Automated Machine Tender

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Statement of Disclaimer

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1.0 Introduction

The use of CNC (Computerized Numerical Control) machines automate production in factories through programs controlling manual operations such as milling or lathing. The manufacturing process can be automated beyond the use of CNC machines through the addition of a robotic tender capable of carrying out the same tasks as a human operator. A tender can be integrated into the CNC machine itself, or purchased separately and configured to autonomously operate a particular machine. Smaller companies making limited production runs may not find it economically feasible to purchase large scale integrated machine tenders, but would benefit from a versatile autonomous machine tender that can be utilized for their specific application. The objective of this project is to design and build a relatively low-cost robotic operator capable of loading and unloading a CNC mill. Our machine tender will come with its own code ready to program instead of requiring connection to another machine to program its function. The tender will be capable of accepting inputs and providing outputs to any Haas CNC VF-2, VF-3, ToolRoom, or minimill connected. The device will also be capable of being swapped out for a human operator for the CNC machine. Our sponsor is Haas Automation, Inc., and our point of contact at Haas is Mr. Bill Tandrow, Vice President of Mechanical Engineering. The engineering advisor for this project is Professor Eileen Rossman of the Cal Poly San Luis Obispo Mechanical Engineering Department.

2.0 Background

2.1 Existing Products

Existing machine tenders and robotic arms typically include multiple servo, permanent magnet, or brushless AC motor positions to provide multi axis movement capabilities. The FANUC Robotics M-20iA Series robotic arms (Figure 2.1.1) use multiple brushless AC motors to provide six degrees of freedom, a repeatability of 0.10mm, and a payload capacity of 20kg. This model houses all wires internally to prevent dress out issues that could potentially lead to downtime [1].

Figure 2.1.1: *FANUC Robotics M-20iA Robotic Arm*

The Versabuilt VBX-160 is an expensive, self contained machine tender that is compatible with many CNC machines (Figure 2.1.2). Although it costs $89,995, this model holds both unmachined metal blanks and finished parts by incorporating shelves within the machine tender apparatus. These shelves provide a versatile number, and shape of parts that this machine can manipulate. In order to grip parts, the Versabuilt requires MultiGrip soft jaws to be machined for each specific part. The soft jaws can then be used to machine parts through up to three different operations. The VBX-160 can load, flip, transfer, and unload a part that undergoes two operations in roughly 30 seconds [2].

Figure 2.1.2: *Versabuilt Machine Tender*

Electronic Pick-Place machines (SMT machines) are typically used for assembling electronics using parts much smaller than a CNC machine tender. These machines, however, have a similar function to a machine tender. They are designed for high speed and high precision by using small vacuum suction cups to manipulate parts. Typically parts are picked from trays or reels before being placed into position [3].

2.2 Existing Information

The current information on machine tending includes what the machine accomplishes, the improvement over current methods of tending to machines, and the likely future such devices will face. The use of robotics in machine tending was originally intended to replace human workers in order to improve machine uptime. Without having as many limitations as human workers, while still having multiple advantages, the machine uptime for operated machinery has increased through the use of these machine tenders. In India, one such facility experienced a "30 percent increase in machine uptime" for devices with machine tenders installed [4].

Despite these advancements, there are still areas where robotics are lacking in function. Humans still need to feed the tenders material for them to use once they run out. Operations where residual material is left over (such as burr in a CNC machine) still needs to be cleared out. Human safety has to be factored into designing and operating machine tending devices. One journal notes how, should machine tenders be left to run overnight, they would not be able to have any unplanned halts in production found out until workers arrived the next day [5]. For all the ways machine tending is lacking, there are people currently designing solutions to such issues. To remove debris, a compressed air gun can be built into a machine tender. Halts in production can be detected by setting up a system to monitor machine operation in a factory. As problems arise, solutions are researched, so that base functions of these devices can be further improved.

2.3 Applicable Codes and Standards

International/Ingress Protection Rating, or IP Code for short, is a weatherproofing standard “developed by a technical committee of the International Electrotechnical Commission” (IEC) under the designation IEC 60529 and “was adopted as an American National Standard” by the American National Standards Institute (ANSI) [6]. IP code describes a machine’s protection against ingress by solid objects and protection against ingress by water. The code takes the form of “IP XY,” where X is an integer from 0 to 6 representing protection against solid objects of various corresponding sizes, and Y is an integer from 0 to 8 representing protection against various levels of contact with water.

The Occupational Safety & Health Administration (OSHA) maintains a chapter in their technical manual (OTM) on Industrial Robots and Robot System Safety (Section IV, Chapter 4). The chapter details types of robots, hazards, investigation guidelines and safety practices [7].

These practices reference the document designated “ANSI/RIA R15.06-1999 standard for Industrial Robots and Robot Systems - Safety Requirements” published by the Robotic Industries Association (RIA) and approved by ANSI in 1999 [8]. The document lists a number of safety requirements, among them that an emergency stop must be compliant with NFPA 79.

Accuracy and Repeatability are measured by standard ISO 230-2, entitled “Test code for machine tools - Part 2: Determination of accuracy and repeatability of positioning of numerically controlled axes,” developed by the International Organization for Standardization (ISO) and last updated in 2014 [9].

2.4 Relevant Patents

Peter Movsesian patented a mobile robotic arm (Figure 2.3.1) designed to retrieve household items from various heights to optimize storage in hard to reach places, ideal for handicapped individuals [10].

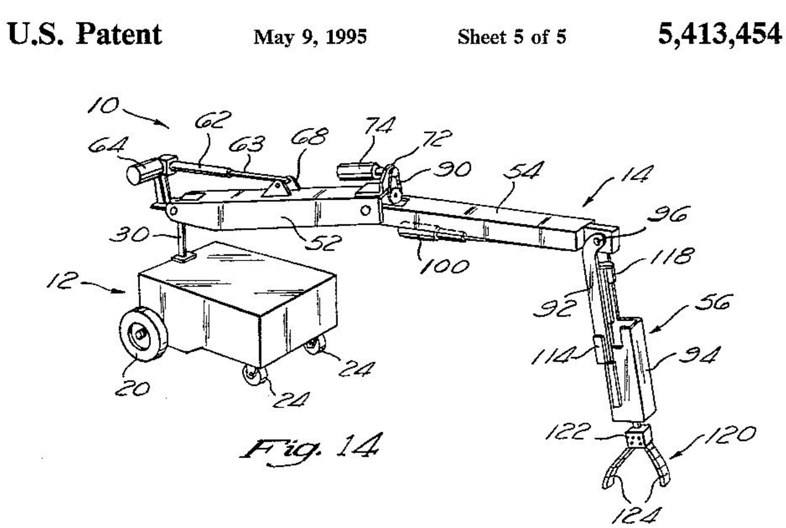

Figure 2.3.1: *Mobile Robotic Arm*

Many robotic arms are specifically designed for industrial applications, which makes this specific design unique. Although it has similar objectives to our application, this robotic arm includes a mobile base which is electronically configured to a remote controller. Our product will stand alone and not require constant human interaction.

Similar to our industrial application, Kabushiki Kaisha Komatsu Seisakusho from Japan owns the patent to a "Robot Arm for an Industrial Robot" (Figure 2.3.2). This robot is designed for "high ability of movement, excellent reachability and high accuracy by virtue of a comparatively small amount of rotational movement and a comparatively large amount of translational movement". At each connection of arms, this robot requires a certain inclination angle and includes an electric motor to drive each arm. Unlike our machine tender, this robotic arm is designed for a variety of applications; therefore, it does not include a part manipulator specifically designed for machine tending [11].

U.S. Patent **Feb. 27, 1990** **Sheet 2 of 6** **4,904,148**

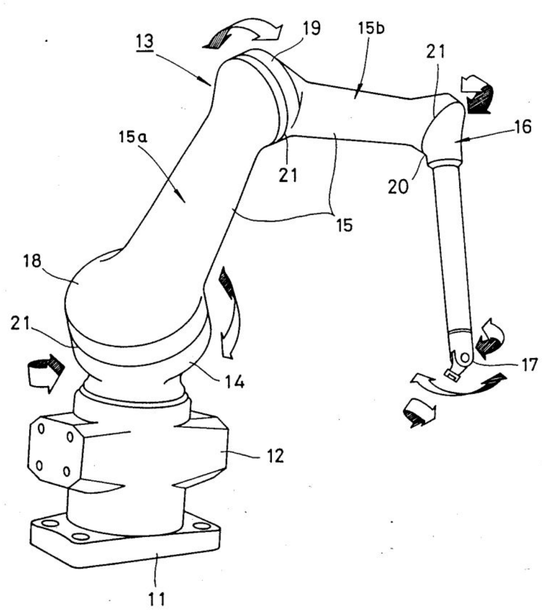

FIG. 2

Figure 2.3.2: *Robot Arm for an Industrial Robot*

Although not specifically a robotic arm, A-dec, Inc., owns the patent to a "Telescoping Extension Arm for Supporting a Monitor" (Figure 2.3.3). Figure 2.3.3 is cropped to show the telescoping action of the arm. This is a simple device with only one telescoping member designed to lock in a specific location rather than be constantly moving. Our robotic arm will undergo substantially more cycles than A-dec, Inc's monitor support. The end attachment of this arm is also designed to support a monitor rather than manipulating metal objects [12].

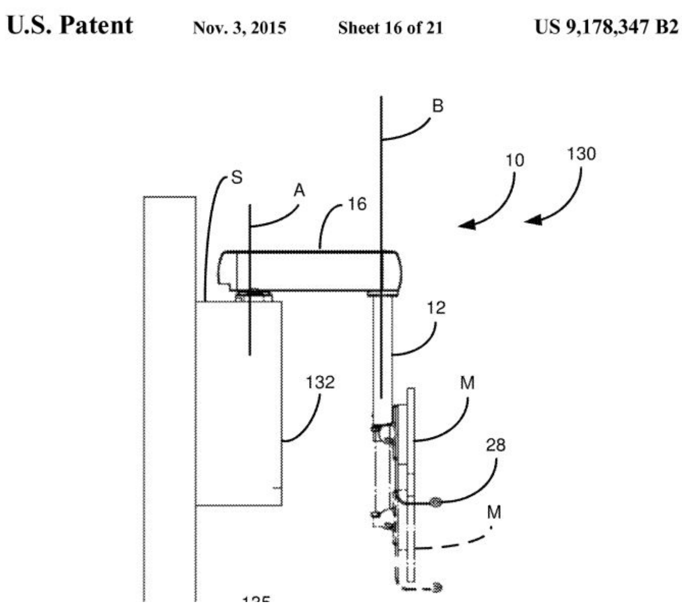

Figure 2.3.3: *Telescoping Extension Arm for Supporting a Monitor*

Kawabuchi Mechanical Engineering Laboratory, Inc., located in Tokyo, Japan has been issued a patent by the World Intellectual Property Organization (WIPO) titled, "Linearly Moving Extendable Mechanism and Robot Arm Equipped with a Linearly Moving Extendable Mechanism" (Figure 2.3.4). Although the patent was filed in Japanese and needed to be translated, this mechanism uses multiple connected blocks to extend and retract. These blocks are not firmly coupled, which allows more freedom of movement and gives the blocks the ability to be stored in a smaller space [13].

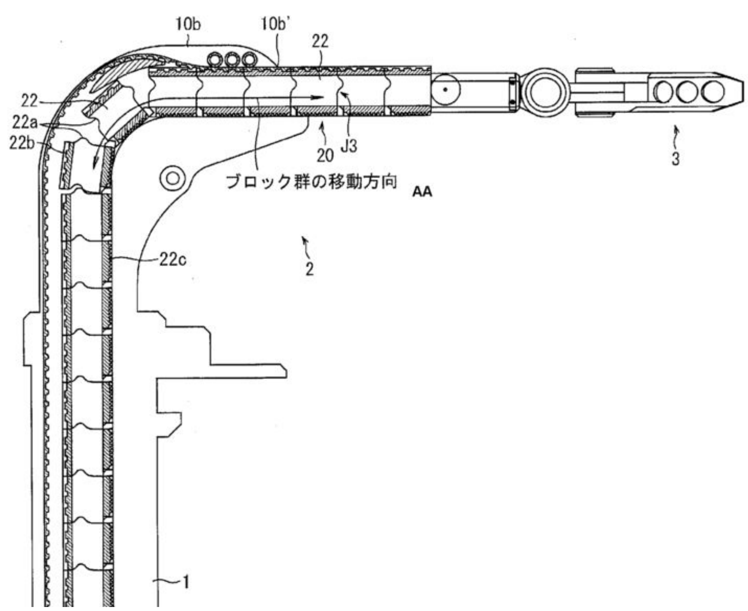

Figure 2.3.3: *Linearly Moving Extendable Mechanism and Robot Arm Equipped with a Linearly Moving Extendable Mechanism*

3.0 Objectives

Our team developed objectives based off of the requirements given to us by our sponsor for the machine tender they wanted developed. Successful machine tending for either a VF-2, VF-3, MiniMill, or ToolRoom mills will require a user friendly and cost effective machine. We will strive to build a durable machine tender within a budget of $3500, to make it relatively affordable compared to the cost of a Haas CNC machine. It will be able to load a 5 lb blank with the dimensions of 2” x 4” x 6” within 30 seconds, and then unload the finished piece within another 30 seconds. Completing this process quickly will allow for the efficient manufacturing of multiple pieces. Additionally, metal blanks of varying size and material need to be accounted for to increase the variety of applications for the machine tender. This process will be completed with a repeatability of 0.025” TIR (Total Indicated Run-Out) in order to guarantee the machine tender can successfully assist in the machining of multiple pieces. Having automated machine tending is only useful if multiple pieces can be machined; therefore our machine tender will be able to store and ultimately insert at least 40 parts into the CNC machine.

Repairability and reconfigurability are also important to our design of a machine tender. Our design will allow for repair with basic electro-mechanical understanding and shop tools. This will be incorporated into our goal of producing a product with a Mean Time To Repair (MTTR) of under 24 hours to limit potential downtime. Any reconfiguration can be completed within 20 minutes to allow the operator to quickly change between different processes for maximum adaptability.

Designing any machine requires safety to be taken into account to prevent injury or damage to either the machine tender or the CNC machine. Our machine tender will incorporate a fast acting emergency stop, and pause if foreign objects are sensed within its range of motion.

Completing successful machine tending will require basic communication with the specified Haas CNC machine. Our machine will use a four flag communication system, while operating independent of the CNC control system.

The risk column in Table 3.1 indicates how difficult the corresponding parameter will be to achieve. The different degrees of risk are divided between three groups, low (L), medium (M), and high (H). The compliance portion indicates how the success of each parameter will be verified. We will institute different methods of verification including analysis (A), testing (T), and Inspection (I).

The engineering specifications in Table 3.1 were incorporated into a quality function deployment (QFD) to compare to our customer requirements (Appendix A). This document also shows how each engineering specification impacts the others, which will allow us to consider the full effects of achieving each specification. For example, keeping our project within a budget of $3500 is directly negatively impacted by having a 1.5 meter reach. The engineering specifications were chosen based on the needs of our customers, specifically Haas Automation, Inc. Our customers required a machine tender that can safely and accurately place and retrieve items from a CNC machine in a timely manner all within a reasonable budget.

Table 3.1: *Engineering Requirements*

Spec.	Parameter Description	Requirement or Target	Tolerance	Risk	Compliance
1	Budget	$3,500	Max	M	A
2	Loading Speed	30 seconds	Max	M	T, A
3	Unloading Speed	30 seconds	Max	M	T, A
4	Repeatability	0.025”	Max	M	T, A, I
5	Fast Acting Emergency Stop	0.5 seconds	Max	H	T, A
6	Payload	5 lbs	Min	M	T, A, I
7	Size of Part Manipulated	2" x 4" x 6"	Min	M	T, A, I
8	Stores Multiple Parts	40 parts	Min	M	T, I
9	Haas Mill Compatibility	VF-2/VF-3/MiniMill/ToolRoom	Absolute	L	T, I
10	Mean Time to Repair (MTTR)	24 hours MTTR	Max	H	A, I
11	Man/Machine Tender Swap Time	10 minutes	± 5 min	M	A, T, I
12	Time to Reconfigure	20 minutes	± 5 min	M	A, T, I
13	Reach	1.5 meters	± 0.5 m	M	A, T, I
14	Basic Communication	4 Flag	Absolute	L	I
15	Sense foreign objects	Within range of motion	Absolute	H	T, A

We were able to analyze how existing products meet our customer requirements and engineering specifications and how they compare to our design. In this QFD we specifically analyzed the Versabuilt machine tender, FANUC robotics M-20iA robotic arm, human CNC machine operator because they represent the most direct competition.

Another function of our QFD is to show the relative importance of each customer requirement to each of our customers. To determine our customers, anyone that could potentially come in contact with the machine tender was considered. As our sponsor, Haas Automation Inc., became our first customer; however the needs of the manufacturer, buyer, and operator were also accounted for.

Within the QFD, we are able to consider correlations between the engineering specifications themselves. A majority of the engineering specifications do not impact each other, except for budget. Budget was impacted by over half of the other engineering specifications.

4.0 Design Development

4.1 Concepts

Initial concept development began with ideation primarily via brainwriting, brainstorming and the SCAMPER method. Existing devices capable of carrying out the same or related tasks were researched to provide insight into the variety of possible solutions to each function. Meetings with Mr. Tandrow helped to refine and clarify project requirements, which aided in narrowing down the selection of concepts. Mr. Tandrow also encouraged the investigation of non-traditional approaches to the challenge of machine tending, which informed the process of developing conceptual designs. Sketches to accompany the following concept descriptions are located in Appendix B.

Collapsing Hanging Arm: This device would consist of a robotic arm capable of folding into itself for storage, hung at the base from a linear track mounted onto a structural arch. This arch would either stand in front of the doors to the mill, or over the entire mill itself. The arm would have a modular gripper that could accept a variety of tools that would grip with the same motion as a parallel gripper.

Crane Model: This device would operate in a manner similar to that of a crane. The inside of the mill would be accessed by rotating a track into the doorway, upon which a carriage could ride back and forth. A gripper would extend down from the carriage to pick up and place the part.

Granular Grabber: This device would consist of a telescoping arm extending from a two-dimensional rectilinear base. A cart containing the parts to be machined would sit between the enclosed structure and the mill. At the end of the arm a flexible membrane filled with a granular material could be formed to a part, and the air within vacuumed out to act as a gripper.

Horizontal Pusher: Unmachined items would be placed in front of a linear actuator via a simple rail system. Next, the linear actuator will push the unmachined blank with one degree of freedom, and place it in the CNC machine. After the CNC machine completes the operation, the attached grabber will pull the finished product out of the CNC machine.

Multi-grabber Rail System: The multi-grabber rail system would consist of two carts that slide along rails. One cart would be designated to pick up and place unmachined parts, while the other would take finished parts from the CNC machine and place them in a holding location. The rail system will be fixed and predefined to only allow motion in specific directions. Each cart would consist of a multi-fingered grabber below to pick up parts.

Rotary Pallet Shelf: This device would load pallets containing an array of parts laid out in a preset orientation. Locating pins extending from parallel jaws would align with corresponding holes in each pallet. These jaws, which would clamp onto and hold the pallet, would be mounted onto a 4-bar linkage actuated to rotate out and set the pallet into the CNC mill fixturing. The shelf would contain two columns of pallets: one side lifting and the other side lowering. Linear actuators would move pallets from one column to another, enabling the rotation of parts.

Multi Axis Robotic Arm: Already the industry standard for multiple applications, the Multi Axis Robotic Arm provides a versatile approach to machine tending.

4.2 Selection

After completing Pugh Matrices (Appendix D) for the functions associated with machine tending, the results were used to develop concepts. By taking the results and organizing them by which categories had multiple positives and negatives (indicated by positive and negative ones) we could quantify the desirability of a design. Within the Pugh Matrices, specific movement and grabbing options were initially analyzed in relation to a human capabilities. This analysis produced a highly rated option for each function, which replaced human capabilities as the new datum. Using an iterative process allowed each option within each function to be directly compared to each other.

With each option given a score based on our engineering specifications, the Pugh Matrices were used to develop complete concepts using a combination of options within the Pugh Matrices. These concepts were entered into a Go/No Go matrix (Appendix E) to verify that each concept has the ability to satisfy the minimum absolute requirements for our product to be considered successful. This Pugh Matrix operated on a simple pass/fail system where a concept had to pass everything listed (which involved the most basic of needed functions) in order to move on to further considerations.

After this verification process, the fully developed concepts were entered into a detailed decision matrix (Appendix F) to provide a structured and less biased way to choose a final design. Each concept was rated given its ability to meet design specifications. The design specifications were also weighted given their respective importance. Multiplying the weight of the design specification by the rating any particular concept received gave the concept a score for the respective design consideration. For each concept, a final score was calculated by summing the scores the concept received for each design requirement.

As can be seen in Table 4.2.1, the Granular Grabber Received the highest total score of 151, though both the Horizontal Pusher and Collapsing Hanging Arm received only a slightly lower total score.

Table 4.2.1: *Detailed Decision Matrix Results for Overall System Concepts*

Concept	Total Score
Granular Grabber	151
Horizontal Pusher	143
Collapsing Hanging Arm	141
Rotary Palette Shelf	136
Multi-Axis Robotic Arm	134
Crane Model	132
Multi Grabber Rail System	123

The Granular Grabber excelled at simplicity of motion and versatility at being able to grip parts of variable contours without replacing the grabbing attachment. It scored poorly in the reconfigurability criteria because of its constricted motion. Ultimately, this concept satisfied criteria that had a high weighting associated with them, such as safety and interchangeability with a human operator, which led to the Granular Grabber receiving the highest score..

The Crane model provided an extremely average solution to most criteria and performed poorly in both the strength and speed category. There were a few criteria the Crane Model excelled at, however, such as minimal mounting and integration because the entire apparatus would operate away from the machine.

By providing a solution that allowed the machine tender to quickly and easily fold, the Collapsing Hanging Arm excelled in its ability to be easily interchanged with a human operator. The Collapsing Hanging Arm’s close proximity to the CNC machine doors would allow for quick movement of parts between the part holding location and inside the CNC machine. This design, however, would inherently require the machine tender be mounted above the CNC doors on the CNC machine, which directly opposes the minimal mounting specification.

With a mostly self enclosed design, the Rotary Palette Shelf provides an extremely safe option for machine tending. The high rating of the concept coupled with the high weighting of the criteria, resulted in a score that had a considerable impact on the final score the option received. This concept also provided substantial strength to lift objects and can function with parts with variable contours. Even with these benefits, the Rotary Palette Shelf still ended with a low final score. This resulted from its high relative cost, difficulty to set up, and difficulty to reconfigure.

The Multi Grabber Rail System excelled in speed to deliver and retrieve parts from the multiple carriage design, which allowed both the delivery and retrieval to occur simultaneously. Having the system physically predefined provided an answer that was not reconfigurable, and extremely difficult to interchange with a human operator. Receiving a low score in the criteria related to interchangeability with a human operator significantly impacted the final score because of the high weight of the criteria. Ultimately this was a primary factor that led to the Multi Grabber Rail System receiving the lowest final score.

Although the Horizontal Pusher never received the highest score in any category, it scored slightly above average in a majority of the criteria. The main benefit of this concept is the simplicity of the design, which resulted in a relatively high rating in cost, safety, and durability. Also, similar to the Multi Grabber Rail System, this design utilizes two simultaneously moving parts, which allow for faster retrieval and delivery of parts. Because of this simplicity, the horizontal pusher scored poorly in reconfigurability and repeatability. With only one degree of freedom, the Horizontal Pusher would not have fine control of the final location of the part within the CNC machine.

Widely used in industrial applications, the Multi Axis Robotic Arm is the standard design for existing machine tenders. This design provides both reconfigurability and a high repeatability. With multi axis movement control, parts can be placed extremely accurately using a variety of motions. Even with the substantial benefits, the final score of the Multi Axis Robotic Arm reflected an average solution. The main contributing factors to the relatively low score were the cost, and difficulty to set up and reconfigure.

4.3 Preliminary Concept

The preliminary design concept is primarily a revision upon the granular grabber concept. The design still consists of a telescoping arm extending from a rectangular frame, but in order to address concerns regarding cantilevering the arm out from the back of the frame, a support was added to the front which can move in two dimensions just as the rear can, and both will have linear actuators to move them along their respective planes. Basic calculations have been done on the statics required to ensure stability (Appendix G). These actuators will be selected after analysis of various options, among which are motors driving rack and pinions, or pneumatic devices which would make use of the compressed air in the machine shops. Two-axis pivots were added to both supports which will now enable the back and the front to be located in different positions, changing the orientation of the robot arm. The gripper at the end of the arm will be operated using a vacuum pump, and oriented using two axes of rotation. It will likely be actuated using servo or stepper motors.

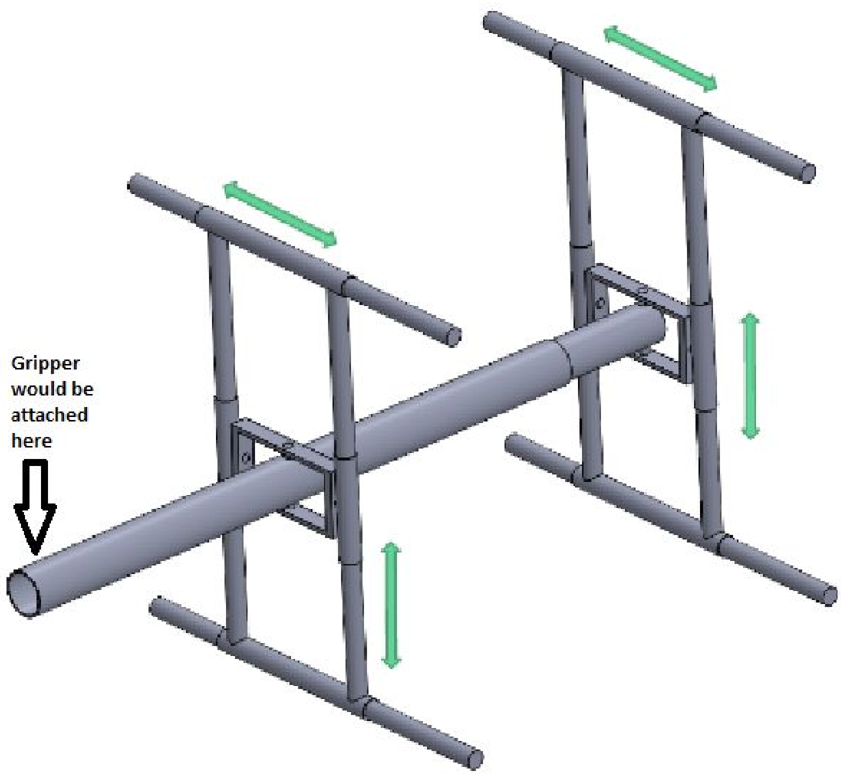

Figure 4.3.1: *Internal Portion of Design Concept*

The component materials have not yet been selected, but certain materials may be projected due to design requirements. The tender will ideally be designed for infinite life, and can be expected to go through many loading cycles, so a large portion of the components are likely to be made of steels or titanium alloys due to having endurance limits. If analysis determines that the deflection caused by using these materials has too large an effect on repeatability, and that particular components are only loaded in a single direction, composite materials may be chosen instead for their specific stiffness. Preliminary calculations have been done analyzing the approximate maximum deflection, which was deemed acceptable (Appendix H).

The main frame is estimated to be three feet on a side and mounted on a cart three feet high. The arm is estimated to extend three to five feet out from the front edge of the frame. Initial deflection calculations were done assuming a round arm three inches in diameter, both solid and with a wall thickness of 0.25 inches.

Although the granular gripper did not receive a high score on the Pugh matrix, the matrix was not weighted and did not reflect the value placed in the ability to accept parts with a variety of contours, which the granular gripper can accomplish better than any other manipulator. The value was made more apparent in the detailed decision matrix, which led to the inclusion of the gripper in the final concept.

The device is mounted on a cart to allow ease of set-up. The cart is rolled into and out of position, minimizing the amount of time it takes to switch between robotic and human operation. The entirety of the motion is limited to within and in front of the frame, giving the design a high safety rating, which is given the most weight in the decision matrix.

Even with a preliminary design, all safety factors needed to be considered. An initial hazard identification checklist has been completing to ensure all potential risks have been accounted for (Appendix J).

4.4 Preliminary Plans for Construction and Testing

Following the detailed design and analysis of the machine, which is planned to be completed in April of 2016, construction is expected to begin in early May. Any parts that can be finalized before the completion of the design may be ordered early on to reduce waiting periods. This will allow manufacturing and construction to begin while the remaining parts and materials are in transit. To reduce the cost of the tender, as many standard components as possible will purchased. The machine shops in the Cal Poly Aero Hangar and Bonderson Projects Center will be utilized to manufacture any custom parts. Manufacturing time estimates will be tripled as instructed by the Cal Poly shop technicians to generate more realistic estimates. During the design process, care will be taken to avoid designing as many features that require CNC machining operations as possible, due to the fact that none of the members of the senior project team are blue-tag certified. If, however, it has been deemed that a part is best manufactured using CNC, the shop technicians may be commissioned to manufacture the part. If the project members are able to familiarize themselves with the CNC machines by the time manufacturing begins, then none of it will have to be outsourced.

Depending on how early on it will be possible to select or create a board, software development may begin well in advance of, and occur parallel to construction. Some of the software may be designed and to a certain degree tested independent of the hardware. Later stages of testing will require the completed mechanical system.

Testing will involve the tender picking up objects of various masses and contours from different locations and placing them at other locations at different distances. As an example, it would provide useful data to test the capabilities of the tender holding an item (of consistent weight) at increasing increments of .25 meters to determine the effect distance has on repeatability for a certain build or material. Likewise the effects weight can be determined by holding an item at a constant distance and increasing its weight in increments of 0.5 lbs. Precision sensors (likely distance sensors mounted at known angles and locations) will be used to measure repeatability, which will determine if the machine meets the project requirements. Multiple tests would be run under different scenarios to determine the range of conditions under which the final product would function within acceptable parameters.

Software testing can be performed to a certain extent without the mechanical design as long as the electronic components are available. Drivers for most sensors and actuators can be written and tested with a power supply and a development board.

5.0 Final Detailed Design

5.1 Introduction

The preliminary design concept provided an exceptionally large and versatile range of motion because of the ability for the front and back supports to move independently. While this design could provide multiple advantages, it also introduced unnecessary complications. This design could be drastically simplified while still completing the customer requirements. By constricting the design to a completely rectilinear range of motion, fewer subsystems and parts were necessary. With the new design, the machine tender can be broken down into four subsystems. These consisted of the gripper, telescoping arm, horizontal motion, and vertical motion..

Before the specific detailed design could begin, the dimensions of the machine tender needed to be finalized. After measuring a Haas VF-3, it was determined that the machine tender would need to be able to reach past a table of parts, plus an additional three feet to reach the back of the table. In order to hold 40 unmachined parts, it was first proposed that they be laid out in a 5x8 grid, the parts being separated by an inch on each side. This was ultimately deemed an inefficient use of space, especially considering the vertical travel capabilities of the tender. To the efficiently utilize the available space, the parts were laid out in a more efficient orientation; two levels of parts, each holding at least 20. By decreasing the dimensions of the part table, the machine tender would only need to maneuver 36" horizontally to reach every part. This 36" was determined by assuming the same 1" gap between each part (Appendix L1). With two levels of parts, plus the 27" vertical distance between the VF-3 table and tool changing arm, the vertical range of motion was set at 50". This increased height raised concerns over the stability of the new design, prompting a decision to weigh down the base of the tender.

5.2 Gripper

The gripping subsystem saw numerous design changes and iterations to its function. Initially the use of a granular gripper was dismissed after research proved its capabilities and cost effectiveness did not correlate with our requirements. The next research delved into 'do it yourself' style grippers, made largely from scratch. While this too proved to be largely cost ineffective and time consuming, it revealed a crucial issue with our prior design; namely how it overcomplicated a simple task by incorporating unnecessary degrees of freedom. Once the degrees of freedom for the device were reduced and the device was modified accordingly. Pneumatics gripping, being actuated by pressure, applies a constant gripping force opposed to the inherent position based gripping of electromechanical actuators. This characteristic makes them ideal for our application.

The pneumatic gripper was determined by the needed gripper force. By calculating the gripping force based off of a maximum weight of 10 lbs (twice the maximum 5 lbs given to ensure it would be capable of doing that), we determined the amount of force needed to successfully grip part. Within the calculations, ensuring a safety factor larger than 2 was necessary to account for dynamic forces. This safety factor was determined from the pneumatic gripper catalog calculations. One particular gripper could be specified using supplied pressure, 70 psi (given to us by our sponsor) and a gripper length, 100 mm, derived from iteration (Appendix K1). Next, an equation solver could be created to calculate the acceptable gripper dimensions (Appendix L2, L3). Two sets of gripper dimensions were made for different gripping scenarios, with the worse case scenario (the 6” wide grip) being calculated for. Depending on the orientation of an unmachined block on the parts tray, the appropriate gripper can easily the attached gripper mechanism.

The resulting gripper designed has fingers able to gip a 6 inch piece at it’s widest while still having a high safety factor, at about 48 (Appendix L2, L3). The fingers are made out of steel due to a low cost compared to the durability and machinability it offers. By making the fingers thin, it allowed for more parts to be placed on a table for the tender to use during operation (Appendix L1). Thin strips of adhesive rubber were selected for the grip to achieve the coefficient of friction necessary to pick up metal blanks easily. Adhesive rubber strips can also be easily applied to the machined fingers made for the pneumatic gripper. Designing a spacer to connect the telescoping arm to the gripper was necessary because the face of the arm was not big enough the be attached to the gripper. Final dimensions for the spacer (.75” x 3.5” x 6”) will allow for adequate space to attach the gripper and telescoping arm to the plate without interference. The size and material were determined through calculations to ensure that the fasteners used on the spacer would not fail (Appendix L18). Selections involving more specific aspects of the pneumatic gripper itself, like the tubing, tubing inserts, and solenoid, were largely driven by compatibility with the gripper and the availability and cost of parts.

