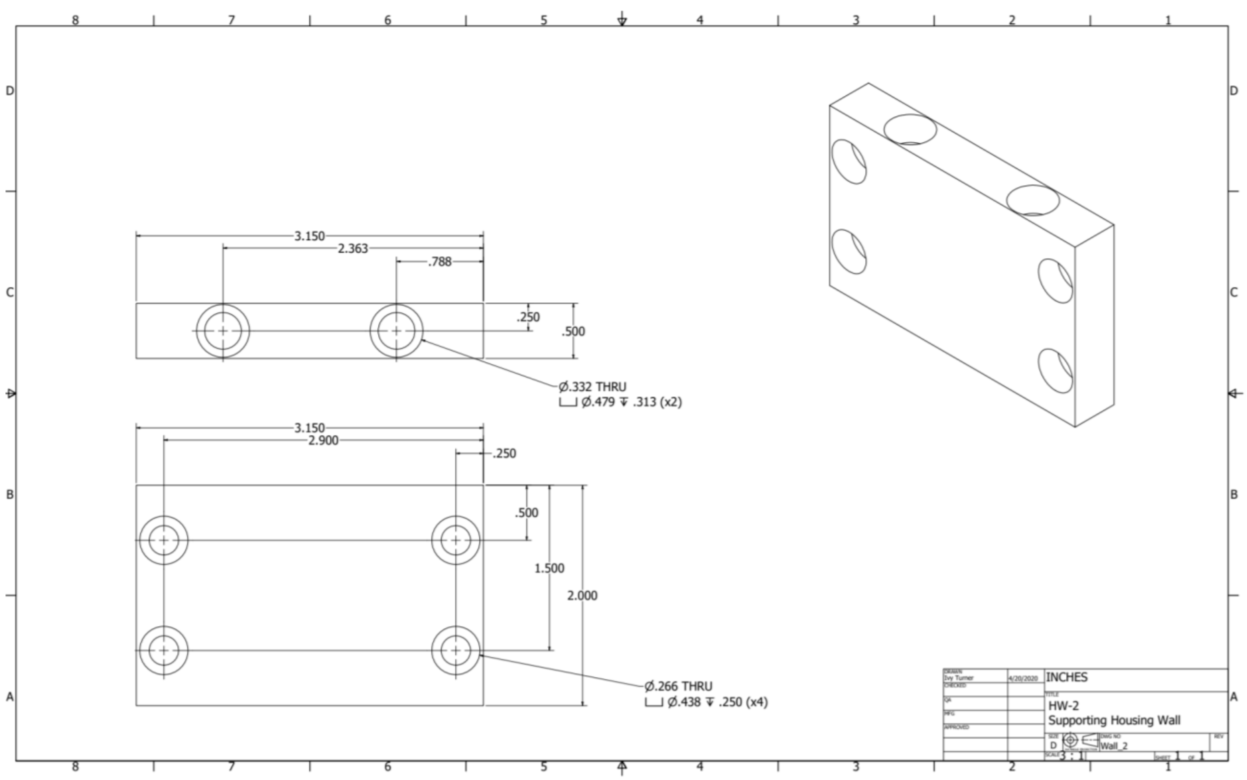


Figure A8: Sliding Block Connecting Wall

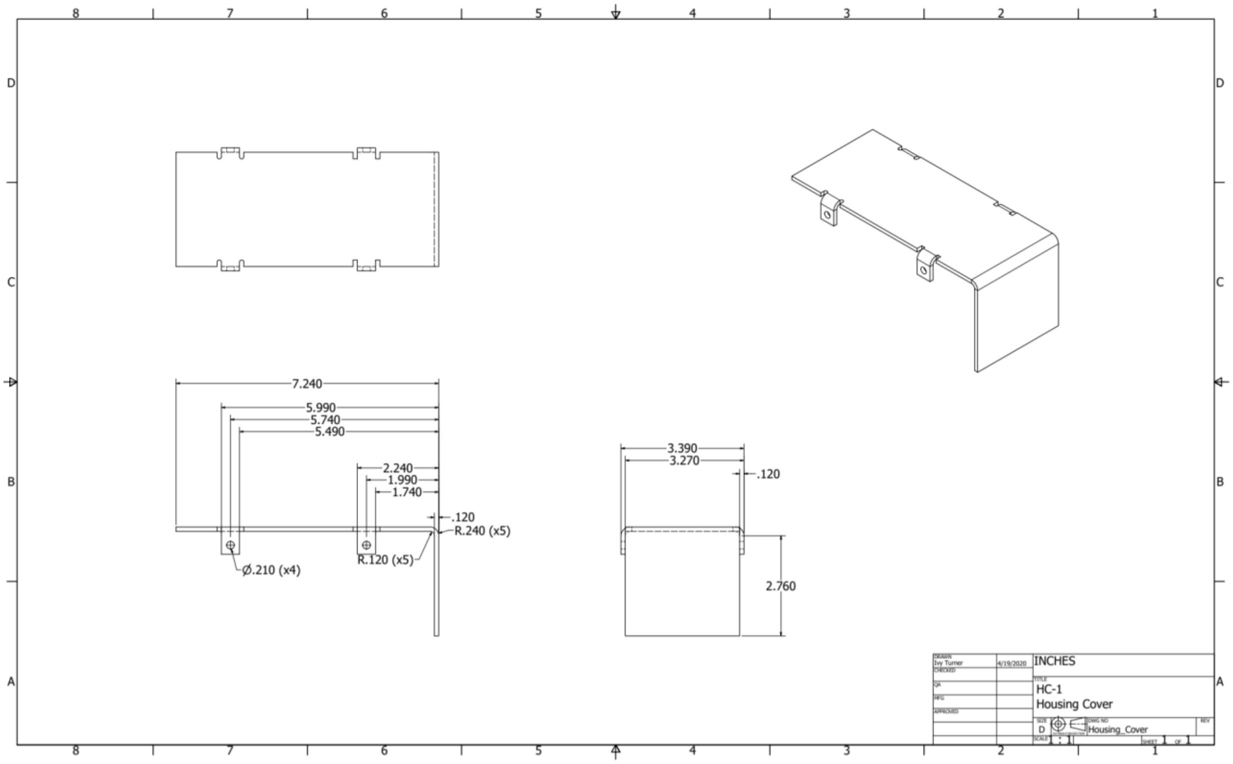