5.3 Telescoping Arm

The fully extended telescoping arm with an unmachined, five pound part gripped on the end represents the situation with both the highest applied stress and maximum deflection. Using a maximum end weight of forty pounds, the inside and outside telescoping arm tube dimensions could be calculated. A maximum of forty pounds was calculated by using the combined weight of an unmachined blank, and a potential gripping apparatus. This would turn out to be a conservative assumption, as the combined weight of the gripper and payload would prove to be less than forty pounds.

Due to the numerous components comprising the telescoping subsystem, and for the sake of clarity, the arm can be broken down into further sub-assemblies. these include the arm cylinder, the V-blocks used for mounting, and the motor mount. The cylinder arm, being already sized for desired length of travel, became the basis for the design of the sub-assemblies.

5.3.1 Cylinder Arm

The arm itself consists of two concentric cylinders - a solid steel rod 2” in diameter, and a 3” steel pipe with an outer diameter of 3.5.” These values were determined after calculating maximum deflection, fatigue factor of safety, and yield factor of safety (Appendix L4) based on superposition of beam diagrams (Appendix K4). Table 5.3.1 represents the analysis results for the inner telescoping arm used to determine acceptable deflections and safety factors for a variety of 1045 Cold Drawn Precision Steel Shafting stock sizes. This material was chosen for the inner telescoping shaft because of the tight straightness tolerances to prevent the linear telescoping bearing from binding. Steel was chosen because of the optimal combination of specific stiffness and endurance limit. This shaft will undergo numerous loading cycles, therefore it is important to guard against fatigue failure. The highlighted row represents the chosen steel shaft size.

Table 5.3.1: *Inner Solid Steel Shaft Analysis Results*

1045 Cold Drawn - Precision Shafting			
Diameter (in)	Max Deflection (in)	Safety Factor	Yield Safety Factor
0.75	11.98	0.4347	0.3878
0.875	6.649	0.6583	0.5941
0.9375	5.122	0.79	0.7166
1	4.02	0.9348	0.8521
1.125	2.597	1.263	1.162
1.25	1.768	1.641	1.522
1.375	1.255	2.065	1.929
1.438	1.071	2.295	2.151
1.5	0.9237	2.531	2.38
1.625	0.7	3.035	2.871
1.75	0.544	3.572	3.398
2	0.3496	4.727	4.544
2.5	0.1734	7.256	7.1
3	0.1014	9.929	9.859

The inner rod is held in position by using a linear bearing at each end. The front bearing is then fit into the outer pipe, and the rear bearing is fixed onto the back end of the inner rod, capable of traveling along with it. As there are no available linear bearings on the market that fix to a shaft and slide along a cylinder, the rear bearing will be fabricated out of nylon tubing. The material was selected due to its machinability and relatively low friction coefficient with steel.

After selecting the dimensions of the interior telescoping tube, the outer housing tube dimensions could be sized. With a 2” diameter interior shaft, this size represented the minimum acceptable value for the inner diameter of the outer telescoping tube. The outer telescoping tube was then analyzed for deflection, fatigue safety factor, and yield safety factor (Appendix L5). The highlighted row represents the steel tube size used in this design. Because every analyzed dimension configuration provided an acceptable maximum deflection and safety factors, the final decision was primarily driven by the lightest option compatible with linear sleeve bearings.

Table 5.3.2: *Outer Telescoping Tube*

1020 DOM A513 Round Steel Tube				
Do (In)	Di (In)	Max Deflection (in)	Safety Factor	Yield Safety Factor
2.25	2.01	0.01578	7.001	12.05
2.5	2.26	0.01122	8.693	15.09
2.5	2	0.00582	15.24	26.27
3	2.624	0.00404	18.23	31.98
3	2.5	0.00307	22.57	39.46
3	2.25	0.00211	29.33	50.97
3	2	0.00162	33.92	58.69
3.5	3.124	0.00240	25.03	44.44
3.5	3	0.00179	31.21	55.21
4	3.5	0.00112	41.09	73.46
5	4.5	0.00050	64.42	117.2
6	5.5	0.00026	92.19	170.2
8	7.5	0.00008	159.9	302.3
10	9.5	0.00003	242.1	466.3
12	11.5	0.00001	337.2	659.4
12	11.25	-0.00001	453.5	881.3

The linear motion is accomplished by driving a rack set into the top of the inner rod. To prevent collision with the forward bearing, the rack sits low in a channel machined into the rod. It is secured by screws that run along its length, kept clear of the pinion teeth by the use of counterbores. Rather than space the screw holes uniformly, they are positioned to sit between the rack teeth so as to minimize material removed from the teeth when creating the counterbores. The screws were selected by using an EES script written to calculate screw safety factors to determine whether screws selected for other subsystems might satisfy the current application (Appendix L6). This was done because the fasteners had to be purchased in packs of 50 or 100, and the other subsystems typically did not require more than a dozen, which resulted in many left over. #6-20 screws were selected for the rack, because their small heads required relatively unobtrusive counterbores. The yield factor of safety is only 1.33, but this is typical for fasteners with preload. The load factor is a much higher 154, and the factor of safety against joint separation is 66.3.

The size of the rack was driven by the diameter of the rod. It was desirable to minimize the size of the rack in order to mitigate the effects of machining the slot. The load on the teeth was calculated by assuming the combined weight of the arm and the payload acted on the nylon bearing. This was used to calculate the static friction force that the motor needed to overcome when accelerating the arm (Appendix L7). The required acceleration of the arm was estimated to be 10.5 in/s^2. The product of the mass of the arm and its desired acceleration was added to the friction force to determine the force required along the direction of the rack. Through the use of AGMA stress equations (Appendix L8) with a desired lifetime of 5 years, a face width of 0.75 inches was selected, and a 24-tooth pinion with a diametral pitch of 12. A larger pinion was initially selected due to the higher safety factor, but it resulted in a required torque that made selecting a motor prohibitively expensive.

The outer pipe was designed with a channel cut out of the top to allow the pinion access to the rack. The channel is as near as possible to the front, because the length of travel is limited by the largest possible distance between the pinion and the nylon bearing. If the arm were to extend further than that, the pinion would collide with the bearing. At both ends of the pipe, two smaller slots are machined at a right angle to each other, as deep at their centers as half the thickness of the pipe. These exist to align the pipe with the V-blocks.

5.3.2 V-Block

The V-blocks are mounting blocks shaped to restrict movement of the arm. When the valley of the V-block lines up with the groove in the arm, translation perpendicular to the axis, rotation about the axis, and travel along the axis is limited. The arm can then be locked in place by screwing down the straps. The V-blocks consist of ½”-thick sections of steel screwed together to increase their stability and stiffness, as they are the means by which the arm is secured to the horizontal rails, and they are the points by which the weight of the arm is supported (Appendix L9).

As with the rack screws, the #10-24 fasteners for the V-blocks were selected from a pool of the same screws used elsewhere in the machine tender in order to standardize sizing and conserve funds. They were verified with an EES script (Appendix L10, L11), and shown to be subject to loads small enough to be considered negligible compared to the preload tension, but the load factor is only 28, and the safety factor against separation dropped to 18.4..

The V-blocks were mounted to the linear bearings with ¼”-20 bolts, and were subject to an even lighter load than the #10-24 fasteners used to assemble the V-blocks.

5.3.3 Motor Mount

The motor which was selected for the telescoping application is the HT34-506, a NEMA 34 frame stepper motor capable of delivering sufficient starting torque at 24VDC (Appendix K3). A direct drive was desired to minimize cost and complexity, but the pinion was made for a larger shaft. Since ½” to ⅝” is not a standard step-up shaft adapter size, a custom adapter was designed and analyzed in EES, which indicated a Modified-Goodman safety factor of 4.8 (Appendix L12, N9). The motor is attached to a NEMA 34 motor mount, which is screwed into a base welded together from ½” steel fastened to a linear bearing which shares the same shaft as the forward V-block (Appendix N11). This structure positions the motor at the proper height to mesh with the rack, but to prevent separation between the rack and pinion when the motor starts running, the telescope pipe is U-bolted to the mounting structure through an intermediate angle iron (Appendix M4).

5.4 Horizontal Motion

In order to traverse the minimum horizontal distance of 36”, a timing belt driven by a stepper motor was attached to move the v-block relative to the vertical actuator. Using a belt system provides an inexpensive linear motion system with the ability to move relatively quickly. Lead, or power screws provide another alternative to a belt system. Although they provide multiple advantages, lead screw generally move slower than belt systems. Quickly traversing the large horizontal range of motion is optimal to ensure the machine tender is capable of loading or unloading the VF-3 within the 30 second requirement. In order to successfully complete this motion, multiple mechanical components were analyzed and specified. .

Typically, timing belt systems attach to linear bearings; which slide along hardened, precision steel shafts. Most designs that feature this motion system operate on a significantly smaller scale. Current designs do not feature a horizontal shaft that undergoes severe deflection, which could potentially result with the linear bearing binding. By using a ¾” diameter shaft, the maximum deflection of the horizontal guide shaft would be 0.77” (Appendix L13). This was calculated with a maximum shaft length of 48” and an applied load of 150 pounds force. The maximum shaft length was determined by adding a factor of safety to the previously calculated minimum range of motion to allow necessary structural elements to be included. The final length of the shaft between supports is 46.59”. From shaft calculations (Appendix L5) the applied force from the base telescoping arm shaft is a maximum of 138 lbf. Although there is deflection, the selected linear sleeve bearings have a misalignment capability of 0.5°, which provides enough clearance to prevent binding. A shaft made from hardened steel was selected because of precise straightness tolerances necessary to ensure the linear sleeve bearing slides smoothly.

The mounted linear sleeve bearings directly attach to the v-block telescoping arm support. Protruding from the v-block assembly is an aluminum piece bolted to two 3-D printed plastic attachments. Utilizing additive manufacture allows intricate, and precise designs to be created. Typically the product of 3-D printing can not withstand forces comparable to standard metal, which is satisfactory because the 3-D printed parts will not be under high loads or stresses. Each of the 3-D printed attachments meshes with a ½” wide L-series timing belt with a ⅜” pitch. By using two separate 3-D printed attachments, the belt can be tensioned to defend on slack inherent in timing belt usage. Using the applied torque from the horizontal motion motor (Appendix L14), and belt sizing charts from Gates Mectrol [14] the proper belt was chosen. According to these charts, an XL series timing belt would have met the specifications for torque, however given the necessary belt pitch length of 105”, a L belt is better suited to this application.

5.5 Vertical Motion

Lead screws provide an excellent method to translate the rotational motion output from a stepper motor into linear motion. Using only one lead screw located precisely between vertical shafts could suffice, however it would not be optimal. In this location the mounted horizontal linear bearings used for horizontal movement would be subjected to a moment that could potentially cause binding because of the distance between the lead screw support and the force applied from the telescoping arm. In order to prevent this, lead screws were positioned next to each vertical guide rail at both the front and back supports for a total of 4 lead screws. Assuming the reaction force at the front is split between both lead screws equally, the torque necessary to raise the load is 20.68 lbf-in using 1” diameter lead screws. Using a 1” diameter lead screws provides resistance to buckling, as buckling would require a force more than three times larger than the applied force. Also, the von mises stress significantly lower than the yield or ultimate strength of steel at only 1565 psi (Appendix L15).

Besides utilizing the linear actuating ability of a lead screw, the self-locking capability is inherently useful in vertical linear actuation. Using self-locking lead screws will prevent the stepper motor from constantly outputting energy to fight gravity and keep the load stationary.

An ACME threaded square nut was positioned within a custom housing (Appendix M8) to translate the linear motion of the nut to the telescoping arm, and ultimately the gripper holding a part. Aluminum was chosen as the material for the bearing housing because it is strong enough to withstand the applied forces, and is significantly cheaper than steel. This housing will be manufactured by welding three separate ¼” thick pieces together with a ¼” thick fillet weld, Using a ¼” fillet weld will provide a maximum shear stress lower than maximum allowable shear stress. The weld specifications were determined by assuming the weld would support the reaction from the telescoping arm entirely alone (Appendix L16). By using a ¼” weld height, the welds along with fasteners will provide sufficient support for the applied load. The major loads are translated to the housing structure through bolts connecting the steel spacing block to the aluminum housing. With proper bolt pre loading of 75% of the bolt proof strength, the load factor of safety and load factor of safety guarding against joint separation are incredibly high. Any fasteners have the potential to undergo shear failure as well. The applied stresses have the potential to lead to failure by bending of the bolt, failure by pure shear, rupture of the connected members, or failure by crushing of the bolt or plate, however they are significantly smaller than the proof strength of the bolts (Appendix L16).

The entire horizontal system is guided along vertical linear guide rails, each with a ½” diameter. Since there is no direct load from the telescoping arm, using a ½” diameter provides sufficient structural support. The force required to induce buckling is drastically larger than any potential applied loads (Appendix L17), which are only a result of housing weight. Any potential compressive failure could only develop after buckling for either ductile or brittle material failure theories. Therefore, any metal will suffice, however, hardened precision steel shafts provide a tight straightness tolerance to prevent the mounted linear bearing from binding.

5.6 Cost Analysis

One significant customer this machine tender is designed for, are small machine shop owner without the capital to invest in the expensive existing machine tenders. Constraining the budget to roughly $3500 will give the aforementioned machine shop owners the ability to experiment with autonomous machine tending. The current budget of the machine tender is $3,509.27 (Appendix O1). Only machine components are included within this budget. Software elements and automatic fixturing will need to be included in the final bill of materials, and will roughly increase the budget by $500. Cheaper alternative materials may replace some expensive components if they sufficiently fulfill every function and requirement of the original part. Detailed drawings of part manufactured specifically for this application, along with exploded assembly drawings can be view in Appendix N and Appendix M, respectively.

5.7 Safety Considerations

Although the components of the device are not moving at particularly high speeds, the fact that the part being moved is a six foot steel shaft is enough to make it a safety concern, due solely to the size and weight of the machine. The tender contains sharp edges to look out for, but the greatest danger is the many potential pinch points created by the travel of the machine. Additionally, the height of the tender makes potential instability a concern; should it fall over, it could hurt someone. A comprehensive list of the concerns we have about the machine are contained within the FMEA and Safety Hazard Checklist.

5.8 Maintenance and Repair Considerations

In order to maximize repairability, as many components of the machine as possible are sized from stock parts. This makes it a simple matter of ordering a new part and replacing the old one when it breaks. We also elected to use fasteners to secure the majority of the machine together, which should further enable maintenance and repair, or even possibly modification of the tender.

6.0 Management Plan

After constructing our team, each member was designated specific tasks. Ryan Canfield will lead communications with our advisor and sponsor and act as manufacturing and prototype fabrication lead. Samuel Adler will be responsible for managing team finances and materials for the project. Louis Roseguo will be the coding lead and maintain the team repository information and any other relevant documents.

As can be seen in Table 6.1, we will first complete our project proposal, which includes information about the process and steps we will follow for successful completion of our project. After completing the project proposal we will produce a preliminary design report which will include our idea generation results. Our final design report will include computer models of our final design, justification for our decisions, and a cost analysis of a prototype. In May, the spring expo will occur; which will include all Cal Poly senior projects that began in September as well as our preliminary prototype or poster describing our design with Solidworks models and analysis. Next we will focus on the optimization of our prototype, and will send updates to our sponsor in September. Finally, we will have our completed product and our final project report finished at the beginning of December.

Table 6.1: *Project Milestones*

Milestone	Date
Project Proposal	2/2/2016
Preliminary Design Report	2/23/2016
Final Design Report	5/3/2016
Spring Senior Project Expo	5/26/2016
Project Update to Sponsor	9/27/2016
Final Project Report	12/1/2016

After establishing a preliminary design, a program evaluation and review technique (PERT) chart was developed to lay out the timeline for the remainder of our project (Appendix I). All design aspects and engineering analysis will be completed in five weeks in preparation for the final design report. Purchasing materials and developing a prototype will take a total of four weeks before testing can begin. After our prototype undergoes a series of tests, the next five weeks will be spent iterating to find the optimal design. When a final design has been decided upon, two weeks will be spent finalizing the product and creating a user manual. A detailed timeline can be seen in the Gantt chart (Appendix I2).

7.0 Manufacturing Plan

In order to complete the upcoming milestones by manufacturing and testing a completed machine tender, each subsystem will be initially assembled independently before being combined into a finished product. A majority of the components used in each subsystem are stock parts bought from directly from the suppliers with no need for further manufacturing. Certain parts, however, do require manufacturing, and a plan to successfully machine them to acceptable tolerances.

7.1 Gripper Subsystem

Gripper Finger [Either 4” (Appendix N1) or 6” (Appendix N2)]

A. Cut to length.
B. Drill 2 X 8mm clearance through holes.
C. Mill Corners Out

Spacer Plate (Appendix N3)

A. Ensure stock dimensions are acceptable
B. Drill 6 X 8mm clearance through holes

7.2 Telescoping Subsystem

Arm (Appendix N4)

A. Cut to length.
B. Slot is too long to machine on a manual mill. Commission shop technicians to CNC slot.
C. Measure, drill five 7/64” pilot holes.
D. Tap five #6-32 holes.

Nylon Sleeve Bearing

A. Cut to length.
B. Bore if necessary.
C. Turn if necessary.

Rack (Appendix N5)

A. Cut to length.
B. Mill five counterbores.
C. Drill five 5/32” holes.

Outer Pipe (Appendix N6)

A. Cut to length.
B. May be unable to use manual mill due to size of part-consult shop technicians.
C. CNC internal grooves.

Angle Iron (Appendix N7)

A. Cut to length.
B. Drill two ½" holes.
C. Drill two ¼" holes.

Strap (Appendix N8)
A. Cut steel strip to length.
B. Lay across section of leftover outer pipe.
C. Strike with mallet until strap conforms to pipe.
D. Drill two 3/16" holes.

Shaft Adapter (Appendix N9)
A. Cut raw shaft to length.
B. Turn to overall diameter.
C. Drill ½" hole 1" deep into center of bore side face.
D. Turn shaft side to ⅝".
E. Mill out flat along shaft.
F. Drill and tap two ¼-20 holes.

Motor Hanger Base (Appendix N10)
A. Using steel left over from V-block center, cut to length.
B. Drill four ¼" through holes.
C. Bandsaw off corner section.

Motor Base Bottom (Appendix N11)
A. Cut sections to length.
B. Cut angled section.
C. Tack-weld perpendicular sections into position.
D. Complete weldments.
E. Tack-weld angled section into position.
F. Complete weldments.
G. Holes should be measured out and drilled after welding in case of warping due to heat. Drill two ¼" through holes.
H. Drill four 3/16" through holes.

V-Block Bottom (Appendix N12)
A. Cut steel bar to length.
B. Drill eight 3/16" through holes.
C. Drill two ¼" through holes.
D. Drill two 13/64" through holes.
E. Mill two 7/16" counterbores.

V-Block Side (Appendix N13)
A. Cut steel bar to length.

B. Drill one 13/64” through hole.
C. Drill two No. 25 holes.
D. Tap two #10-24 holes.
E. Mill 7/16” counterbores.

V-Block Center (Appendix N14)

A. Cut steel stock to length.
B. Drill six No. 25 holes.
C. Tap six #10-24 holes.
D. Mill out V.

7.3 Horizontal Motion Subsystem

Horizontal Motor Support (Appendix N15)

A. Cut each side to the specific dimensions from a ¼” thick sheet
B. On one of the larger faces mill a 3” diameter hole out of the middle.
C. On the same face, drill four ¼” clearance holes for bolts
D. Weld the four sides together with a ¼” weld height

Pulley Support (Appendix N16)

A. Cut each side to the specific dimensions from a ¼” thick sheet
B. Drill a ½” hole through the center of each of the larger pieces
C. Weld the four sides together with a ¼” weld height

Belt Holder (Appendix N17)

A. 3-D Print

Belt Adapter (Appendix N18)

A. 3-D Print

7.4 Vertical Motion Subsystem

Spacer Block (Appendix N19)

A. Cut Block to Dimensions
B. Drill two ¼” clearance holes for bolts

Vertical Weld Plate (Appendix N20)

A. Cut a ⅝” thick sheet of aluminum dimensions
B. Mill thickness down to 0.55”
C. Drill four ¼” clearance bolt holes
D. Drill four 0.16” clearance bolt holes
E. Drill ½” diameter counterbores to a depth of 0.30” concentric with the 0.16” clearance bolt holes.

ACME Nut Housing (Appendix N21)

A. Cut three sheets to length out of ¼” thick aluminum
B. Drill four ¼” clearance bolt holes in one piece.
C. In the two others, drill two ¼” clearance bolt holes
D. In the larger of these two, drill two 0.19” clearance bolt holes
E. In the same two, mill a 1.25” square through all (rounded internal corners are acceptable)
F. Weld parts together with a ¼” weld height

Shaft Coupler (Appendix N22)

A. Using a 2” diameter steel rod, mill around a 1” diameter concentric with the original rod 1.1” down
B. Drill ½” hole ¾” deep concentric with the original rod
C. On the opposite face, mill a 1.5” diameter hole, 0.5” deep, concentric with the original rod.
D. Drill four #6 holes until next, equally spaced 0.25” up from the bottom of the largest diameter.
E. Drill a 0.19” diameter set screw hole equally between the top of the small diameter, and the change between large and small diameter.

ACME 1” - 5 Threaded Rods (Motor Side)

A. Cut threaded rod down to 52.5”
B. On one end, use a lathe to decrease the diameter to 0.75” until 1.5” from the flat edge
C. On the other end, lathe the diameter down to 0.5” until 1” from the flat edge
D. Mill the new 0.5” diameter flat on one side, until the width is 0.35”

ACME 1” - 5 Threaded Rods (Non Motor Side)

A. Cut threaded rod down to 54”
B. On one end, use a lathe to decrease the diameter to 0.75” until 1.5” from the flat edge
C. On the other end, lathe the diameter down to 0.5” until 1” from the flat edge
D. Mill the new 0.5” diameter flat on one side, until the width is 0.35”

Off Motor Threaded Rod Base (Appendix N23)

A. Cut 2.5” steel rod to length
B. Mill a circle with 0.25” diameter to a depth of 0.35” concentric with the original rod.

8.0 Assembly Instructions

8.1 Gripper (Appendix M1)

A. Thread 100mm screws through fingers
B. Screw 100mm screws into the small fingers of the gripper
C. Using two 35 mm screws, fasten the spacer plate to the gripper

8.2 Telescoping

8.2.1 Arm (Appendix M2)

A. Set rack into slot.
B. Screw rack into slot.
C. Insert first retaining ring.
D. Insert bearing.
E. Insert second retaining ring.
F. Slide rod into bearing.
G. Press nylon bearing onto rear of rod.

8.2.2 V-Block (Appendix M3)

A. Screw side plates to center plate.
B. Screw bottom plate to three combined plates.
C. Bolt V-block to bearing.
D. Set arm into B-block (both must be assembled and mounted to shafts).
E. Screw strap down around arm into V-block.

8.2.3 Motor Mount (Appendix M4)

A. Screw motor mount to hanger and base.
B. Screw angle iron into hanger.
C. Bolt base to bearing.
D. Thread U-bolt around arm and tighten nuts.
E. Screw motor to mount.
F. Attach shaft adapter to motor.
G. Attach pinion to shaft adapter.

8.3 Horizontal Assembly (Appendix M5)

A. Attach timing belt pulley to the horizontal motor output shaft with a set screw
B. Bolt the horizontal motor to the horizontal motor support
C. Line up the other timing belt pulley, thrust bearings, and thrust washers with the holes machined into the pulley housing
D. Slide the nylon bearing through the housing holes
    a. Press fit the nylon bearing through the pulley until the pulley is in the middle of the nylon bearing.

E. Attach the timing belt around the circumference of each pulley (Keep the two ends of the timing belt facing up)

8.4 Vertical Assembly

8.4.1 Vertical Linear Guide Shafts (Appendix M6)

A. Slide a mounted linear bearing onto each vertical linear guide shafts.
B. Fit each end of the vertical linear guide shaft into a flange bearing mount.
C. Repeat for all four vertical linear guide shafts.

8.4.2 Vertical Actuator Assembly (Appendix M7)

A. Attach the pulley to the output shaft of the vertical actuating motor with a set screw
B. Secure the threaded rod coupler to the hub of the pulley with four 6-32 socket head cap screws
C. Attach ½" diameter end of the motor side threaded rod (shorter) to the coupler with a set screw
D. Place the thrust washers and thrust bearings on top of the off motor threaded rod base.
E. Place the pulley on the top of the top thrust washer.
F. Attach the off motor threaded rod (longer) to the pulley with a set screw
G. Secure the timing belt around the pulleys
H. Repeat the previous steps for the second vertical actuating system.

8.4.3 Lead Screw Nut Housing (Appendix M8)

A. Secure ACME nut housing to the vertical weld plate with four ¼" - 20 socket head cap screws.
B. Position the lead screw nut within the ACME nut housing.
C. Secure the lead screw nut in place by fastening the spacer block to the housing with two ¼" - 20 socket head cap screws
D. Position the horizontal shaft mount on the ACME nut housing, and bolt the shaft mount to the housing using only the outside holes and two 10 - 24 socket head cap screws.
E. Repeat the previous steps four times
F. Position each lead screw nut on one of the lead screws, while lining them all up horizontally.

8.5 Final Assembly (Appendix M9)

A. Place the front telescoping assembly linear sleeve bearings on one horizontal guide shaft
B. Place the back telescoping assembly linear sleeve bearing on the second horizontal guide shaft.
C. Position one ¾" diameter horizontal linear shaft between each pair of lead screws.
D. Once the horizontal linear shaft is flush against both spacer block, secure the shaft in place with two 10 - 24 socket head cap screws.
E. Attach the timing belt designed for horizontal motion to the v-block by securing the timing belt between the 3-D printed pieces and the aluminum (Appendix M10).

9.0 Final Design

9.1 Manufacturing

In order to manufacture the components of the machine tender multiple different machining processes were necessary. To manufacture parts with machine tool requirements beyond the scope of the available machine shop, the parts were outsourced to our sponsor, Haas Automation, Inc. In order to accomplish this, detailed drawing were sent that specified the critical dimensions. These parts included the telescoping arm (Appendix O1), base shaft (Appendix O2), rack (Appendix 03), and lead screws (Appendix 04).

Only the band saw and manual mill were required to manufacture a majority of the remaining parts. The horizontal band saw cut raw stock roughly to size, while the final dimensions were obtained through the manual mill. After creating either steel or aluminum blocks within the required dimensional tolerance, an edge finder was used to locate each block that required precision hole placement. In order to successfully drill holes larger than ½ In. in steel, pilot holes were first drilled corresponding to the final hole dimensions. Certain components required counterbores, which were achieved by plunging with end mills of the correct dimension. Typically, the diameter of the counterbore was inconsequential which resulted in larger holes depending on the availability of end mills in the machine shop. The parts that require this machining process include the Angle Iron (Appendix O5), Motor Base Bottom (Appendix O6), Motor Hanger (Appendix O7), Horizontal Motor Supports (Appendix O8), V-Block Side (Appendix O9), V-Block Center (Appendix 10), V-Block Base (Appendix O11), Pulley Housing (Appendix O12), horizontal motion supports (Appendix O13), Belt Adapter (Appendix O14) and Lead Screw Support (Appendix O15), The V-Block Center required more intricate milling techniques. Initially a small hole at the bottom of the V was drilled using an edge finder and the Digital Read Out (DRO) on the manual mill. Both parts were then clamped to the manual mill table with toe clamps on a sacrificial piece of wood, and a dial indicator was used to position them at an angle within tolerance. An edge finder could then be used to find the center of the hole. While in the correct position, an end mill could be used to cut out the V by simply moving the table in the X or Y direction.

After completing the milling operations, both the Motor Hanger and Pulley Housings required welding. Using the Metal Inert Gas (MIG) method, the individual components were welded together to create the final parts.

Inside the pulley housing sits the Pulley Shaft (Appendix O16), which supports the horizontal motion timing belt pulleys. A foot long stock of precision, stainless steel, rotary shafts was cut to size using an abrasive saw. Next, they were ground down to within tolerance on a grinding wheel. The timing belt within this housing was cut to size using scissors, and held in place by Timing Belt Holders. These Timing Belt Holders were 3D printed using the fused filament deposition method in the Sandbox Innovation Room on the Cal Poly campus.

To fabricate the strap (Appendix O17) around the base shaft, which connected to the V-block, a thin strip of steel was cut on the abrasive saw, then hammered around the base shaft to ensure an acceptable fit. Holes were then match drilled into the ends to screw it into the V-Block Center.

Only a select few components required the lathe. Both the nylon Slider Rod (Appendix O18) and the Shaft Adapter (Appendix O19) were cut to size on a horizontal band saw, then faced down within tolerance on a lathe. During the manufacturing of the slider rod, a large drill bit made the initial hole, then a boring bar was used to achieve the required tolerances. For the Shaft Adapter, the diametral tolerances were achieved through turning on a lathe. An end mill was then used to create the internal keyway. Also, the outer diameter of the linear sleeve bearings was slightly larger than the inside diameter of the base shaft; therefore, the outside diameter was turned down roughly 0.010” inches to create an acceptable fit.

To construct the frame, connecting plates were cut on the horizontal band saw out of large steel plates. Initially the large plate would not fit on the bandsaw, therefore they were cut on the plasma cutter. The smaller pieces were then cut to size on the horizontal band saw. The drill press was then used to create holes to attach the frame together. The steel tubing could then be cut to size on the abrasive saw, and have holes match drilled with a hand drill to fit with the connecting plates.

In order to tension the timing belt that connects the leads screws, an idler needed to be incorporated into the design. To manufacture the plates that attach the threaded idler sits in, they were simply cut with a horizontal bandsaw with holes match drilled into them because the size of the plates was completely inconsequential. (Appendix O20)

9.2 Assembly

The first assembly created was the telescoping mechanism (Appendix M2), whose major components include the base shaft and telescoping rod. The rack, which mates with the pinion, was screwed into a groove on the telescoping rod. Then the nylon slider rod could be press fit on the back of the telescoping rod. The linear sleeve bearing was inserted into the base base shaft,with retaining rings to constrain it. The end of the telescoping rod was then fit into the base shaft and through the linear sleeve bearing. This mechanism could then be strapped to the V-Block assemblies and mounted on the linear ball bearings that roll on the precision horizontal shafts; however they were not immediately put on the horizontal shafts.

Next, the frame was assembled by bolting the connector plates to steel tubing. The lead screw supports were then bolted to the frame to support the lead screws. With the lead screws ends in a sleeve bearing and the supports, the motor was placed with a rigid coupler to ensure concentric placement. The motor was attached to a stock NEMA 34 motor support, and bolted to a spare piece of steel tubing, which was then bolted to the frame. Mounted on the end of a lead screw with a set screw, the vertical motion pulleys were sandwiched by thrust bearings to ensure minimal friction during movement. Part of the vertical movement included the tensioning idler assembly (Appendix P1) for the timing belt. Threaded idlers were screwed into a plate, while the entire assembly could slide along the frame to either increase or reduce tension in the belt.

With the lead screw subassembly mounted on the frame, the telescoping assembly could be mounted on the lead screws. The horizontal motion pulleys, motor mounts and horizontal shaft mounts, each attached to a flanged nut, were screwed onto the corresponding lead screw. A table holding the heavy telescoping assembly could then be placed inside the frame with the linear ball bearings in line with the shaft mounts. While resting on the table, the horizontal shafts on each were positioned inside the mounts and through the linear ball bearing. The timing belt connecting the lead screw in both the front and back could then be turned by hand to lift the heavy telescoping assembly off of the table. Finally the table could be removed with the telescoping assembly in place.

The last subassembly to be incorporated into the final design was the horizontal motion (Appendix P2, P3). The timing belt pulleys were mounted on the motor output shaft with two set screws. Within the pulley housing, the timing belt pulley rested on the precision pulley shaft, with thrust bearings on both sides. Grease was then applied to decrease friction and decrease the amount of force required to spin the pulley. With the timing belt holders loosely attached to the V-Block assembly, the timing belt could be installed by connecting both pulleys. Then, with the belt under an adequate amount of tension, the timing belt connectors could be tightened to keep the required tension.

9.3 Final Mechanical Design

Successfully constructing the final design (figure 9.1) (Appendix S) required multiple changes from the planned design described in section 5. The current horizontal motion subassembly (Appendix P2, P3) incorporates a round, flanged, plastic acme nut, which required multiple changes in design. First, the housing around the nut required two aluminum blocks, which supported the motor mount. The flanged acme nut sat inside a steel block, which also held the horizontal motion support. Next, the opposing pulley became mounted inside a steel housing, which could be assembled with ¼”-20 screws rather than being welded together.

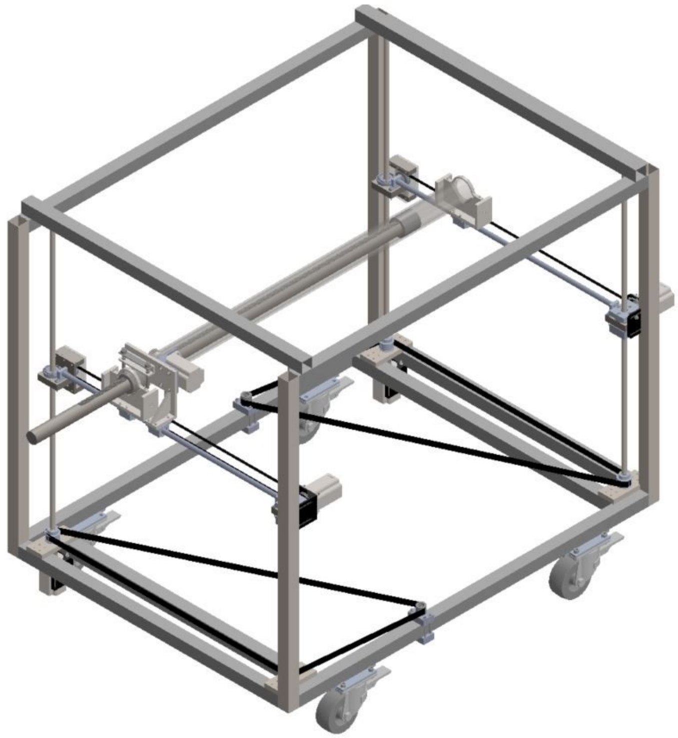

Figure 9.1: SolidWorks Model of Final Design

The vertical motion, (Appendix P4) actuated through lead screws, now includes steel blocks to support each lead screw onto the frame. This requires thrust bearings to sit above and below the lead screw for rotation with minimal friction.

The design in section 5 does not include a frame. Therefore, one was designed using 16 gauge square steel tubing whose outer dimensions were 2” x 2”. Connector plates were cut from ¼” thick steel and holes were drilled into it. Holes in the steel tubing were then match drilled to ensure the connector plates could attach the steel tubing together.

10.0 Testing

10.1 Mechanical Results

After completing the final construction of the machine tender, motion could be manually actuated in all directions without binding. The telescoping rod extends and retracts smoothly, but can rotate under torque so that the gear teeth fail to mesh. The improvised keyway in the shaft adapter is not small enough to firmly secure the adapter to the motor. The telescoping mechanism slides freely on the horizontal rods, but the rails noticeably undergo elastic deformation, which peaks when the arm is loaded at the center of travel. The lead screws can be turned manually if the timing belts are used simultaneously to raise and lower the machine, but the idlers lack a lip around the top edge, which puts the belt at risk of slipping off. The casters attached to the bottom of the frame make the assembly mobile. In its current state, the machine lacks a gripper, and a cable management system, as well as limit switches, have yet to be implemented. The control hardware has not been mounted to the device.

The motors appear to have some difficulty moving their respective axes. The most likely cause of this is that the stepper drivers used in the design limit the current to 4.5 Amps, which is lower than the current draw indicated in the motor data, which was how the motors were selected in the first place.

10.2 Software Results

Despite numerous attempts, actuating the machine tender with the PLC was ultimately unsuccessful. Although the PLC and accompanying software meet the project requirements, a stepper pulse could not be read from the PLC high-speed output in time for the project demonstration. The cause of this has yet to be diagnosed, but there are several areas that could be investigated if afforded more time.

The most likely cause for failure to output a signal is a simple error in programming. Perhaps a location in memory is receiving an incorrect value to properly configure the output, or the configuration data is being written to the wrong register altogether. There is some evidence against the programming being the source of the problem, in that a sample program from the user manual was replicated to test the code, and still failed to generate a signal when connected to an oscilloscope. The next most likely cause is a mistake in the wiring of the test circuit. Reviewing the documentation of the components and rewiring the circuit from scratch may resolve the issue. An additional potential cause is the unsteady output from the power supplies, which did not output the specified 24 volts, and fluctuated with an amplitude of approximately 2 Volts about an average 19-21 Volts. A possible result of this is that insufficient voltage failed to trigger the PLC logic, but if the apparent normal operation of the LCD screen and the ability to communicate between a computer and the PLC are any indication, then the PLC can still run properly on less than the rated voltage. The least likely source of error is that the hardware itself is somehow damaged in a way that allows the PLC to turn on and download programs, but not activate the outputs. This condition would be expensive to test, requiring another controller known to work, and is not worth investigating until all other avenues have been exhausted.

Partial control of the motors may have been possible with the use of the high-speed modules, which were not tested, because testing of the high-speed modules was planned to follow testing of the high-speed outputs on the base unit, which never finished. The system may have worked if controlling the add-on modules was pursued before the base unit pulse output.

In lieu of a working PLC, partial functionality was obtained by programming an arduino Uno to send signals to the stepper drivers. Rather than full automation, the program responded to user input for demonstration purposes (Appendix T). The script would enable pulses to different drivers and control directional outputs based on which switches were activated. In the event of an emergency stop, the program would disable all motion until a reset switch is toggled twice.

11.0 Conclusion and Recommendations

Although the full automation of the machine tender was ultimately beyond the scope of what could be accomplished in the given time frame, the mechanical design can successfully demonstrate motion in all directions. Given more time, automation of the machine would be feasible. If microcontrollers were used from the beginning, it is likely that this objective would have been met given our familiarity with them. The gripper assembly still needs to be constructed, attached, and integrated into the control system. Having a more detailed mechatronic design earlier in the project would have significantly helped the completion of the project. Additionally, starting on the mechatronic system earlier on would have been significantly beneficial. Since portions of the mechatronics can be worked on independently of the rest of the machine in order to achieve basic functionality, it would have been better to start working with the program as soon as the means were decided upon. This would have the added benefit of integrating a mounting system for the mechatronics components into the mechanical design earlier on, which became a problem, as they have yet to be mounted.

It may be worthwhile to investigate altering the size of the machine. The same basic design could be scaled up or down for CNC mills of various sizes. Maintenance of this machine should also be taken into consideration for future design improvements such as developing an easy way to clean out the inside of the telescoping rod. Any residue or dust that builds up will only increase the force required to overcome the friction between the slider rod and steel tube. With the heavy steel rod and tube mounted on the lead screws, it is nearly impossible to service or replace any components of the telescoping arm assembly without disassembling a majority of the machine.

Once all of the mechatronics components are fully integrated into the mechanical design, an operating manual will need to be developed. This will need to include assembly instructions and safety practices in addition to standard machine operation.

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Appendices

Appendix A - Quality Function Deployment Diagram

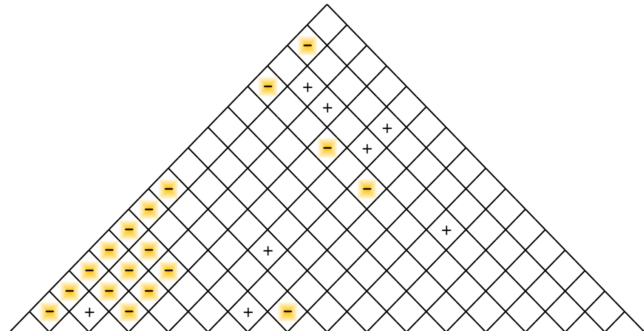
QFD: House of Quality
Project: CNC Machine Tender
Revision: 01
Date: 1/27/2016

Samuel Adler
Ryan Canfield
Louis Roseguo

Correlations	
Positive	+
Negative	−
No Correlation	

Positive Relationships	
Strong	●
Moderate	○
Weak	▽

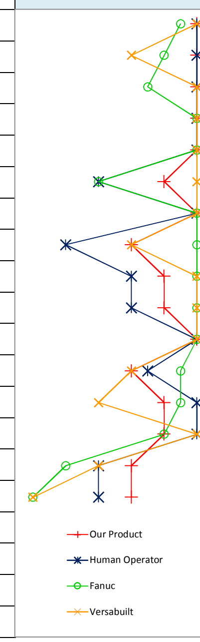
Direction of Improvement	
Maximize	▲
Target	◇
Minimize	▼



Column #	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Direction of Improvement	▼	▼	▼	▼	▼	▲	◇	▲	◇	▼	▼	▼	▲	◇	◇	

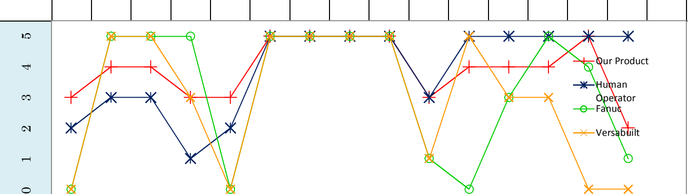
WHO: Customers | HOW: Engineering Specifications | NOW: Current Product Assesment - Customer Requirements

Row #	Weight Chart	Relative Weight	HAAS Automation Inc.	Manufacturing	Buyer	Operator	Maximum Relationship	WHAT: Customer Requirements (explicit & implicit)	Within a budget of $3500	Load Partswithin 30 seconds	Exchange parts within 30 seconds	Repeatability of 0.025 TIR	E-stop to stationary within 0.5 sec	Lift/manipulate 5lb aluminum blank	Lift/manipulat 2"X4"X6" blank	Stores at least 40 parts	VF-2, VF-3, MiniMill or ToolRoom Compatible	24 hour mean time repair	5-10 mins from end of tender operation to beginning of human operation	20 minutes to reconfigure	1.5 meter minimum reach	4 flag communication	Senses foreign objects within range of motion		Our Current Product	Human Operator	FANUC M-20iA	Versabuilt		Row #
1		7%	10	1	8	10	9	Safe for humans to interact with					●						○	▽			●		5	5	4.5	5		1
2		8%	10	2	8	10	9	Emergency Stop Function/Accept stop commands from CNC stop					●				●					●	●		5	5	4	3		2
3		5%	6	2	1	8	9	Minimal Mounting/Integration Necessary	○								●		○						5	5	3.5	5		3
4		6%	10	1	10	3	9	Lift and manipulate up to a 5 lb rectangular blank		▽	▽	○		●	○	▽				▽					5	5	5	5		4
5		6%	10	1	10	3	9	Lift and manipulate up to a 2" x 4" x 6" rectangular blank		▽	▽	●		○	●	○				▽					5	5	5	5		5
6		5%	10	1	6	3	9	Able to store and process at least 40 blanks and 40 finished parts		○	○	▽		●	●	●				○	●				4	2	2	5		6
7		7%	10	3	8	3	9	Compatable with a VF-2, VF-3, MiniMill, and ToolRoom mill	▽	○	○	●	○		○		●		▽	▽	●	●			5	5	5	5		7
8		8%	10	5	8	1	9	Repeatability of 0.025" TIR				●		▽	●	○	○			▽	○				3	1	5	3		8
9		5%	10	1	8	1	9	Deliver and load at new part within 30 seconds		●	●	○		●	●	▽					▽	○			4	3	5	5		9
10		5%	10	1	8	1	9	Exchange finished parts for new parts within 30 seconds		●	●	○		●	●	▽					▽	○			4	3	5	5		10
11		5%	7	1	4	7	9	Operates independently of CNC control	▽			○								○		●	●		5	5	5	5		11
12		10%	8	6	8	8	9	Affordable, reliable, reconfigurable, and repairable motion	●			○		▽	▽			○		●	●				3	3.5	4.5	3		12
13		5%	8	1	2	7	9	Simple to set up and program for part variation						▽	○				●	●	▽	▽			4	5	4.5	2		13
14		6%	10	1	3	10	9	Easily interchangable with human operator											●						4	5	4	5		14
15		7%	6	2	7	9	9	Simple to repair	○									●							3	2	1	2		15
16		7%	9	5	6	1	9	Budget of $3500	●									○							3	2	0	0		16
17		0%																												17
18		0%																												18
19		0%																												19
20		0%																												20



HOW MUCH: Target	Within a budget of $3500	Load Partswithin 30 seconds	Exchange parts within 30 seconds	Repeatability of 0.025 TIR	E-stop to stationary within 0.5 sec	Lift/manipulate 5lb aluminum blank	Lift/manipulat 2"X4"X6" blank	Stores at least 40 parts	VF-2, VF-3, MiniMill or ToolRoom Compatible	24 hour mean time repair	5-10 mins from end of tender operation to beginning of human operation	20 minutes to reconfigure	1.5 meter reach	4 flag communication	Senses foreign objects within range of motion	
Max Relationship	9	9	9	9	9	9	9	9	9	9	9	9	9	9	9	
Technical Importance Rating	200.66	130.07	130.07	275.75	158.52	218.25	308.65	98.155	201.8	112.49	141.07	194.82	230.96	210.37	182.99	0
Relative Weight	7%	5%	5%	10%	6%	8%	11%	4%	7%	4%	5%	7%	8%	8%	7%	0%
Weight Chart																
Our Product	3	4	4	3	3	5	5	5	5	3	4	4	4	5	2	
Human Operator	2	3	3	1	2	5	5	5	5	3	5	5	5	5	5	
FANUC M-20iA	0	5	5	5	0	5	5	5	5	1	0	3	5	4	1	
Versabuilt	0	5	5	3	0	5	5	5	5	1	5	3	3	0	0	

Current Product Assesment - Engineering Specifications



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Template Revision: 0.9 Date: 4/23/2010
Christopher Battles

Appendix B - Concept Sketches

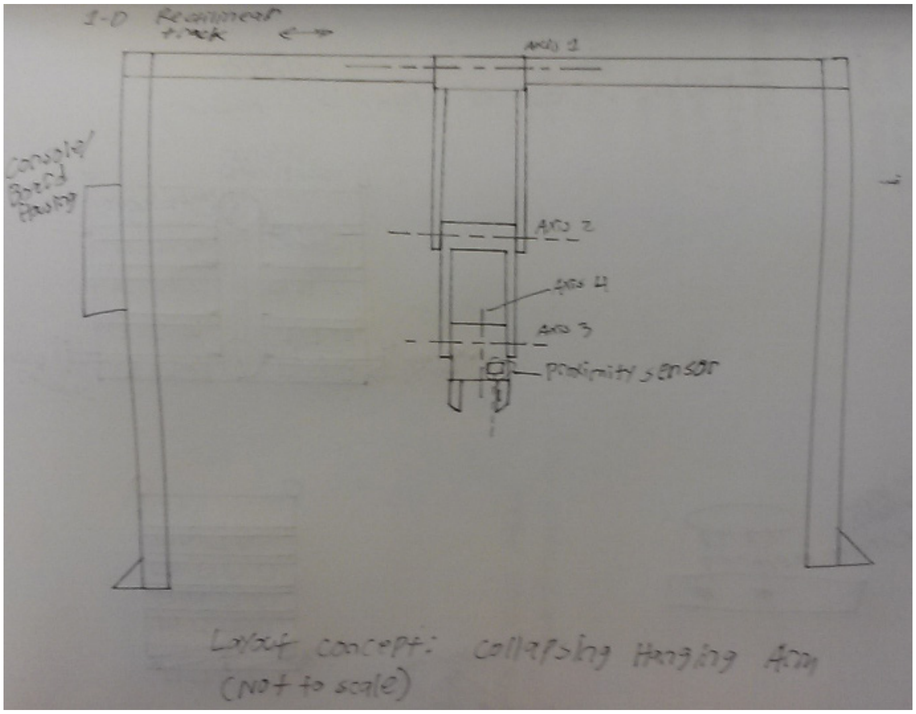


Figure B-1. *Collapsing Hanging Arm*

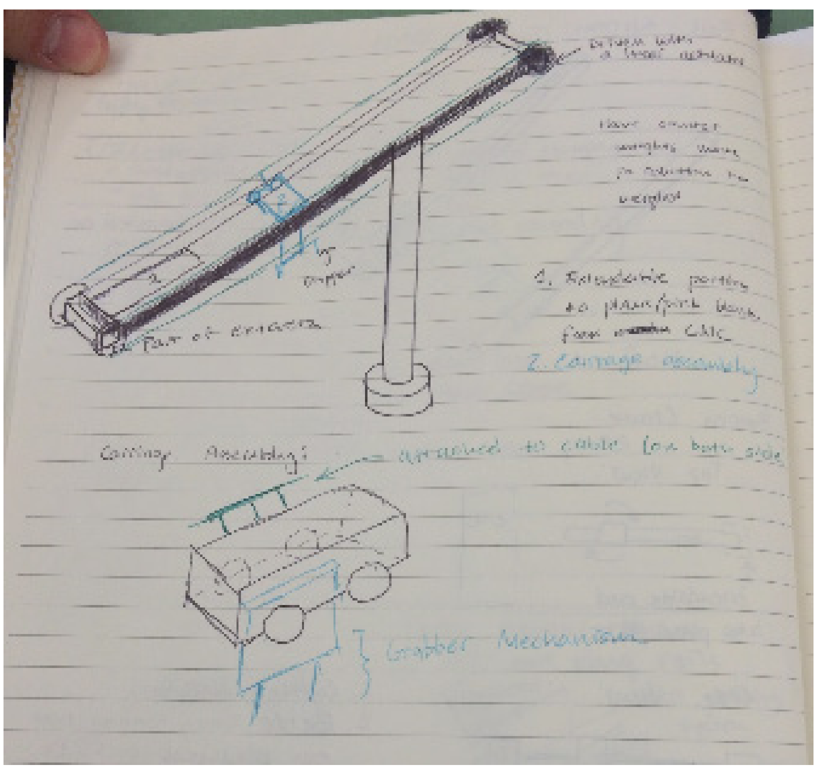


Figure B-2. *Crane Model*

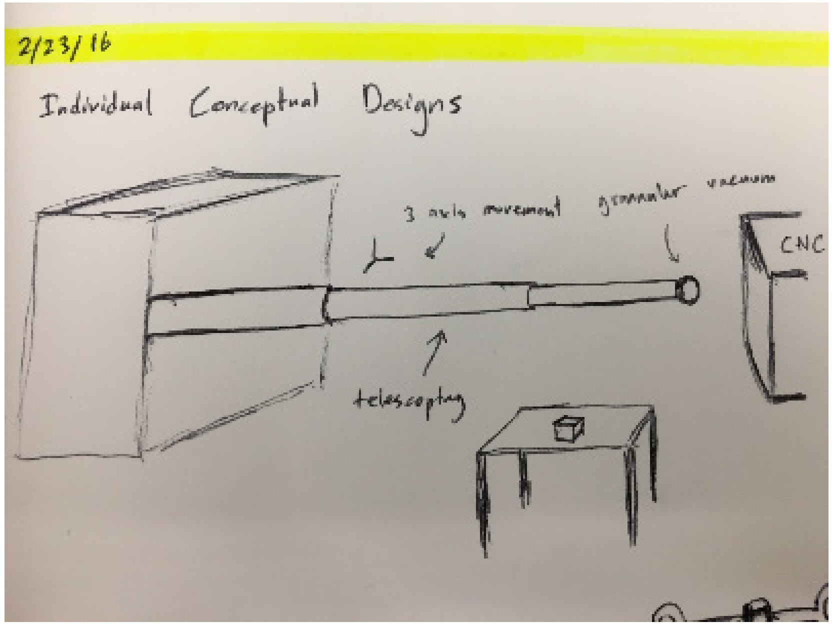
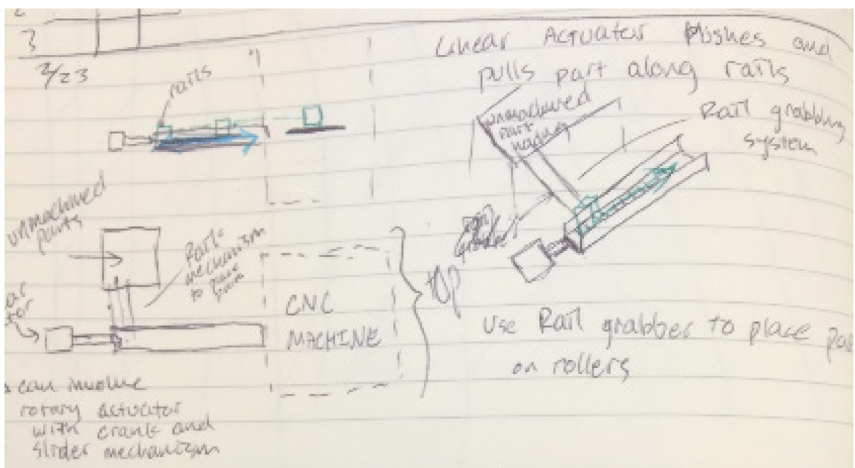
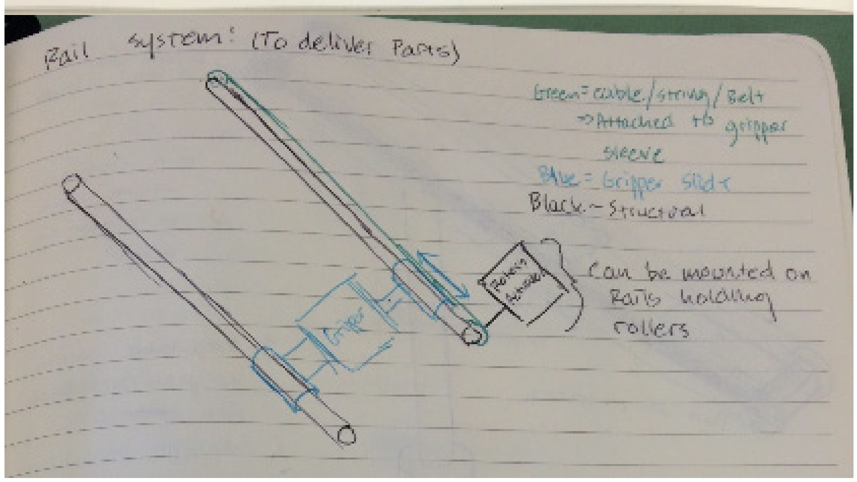


Figure B-3. *Granular Grabber*





Figures B-4 and B-5. *Horizontal Pusher*

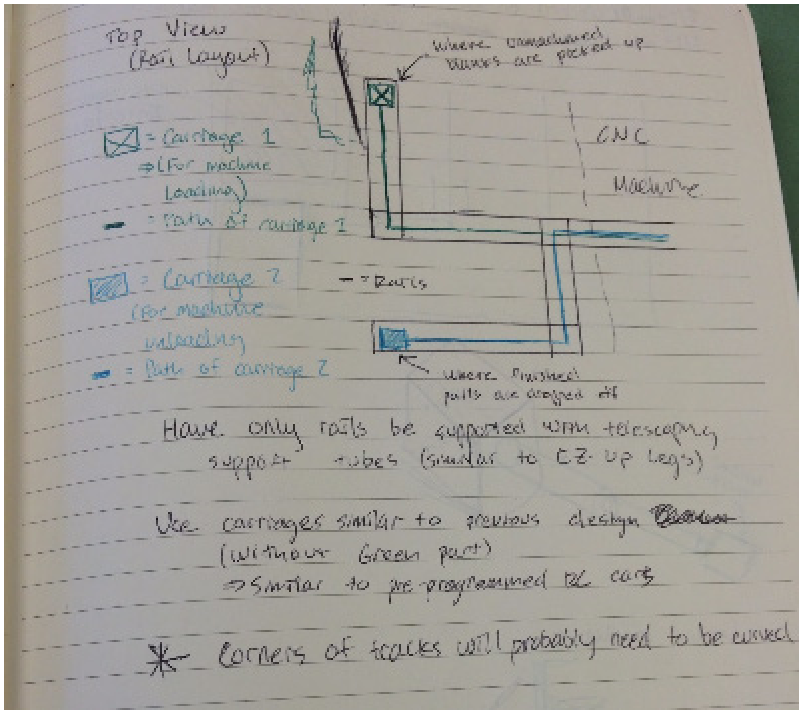


Figure B-6. *Multi-Grabber Rail System*

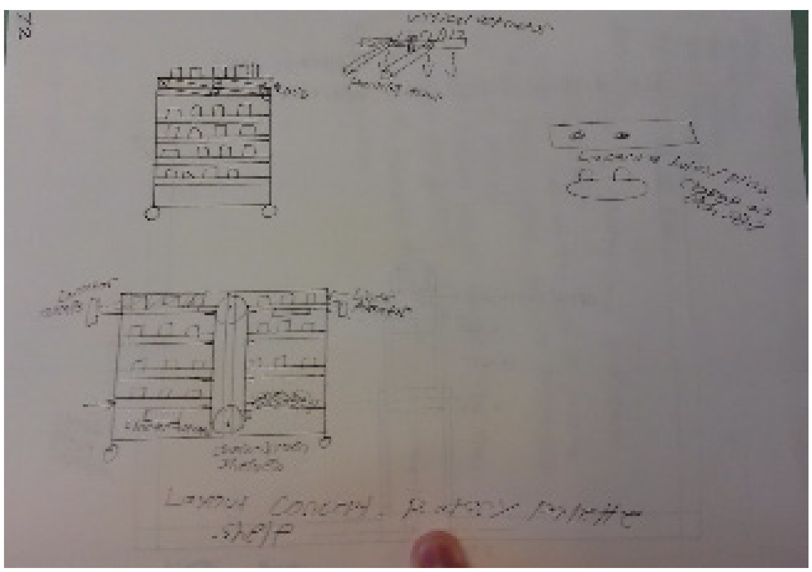


Figure B-7. *Rotary Pallet Shelf*

Appendix C - Glossary

ANSI - American National Standards Institute.

Axis - Either referring to coordinate axes, such as in Cartesian space, or the amount of axes on a robot allowing freedom of movement (e.g., a robot arm made up of two motors with parallel shafts would be considered a two-axis robot despite not technically being able to rotate along two coordinate axes).

Blue Tag Certification - The highest level of certification at the Cal Poly machine shops short of being a shop technician. Grants access to the CNC machines.

Brainstorming - Ideation method involving a focus on producing a large quantity of ideas in a short period of time. Judgement is ideally withheld at this stage.

Brainwriting - Ideation method involving the transcribing of ideas and passing them on with the intention to facilitate new ideas.

CNC - Computer Numerically Controlled. Implemented to automate manufacturing operations.

CNC Machine Tender - A robotic operator designed to carry out the same loading and unloading operations as a human technician.

Degree of Freedom (DOF) - A unique axis of translation or rotation along which a machine can move.

Haas Automation, Inc. - Project sponsor and manufacturer of CNC machines and machine tooling.

IEC - International Electrotechnical Commission.

ISO - International Organization for Standardization.

MTTR - Mean Time To Repair, the average time it takes to repair something when it breaks

OSHA - Occupational Safety and Health Administration.

Pallet - A platform for holding multiple parts.

Payload - Mass or weight of part to be supported.

QFD - Quality Function Deployment. Table indicating the relationship between various functions and customer requirements.

Red Tag Certification - The first level of certification at the Cal Poly machine shops. Grants access to the shops and certain tools.

Repeatability - Precision to which a part may be accurately and reliably placed.

RIA - Robotic Industries Organization.

SCAMPER - Substitute, Combine, Adapt, Modify, Put to other uses, Eliminate, Rearrange/Reverse. Structured ideation method involving brainstorming with triggers to facilitate lateral thinking.

TIR - Total Indicated Runout.

Yellow Tag Certification - The second level of certification at the Cal Poly machine shops. Grants access to the manual lathes and mills, hydraulic presses, welders and plasma cutters.

Appendix D - Pugh Matrices

Pugh Matrix - Movement (Human Datum)

D1

Requirements	Walking (Datum)	Servo Motor	Stepper Motor	Linear Actuator	Gears	Hydraulics	Pneumatics	Belts	Sprockets	Four Bar Linkage	Power Screws	Solenoid Actuator	Vacuum (DP)	Magnetic	Springs	Pulley	Ramp	Conveyor Belt	Voice Coil Actuator	Delta Robot
Safe for humans to interact with		-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1
Emergency stop awareness		-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1
Emergency stop reaction time		1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
Minimal mounting/integration necessary		-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1
Strength to lift items		-1	-1	1	1	1	1	1	1	1	1	-1	0	-1	0	1	-1	-1	-1	0
Durability		-1	-1	-1	1	0	0	-1	1	1	-1	-1	-1	1	-1	-1	1	0	-1	0
Lift/manipulate up to a 2" x 4" x 6" rectangular blank		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Can store/process at least 40 blanks and 40 finished parts		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Compatable with VF-2, VF-3, Minimill, and/or ToolRoom Mill		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Repeatability		1	1	1	1	1	1	1	1	1	1	-1	-1	-1	-1	-1	-1	-1	1	1
Speed to deliver and retreive parts		0	1	1	1	-1	-1	0	0	0	-1	0	1	1	0	-1	0	-1	0	1
Operates independently of CNC control		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Reliable motion		1	1	1	1	1	1	1	1	1	1	1	-1	-1	0	1	1	1	1	1
Reconfigurable motion (Physical)		-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	0	-1	-1
Simple to set up and program for part variation (Teaching)		-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1
Easily interchangable with human operator		-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1
Simple to repair		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Manufacturing budget		1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
Cost		1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
Totals		-3	-2	0	2	-1	-1	-1	1	1	-2	-5	-5	-4	-5	-4	-3	-4	-3	0

Pugh Matrix - Gripping (Human Datum)

D2

Requirements	Hand (Datum)	Soft Robotics	Suction Cups	Vise	Part-specific Grippers	3 Pronged Jaws	Scooping	Multijointed Fingers	Adhesive	Forklifting	Magnetics	Granular Grip
Safe for humans to interact with		0	0	-1	-1	-1	-1	-1	-1	-1	-1	-1
Emergency stop awareness		-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1
Emergency stop reaction time		1	1	1	1	1	1	1	0	1	1	1
Minimal mounting/integration necessary		-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1
Strength to lift items		-1	1	1	1	1	-1	0	1	1	1	-1
Durability		-1	-1	1	1	0	1	-1	-1	-1	1	-1
Lift/manipulate up to a 2" x 4" x 6" rectangular blank		0	0	0	0	0	-1	0	0	0	-1	-1
Can store/process at least 40 blanks and 40 finished parts		0	0	0	0	0	0	0	0	0	0	0
Compatable with VF-2, VF-3, Minimill, and/or ToolRoom Mill		0	0	0	0	0	0	0	0	0	0	0
Repeatability		0	1	1	1	1	-1	1	-1	-1	-1	0
Speed to deliver and retreive parts		0	0	0	0	0	0	0	-1	0	0	0
Operates independently of CNC control		0	0	0	0	0	0	0	0	0	0	0
Reliable motion		0	1	1	1	1	-1	0	-1	-1	1	0
Reconfigurable motion (Physical)		-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	1
Simple to set up and program for part variation (Teaching)		-1	-1	-1	-1	-1	-1	-1	-1	-1	-1	-1
Easily interchangable with human operator		0	0	0	0	0	0	0	0	0	0	0
Simple to repair		0	0	0	0	0	0	0	0	0	0	0
Manufacturing budget		1	1	1	1	1	1	1	1	1	1	1
Cost		1	1	1	1	1	1	1	1	1	1	1
Totals		-4	0	1	1	0	-6	-3	-7	-5	-2	-4

Pugh Matrix - Movement (Machine Datum)

D3

Requirements	Gears (Datum)	Servo Motor	Stepper Motor	Linear Actuator	Pneumatics	Hydraulics	Sprockets	Belts	Power Screws	Four Bar Linkage	Vacuum (DP)	Solenoid Actuator	Springs	Magnetic	Ramp	Pulley	Voice Coil Actuator	Conveyor Belt	Delta Robot
Safe for humans to interact with		1	1	1	0	0	0	0	0	0	0	1	-1	1	1	0	1	0	1
Emergency stop awareness		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Emergency stop reaction time		1	1	0	0	0	0	-1	1	0	1	1	-1	1	-1	-1	1	-1	1
Minimal mounting/integration necessary		1	1	1	-1	0	0	0	0	-1	0	1	0	0	1	0	1	0	-1
Strength to lift items		-1	-1	1	1	1	0	-1	1	0	0	-1	-1	0	0	-1	1	-1	0
Durability		-1	-1	-1	-1	-1	0	-1	-1	-1	-1	-1	-1	1	1	-1	-1	-1	-1
Lift/manipulate up to a 2" x 4" x 6" rectangular blank		0	0	0	0	0	0	0	0	0	0	-1	0	0	-1	0	0	0	0
Can store/process at least 40 blanks and 40 finished parts		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	0
Compatable with VF-2, VF-3, Minimill, and/or ToolRoom Mill		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Repeatability		0	0	1	0	0	1	0	1	-1	-1	-1	-1	0	-1	0	0	-1	1
Speed to deliver and retreive parts		0	0	-1	0	0	0	0	0	0	0	1	-1	0	0	0	0	1	1
Operates independently of CNC control		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Reliable motion		0	0	0	0	0	0	0	0	0	-1	1	-1	-1	1	0	0	0	1
Reconfigurable motion (Physical)		1	1	0	1	1	0	0	0	-1	-1	1	-1	0	-1	1	1	0	1
Simple to set up and program for part variation (Teaching)		1	1	0	0	0	0	0	0	0	1	1	0	0	-1	0	1	0	0
Easily interchangable with human operator		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Simple to repair		0	0	-1	-1	-1	0	1	1	1	-1	-1	1	0	1	0	-1	1	-1
Manufacturing budget		0	0	0	0	-1	-1	1	-1	1	-1	-1	0	0	1	0	-1	-1	-1
Cost		0	0	-1	-1	-1	-1	1	-1	1	-1	-1	1	-1	1	0	-1	-1	-1
Totals		3	3	0	-2	-2	-1	0	1	-1	-5	0	-6	1	2	-2	2	-3	1

Pugh Matrix - Gripping (Machine Datum)

D4

Requirements	Vise (Datum)	Soft Robotics	Suction Cups	3 Pronged Jaws	Scooping	Multijointed Fingers	Adhesive	Forklifting	Magnetics	Granular Grip	Part-specific Grippers
Safe for humans to interact with		1	1	0	0	0	0	0	1	1	0
Emergency stop awareness		0	0	0	0	0	0	0	0	0	0
Emergency stop reaction time		0	0	0	0	0	-1	0	1	0	0
Minimal mounting/integration necessary		0	0	-1	0	0	0	0	0	0	-1
Strength to lift items		0	-1	-1	1	0	0	1	-1	-1	0
Durability		-1	-1	-1	1	-1	-1	1	1	-1	0
Lift/manipulate up to a 2" x 4" x 6" rectangular blank		1	0	0	0	1	0	0	0	1	1
Can store/process at least 40 blanks and 40 finished parts		0	0	0	0	0	0	0	0	0	0
Compatable with VF-2, VF-3, Minimill, and/or ToolRoom Mill		0	0	0	0	0	0	0	0	0	0
Repeatability		1	0	0	-1	1	-1	-1	0	0	0
Speed to deliver and retreive parts		0	0	0	-1	0	0	0	0	0	0
Operates independently of CNC control		0	0	0	0	0	0	0	0	0	0
Reliable motion		1	-1	0	0	1	-1	0	0	0	0
Reconfigurable motion (Physical)		0	0	1	0	-1	0	1	0	1	-1
Simple to set up and program for part variation (Teaching)		-1	0	1	1	-1	0	0	0	1	-1
Easily interchangable with human operator		0	0	0	0	0	0	0	0	0	0
Simple to repair		-1	0	0	0	-1	1	0	0	-1	0
Manufacturing budget		-1	0	-1	0	-1	0	0	-1	0	-1
Cost		-1	0	-1	0	-1	0	0	-1	-1	-1
Totals		-1	-2	-3	1	-3	-3	2	0	0	-4