Figure A9: Housing Cover Dimensions

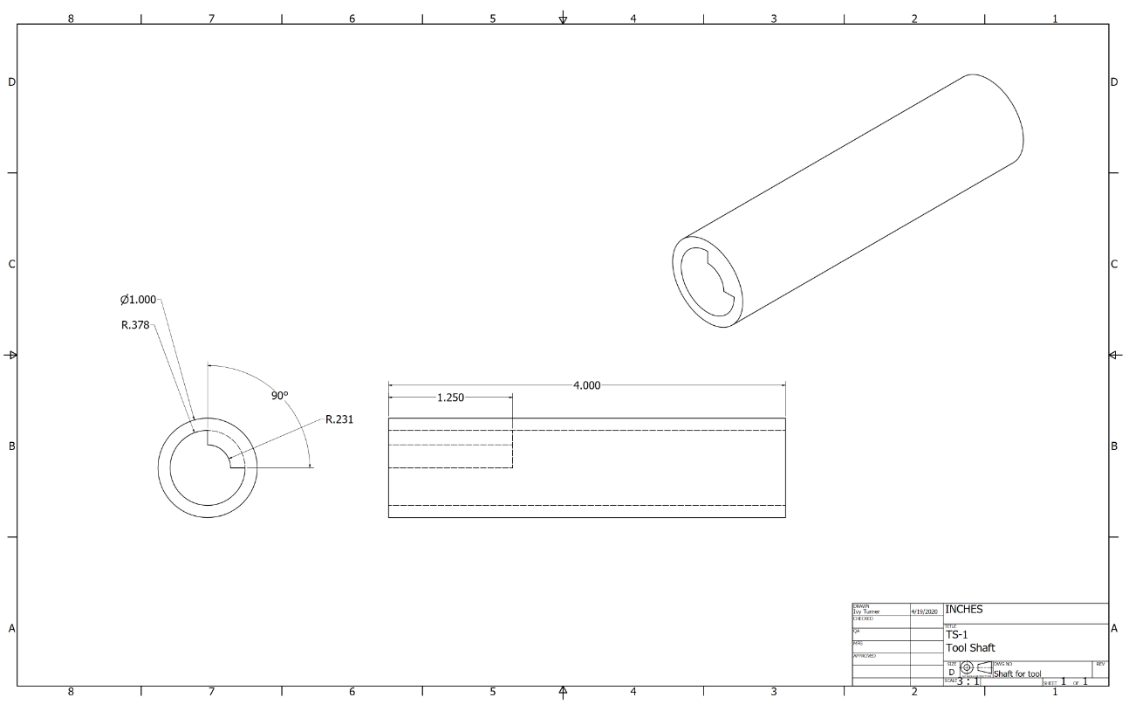