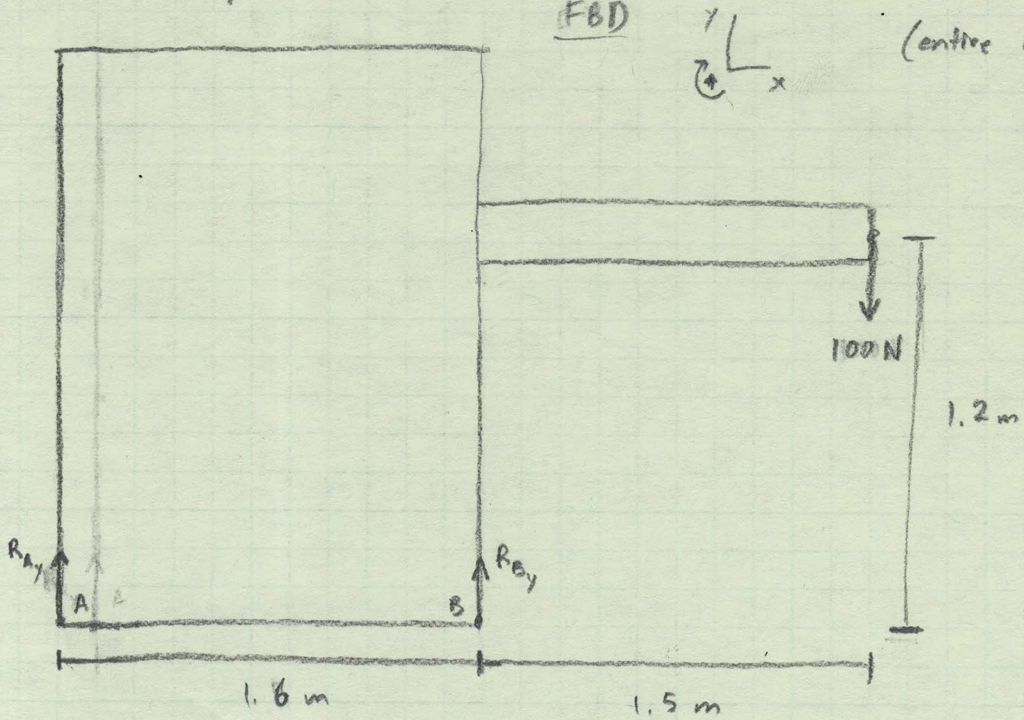
Go/No Go Matrix	Granular Grabber	Crane Model	Collapsing Hanging Arm	Rotary Palette Shelf	Multi Grabber Rail System	Horizontal Pusher	Multi-Axis Robotic Arm
Safe for humans to interact with	Pass	Pass	Pass	Pass	Pass	Pass	Pass
Emergency stop	Pass	Pass	Pass	Pass	Pass	Pass	Pass
Can be interchanged with human operator	Pass	Pass	Pass	Pass	Pass	Pass	Pass
Can lift/manipulate 5lb blocks	Pass	Pass	Pass	Pass	Pass	Pass	Pass
Lift/manipulate up to a 2" x 4" x 6" rectangular blank	Pass	Pass	Pass	Pass	Pass	Pass	Pass
Able to store/process at least 40 blanks & 40 finished parts	Pass	Pass	Pass	Pass	Pass	Pass	Pass
Compatable with VF-2, VF-3, Minimill, and/or ToolRoom Mill	Pass	Pass	Pass	Pass	Pass	Pass	Pass
0.025" Repeatability	Pass	Pass	Pass	Pass	Pass	Pass	Pass
30 seconds to deliver and retreive parts	Pass	Pass	Pass	Pass	Pass	Pass	Pass
Operates independently of CNC control	Pass	Pass	Pass	Pass	Pass	Pass	Pass

Appendix F - Detailed Decision Matrix

Criteria	Weight (1-5)	Granular Grabber		Crane Model		Collapsing Hanging Arm		Rotary Palette Shelf		Multi Grabber Rail System		Horizontal Pusher		Multi-Axis Robotic Arm	
		Rating	Score	Rating	Score	Rating	Score	Rating	Score	Rating	Score	Rating	Score	Rating	Score
Safe for humans to interact with	5	4	20	3	15	3	15	5	25	4	20	4	20	3	15
Emergency stop reaction time	4	4	16	4	16	4	16	3	12	4	16	4	16	4	16
Minimal necessary mounting/integration	2	5	10	5	10	1	2	3	6	3	6	4	8	5	10
Strength to lift items	3	4	12	2	6	4	12	5	15	2	6	4	12	5	15
Durability	2	3	6	3	6	4	8	4	8	3	6	4	8	5	10
Can accept parts with variable contours	3	5	15	4	12	4	12	5	15	4	12	4	12	4	12
Accuracy of placing part (Repeatability)	3	3	9	4	12	4	12	3	9	3	9	2	6	5	15
Speed to deliver and retreive parts	3	3	9	2	6	4	12	4	12	5	15	4	12	4	12
Reconfigurable motion (Physical)	2	2	4	3	6	4	8	1	2	1	2	1	2	4	8
Easy to set up/program for part variation (Teaching)	2	5	10	3	6	3	6	1	2	4	8	4	8	1	2
Easily interchangable with human operator	5	4	20	3	15	5	25	4	20	2	10	3	15	1	5
Simple to repair	1	4	4	4	4	3	3	2	2	3	3	4	4	2	2
Size of machine tender	2	2	4	3	6	1	2	2	4	1	2	4	8	4	8
Manufacturing cost	4	3	12	3	12	2	8	1	4	2	8	3	12	1	4
Final Score		**151**		**132**		**141**		**136**		**123**		**143**		**134**	

Basic Statics Analysis

FBD (entire machine)



$$\sum M_B : (100\text{ N})(1.5\text{ m}) + (R_{A_y})(1.6\text{ m}) = 0$$

$$\boxed{R_{A_y} = -93.75\text{ N}}$$

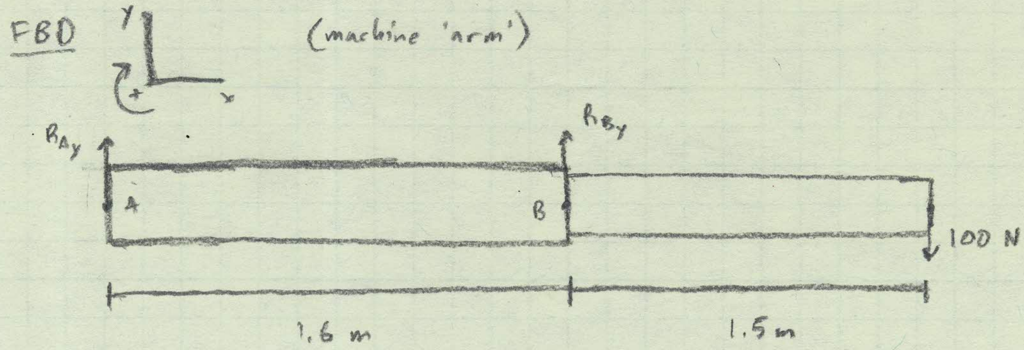
$$\sum F_y : -100\text{N} - 93.75\text{ N} + R_{B_y} = 0$$

$$\boxed{R_{B_y} = 193.75\text{ N}}$$

For ~ 20 lb of material held by 1.5 m arm, would get the above forces for the example dimensions chosen.

Basic Statics Analysis (cont)

FBD (machine 'arm')



ΣM_B : $(100\ N)(1.5\ m) + (R_{Ay})(1.6\,m) = 0$

$\boxed{R_{A_y} = -93.75\ N}$

ΣF_y : $-100\ N - 93.75\ N + R_{B_y} = 0$

$\boxed{R_{B_y} = 193.75\ N}$

For ~20 lb of material held by 1.5 m extended machine arm, would get the above forces for the example dimensions shown.

Pipe Sizing For Cantilever Arm

```
format compact
% Solid Pipe

P = 5;                         % Lbf, Applied Weight at End of Arm
l = 3;                         % Ft, Length of Arm
E = 29*10^6;                   % Mpsi, Modulus of Elasticity (Steel: 29)
d = 3;                         % In, Outside Diameter of Pipe
I = 3.1415*(d^4)/64;           % Calculate Area Moment of Inertia

rho = 0.283;                   % lb/in^3, Density of Material(Steel: 0.283)

omega1 = rho*3.1415*((d/2)^2); % lb/in force from beam mass

SolidPipeDeflection = P*(l^3)*(12^3)/(3*E*I)+(omega1*(l^4)*(12^4))/(8*E*I)

% Hollow Pipe

P = 20;                        % Lbf, Applied Weight at End of Arm
l = 3;                         % Ft, Length of Arm
E = 29*10^6;                   % Mpsi, Modulus of Elasticity (Steel: 29)
do = 3;                        % In, Outside Diameter of Pipe
t = 0.25;                      % In, Thickness of Wall

di = do-(2*t);                 % In, Inside Diameter of Pipe
I = 3.1415*((do^4)-(di^4))/64; % Calculate Area Moment of Inertia in^4

rho = 0.283;                   %lb/in^3, Density of Material (Steel: 0.283)

omega2 = rho*3.1415*((do/2)^2-(di/2)^2); %lb/in force from beam mass

HollowPipeDeflection = P*(l^3)*(12^3)/(3*E*I)+(omega2*(l^4)*(12^4))/(8*E*I)
```

```
SolidPipeDeflection =
    0.0043
HollowPipeDeflection =
    0.0074
```

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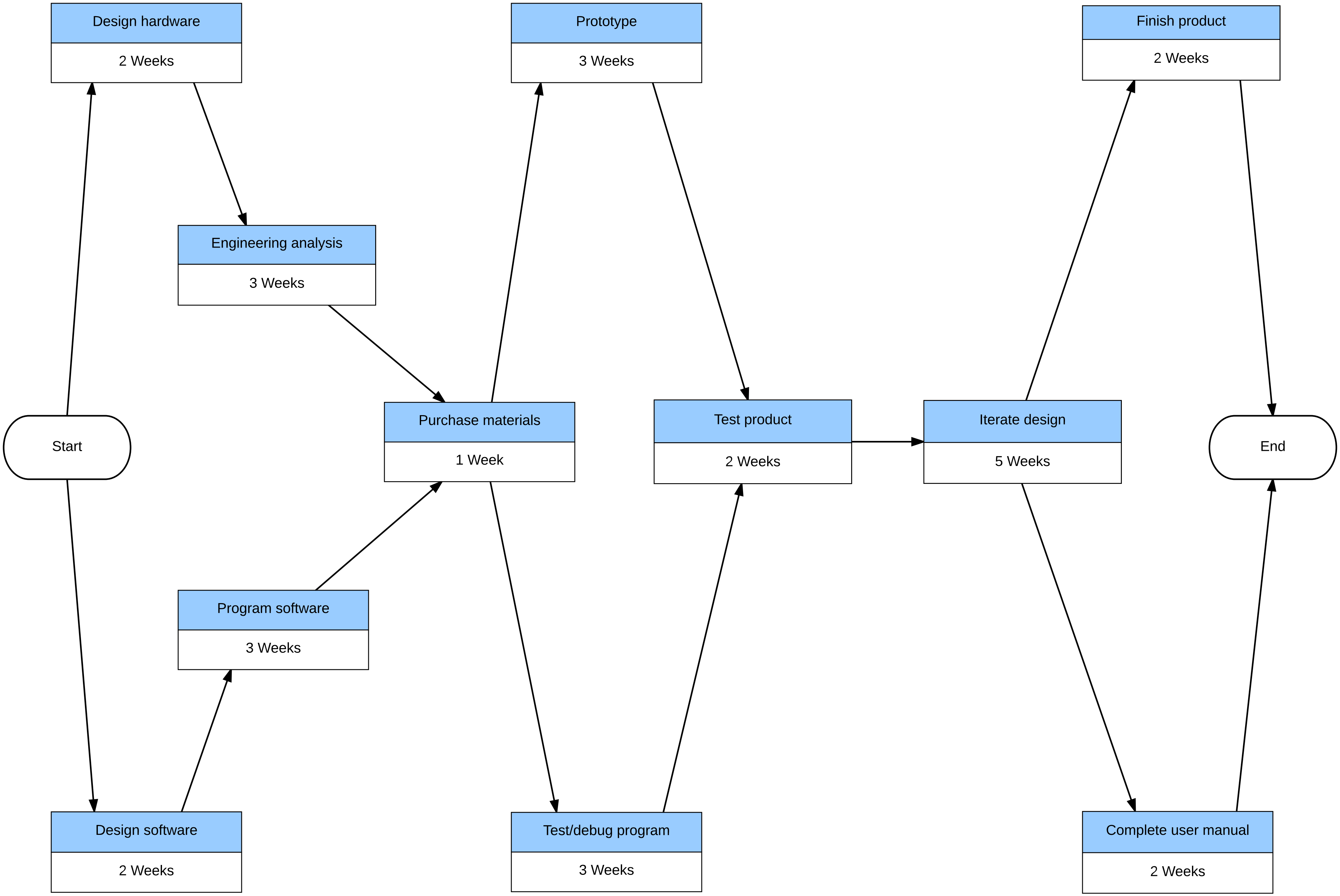
Appendix I - Pert Chart

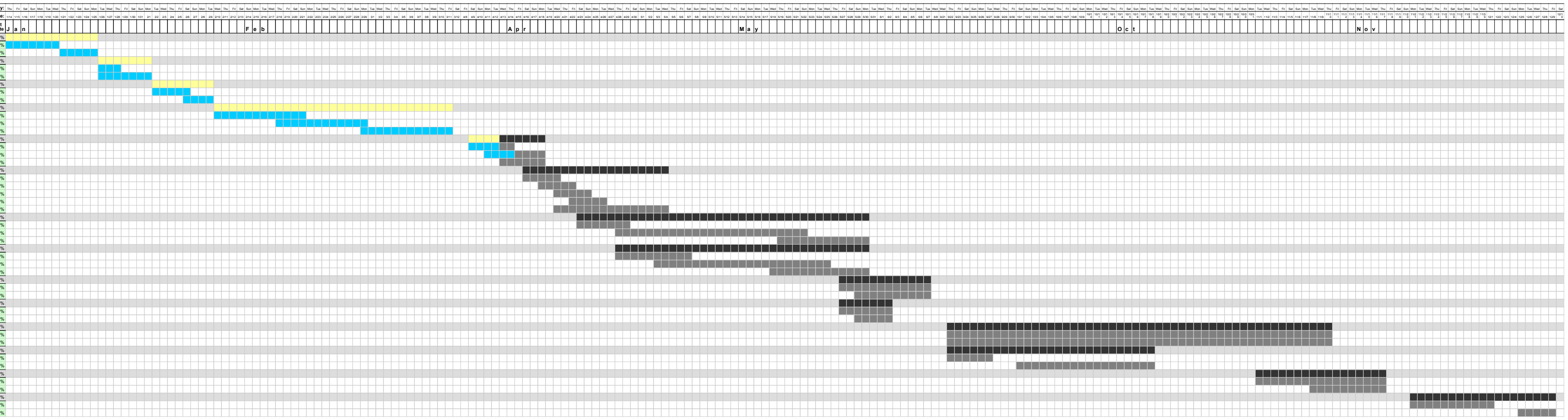


I2

Automated Machine Tender - Tender Cuts

Start Date: January 14, 2016

Task	Start Date	End Date	Duration (days)	Percent Complete
1.0 Research for Project	1/14/2016	1/25/2016	12	100.00%
1.1 Individual Research	1/14/2016	1/20/2016	7	100.00%
1.2 Compile Research	1/21/2016	1/25/2016	5	100.00%
2.0 Project Proposal	1/26/2016	2/1/2016	7	100.00%
2.1 Define Problem	1/26/2016	1/28/2016	3	100.00%
2.2 Completed Proposal	1/26/2016	2/1/2016	7	100.00%
3.0 Brainstorming	2/2/2016	2/9/2016	8	100.00%
3.1 Individual Brainstorming	2/2/2016	2/6/2016	5	100.00%
3.2 Group Brainstorming	2/6/2016	2/9/2016	4	100.00%
4.0 Preliminary Design Report	2/10/2016	3/11/2016	31	100.00%
4.1 Analysis of Concepts	2/10/2016	2/21/2016	12	100.00%
4.2 Completed Report	2/18/2016	2/29/2016	12	100.00%
4.3 Design Review	2/29/2016	3/11/2016	12	100.00%
5.0 Technical Responsibilities	4/9/2016	4/18/2016	10	45.00%
5.1 Size Mechanical Parts	4/9/2016	4/14/2016	6	70.00%
5.2 Computer Modeling	4/11/2016	4/18/2016	8	60.00%
5.2 Coding Diagrams	4/13/2016	4/18/2016	6	0.00%
6.0 Critical Design Review	4/16/2016	5/4/2016	19	0.00%
6.1 Detailed Drawings	4/16/2016	4/20/2016	5	0.00%
6.2 Material Selection	4/18/2016	4/22/2016	5	0.00%
6.3 Assembly Instructions	4/20/2016	4/24/2016	5	0.00%
6.4 Cost Analysis	4/22/2016	4/26/2016	5	0.00%
6.5 Presentation/Report	4/20/2016	5/4/2016	15	0.00%
7.0 Machining/Part Acquisition	4/23/2016	5/30/2016	38	0.00%
7.1 Purchase Materials	4/23/2016	4/29/2016	7	0.00%
7.1 Fabricate Parts	4/28/2016	5/22/2016	25	0.00%
7.3 Prototype Assembly	5/19/2016	5/30/2016	12	0.00%
8.0 Coding	4/28/2016	5/30/2016	33	0.00%
8.1 Initial Code Complete	4/28/2016	5/7/2016	10	0.00%
8.2 Debugging/Fixing Code	5/3/2016	5/25/2016	23	0.00%
8.3 Working Code	5/18/2016	5/30/2016	13	0.00%
9.0 Manufacturing/Test Review	5/27/2016	6/7/2016	12	0.00%
9.1 Current Status/Test Plan	5/27/2016	6/7/2016	12	0.00%
9.2 Safety Check/Schedule	5/29/2016	6/7/2016	10	0.00%
10.0 Project Update Report	5/27/2016	6/2/2016	7	0.00%
10.1 Compile Building Photos	5/27/2016	6/2/2016	7	0.00%
10.2 Report Time Spent	5/29/2016	6/2/2016	5	0.00%
11.0 Test Prototype	9/22/2016	11/10/2016	50	0.00%
11.1 Test Hardware	9/22/2016	11/10/2016	50	0.00%
11.2 Test Software	9/22/2016	11/10/2016	50	0.00%
12.0 Project Reports/Tasks	9/22/2016	10/18/2016	27	0.00%
12.1 Project Update Memo	9/22/2016	9/27/2016	6	0.00%
12.2 Hardware/Safety Demo	10/1/2016	10/18/2016	18	0.00%
13.0 Senior Design Expo	11/1/2016	11/17/2016	17	0.00%
13.1 Final Prototype	11/1/2016	11/17/2016	17	0.00%
13.2 Poster	11/8/2016	11/17/2016	10	0.00%
14.0 Final Tasks	11/21/2016	12/9/2016	19	0.00%
14.1 Final Project Report	11/21/2016	12/1/2016	11	0.00%
14.2 Completion Checklist	12/5/2016	12/9/2016	5	0.00%



SENIOR PROJECT CRITICAL DESIGN HAZARD IDENTIFICATION CHECKLIST

Team: Tender Cuts Advisor: Rossman

Y	N	
☑	☐	Do any parts 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 adequately guarded?
☐	☑	Does any part of the design undergo high accelerations/decelerations that are exposed to the user?
☑	☐	Does the system have any large moving masses or large forces that can contact the user?
☐	☑	Does the system produce a projectile?
☑	☐	Can the system to fall under gravity creating injury?
☐	☑	Is the user exposed to overhanging weights as part of the design?
☑	☐	Does the system have any sharp edges exposed?
☐	☑	Are there any ungrounded electrical systems in the design?
☐	☑	Are there any large capacity batteries or is there electrical voltage in the system above 40 V either AC or DC?
☑	☐	Is there be any stored energy in the system such as batteries, flywheels, hanging weights or pressurized fluids when the system is either on or off?
☐	☑	Are there any explosive or flammable liquids, gases, dust, or fuel in the system?
☐	☑	Is the user of the design required to exert any abnormal effort and/or assume a an abnormal physical posture during the use of the design?
☐	☑	Are there any materials known to be hazardous to humans involved in either the design or the manufacturing of the design?
☐	☑	Will the system generate high levels of noise?
☐	☑	Will the product be subjected to extreme environmental conditions such as fog, humidity, cold, high temperatures ,etc. that could create an unsafe condition?
☑	☐	Is it easy to use the system unsafely?
☐	☑	Are there any other potential hazards not listed above? If yes, please explain on the back of this checklist.

For any “Y” responses, add a complete description on the reverse side. DO NOT fill in the corrective actions or dates until you meet with the mechanical and electrical technicians.

Description of Hazard	Corrective Actions to Be Taken	Planned Completion Date	Actual Completion Date
The machine has pinch points at the ends of every axis of travel.			
There is a 6 foot long 2 inch steel rod that could come in contact with the user.			
Due to its height and the weight of the materials, if the system were to fall over, it might cause injury.			
The system has corners the user could come into contact with.			
The gripper is actuated by a pneumatic cylinder, which runs on pressurized air.			
It would be a simple matter for a user to stick a limb into the arm's area of travel.			

Appendix J - Safety

Item/Function	Potential Failure Mode	Potential Failure Effect(s)	Severity (1-10)	Potential Cause(s)/ Mechanism(s) of Failure	Occurance (1-10)	Criticality	Recommended Action(s)	Responsibility & Target Completion Date	Action Results				
									Action Taken	Severity	Occurance	Detection	Rpn
Telescoping	Gear Misalignment	Inefficient Motion	4	High Shock, Poor Material	3	12	Limit Acceleration	Louis, 6/7/2016					
	Debris Entry	Jamming	7	Foreign Intrusion, Overextending Arm	3	21	Cover openings	Louis, 6/7/2017					
	Shaft Misalignment	Unintended Rubbing	5	Yield, Rough Surface Finish, Foreign Intrusion, High Shock	4	20	Use self-aligning bearing	Louis, 6/7/2018					
Gripping	Weak Grip	Unable to Lift Part	7	Low Friction Factor, Low Force, Incompatible Gripping Angle	1	7	Layer with high-friction material, switch to pneumatic	Sam, 6/7/2016					
	Sensor Malfunction	Unable to Detect Part	7	Compressive Damage, Sensor Misalignment, Power to sensor lost	2	14	switch to pneumatic	Sam, 6/7/2017					
Rectilinear Motion	Actuator Failure	No Movement	7	Overtorque, High Speed, Loss of power, Overheat, Short	2	14	Implement feedback	Ryan, 6/7/2016					
	Bearing Misalignment	Magnify Stresses	5	Yield, High Shock, Deflection	1	5	Change geometry	Ryan, 6/7/2017					
	Move too slow	Fail to Meet Time Requirement	6	Low Current, Low pulse rate	4	24	Use appropriate power supply for entire system, increase pulse rate	Ryan, 6/7/2018					
	Fatigue	Fracture	8	High amplitude, poor material	1	8	Use large safety factor	Ryan, 6/7/2019					
User Input	Coding Error	Doesn't Work	7	Numerous	7	49	Good documentation	Louis, Sam 6/7/2016					
	Impossible Instruction	Perpetual Loop	7	Fails to respond to commands	4	28	Reset	Louis, Sam 6/7/2017					
	Self Harming Instruction	Damage to Tender	8	Lack of failsafes	2	16	Introduce Failsafe	Louis, Sam 6/7/2018					
	Overloading the Machine	Yield in Arm, Can't Lift Part	8	High-density part, Obstruction	2	16	Attach sensor	Louis, Sam 6/7/2019					
Zeroing Mechanism	Faulty Sensors	Wrong Position	6	No power to sensors, Uncalibrated	2	12	Calibrate sensors	Louis, Sam 6/7/2020					
	Unintended obstructions	Damage to Tender	8	Unable to detect obstructions	3	24	Attach sensor	Louis, Sam 6/7/2021					
	Zeros in wrong location	Wrong Position	6	Too fast, sensor in wrong location	3	18	Use two sensors	Louis, Sam 6/7/2022					
Power Supply	Overheat	Loss of Electrical Components	8	Lack of heat transfer away from power supply	3	24	Fins, access to open air, fans	Louis, 6/7/2016					
	Interrupted Power Supply	Part Drop, Tender out of Order With CNC	8	Power outage, damaged wiring	4	32	Uninterruptible power supply	Louis, 6/7/2017					
	Surge/Excess	Loss of Electrical Components	8	Power surge, short	3	24	fuse/breaker	Louis, 6/7/2018					
	Short	Loss of Electrical Components	8	Lack of insulation, no grounding	3	24	use insulation, ground.	Louis, 6/7/2019					
Structural Integrity	Buckling	Shafts Buckle	8	Not enough distribution, too few shafts, too thin	3	24	Change geometry	Ryan, 6/7/2016					
	Yield	Bending	7	Under sizing, wrong material, Bad geometry	3	21	Change geometry	Ryan, 6/7/2017					
	Fatigue	Fracture	8	High amplitute, under sized, wrong material, bad geometry	3	24	high factor of safety	Ryan, 6/7/2018					
Programming	Unanticipated Overflow	Varies Significantly	8	Failure to account for overflow	5	40	Saturate Values	All, 6/7/2016					
	Faulty Task Diagrams	Time Lost Fixing Logic During Programming	6	Logical Error	5	30	Do not rush state diagrams	All, 6/7/2017					

Air gripper(parallel style)

HFZ Series

Specification

Bore size (mm)			6	10	16	20	25	32	40
Acting type			Double acting Single acting						
Fluid			Air(to be filtered by 40 μ m filter element)						
Operating pressure	Double acting	Φ6, Φ10	0.2~0.7MPa(28~100psi)(2.0~7.0bar)						
		Others	0.1~0.7MPa(15~100psi)(1.0~7.0bar)						
	Single acting	Φ6, Φ10	0.35~0.7MPa(50~100psi)(3.5~7.0bar)						
		Others	0.25~0.7MPa(36~100psi)(2.5~7.0bar)						
Temperature ℃			-20~70						
Lubrication			Not required						
Repeatability mm			±0.01					±0.02	
Max. frequency			180(c.p.m)					60(c.p.m)	
Sensor switches ①			DS1-H	CS1-G DS1-G	CS1-G, DS1-G, DS1-H				
Port size			M3×0.5		M5×0.8				

① Sensor switch should be ordered additionally, please refer to P397~420 for detail of sensor switch.

Ordering code

HFZ 20 □

Model

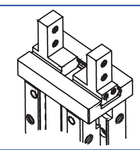
HFZ: Air finger(Double acting)
HFSZ: Air finger(Single acting and nrmally closed)
HFTZ: Air finger(Single acting and normally opened)

Bore size

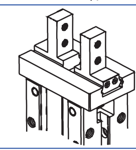
6: Φ6mm
10: Φ10mm
16: Φ16mm
20: Φ20mm
25: Φ25mm
32: Φ32mm
40: Φ40mm

Finger type

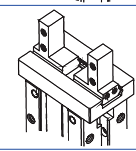
Blank:Standard



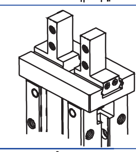
R:Narrow type



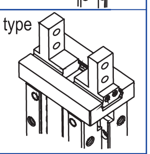
B:Side mounting type



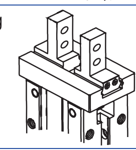
W:Side mounting and narrow type



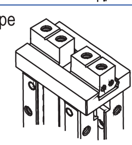
N:Thru.hole mounting type



M:Thru.hole mounting and narrow type



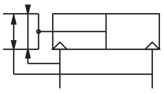
F:Bottom mounting type



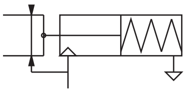
① Φ6, Φ32, Φ40 bore size don't have R, W & M type. Add) HFZ series are all attached with magnet.

Symbol

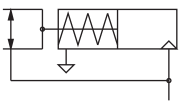
HFZ: Double acting



HFTZ: Single acting and normally opened

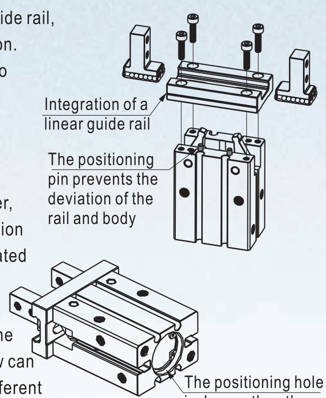


HFSZ: Single acting and normally closed



Product feature

1. Integrated design of linear guide rail, high rigidity and high precision.
2. A positioning pin is attached to the bottom of the linear guide rail, which can prevent the deviation of the positioning rail and body.
3. The hole of the body is deeper, which can improve the precision and the consistency of repeated dismounting and positioning.
4. According to the actual using requirements of customers, the initial position of clamping jaw can be customized to meet the different needs under different working conditions.

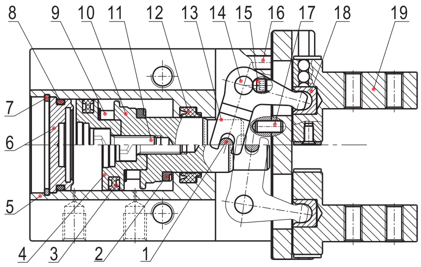


Back
HFZ

Air gripper(parallel style)

HFZ Series

Inner structure and material of major parts



NO.	Item	Material
1	Pin	Stainless steel
2	Bumper	TPU
3	Piston seal	NBR
4	Piston	Aluminum alloy/Stainless steel
5	Body	Aluminum alloy
6	Back cover	Aluminum alloy
7	C clip	Spring steel
8	O－ring	NBR
9	Magnet	Sintered metal(Neodymium-iron-boron)
10	Piston rod	Aluminum alloy/Stainless steel
11	Screw	Carbon steel
12	Rod packing	NBR
13	Curved bar	Stainless steel
14	Pin	Stainless steel
15	Countersink screw	Carbon steel
16	Hexagon screw	Carbon steel
17	Pin	Stainless steel
18	Guide sleeve	Stainless steel
19	Assembly of clamping jaw and guide rail	Stainless steel

Gripping force and stroke

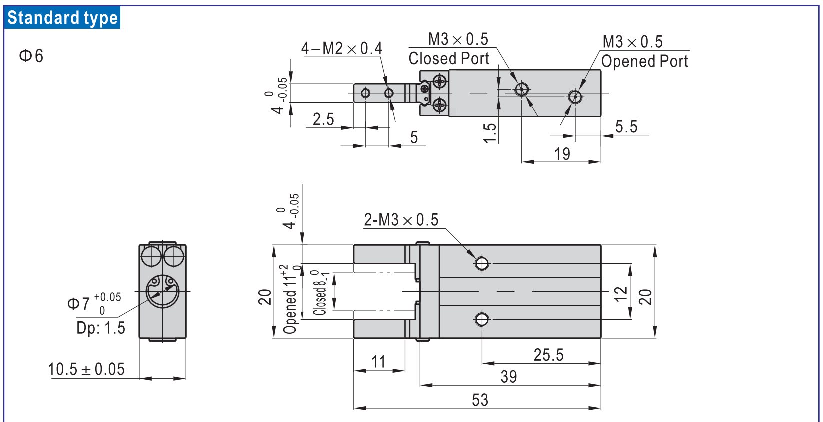
Acting		Model	Gripping force per finger Effective valve(N)		Opening/Closing stroke (Both sides)(mm)	Weight (g)	
			External	Internal		F Type	Others
Double acting		HFZ6	3.3	6.1	4	24	25
		HFZ10	11	17	4	56	56
		HFZ16	34	45	6	124	124
		HFZ20	45	68	10	236	236
		HFZ25	69	102	14	418	428
		HFZ32	160	195	22	750	729
		HFZ40	255	320	30	1340	1268
Single acting	Normally opened	HFTZ6	1.9	-	4	25	26
		HFTZ10	7	-	4	57	57
		HFTZ16	27	-	6	125	125
		HFTZ20	35	-	10	238	238
		HFTZ25	55	-	14	420	430
		HFTZ32	133	-	22	799	778
		HFTZ40	220	-	30	1437	1365
	Normally closed	HFSZ6	-	3.7	4	25	26
		HFSZ11	-	13	4	57	57
		HFSZ16	-	38	6	125	125
		HFSZ20	-	59	10	238	238
		HFSZ25	-	87	14	420	430
		HFSZ32	-	163	22	799	778
		HFSZ40	-	270	30	1437	1365

Note) The gripping force in the above table is in the working pressure of 0.5MPa, and with a gripping point of L=20mm.
Add) Please refer to page 355 for the definition of "L".

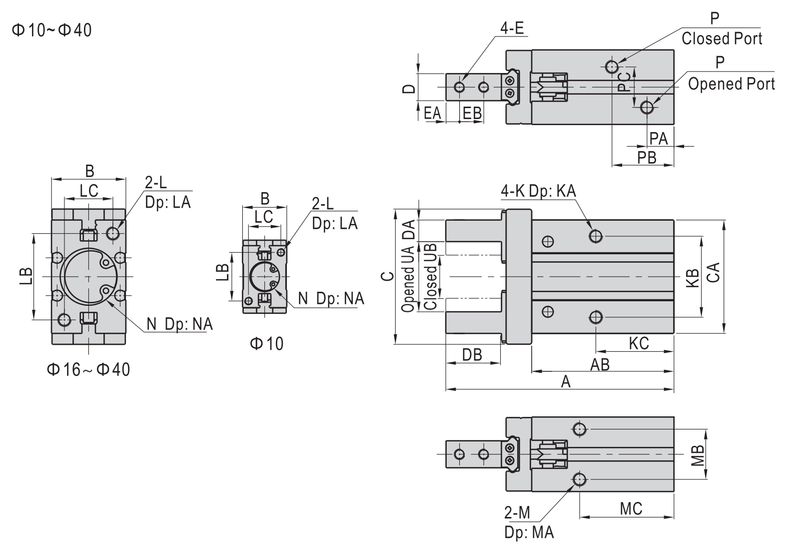
Dimensions

Standard type

Φ6



Φ10~Φ40



Model\Item	A	AB	B	C	CA	D	DA	DB	E	EA	EB
HFZ10	57	37.5	16.4	29	23	$5^{0}_{-0.05}$	$4^{0}_{-0.05}$	12	M2.5×0.45	3	5.7
HFZ16	67.5	42.5	23.6	38	30.5	$8^{0}_{-0.05}$	$5^{0}_{-0.05}$	15	M3×0.5	4	7
HFZ20	84.5	53	27.6	50	42	$10^{0}_{-0.05}$	$8^{0}_{-0.05}$	20	M4×0.7	5	9
HFZ25	102.5	63.5	33.6	63	52	$12^{0}_{-0.05}$	$10^{0}_{-0.05}$	25	M5×0.8	6	12
HFZ32	113(122)	67(76)	40	97	60	$15^{0}_{-0.05}$	$12^{0}_{-0.05}$	29	M6×1.0	7	14
HFZ40	139(152)	83(96)	48	119	72	$18^{0}_{-0.05}$	$14^{0}_{-0.05}$	36	M8×1.25	9	17

Model\Item	K	KA	KB	KC	L	LA	LB	LC	M	MA	MB	MC
HFZ10	M3×0.5	5.5	16	23	M3×0.5	6	18	12	M3×0.5	6	11.5	27
HFZ16	M4×0.7	8	24	24.5	M4×0.7	8	22	15	M4×0.7	4.5	16	30
HFZ20	M5×0.8	10	30	29	M5×0.8	10	32	18	M5×0.8	8	18.5	35
HFZ25	M6×1.0	12	36	30	M6×1.0	12	40	22	M6×1.0	10	22	36.5
HFZ32	M6×1.0	13	46	40(49)	M6×1.0	13	46	26	M6×1.0	10	26	48(57)
HFZ40	M8×1.25	16	56	49(62)	M8×1.25	17	56	32	M8×1.25	13	32	58(71)

Model\Item	N	NA	P	PA	PB	PC	UA(Opened)	UB(Closed)
HFZ10	$\Phi 11^{+0.05}_{0}$	2	M3×0.5	7.5	19	10	15.5^{+2}_{0}	11.5^{0}_{-1}
HFZ16	$\Phi 17^{+0.05}_{0}$	2	M5×0.8	7.5	19	13	21^{+2}_{0}	15^{0}_{-1}
HFZ20	$\Phi 21^{+0.05}_{0}$	3	M5×0.8	9.5	23	15	26.5^{+2}_{0}	16.5^{0}_{-1}
HFZ25	$\Phi 26^{+0.05}_{0}$	3.5	M5×0.8	10	24	20	33.5^{+2}_{0}	19.5^{0}_{-1}
HFZ32	$\Phi 34^{+0.05}_{0}$	4	M5×0.8	11	31(40)	24	$48^{+2.5}_{0}$	26^{0}_{-1}
HFZ40	$\Phi 42^{+0.05}_{0}$	4	M5×0.8	12	38(50)	28	$60^{+2.5}_{0}$	30^{0}_{-1}

Note) The values in "()" in the above table are single acting type sizes.

HFZ

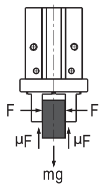
Air gripper(parallel style)

HFZ Series

How to select product

Please select pneumatic finger according to the following steps:
The selection of the effective gripping force → the confirmation of the gripping point → the confirmation of the external force put on the gripping jaw.

1. The selection of the gripping force



The gripping work-pieces shown above, on the impact condition of ordinary handling state, taking safety coefficient a=4, have a gripping force that is more than 10-20 times of the mass of the gripped objects.

μ =0.2	μ =0.1
$F=\frac{mg}{2\times0.2}\times4$ $=10\times mg$	$F=\frac{mg}{2\times0.1}\times4$ $=20\times mg$
10 times of the mass of the gripped objects	20 times of the mass of the gripped objects

The work-pieces as shown in the left :
F: Gripping force (N)
μ : friction coefficient between fittings and work-pieces.
m: mass of work-pieces
g: acceleration of gravity (=9.8m/s^2)

The condition that the work-pieces won't drop is: $2\times\mu F>mg$

so: $F>\frac{mg}{2\times\mu}$

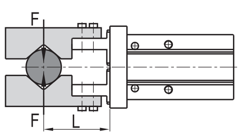
Safety coefficient is a, so F is:

$F=\frac{mg}{2\times\mu}\times a$

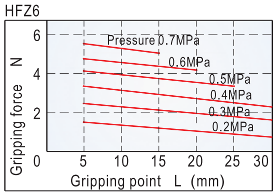
Note) If the friction coefficient μ >0.2, for safety, please also select clamping force according to the principle of 10~20 times of the mass of the clamped objects. As for large acceleration and shock, it requires for greater safety coefficient.

1.1) The actual gripping force must be within the effective gripping forces of different pneumatic fingers specifications shown in the below chart.

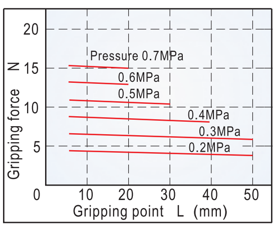
Double acting type closed gripping force



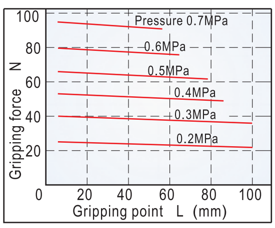
HFZ6



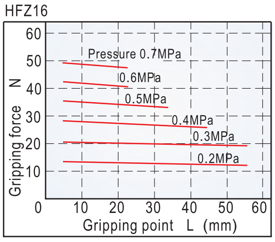
HFZ10



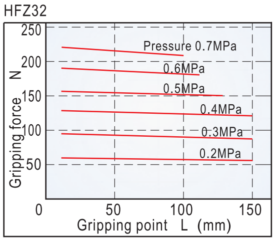
HFZ25



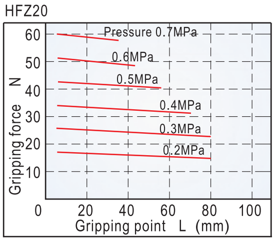
HFZ16



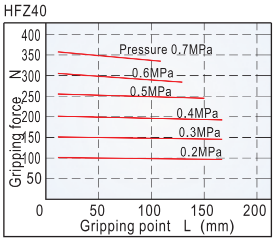
HFZ32



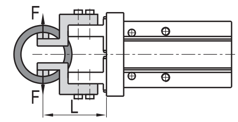
HFZ20



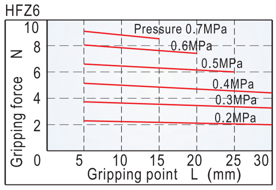
HFZ40



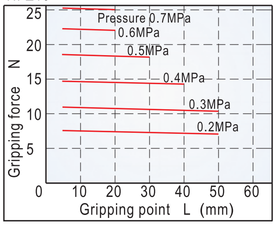
Double acting type opened gripping force



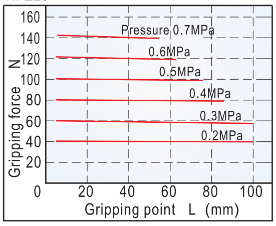
HFZ6



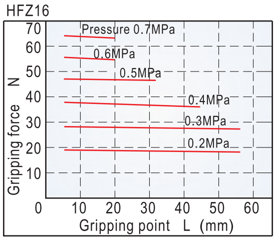
HFZ10



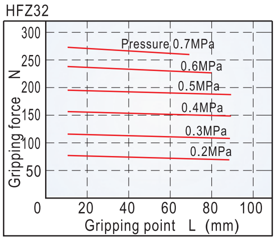
HFZ25



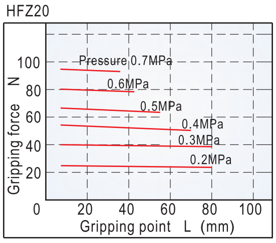
HFZ16



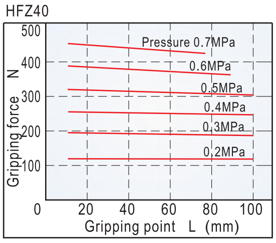
HFZ32



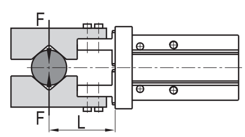
HFZ20



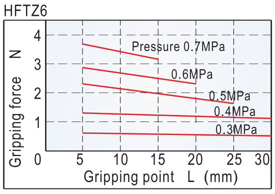
HFZ40



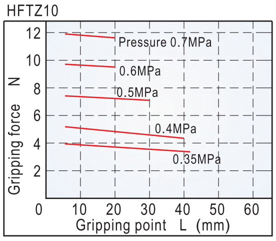
Single acting normally opened gripping force



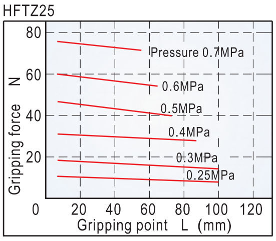
HFTZ6



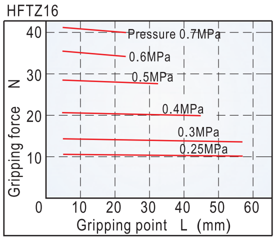
HFTZ10



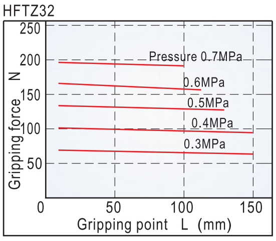
HFTZ25



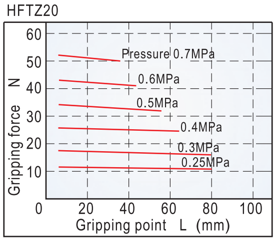
HFTZ16



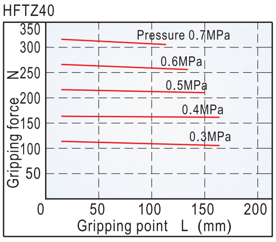
HFTZ32



HFTZ20



HFTZ40

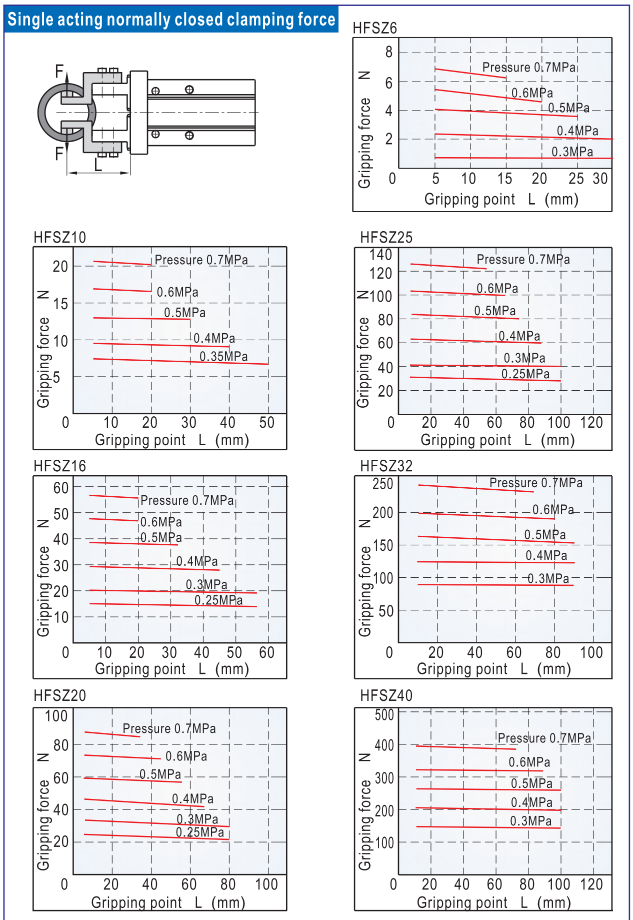


Back
HFZ

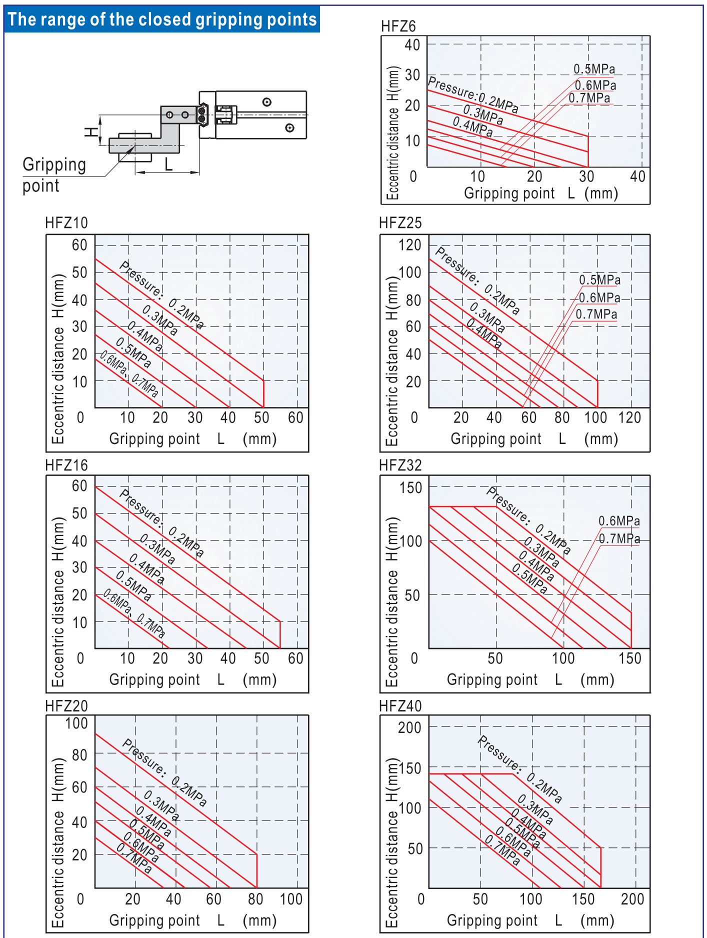
Air gripper(parallel style)

HFZ Series

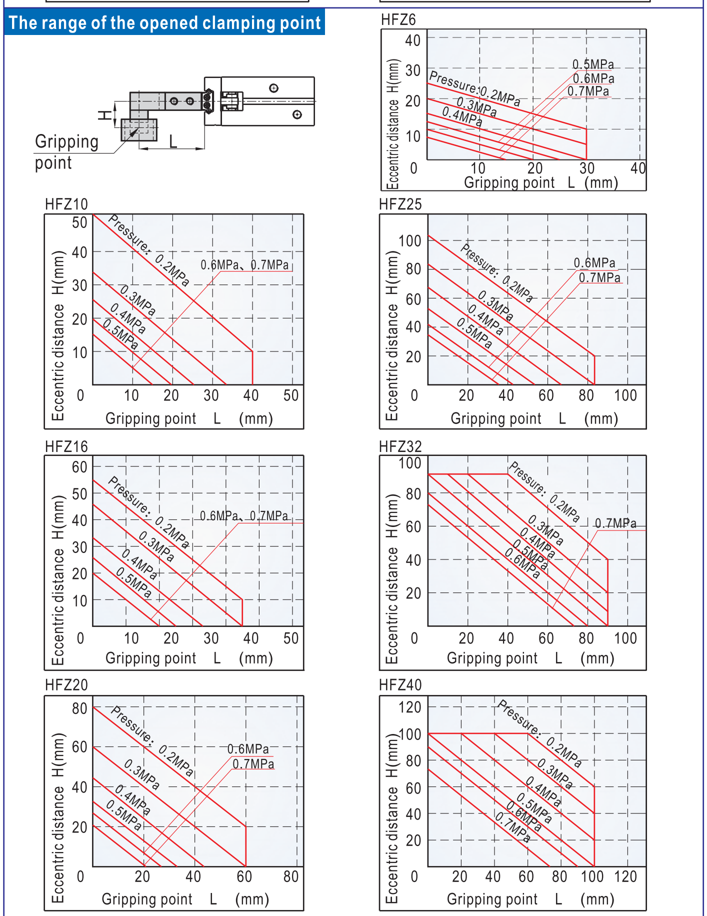
Single acting normally closed clamping force



The range of the closed gripping points

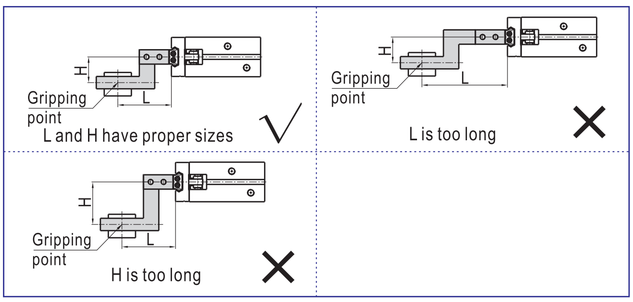


The range of the opened clamping point



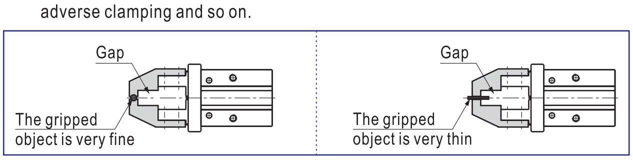
2. The selection of the gripping point

2.1) Please select the gripping point within the limited field shown below.
Over the limits, gripping jaws would be subjected to excessive torque loads, and lead to short life of the air gripper.



2.2) In the allowable range of gripping point, it is better to design for short and light fittings. If the fittings are long and heavy, the inertia force when the finger is open and close will become larger, and the performance of gripping jaw will be degraded, at the same time it will affect the life.

2.3) When the gripped object is very fine and thin, you have to equip with gap between fittings. If not, there will be unstable clamp, resulting in a position offset and adverse clamping and so on.

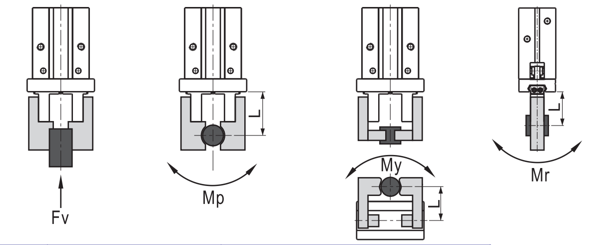


Back
HFZ

Air gripper(parallel style)

HFZ Series

3. The confirmation of the external force put on the gripping jaw.



Bore size	The allowed vertical loads Fv(N)	Max. permissible torque (Nm)		
		Mp	My	Mr
6	10	0.04	0.04	0.08
10	58	0.26	0.26	0.53
16	98	0.68	0.68	1.36
20	147	1.32	1.32	2.65
25	255	1.94	1.94	3.88
32	343	3	3	6
40	490	4.5	4.5	9

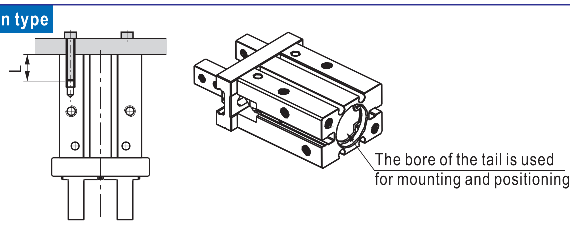
Note) The loads and torque values of said are all static values.

The calculation of allowable forces when moment loads work	Examples of calculation
Allowable load(N) $= \frac{\text{M(Maximum permissible moment)(N.m)}}{L \times \boxed{10^{-3}}}$ Unit conversion constant	In the guide rail of HFZ16, the external force of the pitching moment static loads put on the point of L=30mm is f=10 N, Allowable load F= $\frac{0.68}{30 \times 10^{-3}}$ = 22.7(N) Actual load f=10(N)<22.7(N) To meet the using requirements

Installation and application

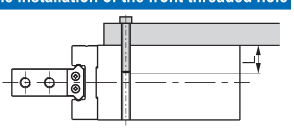
1. Due to the abrupt changes, the circuit pressure is low, which will lead to the decrease of the gripping force and falling of the work-pieces. In order to avoid the harm to the human body and damage to the equipment, anti-dropping device must be equipped.
2. Don't use the air gripper under strong external force and impact force.
3. Please contact with us when the single acting type clamps only with the spring force.
4. When install and fix the air gripper, avoid falling down, collision and damage.
5. When fixing the gripping jaw parts, don't twist the gripping jaw.
6. There are several kinds of installation method, and the locking torgue of fastening screw must be within the prescribed torque range shown in the below chart. If the locking torque is too large, it will cause the dysfunctional. If the locking torque is too small, it will cause the position deviation and fall.

Tail installation type



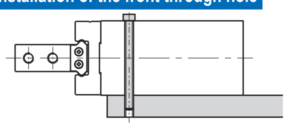
Bore size	The bolts type	Max. locking moment (Nm)	Max. screwed depth (mm)	The aperture of the positioning bore(mm)	The depth of the positioning bore(mm)
10	M3×0.5	0.88	6	$\Phi 11^{+0.05}_{0}$	2
16	M4×0.7	2.1	8	$\Phi 17^{+0.05}_{0}$	2
20	M5×0.8	4.3	10	$\Phi 21^{+0.05}_{0}$	3
25	M6×1.0	7.3	12	$\Phi 26^{+0.05}_{0}$	3.5
32	M6×1.0	7.9	13	$\Phi 34^{+0.05}_{0}$	4
40	M8×1.25	17.7	17	$\Phi 42^{+0.05}_{0}$	4

The installation of the front threaded hole



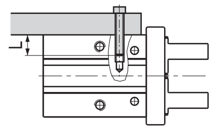
Bore size	The bolts type	Max. locking moment (Nm)	Max. screwed depth (mm)
6	M3×0.5	0.88	10
10	M3×0.5	0.69	5
16	M4×0.7	2.1	8
20	M5×0.8	4.3	10
25	M6×1.0	7.3	12
32	M6×1.0	7.9	13
40	M8×1.25	17.7	16

The installation of the front through hole



Bore size	The bolts type	Max. locking moment (Nm)	Max. screwed depth (mm)
6	M2.5×0.45	0.49	-
10	M2.5×0.45	0.49	5
16	M3×0.5	0.88	8
20	M4×0.7	2.1	10
25	M5×0.8	4.3	12
32	M5×0.8	4.3	13
40	M6×1.0	7.3	16

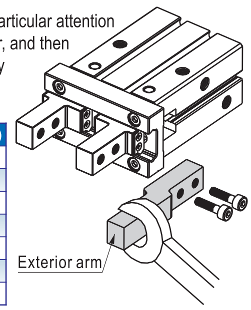
Surface installation type



Bore size	The bolts type	Max. locking moment (Nm)	Max. screwed depth (mm)
10	M3×0.5	0.9	6
16	M4×0.7	1.6	4.5
20	M5×0.8	3.3	8
25	M6×1.0	5.9	10
32	M6×1.0	5.9	10
40	M8×1.25	13.7	13

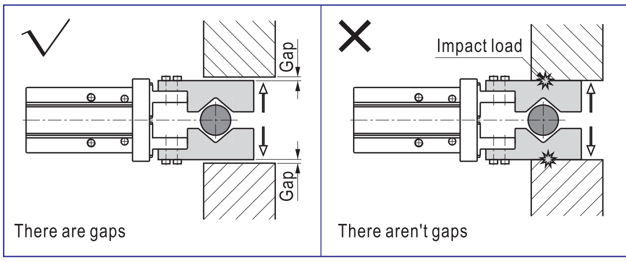
7. The installation method of the gripping jaw fittings
When install the gripping jaw fittings, you have to pay particular attention that you can only hold the gripping jaw by using spanner, and then lock the screws with allen wrench. Never clamp the body directly and then lock the screws, otherwise the parts will be easily damaged.

Bore size	The bolts type	Max. locking moment (Nm)
6	M2×0.4	0.15
10	M2.5×0.45	0.31
16	M3×0.5	0.59
20	M4×0.7	1.4
25	M5×0.8	2.8
32	M6×1.0	4.9
40	M8×1.25	11.8

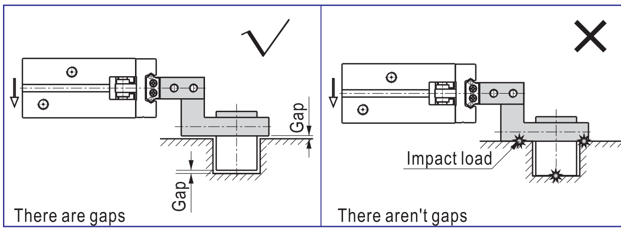


8. Confirm that there is no external forces exerted on the gripping jaw.
Transverse load acts on the gripping jaw, which will cause impact load and leads to the shaking and damage of gripping jaw. Equip with gaps so that the air gripper will not crash into work-pieces and accessories at the end of its trip.

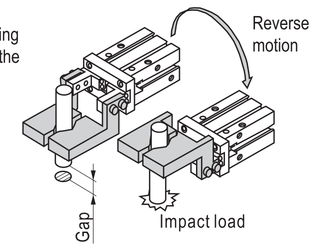
8.1) The end of stroke under the open state of air gripper



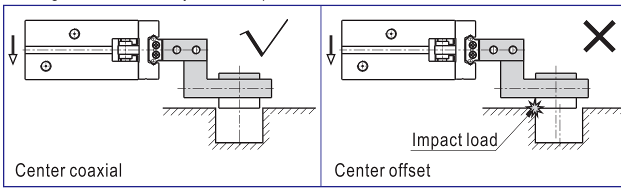
8.2) The end of stroke under the move state of air gripper



8.3) Reverse motion state
When reverse motion state, the gripping point must be precision, otherwise in the reverse motion state the air gripper maybe impact with ambience and will cause impact load .



9. When the work-pieces are inserted, the center line should be coaxial, no offset, in case there are external force generated on gripping jaw. When testing, it is specially required that the manual operation should be reduced, the pressure should be used to run it at a low speed, and guarantee the safety and no impact.



10. Please use the flow control valve to adjust the opening and closing speed of gripping jaw if too fast.
11. People can not enter the movement path of air gripper and articles can not be placed on the path too.
12. Before removing the air gripper, please confirm that it is out of working state, and then discharge of compressed air.

Belt Sizing Guide

Many conveying timing belts operate at low speeds and minimal loads. This eliminates the need for extensive calculations and a simplified approach to belt selection can be used. For these lightly loaded applications, the belt can be selected according to the dimensional requirements of the system, product size, desired pulley diameter, conveyor length, etc.

The belt width ***b*** is often determined according to the size of the product conveyed, and as a rule, the smallest available belt pitch is used. For proper operation, the pre-tension T_i should be set as follows:

$$T_i \approx 0.3 \cdot b \cdot T_{1all}$$

where: T_i = belt pre-tension
T_{1all} = max allowable belt tension for 1" or 25mm wide belt (see Table 1 or Table 2)

U.S. customary units: T_i [lb], T_{1all} [lb/in], b [in]
Metric units: T_i [N], T_{1all} [N/25mm], b [mm].

For all applications where the loads are significant, the following step-by-step procedure should be used for proper belt selection.

Step 1. Determine Effective Tension

The effective tension T_e at the driver pulley is the sum of all individual forces resisting the belt motion. The individual loads contributing to the effective tension must be identified and calculated based on the loading conditions and drive configuration. However, some loads cannot be calculated until the layout has been decided.

To determine the effective tension T_e use one of the following methods for either conveying or linear positioning.

Conveying

T_e for conveying application is primarily the sum of the following forces (see Figs. 1 and 2).

1. The friction force F_f between the belt and the slider bed resulting from the weight of the conveyed material.

$$F_f = \mu \cdot w_m \cdot L_m \cdot \cos\beta$$

where: μ = coefficient of friction between the slider bed and the belt (see Table 1A)
w_m = load weight per unit length over conveying length
L_m = conveying length
ß = angle of conveyor incline

U.S. customary units: F_f [lb], w_m [lb/ft], L_m [ft].
Metric units: F_f [N], w_m [N/m], L_m [m].

2. The gravitational load F_g to lift the material being transported on an inclined conveyor.

$$F_g = w_m \cdot L_m \cdot \sin\beta$$

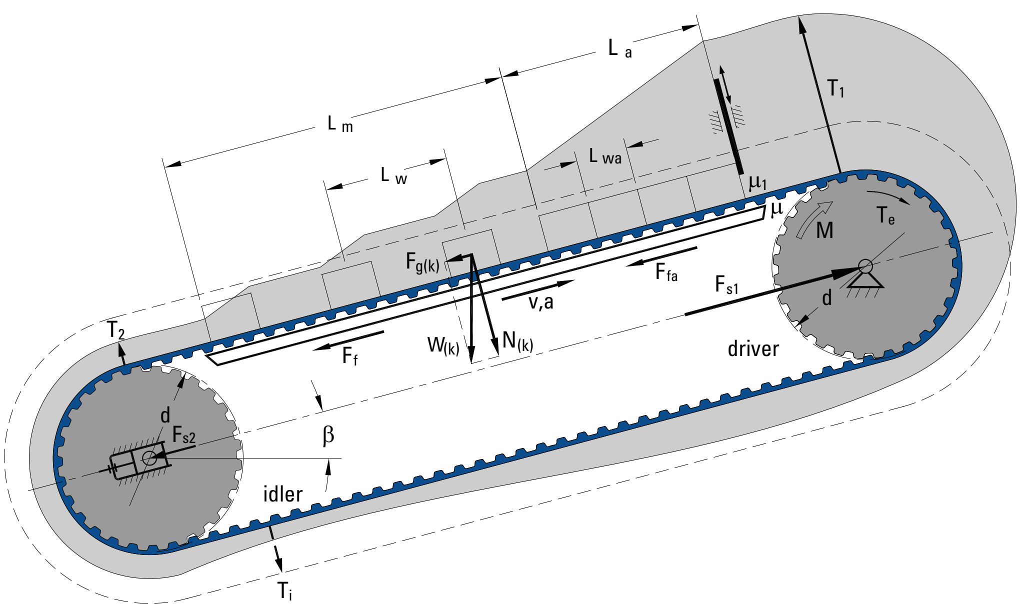


Fig. 1

Belt Sizing Guide

5. The force F_{ai} required to accelerate the idler.

$$F_{ai} = \frac{J_i \cdot \alpha}{r_o} = \frac{m_i \cdot r_o^2}{2 \cdot r_o} \cdot \frac{a}{r_o} = \frac{m_i \cdot a}{2}$$

where: $J_i = \frac{m_i \cdot r_o^2}{2}$ = inertia of the idler
m_i = mass of the idler
r_o = idler outer radius
$\alpha = \frac{a}{r_o}$ = angular acceleration

In the formula above, the mass of the idler m_i is approximated by the mass of a full disk.

$$m_i = \rho \cdot b_i \cdot \pi \cdot r_o^2$$

where: ρ = density of idler material
b_i = width of the idler

U.S. units: ρ [lb•s²/ft⁴], b_i and r_o [ft].
Metric units: ρ [kg/m³], b_i and r_o [m].

6. The force F_{ab} required to accelerate the belt mass.

$$F_{ab} = m_b \cdot a$$

The belt mass m_b is obtained from the specific belt weight w_b and belt length and width.

$$m_b = \frac{w_b \cdot L \cdot b}{g}$$

U.S. units: F_{ab} [lb], m_b [lb•s²/ft], a [ft/s²], w_b [lb/ft²], L and b [ft],
$g = 32.2$ ft/s².
Metric units: F_{ab} [N], m_b [kg], a [m/s²], w_b [N/m²], L and b [m],
$g = 9.81$ m/s².*

Thus for linear positioners, T_e is expressed by:

$$T_e = F_a + F_f + F_w + W_s + [F_{ai}] + [F_{ab}]$$

Note that the forces in brackets can be calculated by estimating the belt mass and idler dimensions. In most cases, however, they are negligible and can be ignored.

Step 2. Select Belt Pitch

Use Graphs 2a, 2b, 2c or 2d to select the nominal belt pitch p according to T_e. The graphs also provide an estimate of the required belt width. (For H pitch belts wider than 6" (152.4mm) and T10 pitch belts wider than 150mm, use Graph 1).

Step 3. Calculate Pulley Diameter

Use the preliminary pulley diameter $\tilde{d}$ desired for the design envelope and the selected nominal pitch p to determine the preliminary number of pulley teeth $\tilde{z}_p$.

$$\tilde{z}_p = \frac{\pi \cdot \tilde{d}}{p}$$

Round to a whole number of pulley teeth z_p. Give preference to stock pulley diameters. Check against the minimum number of pulley teeth z_{min} for the selected pitch given in Table 1 or Table 2.

Determine the pitch diameter d according to the chosen number of pulley teeth z_p.

$$d = \frac{p \cdot z_p}{\pi}$$

Step 4. Determine Belt Length and Center Distance

Use the preliminary center distance $\tilde{C}$ desired for the design envelope to determine a preliminary number of belt teeth $\tilde{z}_b$.

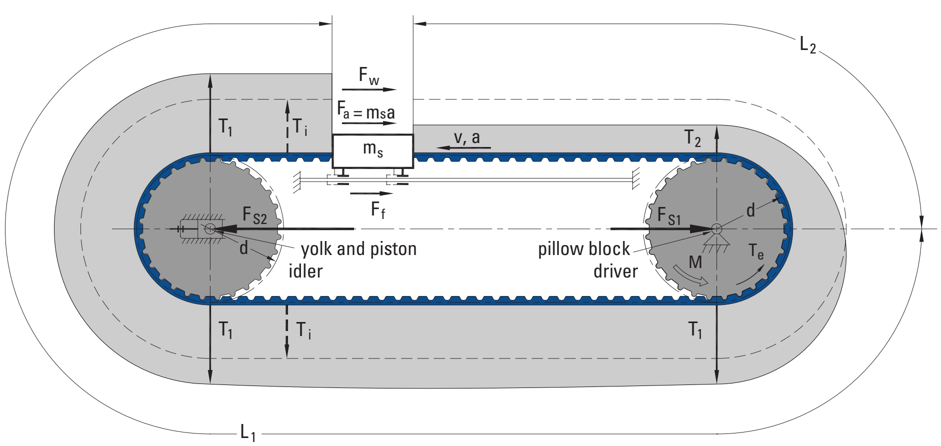


Fig. 3

Belt Sizing Guide

3. The friction force F_{fv} resulting from vacuum in vacuum conveyors.

$$F_{fv} = \mu \bullet P \bullet A_v$$

where: P = pressure (vacuum) relative to atmospheric
A_v = total area of vacuum openings

U.S. units: F_{fv} [lb], P [lb/ft^2], A_v [ft]
Metric units: F_{fv} [N], P [Pa], A_v [m]

The formula above assumes a uniform pressure and a constant coefficient of friction.