Figure A10: Tooling Shaft

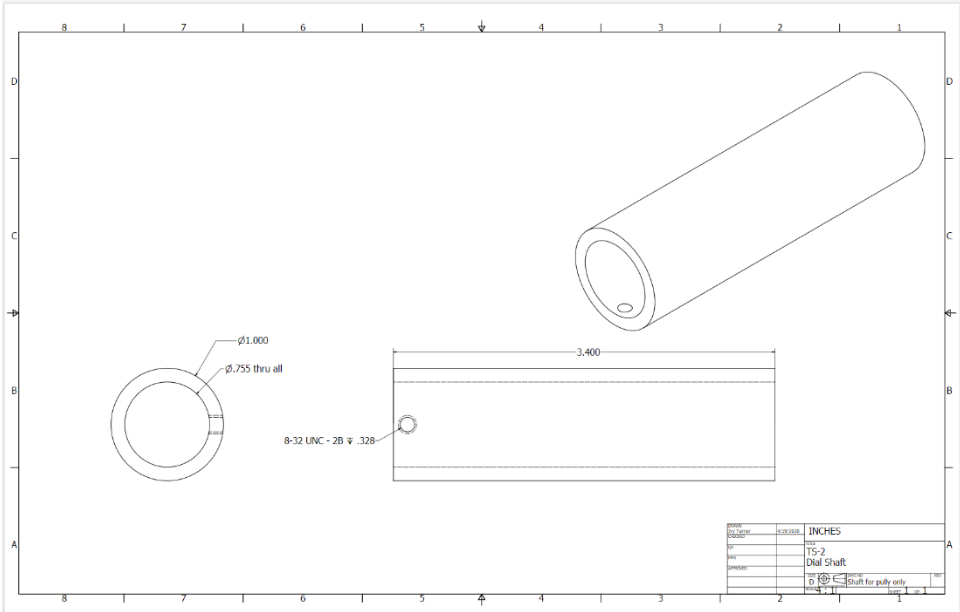
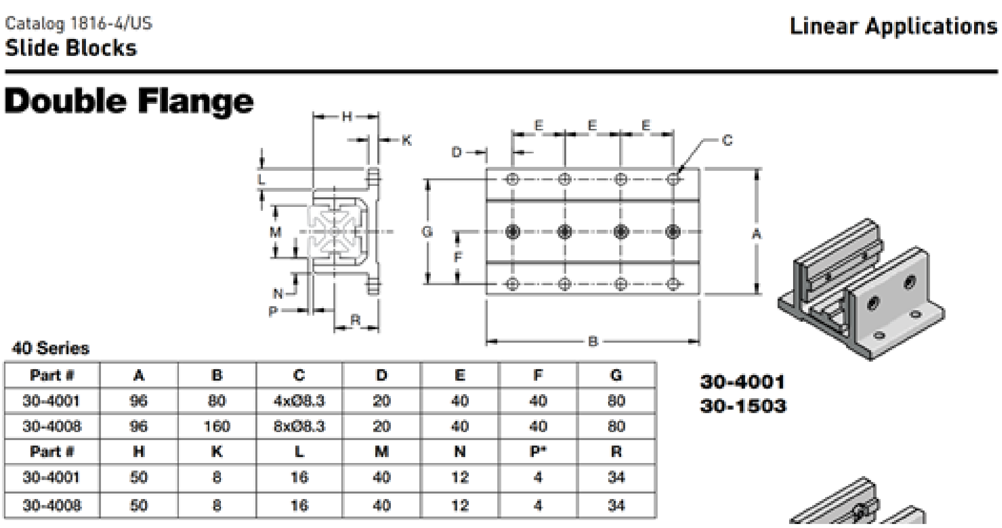