4. The friction force F_{fa} over the accumulation length in material accumulation applications.

$$F_{fa} = (\mu + \mu_a) \bullet w_{ma} \bullet L_a \bullet \cos ß$$

where: L_a = accumulation length
μ_a = friction coefficient between accumulated material and the belt (see Table 1A)
w_{ma} = material weight per unit length over the accumulation length

U.S. customary units: L_a [ft], w_{ma} [lb/ft].
Metric units: L_a [m], w_{ma} [N/m].

5. The inertial force F_a caused by the acceleration of the conveyed load (see linear positioning).

6. The friction force F_{fb} between belt and slider bed caused by the belt weight.

$$F_{fb} = \mu \bullet w_b \bullet b \bullet L_c \bullet \cos ß$$

where: w_b = specific belt weight
b = belt width
L_c = conveying length

U.S. customary units: w_b [lb/ft^2], b [ft], L_c [ft].
Metric units: w_b [N/m^2], b [m], L_c [m].*

For initial calculations, use belt width which is required to handle the size of the conveyed product.

Thus for conveyors, T_e is expressed by:

$$T_e = F_f + F_g + F_{fv} + F_{fa} + F_a + (F_{fb}) + \ldots$$

F_{fb} can be calculated by estimating the belt mass. In most cases, this weight is insignificant and can be ignored.

Note that other factors, such as belt supporting idlers, or accelerating the material fed onto the belt, may also account for some power requirement. In start-stop applications, acceleration forces as presented for linear positioning, may have to be evaluated.

Linear Positioning

T_e for a linear positioning application is primarily the sum of the following six factors (see Fig. 3).

1. The force F_a required for the acceleration of a loaded slide with the mass m_s (replace the mass of the slide with the mass of the package in conveying).

$$F_a = m_s \bullet a$$

The average acceleration a is equal to the change in velocity per unit time.

$$a = \frac{v_f - v_i}{t}$$

where: v_f = final velocity
v_i = initial velocity
t = time

U.S. customary units: F_a [lb], a [ft/s^2], v_f and v_i [ft/s] t[s]. The mass is derived from the weight W_s [lb] and the acceleration due to gravity g (g = 32.2 ft/s^2):

$$m_s = \frac{W_s}{g} = \frac{W_s}{32.2}\left[\frac{lb \bullet s^2}{ft}\right]$$

Metric units: F_a [N], a [m/s^2], v_f and v_i [m/s], t [s], m_s [kg].

2. The friction force F_f between the slide and the linear rail is determined experimentally, or from data from the linear bearing manufacturer. Other contributing factors to the friction force are bearing losses from the yolk, piston and pillow blocks (see Fig. 3).

3. The externally applied working load F_w (if existing).

4. The weight W_s of the slide (not required in horizontal drives).

* If working in US units, w_b found in the belt specifications must be converted to the units lb/ft^2. If working in metric units, w_b must be converted to the units N/m^2.

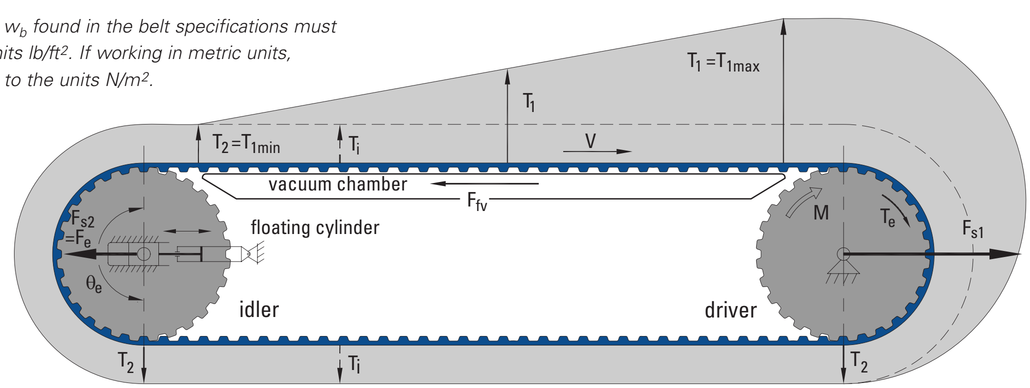
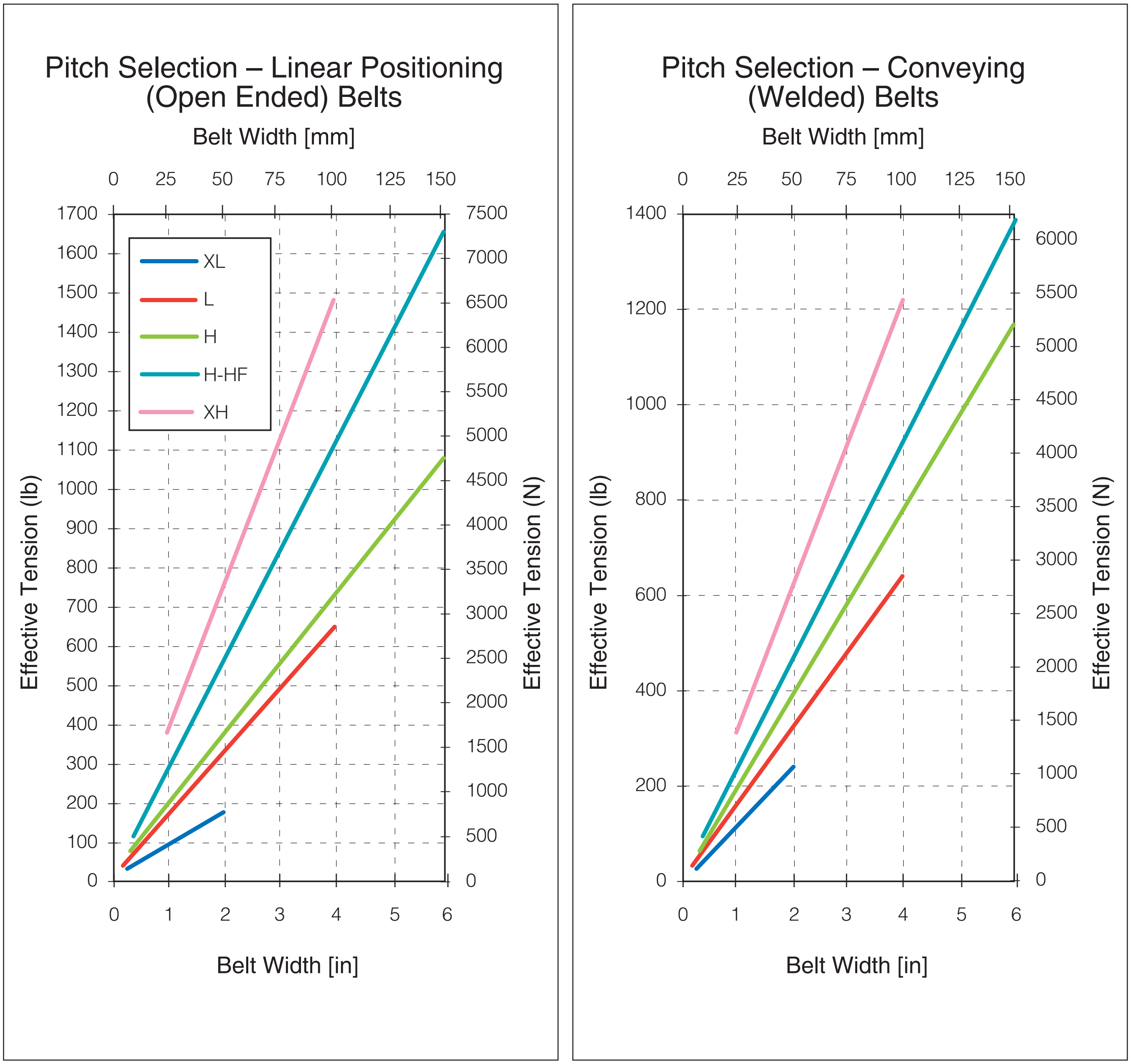
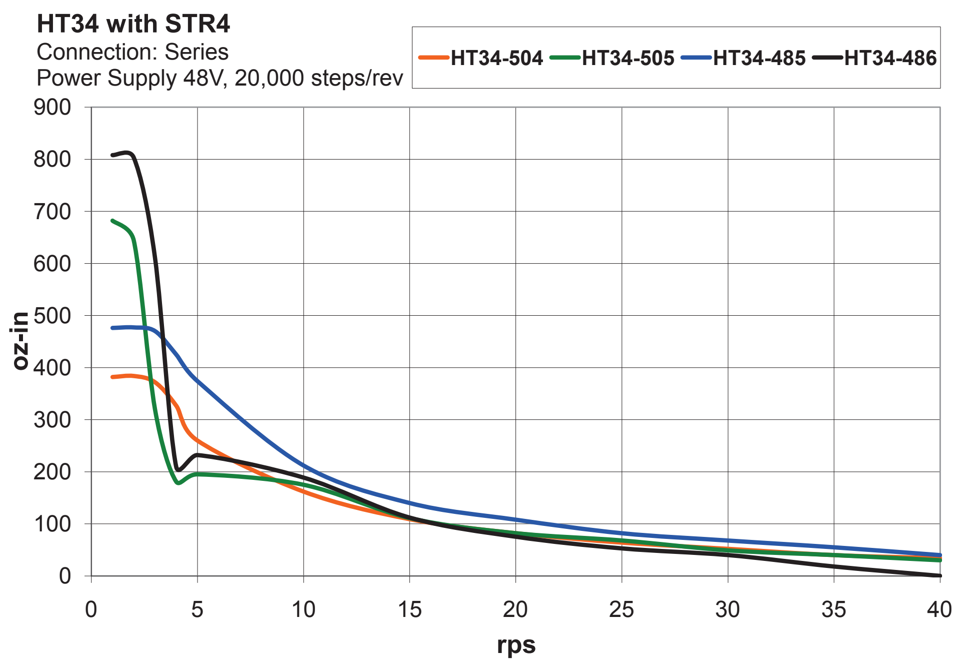


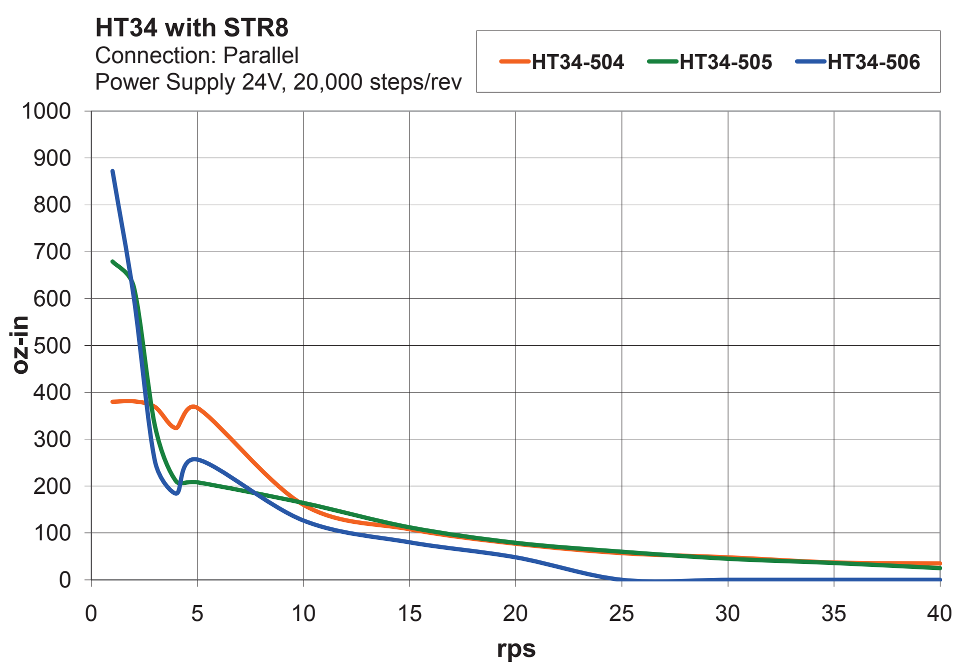
Fig. 2

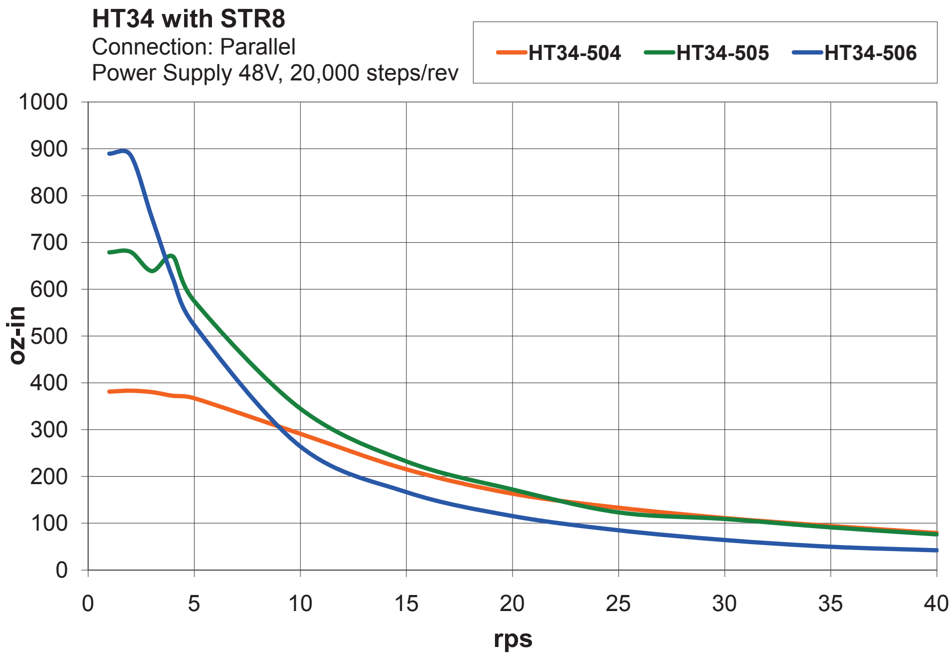
Technical Design Tools



GRAPH 2a


HT34 with STR4
Connection: Series
Power Supply 48V, 20,000 steps/rev
HT34-504
HT34-505
HT34-485
HT34-486
oz-in
rps
900
800
700
600
500
400
300
200
100
0
0
5
10
15
20
25
30
35
40


HT34 with STR8
Connection: Parallel
Power Supply 24V, 20,000 steps/rev
HT34-504
HT34-505
HT34-506
oz-in
rps
1000
900
800
700
600
500
400
300
200
100
0
0
5
10
15
20
25
30
35
40


HT34 with STR8
Connection: Parallel
Power Supply 48V, 20,000 steps/rev
HT34-504
HT34-505
HT34-506
1000
900
800
700
600
500
400
300
200
100
0
oz-in
0
5
10
15
20
25
30
35
40
rps

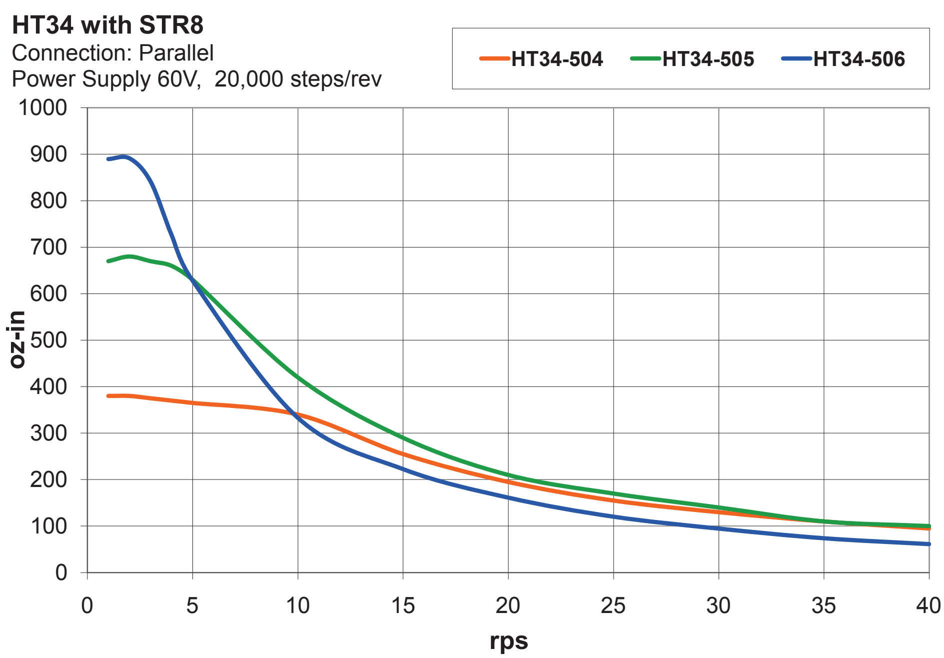
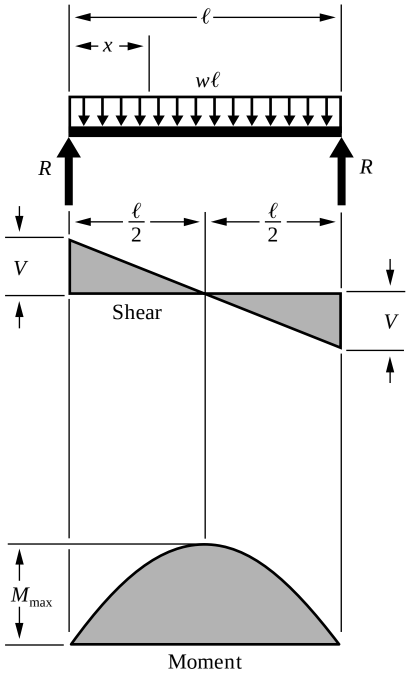

HT34 with STR8
Connection: Parallel
Power Supply 60V, 20,000 steps/rev
HT34-504
HT34-505
HT34-506
1000
900
800
700
600
500
400
300
200
100
0
oz-in
0
5
10
15
20
25
30
35
40
rps

Figure 1 Simple Beam – Uniformly Distributed Load



$$R = V \;\ldots\ldots\ldots\ldots\ldots = \frac{w\ell}{2}$$

$$V_x \;\ldots\ldots\ldots\ldots\ldots = w\left(\frac{\ell}{2} - x\right)$$

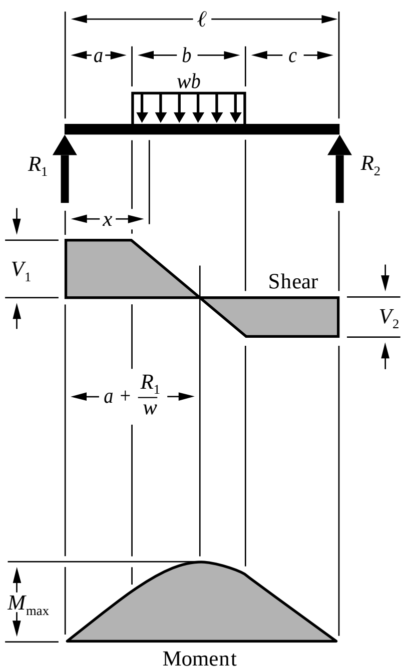
$$M_{max}\ (\text{at center}) \;\ldots\ldots\ldots = \frac{w\ell^2}{8}$$

$$M_x \;\ldots\ldots\ldots\ldots\ldots = \frac{wx}{2}(\ell - x)$$

$$\Delta_{max}\ (\text{at center}) \;\ldots\ldots\ldots = \frac{5w\ell^4}{384\,EI}$$

$$\Delta_x \;\ldots\ldots\ldots\ldots\ldots = \frac{wx}{24\,EI}(\ell^3 - 2\ell x^2 + x^3)$$

Figure 2 Simple Beam – Uniform Load Partially Distributed



$$R_1 = V_1\ (\text{max when } a < c) \;\ldots = \frac{wb}{2\ell}(2c + b)$$

$$R_2 = V_2\ (\text{max when } a > c) \;\ldots = \frac{wb}{2\ell}(2a + b)$$

$$V_x\ \left(\text{when } x > a \text{ and } < (a+b)\right) \;\ldots = R_1 - w(x - a)$$

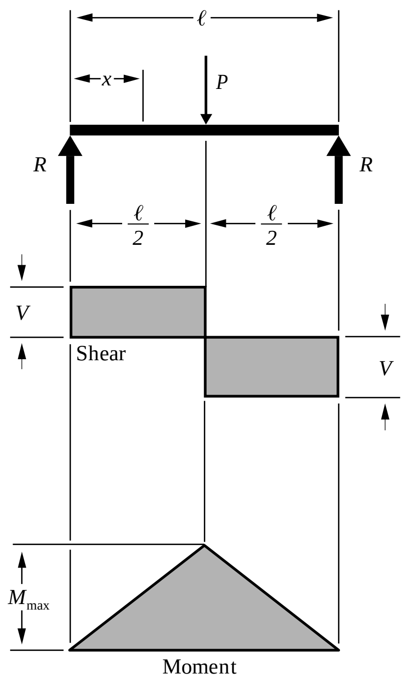
$$M_{max}\ \left(\text{at } x = a + \frac{R_1}{w}\right) \;\ldots\ldots = R_1\left(a + \frac{R_1}{2w}\right)$$

$$M_x\ (\text{when } x < a) \;\ldots\ldots\ldots = R_1 x$$

$$M_x\ \left(\text{when } x > a \text{ and } < (a+b)\right) \;\ldots = R_1 x - \frac{w}{2}(x - a)^2$$

$$M_x\ \left(\text{when } x > (a+b)\right) \;\ldots\ldots = R_2(\ell - x)$$

Figure 7 Simple Beam – Concentrated Load at Center



$$R = V \quad \ldots\ldots\ldots\ldots \quad = \frac{P}{2}$$

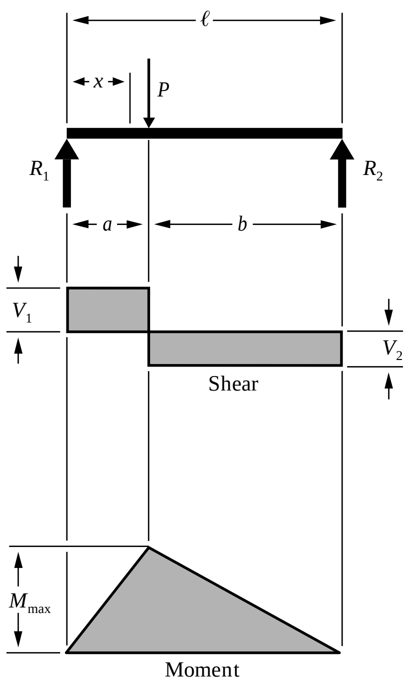
$$M_{max} \text{ (at point of load)} \quad \ldots\ldots \quad = \frac{P\ell}{4}$$

$$M_x \left(\text{when } x < \frac{\ell}{2}\right) \quad \ldots\ldots \quad = \frac{Px}{2}$$

$$\Delta_{max} \text{ (at point of load)} \quad \ldots\ldots \quad = \frac{P\ell^3}{48EI}$$

$$\Delta_x \left(\text{when } x < \frac{\ell}{2}\right) \quad \ldots\ldots \quad = \frac{Px}{48EI}(3\ell^2 - 4x^2)$$

Figure 8 Simple Beam – Concentrated Load at Any Point



$$R_1 = V_1 \text{ (max when } a < b) \quad \ldots\ldots \quad = \frac{Pb}{\ell}$$

$$R_2 = V_2 \text{ (max when } a > b) \quad \ldots\ldots \quad = \frac{Pa}{\ell}$$

$$M_{max} \text{ (at point of load)} \quad \ldots\ldots \quad = \frac{Pab}{\ell}$$

$$M_x \text{ (when } x < a) \quad \ldots\ldots \quad = \frac{Pbx}{\ell}$$

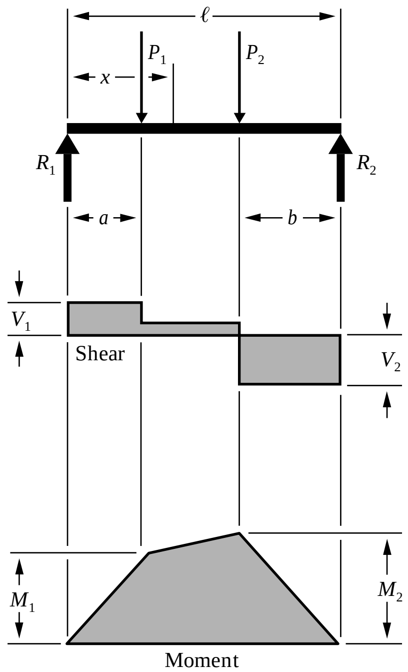
$$\Delta_{max} \left(\text{at } x = \sqrt{\frac{a(a+2b)}{3}} \text{ when } a > b\right) \quad . \quad = \frac{Pab(a+2b)\sqrt{3a(a+2b)}}{27EI\ell}$$

$$\Delta_a \text{ (at point of load)} \quad \ldots\ldots \quad = \frac{Pa^2b^2}{3EI\ell}$$

$$\Delta_x \text{ (when } x < a) \quad \ldots\ldots \quad = \frac{Pbx}{6EI\ell}(\ell^2 - b^2 - x^2)$$

$$\Delta_x \text{ (when } x > a) \quad \ldots\ldots \quad = \frac{Pa(\ell - x)}{6EI\ell}(2\ell x - x^2 - a^2)$$

Figure 11 Simple Beam – Two Unequal Concentrated Loads Unsymmetrically Placed



$$R_1 = V_1 \quad \ldots\ldots\ldots\ldots \quad = \frac{P_1(\ell - a) + P_2 b}{\ell}$$

$$R_2 = V_2 \quad \ldots\ldots\ldots\ldots \quad = \frac{P_1 a + P_2(\ell - b)}{\ell}$$

$$V_x \left(\text{when } x > a \text{ and } < (\ell - b)\right) \ldots = R_1 - P_1$$

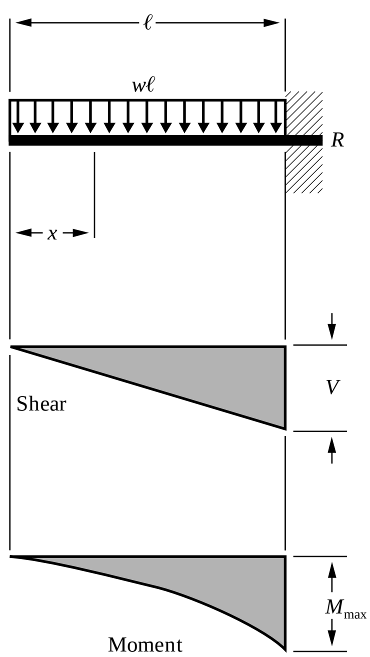
$$M_1 \ (\text{max when } R_1 < P_1) \ldots\ldots = R_1 a$$

$$M_2 \ (\text{max when } R_2 < P_2) \ldots\ldots = R_2 b$$

$$M_x \ (\text{when } x < a) \ldots\ldots\ldots = R_1 x$$

$$M_x \left(\text{when } x > a \text{ and } < (\ell - b)\right) \ldots = R_1 x - P_1(x - a)$$

Figure 12 Cantilever Beam – Uniformly Distributed Load



$$R = V \quad \ldots\ldots\ldots\ldots\ldots \quad = w\ell$$

$$V_x \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad = wx$$

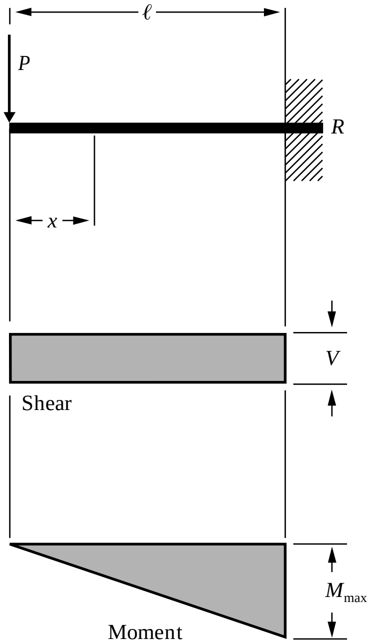
$$M_{max} \ (\text{at fixed end}) \ldots\ldots\ldots = \frac{w\ell^2}{2}$$

$$M_x \quad \ldots\ldots\ldots\ldots\ldots \quad = \frac{wx^2}{2}$$

$$\Delta_{max} \ (\text{at free end}) \ldots\ldots\ldots = \frac{w\ell^4}{8EI}$$

$$\Delta_x \quad \ldots\ldots\ldots\ldots\ldots \quad = \frac{w}{24EI}(x^4 - 4\ell^3 x + 3\ell^4)$$

Figure 13 Cantilever Beam – Concentrated Load at Free End



$R = V$ . . . . $= P$

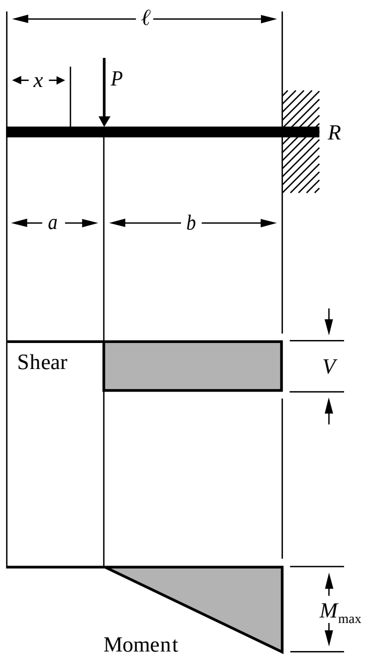
M_{max} (at fixed end) . . . . $= P\ell$

M_x . . . . $= Px$

Δ_{max} (at free end) . . . . $= \dfrac{P\ell^3}{3EI}$

Δ_x . . . . $= \dfrac{P}{6EI}(2\ell^3 - 3\ell^2 x + x^3)$

Figure 14 Cantilever Beam – Concentrated Load at Any Point



$R = V$ . . . . $= P$

M_{max} (at fixed end) . . . . $= Pb$

M_x (when $x > a$) . . . . $= P(x - a)$

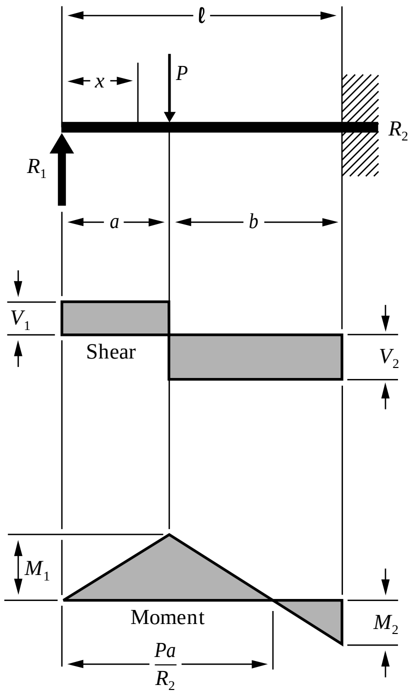
Δ_{max} (at free end) . . . . $= \dfrac{Pb^2}{6EI}(3\ell - b)$

Δ_a (at point of load) . . . . $= \dfrac{Pb^3}{3EI}$

Δ_x (when $x < a$) . . . . $= \dfrac{Pb^2}{6EI}(3\ell - 3x - b)$

Δ_x (when $x > a$) . . . . $= \dfrac{P(\ell - x)^2}{6EI}(3b - \ell + x)$

Figure 17 Beam Fixed at One End, Supported at Other – Concentrated Load at Any Point



$$R_1 = V_1 \dots = \frac{Pb^2}{2\ell^3}(a + 2\ell)$$

$$R_2 = V_2 \dots = \frac{Pa}{2\ell^3}(3\ell^2 - a^2)$$

$$M_1 \text{ (at point of load)} \dots = R_1 a$$

$$M_2 \text{ (at fixed end)} \dots = \frac{Pab}{2\ell^2}(a + \ell)$$

$$M_x \text{ (when } x < a) \dots = R_1 x$$

$$M_x \text{ (when } x > a) \dots = R_1 x - P(x - a)$$

$$\Delta_{max}\left(\text{when } a < .414\ell \text{ at } x = \ell\frac{\ell^2 + a^2}{3\ell^2 - a^2}\right) = \frac{Pa}{3EI}\frac{(\ell^2 - a^2)^3}{(3\ell^2 - a^2)^2}$$

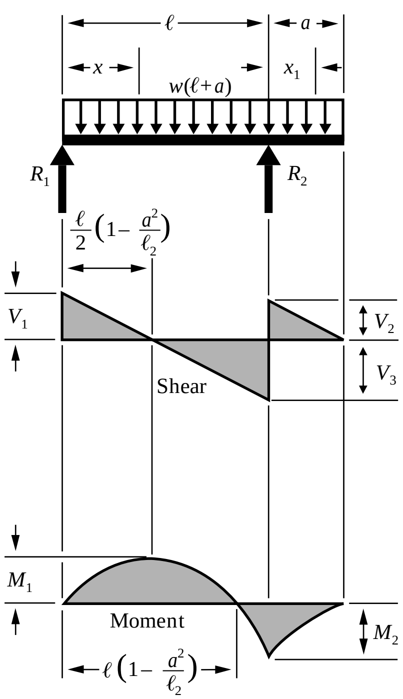
$$\Delta_{max}\left(\text{when } a > .414\ell \text{ at } x = \ell\sqrt{\frac{a}{2\ell + a}}\right) = \frac{Pab^2}{6EI}\sqrt{\frac{a}{2\ell + a}}$$

$$\Delta_a \text{ (at point of load)} \dots = \frac{Pa^2b^3}{12EI\ell^3}(3\ell + a)$$

$$\Delta_x \text{ (when } x < \text{a)} \dots = \frac{Pb^2x}{12EI\ell^3}(3a\ell^2 - 2\ell x^2 - ax^2)$$

$$\Delta_x \text{ (when } x > \text{a)} \dots = \frac{Pa}{12EI\ell^3}(\ell - x)^2(3\ell^2 x - a^2 x - 2a^2\ell)$$

Figure 18 Beam Overhanging One Support – Uniformly Distributed Load



$$R_1 = V_1 \dots = \frac{w}{2\ell}(\ell^2 - a^2)$$

$$R_2 = V_2 + V_3 \dots = \frac{w}{2\ell}(\ell + a)^2$$

$$V_2 \dots = wa$$

$$V_3 \dots = \frac{w}{2\ell}(\ell^2 + a^2)$$

$$V_x \text{ (between supports)} \dots = R_1 - wx$$

$$V_{x_1} \text{ (for overhang)} \dots = w(a - x_1)$$

$$M_1\left(\text{at } x = \frac{\ell}{2}\left[1 - \frac{a^2}{\ell^2}\right]\right) \dots = \frac{w}{8\ell^2}(\ell + a)^2(\ell - a)^2$$

$$M_2 \text{ (at } R_2) \dots = \frac{wa^2}{2}$$

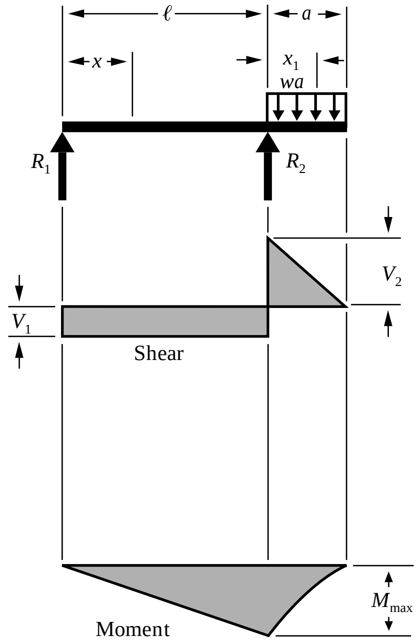
$$M_x \text{ (between supports)} \dots = \frac{wx}{2\ell}(\ell^2 - a^2 - x\ell)$$

$$M_{x_1} \text{ (for overhang)} \dots = \frac{w}{2}(a - x_1)^2$$

$$\Delta_x \text{ (between supports)} \dots = \frac{wx}{24EI\ell}(\ell^4 - 2\ell^2x^2 + \ell x^3 - 2a^2\ell^2 + 2a^2x^2)$$

$$\Delta_{x_1} \text{ (for overhang)} \dots = \frac{wx_1}{24EI}(4a^2\ell - \ell^3 + 6a^2x_1 - 4ax_1^2 + x_1^3)$$

Figure 19 Beam Overhanging One Support – Uniformly Distributed Load on Overhang



$$R_1 = V_1 \ldots\ldots = \frac{wa^2}{2\ell}$$

$$R_2 = V_1 + V_2 \ldots\ldots = \frac{wa}{2\ell}(2\ell + a)$$

$$V_2 \ldots\ldots = wa$$

$$V_{x_1} \text{ (for overhang)} \ldots\ldots = w(a - x_1)$$

$$M_{max} \text{ (at } R_2\text{)} \ldots\ldots = \frac{wa^2}{2}$$

$$M_x \text{ (between supports)} \ldots\ldots = \frac{wa^2x}{2\ell}$$

$$M_{x_1} \text{ (for overhang)} \ldots\ldots = \frac{w}{2}(a - x_1)^2$$

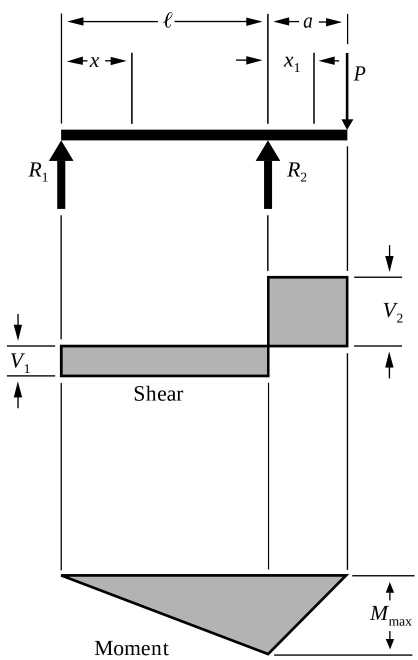
$$\Delta_{max} \left(\text{between supports at } x = \frac{\ell}{\sqrt{3}}\right) = \frac{wa^2\ell^2}{18\sqrt{3}EI} = .03208\frac{wa^2\ell^2}{EI}$$

$$\Delta_{max} \text{ (for overhang at } x_1 = a\text{)} \ldots = \frac{wa^3}{24EI}(4\ell + 3a)$$

$$\Delta_x \text{ (between supports)} \ldots\ldots = \frac{wa^2x}{12EI\ell}(\ell^2 - x^2)$$

$$\Delta_{x_1} \text{ (for overhang)} \ldots\ldots = \frac{wx_1}{24EI}(4a^2\ell + 6a^2x_1 - 4ax_1^{\,2} + x_1^{\,3})$$

Figure 20 Beam Overhanging One Support – Concentrated Load at End of Overhang



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

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

$$V_2 \ldots\ldots = P$$

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

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

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

$$\Delta_{max} \left(\text{between supports at } x = \frac{\ell}{\sqrt{3}}\right) = \frac{Pa\ell^2}{9\sqrt{3}EI} = .06415\frac{Pa\ell^2}{EI}$$

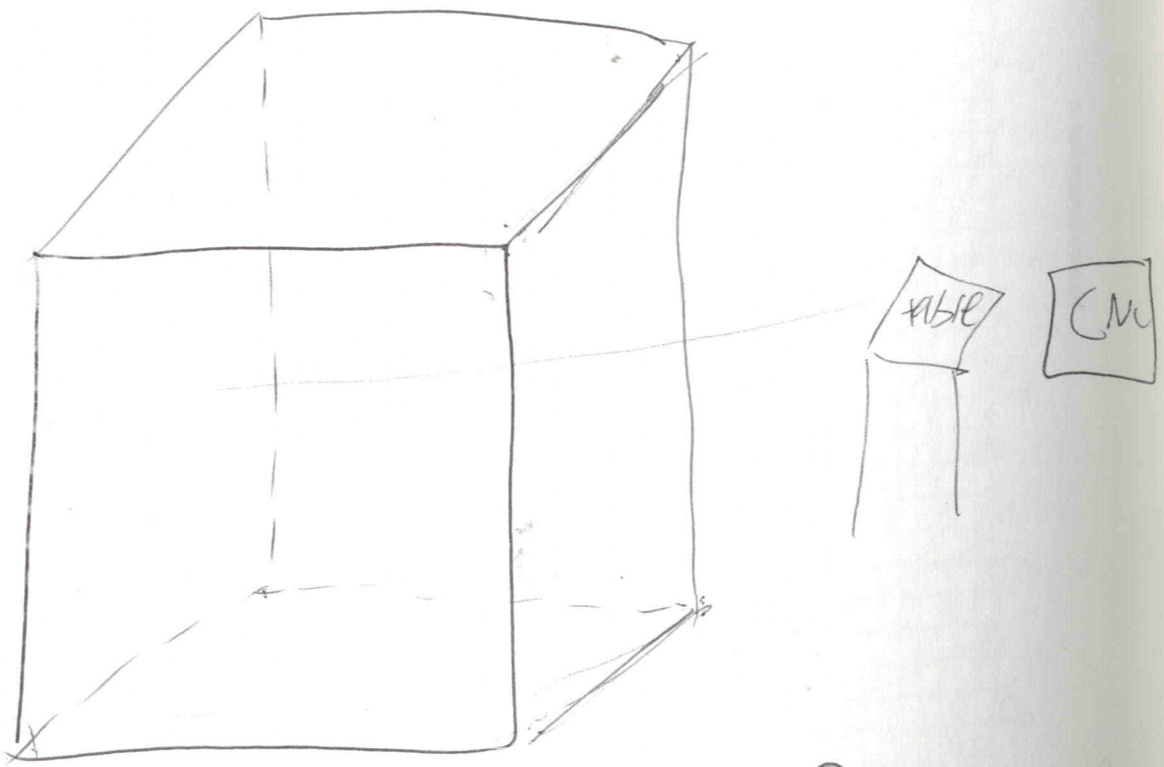
$$\Delta_{max} \text{ (for overhang at } x_1 = a\text{)} \ldots = \frac{Pa^2}{3EI}(\ell + a)$$

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

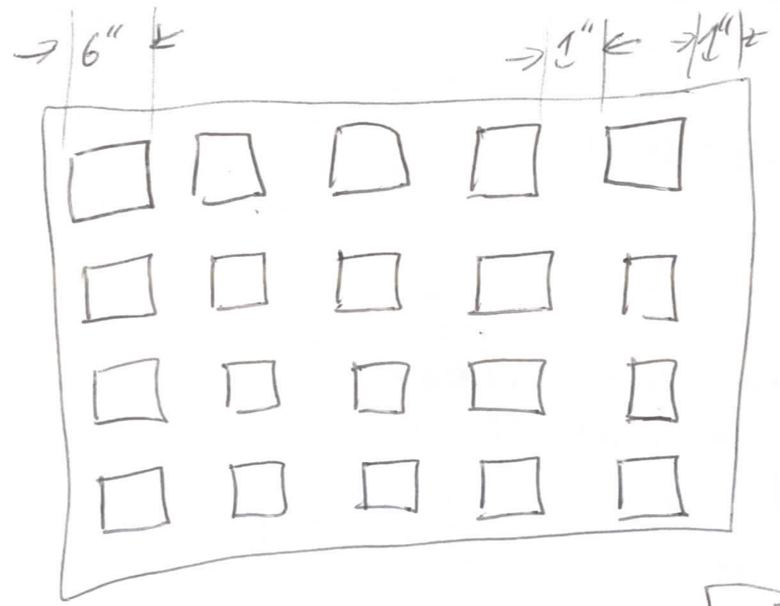
$$\Delta_{x_1} \text{ (for overhang)} \ldots\ldots = \frac{Px_1}{6EI}(2a\ell + 3ax_1 - x_1^{\,2})$$

parts Left to spec

- All Fasteners
- How to mount the rails
- Lead screw mounts
- Belt system
- Telescoping Motor
- Telescoping Motor Mount



- Pressure coming out of VF-3?
- Can we ~~have~~ send more than 1 signal?
- Does the vise open automatically when the ~~door or~~ operation is done?



$6'' \times 5 + 1'' \times 4 + 1'' \times 2 = \boxed{36''}$

$4'' \times 4 + 1'' \times 3 + 1'' \times 2 = \boxed{24''}$

New Dimensions

Table

Find safety factor via mod - Goodman

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

The equations for σ_a, σ_m, and S_e are as follows

$$\sigma_a = \left\{\left[(K_f)_{bend}(\sigma_a)_{bend} + (K_f)_{axial}\frac{(\sigma_a)_{axial}}{0.85}\right]^2 + 3\left[(\tau_a)_{torsion}^2 + (\tau_a)_{shear}^2\right]\right\}$$

$$\sigma_m = \left\{\left[(K_f)_{bend}(\sigma_a)_{bend} + (K_f)_{axial}(\sigma_a)_{axial}\right]^2 + 3\left[(\tau_a)_{torsion}^2 + (\tau_a)_{shear}^2\right]\right\}$$

$$S_e = k_a k_b k_c k_d k_e k_e S_e'$$

Finding S_e' for $S_{ut} < 200$ ksi gets

$$S_e' = 0.5 \cdot S_{ut}$$

Stress eqns are

$$\sigma_{bend} = \frac{My}{I}$$

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

$$\tau_{shear} = \frac{3V}{2A} \quad \text{(rectangular cross section)}$$

$$\tau_{torsion} = \frac{T}{bc^2}\cdot\left(3 + \frac{1.8}{b/c}\right) \quad \text{(c is longer side)}$$

Endurance limit modifying factors are

$$k_a = a\,S_{ut}^{\,b}$$

$$d_e = 0.808(h\cdot b)^{1/2}$$

$$k_b = 0.879\cdot d_e^{-0.107}$$

$k_c = 1$ (because there is bending)

$k_d = 1$ (because normal operating temperatures)

$k_e = 0.702$ (because 99.99% reliability)

$k_f = 1$ (for no other significant miscelaneous issues)

Pneumatic Gripper			Gripper Finger					
Part/Material Selection			Part/Material Selection			Given Info		
Model	HFZ40	N/A	Material (Steel)	1018 CD	N/A	Force Applied	44.12	lbf
Grip Material	Rubber	N/A	Thickness	0.7	in	Force Reacting	10.00	lbf
Grip Point	100	mm	Width	2	in	Finger 'Height'	3.94	in
Pressure	0.5	MPa	Surface Finish	Ground	N/A	Cross Section	1.40	in^2
Given Info			Tool Temp	70	°F	Moment Applied	86.85	lbf*in
Part Weight	10.00	lbf	Reliability	99.99	N/A	Torque Applied	44.12	lbf*in
Friction Factor	0.68	N/A	Notch Radius	0.25	in	Inertia Moment	10.17	in^4
Safety Factor	6.00	N/A	Modifying Factors			S_ut	64.00	ksi
Grip Pressure	72.52	psi	a	1.340	ksi	S_e'	32.00	ksi
Results			b	0.085	N/A	r/d	0.357	N/A
Needed Force	44.12	lbf	k_a	0.941	N/A	Top Thickness	3.44	in
Provided Force	56.20	lbf	d_e	0.956	in	D/d	4.914	N/A
			k_b	0.883	N/A	Various Stresses		
			k_c	1.000	N/A	N.S. (Axial)	7.14	psi
			k_d	1.000	N/A	N.S. (Moment)	-16.81	psi
			k_e	0.702	N/A	S.S. (Torsional)	163.42	psi
			k_f	1.000	N/A	S.S. (Beam)	47.27	psi
			Stress Concentration Factors			Alt. Stress	296.69	psi
			q	0.80	N/A	Mid. Stress	296.45	psi
			K_t (axial)	2.30	N/A	S_e	18.67	ksi
			K_f (axial)	2.04	N/A	Results		
			K_t (moment)	1.45	N/A	SF (mod-Good)	48.72	N/A
			K_f (moment)	1.36	N/A			

Shaft Analysis (Telescoping Arm)

Constants

Dimensions

$L = 72$ [in] *Length of Beam*

$d_o = 2$ [in] *Outer Diameter*

$d_i = 0$ [in] *Inner Diameter*

$A_s = \frac{\pi}{4} \cdot \left[d_o^2 - d_i^2 \right]$ *Cross Sectional Area of Beam*

$I = \frac{\pi}{64} \cdot \left[d_o^4 - d_i^4 \right]$ *Moment of Inertia*

$y = \frac{d_o}{2}$ *Perpendicular Distance to the Neutral Axis*

$\bar{y} = \frac{d_o^3 - d_i^3}{3 \cdot \left[d_o^2 - d_i^2 \right]}$ *Centroid of top half, used for Q calculation*

Material Properties

$E = 29 \cdot 10^6$ *Elastic Modulus*

$\rho = 0.283$ [lbf/in^3] *Density of beam material*

$\sigma_{yield} = 30000$ [psi] *Yeild Strength*

$s_{ut} = 60000$ [psi]

$mass = A_s \cdot \rho \cdot L$ *Total Mass of Material*

Applied Forces

$P = 40$ [lbf] *Applied Point Load at End*

$\omega = \rho \cdot A_s$ *Distributed Load from Beams Mass*

Stress Concentration Constants

$K_{fb} = 1$ *Bendisng Stress Concentration Factor*

$K_{fs} = 1$ *Tortional Stress Concentration Factor*

Reaction Forces

Back Bearing (Ay)

$R_y = \omega \cdot L + P$ *Reaction Force at Support*

Deflection Analysis

Deflection at End

$$dx_p = \frac{P \cdot L^3}{3 \cdot E \cdot I}$$ *Deflection from Point Load*

$$dx_d = \frac{\omega \cdot L^4}{8 \cdot E \cdot I}$$ *Deflection from Distributed Load*

$$dx_{max} = dx_p + dx_d$$ *Total Maximum Deflection at Free End*

Stresses

$$M_{max} = P \cdot L + 0.5 \cdot \omega \cdot L^2$$

Bending Stresses

$$\sigma_x = M_{max} \cdot \frac{y}{I}$$ *Maximum Static Beam Bending Stress in the x-direction*

$$\sigma_y = 0$$

$$\sigma_z = 0$$

Shear Stress

$$\tau_{xy} = \frac{P \cdot A_s \cdot \bar{y}}{I \cdot d_o}$$ *Maximum Transverse Shear Stress in the xy plane*

$$\tau_{yz} = 0$$

$$\tau_{zx} = 0$$

Von Mises

$$\sigma_{max} = \frac{1}{2^{0.5}} \cdot \left[\left(\sigma_x - \sigma_y \right)^2 + \left(\sigma_y - \sigma_z \right)^2 + \left(\sigma_z - \sigma_x \right)^2 + 6 \cdot \left(\tau_{xy}{}^2 + \tau_{yz}{}^2 + \tau_{zx}{}^2 \right) \right]^{0.5}$$ *Von Mises Stress*

$$\sigma_{min} = \tau_{xy}$$ *Assume no bending stress at minimum stress*

Failure Theories

Endurance Limit

$$S_{e'} = 0.5 \cdot s_{ut}$$

Surface Factor Ka

$a = 1.34$ [kpsi] *Ground Surface Finish*

$b = -0.085$

$k_a = a \cdot \left[\frac{s_{ut}}{1000} \right]^b$ *Surface Factor*

Size Factor Kb

$d_e = 0.37 \cdot d_o$ *Equivalent Diameter of a nonrotating hollow round*

$k_b = 0.879 \cdot d_e^{-0.107}$ *Size Factor Assumeing 0.11<= d_e <= 2 in*

Loading Factor Kc

$K_c = 1$ *Loading is Bending*

Temperature Factor Kd

$K_d = 1$ *Assume Room Temperature Operation*

Reliability Facter Ke

$k_e = 0.814$ *Assume 99% Reliability*

Miscellaneous Effects Factor Kf

$k_f = 1$ *No Miscellaneous Effects*

$S_e = k_a \cdot k_b \cdot K_c \cdot K_d \cdot k_e \cdot k_f \cdot S_e'$ *Endurance Limit*

Loads and Failure Criteria

$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}$ *Alternating Stress*

$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2}$ *Midrange Stress*

$\frac{\sigma_a}{S_e} + \frac{\sigma_m}{s_{ut}} = \frac{1}{n_{modGoodman}}$ *Modified Goodman Fatigue Failure Theory*

$\sigma_a + \sigma_m = \frac{\sigma_{yield}}{n_{yield}}$ *Yield Factor of Safety Analysis*

SOLUTION

Unit Settings: SI C kPa kJ mass deg

$a = 1.34$ [kpsi]
$b = -0.085$
$\boxed{dx_{max} = 0.3496 \text{ [in]}}$
$d_e = 0.74$
$d_o = 2$ [in]
$I = 0.7854$ [in^4]
$k_b = 0.9078$ [-]
$K_d = 1$ [-]
$k_f = 1$ [-]
$K_{fs} = 1$ [-]

$A_s = 3.142$ [in^2]
$dx_d = 0.1311$ [in]
$dx_p = 0.2185$ [in]
$d_i = 0$ [in]
$E = 2.900E+07$ [lbf/in^2]
$k_a = 0.9462$ [-]
$K_c = 1$ [-]
$k_e = 0.814$ [-]
$K_{fb} = 1$ [-]
$L = 72$ [in]

$mass$ = 64.01
$n_{modGoodman}$ = 4.727 [-]
ω = 0.8891 [lbf/in]
ρ = 0.283 [lbf/in^3]
σ_a = 3274 [psi]
σ_{max} = 6602 [psi]
σ_x = 6601 [psi]
σ_{yield} = 30000 [psi]
S_e = 20974 [psi]
s_{ut} = 60000 [psi]
τ_{yz} = 0 [psi]
y = 1 [in]
M_{max} = 5184 [lbf*in]
n_{yield} = 4.544 [-]
P = 40 [lbf]
R_y = 104 [lbf]
σ_m = 3328 [psi]
σ_{min} = 53.33 [psi]
σ_y = 0 [psi]
σ_z = 0 [psi]
$\overline{S}_{e'}$ = 30000 [psi]
τ_{xy} = 53.33 [psi]
τ_{zx} = 0 [psi]
$\overline{y}$ = 0.6667 [in]

4 potential unit problems were detected.

Parametric Table: Table 1

	d_o [in]	dx_{max} [in]	$n_{modGoodman}$ [-]	n_{yield} [-]
Run 1	0.75	11.98	0.4347	0.3878
Run 2	0.875	6.649	0.6583	0.5941
Run 3	0.9375	5.122	0.79	0.7166
Run 4	1	4.02	0.9348	0.8521
Run 5	1.125	2.597	1.263	1.162
Run 6	1.25	1.768	1.641	1.522
Run 7	1.375	1.255	2.065	1.929
Run 8	1.438	1.071	2.295	2.151
Run 9	1.5	0.9237	2.531	2.38
Run 10	1.625	0.7	3.035	2.871
Run 11	1.75	0.544	3.572	3.398
Run 12	2	0.3496	4.727	4.544
Run 13	2.5	0.1734	7.256	7.1
Run 14	3	0.1014	9.929	9.859

Shaft Analysis (Base Shaft)

Constants

Dimensions

$x1 = 19.59$ [in] *Distance from support to point analyzed*

$a1 = 19.59$ [in] *Distance between middle support and free end*

$d_o = 3.5$ [in] *Outer Diameter*

$d_i = 3$ [in] *Inner Diameter*

$A_s = \frac{\pi}{4} \cdot \left[d_o^2 - d_i^2 \right]$ *Cross Sectional Area of Beam*

$I = \frac{\pi}{64} \cdot \left[d_o^4 - d_i^4 \right]$ *Moment of Inertia*

$L = 44.41$ [in] *Distance Between Supports*

$y = \frac{d_o}{2}$ *Perpendicular Distance to the Neutral Axis*

$\bar{y} = \frac{d_o^3 - d_i^3}{3 \cdot \left[d_o^2 - d_i^2 \right]}$ *Centroid of top half, used for Q calculation*

Material Properties (Schedule 40 Steel Pipe: ASME SA53)

$E = 29 \cdot 10^6$ *Elastic Modulus*

$\rho = 0.283$ [lbf/in^3] *Density of beam material*

$\sigma_{yield} = 70000$ [psi] *Yeild Strength*

$S_{ut} = 80000$ [psi] *Ultimate Strength of Steel*

$mass = A_s \cdot \rho \cdot \left[L + a1 \right]$ *Total Mass of Material*

Applied Forces

$P = 118.1$ [lbf] *Applied Point Load at End*

$\omega = \rho \cdot A_s$ *Distributed Load from Beams Mass*

Stress Concentration Constants

$K_{fb} = 1$ [-] *Bendisng Stress Concentration Factor*

$K_{fs} = 1$ [-] *Tortional Stress Concentration Factor*

Reaction Forces

Back Bearing (Ay)

$A_y = \omega \cdot [L + a] + P - B_y$ *Back Bearing Reaction Force*

Front Bearing (By)

$B_y = \frac{\omega}{2 \cdot L} \cdot [L + a]^2 + \frac{P}{L} \cdot [L + a]$ *Front Bearing Reaction Force*

Deflection Analysis

Deflection at End

$dx_p = \frac{P \cdot x1}{6 \cdot E \cdot I} \cdot [2 \cdot a \cdot L + 3 \cdot a \cdot x1 + x1^2]$ *Deflection from Point Load*

$dx_d = \frac{\omega \cdot x1}{24 \cdot E \cdot I} \cdot [4 \cdot a^2 \cdot L - L^3 + 6 \cdot a^2 \cdot x1 - 4 \cdot a \cdot x1^2 + x1^3]$ *Deflection from Distributed Load*

$dx_{max} = dx_p + dx_d$ *Total Maximum Deflection at Free End*

Stresses

$M_{max} = x1 \cdot [P + 0.5 \cdot \omega \cdot x1]$

Bending Stresses

$\sigma_x = M_{max} \cdot \frac{y}{I}$ *Maximum Static Beam Bending Stress in the x-direction*

$\sigma_y = 0$

$\sigma_z = 0$

Shear Stress

$\tau_{xy} = \frac{P \cdot A_s \cdot \bar{y}}{I \cdot d_o}$ *Maximum Transverse Shear Stress in the xy plane*

$\tau_{yz} = 0$

$\tau_{zx} = 0$

Von Mises

$\sigma_{max} = \frac{1}{2^{0.5}} \cdot [(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6 \cdot (\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2)]^{0.5}$ *Von Mises Stress*

$\sigma_{min} = \tau_{xy}$ *Assume no bending stress at minimum stress*

Failure Theories

Endurance Limit

$S_e' = 0.5 \cdot S_{ut}$

Surface Factor Ka

$a = 1.34$ [kpsi] *Ground Surface Finish*

$b = -0.085$

$k_a = a \cdot \left[\frac{S_{ut}}{1000}\right]^b$ *Surface Factor*

Size Factor Kb

$d_e = 0.37 \cdot d_o$ *Equivalent Diameter of a nonrotating hollow round*

$k_b = 0.879 \cdot d_e^{-0.107}$ *Size Factor Assumeing 0.11<= d_e <= 2 in*

Loading Factor Kc

$K_c = 1$ *Loading is Bending*

Temperature Factor Kd

$K_d = 1$ *Assume Room Temperature Operation*

Reliability Facter Ke

$k_e = 0.814$ *Assume 99% Reliability*

Miscellaneous Effects Factor Kf

$k_f = 1$ *No Miscellaneous Effects*

$S_e = k_a \cdot k_b \cdot K_c \cdot K_d \cdot k_e \cdot k_f \cdot S_{e'}$ *Endurance Limit*

Loads and Failure Criteria

$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}$ *Alternating Stress*

$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2}$ *Midrange Stress*

$\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{ut}} = \frac{1}{n_{modGoodman}}$ *Modified Goodman Fatigue Failure Theory*

$\sigma_a + \sigma_m = \frac{\sigma_{yield}}{n_{yield}}$

Parametric Table: Table 4

	d_o [in]	d_i [in]	dx_{max} [in]	$n_{modGoodman}$ [-]	n_{yield} [-]
Run 1	2.25	2.01	0.01578	7.001	12.05
Run 2	2.25	1.75	0.008414	12.08	20.66
Run 3	2.5	2.26	0.01122	8.693	15.09
Run 4	2.5	2	0.005816	15.24	26.27
Run 5	2.5	1.75	0.00419	19.38	33.23
Run 6	2.5	1.5	0.00341	21.94	37.49
Run 7	3	2.624	0.004035	18.23	31.98
Run 8	3	2.5	0.003074	22.57	39.46
Run 9	3	2.25	0.002105	29.33	50.97
Run 10	3	2	0.001623	33.92	58.69
Run 11	3.5	3.124	0.002401	25.03	44.44
Run 12	3.5	3	0.001791	31.21	55.21
Run 13	4	3.5	0.001119	41.09	73.46
Run 14	5	4.5	0.0005031	64.42	117.2
Run 15	6	5.5	0.0002565	92.19	170.2
Run 16	8	7.5	0.00008255	159.9	302.3
Run 17	10	9.5	0.00003026	242.1	466.3
Run 18	12	11.5	0.00001088	337.2	659.4
Run 19	12	11.25	-0.000005048	453.5	881.3

Threaded Faster Analysis

Constants

$d = 0.138$ [in] *Nominal Diameter of Screw*

$L_t = 0.75$ [in] *Threaded Length*

$t = 0$ [in] *Washer Thickness (Shigley's A-32)*

$H = 0$ *Nut Thickness (Shigley's A-31)*

$L = 0.75$ [in] *Fastener Length*

$w = 0.42$ [in] *Width of Material Being Fastened*

$l_1 = w + \frac{d}{2}$ *Grip Length*

$l_d = L - L_t$ *Length of Unthreaded Portion in Grip*

$l_{t,1} = l_1 - l_d$ *Length of Threaded Portion in Grip*

$A_d = \pi \cdot \frac{d^2}{4}$ *Area of Unthreaded Portion*

$A_t = 0.00909$ [in^2] *Area of Threaded Portion (Shigley's Table 8-1,8-2)*

$E_1 = 29.7 \cdot 10^6$ *Material Elastic Modulus (Steel)*

Material Elastic Modulus (Steel)

$\alpha = 30$ [Degrees] *Half-Apex Angle (Always use 30 degrees)*

Stiffness

$t_1 = 0.42$ [in] *Thickness of 1st Material*

$t_2 = 0.07$ [in] *Thickness of 2nd Material*

Frustum Diameter at Material Break

$k_b = \frac{A_d \cdot A_t \cdot E_1}{A_d \cdot l_{t,1} + A_t \cdot l_d}$ *Fastener Stiffness*

$\frac{k_m}{E_1 \cdot d} = 0.78715 \cdot \mathbf{exp}\left[0.62873 \cdot \frac{d}{l_1}\right]$ *This equation only works if the entire joint is made of the same material.*

Tension Joints

$F_i = 0.75 \cdot S_p \cdot A_t$ *Preload Force*

$P_{total} = 14.1$ [lbf] *Total External Tensile Force*

$N = 1$ [-] *Number of Screws* *Assume total load taken by single screw*

$P = \dfrac{P_{total}}{N}$ *External Tensile Load per Bolt*

$C = \dfrac{k_b}{k_b + k_m}$ *Stiffness Constant of the Joint*

$P_b = C \cdot P$ *Portion of Force Taken by Bolt*

$P_m = [1 - C] \cdot P$ *Portion of Force Taken by Members*

$F_b = P_b + F_i$ *Resultant Bolt Load*

$F_m = P_m - F_i$ *Resultant Load on Members*

Static Failure Analysis

$\sigma_b = \dfrac{F_b}{A_t}$ *Tensile Stress in the Bolt*

$S_p = 120000$ [psi] *Rated Proof Load of Bolt*

$n_p = \dfrac{S_p}{\sigma_b}$ *Yielding Factor of Safety*

$n_l = \dfrac{S_p \cdot A_t - F_i}{C \cdot P}$ *Load Factor*

$n_o = \dfrac{F_i}{P \cdot [1 - C]}$ *Load Factor Guarding Against Joint Separation*

Shear Failure Analysis

$F = 55$ [lbf] *Shearing Force*

$I = \dfrac{\pi}{64} \cdot d^4$ *Moment of Inertia*

$A_s = \pi \cdot \left[\dfrac{d}{2}\right]^2$ *Cross Sectional Area of Bolt*

$\sigma_{ba} = F \cdot d \cdot \dfrac{w}{4 \cdot I}$

Failure by Bending of the Bolt (Typically compensated for with increased factor of safety

$\tau = \dfrac{F}{N \cdot A_s}$ *Failure by Pure Shear*

$\sigma' = [\sigma_b^2 + 3 \cdot \tau^2]^{0.5}$ *Von Mises*

$\sigma' = \dfrac{120000}{n_{total}}$

$\sigma_{plate} = \dfrac{F}{2.5^2 - 4 \cdot A_s}$ *Rupture of Connected Members*

$$\sigma_{crushing} = \frac{-F}{t_1 \cdot d}$$ *Failure by Crushing of the Bolt of Plate*

SOLUTION

Unit Settings: SI C kPa kJ mass deg

α = 30 [Degrees]	A_d = 0.01496	A_s = 0.01496
A_t = 0.00909 [in^2]	C = 0.1253	d = 0.138 [in]
E_1 = 2.970E+07	F = 55 [lbf]	F_b = 819.9
F_i = 818.1	F_m = -805.8	H = 0
I = 0.0000178	k_b = 552092	k_m = 3.853E+06
L = 0.75 [in]	l_1 = 0.489	l_d = 0
L_t = 0.75 [in]	$l_{t,1}$ = 0.489	N = 1 [-]
$\boxed{n_l = 154.3}$	$\boxed{n_o = 66.34}$	$\boxed{n_p = 1.33}$
$\boxed{n_{total} = 1.327}$	P = 14.1	P_b = 1.767
P_m = 12.33	P_{total} = 14.1 [lbf]	σ_b = 90194
σ_{ba} = 44766	$\sigma_{crushing}$ = -948.9	σ_{plate} = 8.885
σ' = 90419	S_p = 120000 [psi]	t = 0 [in]
$\boxed{\tau = 3677}$	t_1 = 0.42 [in]	t_2 = 0.07 [in]
w = 0.42 [in]		

16 potential unit problems were detected.