Figure A11: Dial Shaft



Catalog 1816-4/US

Slide Blocks

Linear Applications

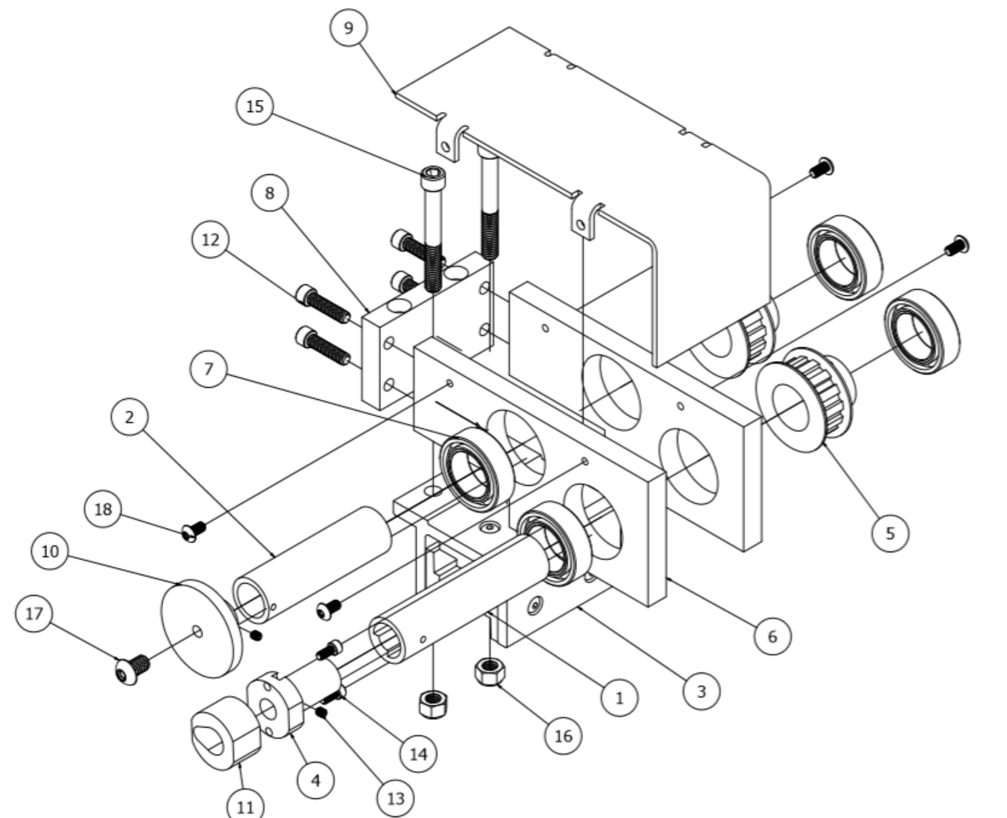
Double Flange

40 Series

Part #	A	B	C	D	E	F	G
30-4001	96	80	4xØ8.3	20	40	40	80
30-4008	96	160	8xØ8.3	20	40	40	80
Part #	**H**	**K**	**L**	**M**	**N**	**P***	**R**
30-4001	50	8	16	40	12	4	34
30-4008	50	8	16	40	12	4	34

Figure A12: Double Flange Slide Block

PARTS LIST			
ITEM	QTY	PART NUMBER	DESCRIPTION
1	1	TS-1	Tool Shaft
2	1	TS-2	Dial Shaft
3	1	30-4001	Slide Block
4	1	TB-1	Tool Body Handle
5	2	6495K140	1" DIA Pulley
6	2	BHW-1	Bearing Housing Wall
7	4	6455K880	Ball Bearing
8	1	HW-2	Supporting Housing Wall
9	1	HC-1	Housing Cover
10	1	DK-1	Dial Knob
11	1	TBP-756024	Block Profile
12	4	91251A542	.25-20x1 SHCS
13	2	91375A204	8-36 X 3/16 Set Screw
14	2	91251A192	8-32 X 3/8 SHCS
15	2	91251A593	5.16-18 X 2.5 SHCS
16	2	90640A130	5/16-18 Lock Nut
17	1	97763A357	5/16-18 X 1/2 BHCS
18	4	97763A231	10-32 X 3/8 BHCS



DRAWN Ivy Turner	4/20/2020	INCHES
CHECKED		TITLE Assembly: Block-to-Valve Body Orientation Device
QA		
MFG		SIZE D DWG NO Tooling_ASMv2_EXP-1 REV
APPROVED		SCALE SHEET 1 OF 1