Assume the same $12.4 \frac{in}{s^2}$ acceleration as before

$$F_p - 51.7\ lbf = m\left(12.4 \frac{in}{s^2}\right)$$

$$w = mg$$

$$m = \frac{w}{g}$$

$$= \left(\frac{103.33\ lbf}{\left(32.174 \frac{ft}{s^2}\right)}\right)\left[\frac{32.174 \frac{lbm\ ft}{s^2}}{lbf}\right]$$

$$m = 103.33\ lbm$$

$$\therefore ma = (103.33\ lbm)\left(12.4 \frac{in}{s^2}\right)\left[\frac{ft}{12\ in}\right]\left[\frac{lbf}{32.174 \frac{lbm\ ft}{s^2}}\right]$$

$$= 3.318602742$$

$$ma = 3.32\ lbf$$

$$\therefore F_p = 3.32\ lbf + 51.7\ lbf$$

$$\boxed{F_p = 55.02\ lbf}$$

which is a lot better than the 100 lbf load we had before.

$$\sigma_P = W_t \cdot K_o \cdot K_v \cdot K_s \cdot \frac{pitch_d}{F} \cdot \frac{K_m \cdot K_B}{J}$$ *Pinion bending stress*

$$\sigma_{all,P} = \frac{S_{t,P}}{S_{F,P}} \cdot \frac{Y_{N,P}}{K_T \cdot K_R}$$ *Pinion allowable bending stress*

$$\sigma_{c,P} = C_p \cdot \left[W_t \cdot K_o \cdot K_v \cdot K_s \cdot \frac{K_m}{d_{p,pinion} \cdot F} \cdot \frac{C_f}{I} \right]^{0.5}$$ *Pinion contact stress*

$$\sigma_{c,all,P} = \frac{S_{c,P}}{S_{H,P}} \cdot \frac{Z_{N,P} \cdot 1}{K_T \cdot K_R}$$ *Pinion allowable contact stress. For the pinion, C_H = 1.*

$S_{t,P} = 77.3 \cdot H_{B,P} + 12800$ *Given by figure 14-2*

$S_{c,P} = 322 \cdot H_{B,P} + 29100$ *Given by figure 14-5*

$H_{B,P} = 150$ *Pinion Brinell hardness. This should be a fairly conservative assumption.*

$H_{B,G} = 150$ *Gear Brinell hardness. This should be a fairly conservative assumption.*

$\phi = 14.5$ [deg] *Pressure angle*

$$speed_{ratio} = \frac{Teeth_{gear}}{Teeth_{pinion}}$$

$Teeth_{pinion} = 24$ *Pinion Teeth*

$Teeth_{gear} = 1 \times 10^{165}$

$pitch_d = 12$ *Diametral pitch*

$length_{rack} = 62$ *Rack length in inches*

$$pitch_c = \frac{3.142}{pitch_d}$$ *Circular pitch*

$$d_{p,gear} = \frac{Teeth_{gear}}{pitch_d}$$ *Gear pitch diameter (infinite for a rack)*

$$d_{p,pinion} = \frac{Teeth_{pinion}}{pitch_d}$$ *Pinion pitch diameter*

$F = 0.75$ *Face width*

hp = ?

$W_t = 54.51$

$$hp \cdot 33000 = Torque_{pinion} \cdot n_{pinion} \cdot 3.142 \cdot \frac{2}{12}$$

$$n_{pinion} = \frac{12.4}{\pi \cdot d_{p,pinion}} \cdot 60$$

$$W_t = \frac{Torque_{pinion}}{\frac{d_{p,pinion}}{12}} \cdot \cos[\phi]$$ *Tangential transmitted load*

$$W_r = \frac{Torque_{pinion}}{\frac{d_{p,pinion}}{12}} \cdot \sin[\phi]$$ *Radial transmitted load*

Face Contact Ratio m_F

m_F = 0 {(Spur Gears)}

$J = 0.36$ *Bending strength geometry factor*

$$I = \frac{\cos[\phi] \cdot \sin[\phi]}{2 \cdot m_N} \cdot \left[\frac{m_G}{m_G + 1}\right]$$ *Surface strength geometry factor*

$m_N = 1$ *Load-sharing ratio*

$m_G = speed_{ratio}$ *Speed ratio*

$C_p = 2300$ *From table 14-8 for steel-on-steel*

$K_T = 1$ *Standard work environment will not involve extreme temperatures*

$$K_v = \left[\frac{A + V^{0.5}}{A}\right]^B$$

$A = 50 + 56 \cdot [1 - B]$

$B = 0.25 \cdot [12 - Q_v]^{[2 / 3]}$

$Q_v = 6$ *Quality number. Assume fairly low, even given the precision application.*

$$V = n_{pinion} \cdot 3.142 \cdot 2 \cdot \frac{d_{p,pinion}}{2 \cdot 12}$$ *Pitch line velocity must be given in ft/min*

$K_o = 1.75$

K_o = 1.0 {(Uniform Loading)}

$C_f = 1$

$K_s = 1$

$K_s = 1.192*(F*(Y^{0.5)/pitch_d})^{0.0535}$
Y = 0.245 {(adjust with every trial)}

$Y = 2*x*pitch_d/3$
$x = (t^2)/(4*l)$
$l = 2.25/pitch_d$

$K_m = C_{mf}$

$K_m = 1 + C_{mc} \cdot [C_{pf} \cdot C_{pm} + C_{ma} \cdot C_e]$

$C_{mc} = 1$

$C_{pf} = \frac{F}{10 \cdot d_{p,pinion}} - 0.025$ *F<=1*

*C_{pf} = F/(10*$d_{p,pinion}$)-0.0375+0.0125*F*

1<F<=17

*C_{pf} = F/(10*$d_{p,pinion}$)-0.1109+0.0207*F-0.000228*F^2*

17<F<=40

C_{pm} = 1

S_1/S<0.175

$C_{pm} = 1.1$ *S_1/S>=0.175*

$C_e = 1$

$C_{ma} = A_c + B_c \cdot F + C_c \cdot F^2$

$A_c = 0.247$

$B_c = 0.0167$

$C_c = -0.765 \cdot 10^{-4}$

$C_H = 1 + A' \cdot [m_G - 1]$

$A' = 0$ *(($H_{B,P/H,B,G}$)<1.2)*

A_{prime} = 0.00698

($H_{B,P/H,B,G}$)>1.7

Life = 5 [yr]

$$Use = \frac{4}{60\ [sec]} \cdot \frac{60\ [sec]}{1\ [min]} \cdot \frac{60\ [min]}{1\ [hr]} \cdot \frac{24\ [hr]}{1\ [day]} \cdot \frac{365.25\ [day]}{1\ [yr]}$$

$Output_{cycles} = Life \cdot Use$

$$N_P = \frac{length_{rack}}{d_{p,pinion} \cdot \pi} \cdot Output_{cycles}$$

$N_G = Output_{cycles}$

$Y_{N,P} = 1.3558 \cdot N_P^{-0.0178}$

$Z_{N,P} = 1.4488 \cdot N_P^{-0.023}$

$K_R = 0.5 - 0.109 \cdot \ln[1 - R]$ *0.99<=R<=0.9999*

*K_R = 0.658 - 0.0759*ln(1-R)*

0.5<R<0.99

$R = 0.99$

$K_B = 1$

$$S_{F,P} = \frac{S_{t,P} \cdot Y_{N,P}}{K_T \cdot K_R \cdot \sigma_P}$$

$$S_{H,P} = \left[\frac{S_{c,P} \cdot Z_{N,P} \cdot C_H}{K_T \cdot K_R \cdot \sigma_{c,P}}\right]^2$$

$$\sigma_G = W_t \cdot K_o \cdot K_v \cdot K_{s,G} \cdot \frac{pitch_d}{F} \cdot \frac{K_m \cdot K_B}{J}$$

$$\sigma_{all,G} = \frac{S_{t,G}}{S_{F,G}} \cdot \frac{Y_{N,G}}{K_T \cdot K_R}$$

$$\sigma_{c,G} = C_p \cdot \left[W_t \cdot K_o \cdot K_v \cdot K_{s,G} \cdot \frac{K_m}{d_{p,pinion} \cdot F} \cdot \frac{C_f}{I}\right]^{0.5}$$

$$\sigma_{c,all,G} = \frac{S_{c,G}}{S_{H,G}} \cdot \frac{Z_{N,G} \cdot C_H}{K_T \cdot K_R}$$

$$S_{F,G} = \frac{S_{t,G} \cdot Y_{N,G}}{K_T \cdot K_R \cdot \sigma_G}$$

$$S_{H,G} = \left[\frac{S_{c,G} \cdot Z_{N,G} \cdot C_H}{K_T \cdot K_R \cdot \sigma_{c,G}}\right]^2$$

$S_{t,G} = 77.3 \cdot H_{B,G} + 12800$

$S_{c,G} = 322 \cdot H_{B,G} + 29100$

*$W_{t,G}$ = (Torque$_{Gear/(d,p,Gear/12)}$)*cos(phi)*
*$W_{r,G}$ = (Torque$_{Gear/(d,p,Gear/12)}$)*sin(phi)*

$J_G = 0.47$

K_s = 1

$K_{s,G}$ = 1.192(F*($Y_{G0.5)/pitch,d}$)$^{0.0535}$*
Y_G = 0.331 {(adjust with every trial)}

$K_{s,G} = 1$

*Y = 2*x*pitch$_d$/3*
*x = (t^2)/(4*l)*
l = 2.25/pitch$_d$

$Y_{N,G} = 1.3558 \cdot N_G^{-0.0178}$

$Z_{N,G} = 1.4488 \cdot N_G^{-0.023}$

SOLUTION

Unit Settings: Eng F psia mass deg

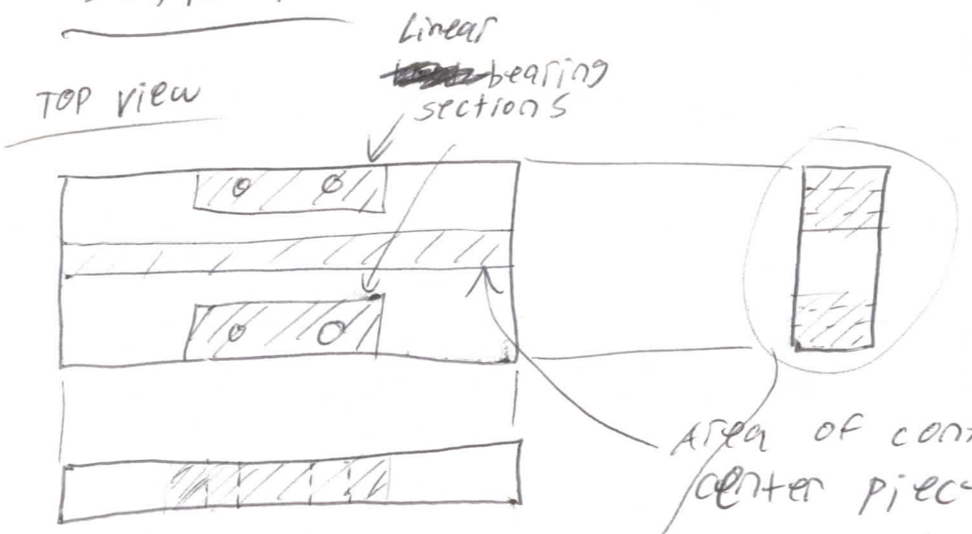
$A = 59.77$	$A_c = 0.247$	$A' = 0$
$B = 0.8255$	$B_c = 0.0167$	$C_c = -0.0000765$
$C_e = 1$	$C_f = 1$	$C_H = 1$
$C_{ma} = 0.2595$	$C_{mc} = 1$	$C_p = 2300$
$C_{pf} = 0.0125$	$C_{pm} = 1.1$	$d_{p,gear} = 8.333E+163$
$d_{p,pinion} = 2$	$F = 0.75$	$hp = 0.01763$
$H_{B,G} = 150$	$H_{B,P} = 150$	$I = 0.1212$
$J = 0.36$	$J_G = 0.47$	$K_B = 1$
$K_m = 1.273$	$K_o = 1.75$	$K_R = 1.002$
$K_s = 1$	$K_{s,G} = 1$	$K_T = 1$
$K_v = 1.108$	$length_{rack} = 62$	$Life = 5$ [yr]
$m_G = 4.167E+163$	$m_N = 1$	$N_G = 1.052E+07$
$N_P = 1.038E+08$	$n_{pinion} = 118.4$	$Output_{cycles} = 1.052E+07$
$\phi = 14.5$ [deg]	$pitch_c = 0.2618$	$pitch_d = 12$
$Q_v = 6$	$R = 0.99$	$\sigma_{all,G} = 5979$
$\sigma_{all,P} = 5979$	$\sigma_{c,all,G} = 50728$	$\sigma_{c,all,P} = 53470$
$\sigma_{c,G} = 62563$	$\sigma_{c,P} = 62563$	$\sigma_G = 5979$
$\sigma_P = 5979$	$speed_{ratio} = 4.167E+163$	$S_{c,G} = 77400$
$S_{c,P} = 77400$	$\boxed{S_{F,G} = 4.14}$	$\boxed{S_{F,P} = 3.975}$
$\boxed{S_{H,G} = 1.521}$	$\boxed{S_{H,P} = 1.369}$	$S_{t,G} = 24395$
$S_{t,P} = 24395$	$Teeth_{gear} = 1.000E+165$	$Teeth_{pinion} = 24$
$Torque_{pinion} = 9.384$	$Use = 2.104E+06$ [1/yr]	$V = 62$
$\boxed{W_r = 14.1}$	$W_t = 54.51$	$Y_{N,G} = 1.017$
$Y_{N,P} = 0.9761$	$Z_{N,G} = 0.9989$	$Z_{N,P} = 0.9476$

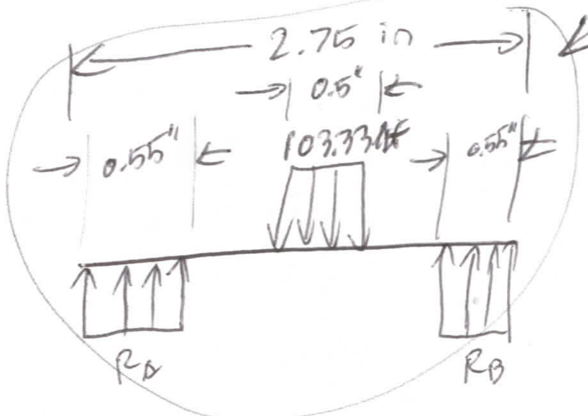
No unit problems were detected.

V-block (assume all the weight of the arm sits on one block)

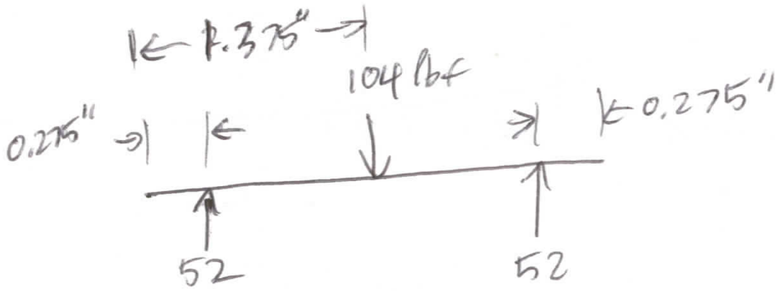
Bottom

Top view





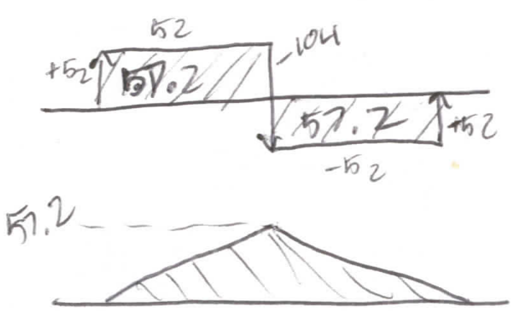
$$R_A = R_B = \frac{104\text{ lbf}}{2} = 52\text{ lbf}$$

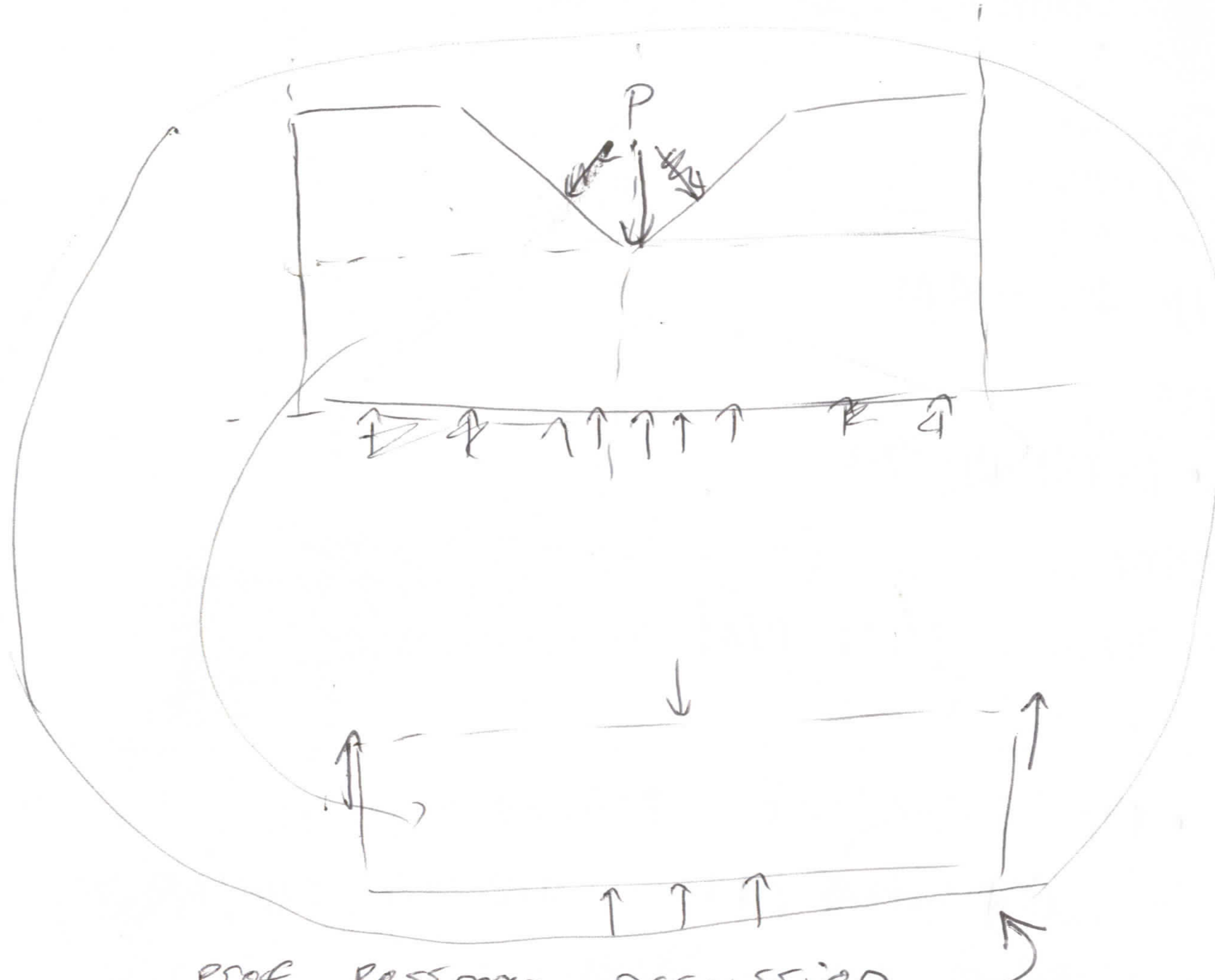


$$\sigma = \frac{Mc}{I}$$

$$= \frac{(57.2\text{ lbf}\cdot\text{in})(0.5\text{ in})}{\left[\frac{(2.06\text{ in})(0.5\text{ in})^3}{12}\right]}$$

$$\boxed{\sigma = 1333\text{ psi}}$$





Prof. Rossman Discussion

$$\tau = \frac{3V}{2A}$$

$$= \frac{3(52\text{ lbf})}{2(0.5\text{ in})(2.06\text{ in})}$$

$$\boxed{\tau = 75.73\text{ psi}}$$

$$\sigma' = (\sigma_x^2 - \cancel{\sigma_x\sigma_y} + \cancel{\sigma_y^2} + 3\tau_{xy}^2)^{1/2}$$

$$= ((1333\text{ psi})^2 + 3(75.73\text{ psi})^2)^{1/2}$$

$$\boxed{\sigma' = 1340\text{ psi}}$$

Meeting Minutes Wednesday 4/27/16

Present

Ryan Canfield
Louis Roseguo
Samuel Adler [11:00 AM]

BEGIN 10:25 AM

Business

- Detail Design

Present

Bill Tandrow [1:11 PM]

Business

- Plan Presentation for meeting the week after CDR 1:00PM, May 11th
- Discuss Design changes
  - Plan around 70 psi
  - Don't plan on 100
- Discuss signals and availability
  - E-stop or fault
  - OK to grip
  - OK to close door
- Discuss Haas internship opportunity
  - Acceptance is likely
  - End of Quarter
- Discuss new simplified design
  - Motion
  - Budget/BOM
- Explain the focus on mechanical design for CDR

END 2:00 PM

$$\sigma' = \frac{S_y}{n}$$

For 1018 Steel, $S_y = 36{,}000$ psi

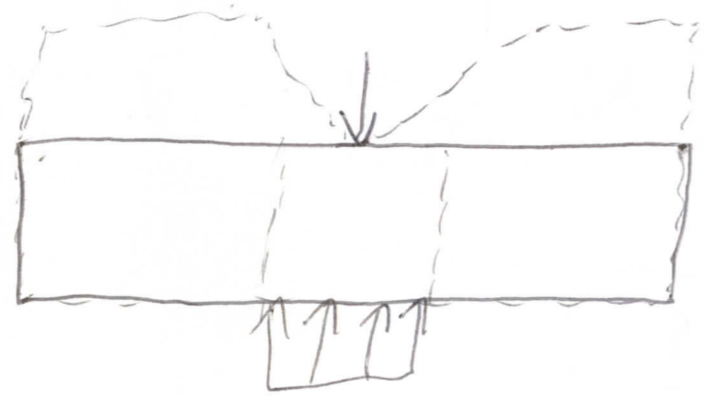
$$n = \frac{S_y}{\sigma'}$$

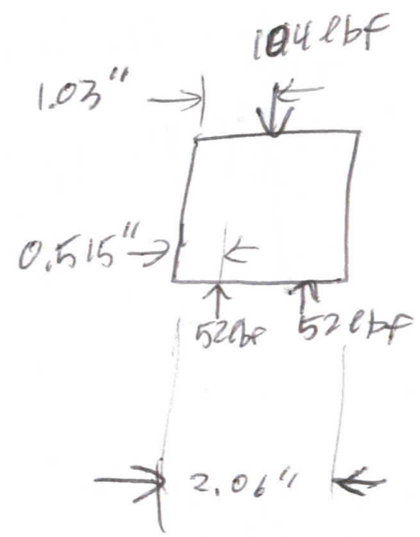
$$= \frac{36{,}000 \text{ psi}}{1{,}340 \text{ psi}}$$

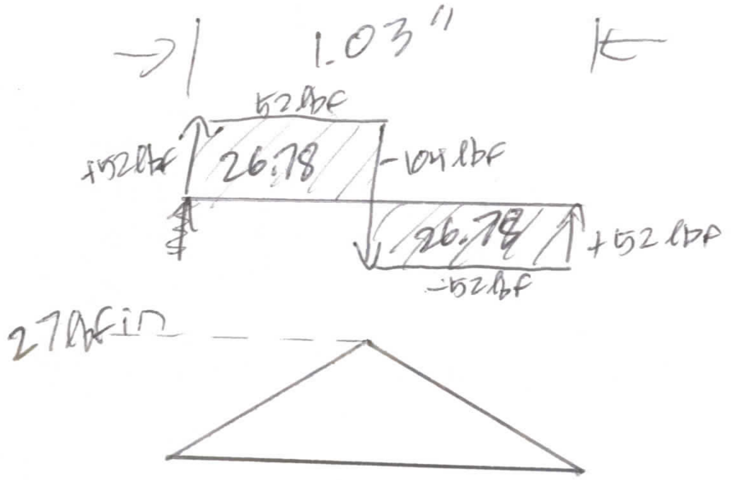
$\boxed{n = 26.86}$ for V-block bottom

V-block center

Take Prof. Rossman's advice and reduce the V into a rectangular beam in bending.







$b = 0.5''$

$h = \frac{3.5}{2}$ in

$h = 1.75$ in

Estimate a stress concentration due to the V shape:

Figure A-15-4

$d = 1.75$ in $\quad r = 0.25$ in

$w = d + 2r = 1.75 + 0.5 = 2.25$ in

$\frac{r}{d} = 1/7 = 0.143$

$\frac{w}{d} = \cancel{\frac{2.25}{0.25}} \quad \frac{2.25}{1.75} = 1.29$

$\therefore \boxed{K_t \simeq 1.8}$

$$\sigma = \sigma_0 K_t = \frac{Mc}{I} K_t = \frac{(27\ \text{lbf-in})\left(\frac{1.75\ \text{in}}{2}\right)}{\left[\frac{(0.5\ \text{in})(1.75\ \text{in})^3}{12}\right]}$$

$\boxed{\sigma = 106\ \text{psi}}$

$$\tau = \frac{3V}{2A} = \frac{3}{2}\frac{(52\ \text{lbf})}{(0.5\ \text{in})(1.75\ \text{in})}$$

$\boxed{\tau = 90\ \text{psi}}$

$$\sigma' = \left(\sigma_x^2 - \cancel{\sigma_x \sigma_y} + \cancel{\sigma_y^2} + 3\tau_{xy}^2\right)^{1/2}$$

$$= \left[(106\ \text{psi})^2 + 3(90\ \text{psi})^2\right]^{1/2}$$

$\boxed{\sigma' = 189\ \text{psi}}$

$\sigma' = \frac{S_y}{n}$

$n = \frac{36{,}000}{189}$

$\boxed{n = 190}$ for V-block center plate, given 1018 steel

will need wider bars than the parts so we can machine them down,

so 0.5" x 4"

Length: 2.75"x4 + 7"x2 = 25" of 4" wide bars Buy 3'

16" of 3" wide bars, buy 2'

Meeting Minutes 4/30/16

Present

Louis Rosegco
Samuel Adler [1:30 PM]

Begin 1:20 PM

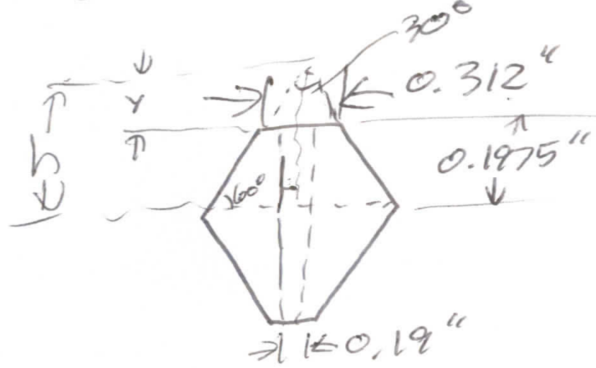
Business

- Detail design

END 4:00PM

$l = 0.395''$

$\frac{l}{2} = 0.1975''$



$\cancel{\tan 60° = h}$

$\tan 60° = \frac{Y}{(0.312/2)}$

$Y = 0.27''$

$h = Y + 0.1975''$

$h = 0.4677''$

$\cancel{\tan 60° = 0.4677}$

simplify: since it's the same material just ~~assume~~ ~~2 cones of the~~

use eq. (8-23)

$\frac{K_m}{Ed} = A e^{(Bd/l)}$ $A = 0.78715$
$B = 0.62873$

Threaded Faster Analysis

Constants

$d = 0.19$ [in] *Nominal Diameter of Screw*

$L_t = 0.75$ [in] *Threaded Length*

$t = 0$ [in] *Washer Thickness (Shigley's A-32)*

$H = 0$ *Nut Thickness (Shigley's A-31)*

$L = 0.75$ [in] *Fastener Length*

$w = 0.5$ [in] *Width of Material Being Fastened*

$l_1 = w + \frac{d}{2}$ *Grip Length*

$l_d = L - L_t$ *Length of Unthreaded Portion in Grip*

$l_{t,1} = l_1 - l_d$ *Length of Threaded Portion in Grip*

$A_d = \pi \cdot \frac{d^2}{4}$ *Area of Unthreaded Portion*

$A_t = 0.0175$ [in^2] *Area of Threaded Portion (Shigley's Table 8-1,8-2)*

$E_1 = 29.7 \cdot 10^6$ *Material Elastic Modulus (Steel)*

Material Elastic Modulus (Steel)

$\alpha = 30$ [Degrees] *Half-Apex Angle (Always use 30 degrees)*

Stiffness

$t_1 = 0.5$ [in] *Thickness of 1st Material*

$t_2 = 0.25$ [in] *Thickness of 2nd Material*

Frustum Diameter at Material Break

$k_b = \frac{A_d \cdot A_t \cdot E_1}{A_d \cdot l_{t,1} + A_t \cdot l_d}$ *Fastener Stiffness*

$\frac{k_m}{E_1 \cdot d} = 0.78715 \cdot \mathbf{exp}\left[0.62873 \cdot \frac{d}{l_1}\right]$ *This equation only works if the entire joint is made of the same material.*

Tension Joints

$F_i = 0.75 \cdot S_p \cdot A_t$ *Preload Force*

$P_{total} = 104$ [lbf]

Total External Tensile Force

Assume the extreme case that the tender is upside down and the entire weight of the arm is hanging by the screws.

$N = 6$ [-] *Number of Bolts*

$P = \dfrac{P_{total}}{N}$ *External Tensile Load per Bolt*

$C = \dfrac{k_b}{k_b + k_m}$ *Stiffness Constant of the Joint*

$P_b = C \cdot P$ *Portion of Force Taken by Bolt*

$P_m = [1 - C] \cdot P$ *Portion of Force Taken by Members*

$F_b = P_b + F_i$ *Resultant Bolt Load*

$F_m = P_m - F_i$ *Resultant Load on Members*

Static Failure Analysis

$\sigma_b = \dfrac{F_b}{A_t}$ *Tensile Stress in the Bolt*

$S_p = 120000$ [psi] *Rated Proof Load of Bolt*

$n_p = \dfrac{S_p}{\sigma_b}$ *Yielding Factor of Safety*

$n_l = \dfrac{S_p \cdot A_t - F_i}{C \cdot P}$ *Load Factor*

$n_o = \dfrac{F_i}{P \cdot [1 - C]}$ *Load Factor Guarding Against Joint Separation*

Shear Failure Analysis

$F = 104$ [lbf]

Shearing Force

Assume the extreme case that the tender is on its back and the weight of the arm is applying a shear force to the screws.

$I = \dfrac{\pi}{64} \cdot d^4$ *Moment of Inertia*

$A_s = \pi \cdot \left[\dfrac{d}{2}\right]^2$ *Cross Sectional Area of Bolt*

$\sigma_{ba} = F \cdot d \cdot \dfrac{w}{4 \cdot I}$

Failure by Bending of the Bolt (Typically compensated for with increased factor of safety

$\tau = \dfrac{F}{N \cdot A_s}$ *Failure by Pure Shear*

$$\sigma' = \left[\sigma_b^2 + 3 \cdot \tau^2\right]^{0.5}$$ *Von Mises*

$$\sigma' = \frac{120000}{n_{total}}$$

$$\sigma_{plate} = \frac{F}{2.5^2 - 4 \cdot A_s}$$ *Rupture of Connected Members*

$$\sigma_{crushing} = \frac{-F}{t_1 \cdot d}$$ *Failure by Crushing of the Bolt of Plate*

SOLUTION

Unit Settings: SI C kPa kJ mass deg

α = 30 [Degrees]	A_d = 0.02835	A_s = 0.02835
A_t = 0.0175 [in^2]	C = 0.1386	d = 0.19 [in]
E_1 = 2.970E+07	F = 104 [lbf]	F_b = 1577
F_i = 1575	F_m = -1560	H = 0
I = 0.00006397	k_b = 873529	k_m = 5.430E+06
L = 0.75 [in]	l_1 = 0.595	l_d = 0
L_t = 0.75 [in]	$l_{t,1}$ = 0.595	N = 6 [-]
n_l = 218.5	n_o = 105.5	n_p = 1.331
n_{total} = 1.331	P = 17.33	P_b = 2.402
P_m = 14.93	P_{total} = 104 [lbf]	σ_b = 90137
σ_{ba} = 38611	$\sigma_{crushing}$ = -1095	σ_{plate} = 16.95
σ' = 90143	S_p = 120000 [psi]	t = 0 [in]
τ = 611.3	t_1 = 0.5 [in]	t_2 = 0.25 [in]
w = 0.5 [in]		

16 potential unit problems were detected.

Threaded Faster Analysis

Constants

$d = 0.19$ [in] *Nominal Diameter of Screw*

$L_t = 0.75$ [in] *Threaded Length*

$t = 0$ [in] *Washer Thickness (Shigley's A-32)*

$H = 0$ *Nut Thickness (Shigley's A-31)*

$L = 0.75$ [in] *Fastener Length*

$w = 0.3$ [in] *Width of Material Being Fastened*

$l_1 = w + \frac{d}{2}$ *Grip Length*

$l_d = L - L_t$ *Length of Unthreaded Portion in Grip*

$l_{t,1} = l_1 - l_d$ *Length of Threaded Portion in Grip*

$A_d = \pi \cdot \frac{d^2}{4}$ *Area of Unthreaded Portion*

$A_t = 0.0175$ [in^2] *Area of Threaded Portion (Shigley's Table 8-1,8-2)*

$E_1 = 29.7 \cdot 10^6$ *Material Elastic Modulus (Steel)*

Material Elastic Modulus (Steel)

$\alpha = 30$ [Degrees] *Half-Apex Angle (Always use 30 degrees)*

Stiffness

$t_1 = 0.3$ [in] *Thickness of 1st Material*

$t_2 = 0.45$ [in] *Thickness of 2nd Material*

Frustum Diameter at Material Break

$k_b = \frac{A_d \cdot A_t \cdot E_1}{A_d \cdot l_{t,1} + A_t \cdot l_d}$ *Fastener Stiffness*

$\frac{k_m}{E_1 \cdot d} = 0.78715 \cdot \mathbf{exp}\left[0.62873 \cdot \frac{d}{l_1}\right]$ *This equation only works if the entire joint is made of the same material.*

Tension Joints

$F_i = 0.75 \cdot S_p \cdot A_t$ *Preload Force*

$P_{total} = 104$ [lbf]

Total External Tensile Force

Assume the extreme case that the tender is on its side and the entire weight of the arm is hanging by the screw.

$N = 1$ [-] *Number of Bolts*

$P = \dfrac{P_{total}}{N}$ *External Tensile Load per Bolt*

$C = \dfrac{k_b}{k_b + k_m}$ *Stiffness Constant of the Joint*

$P_b = C \cdot P$ *Portion of Force Taken by Bolt*

$P_m = [1 - C] \cdot P$ *Portion of Force Taken by Members*

$F_b = P_b + F_i$ *Resultant Bolt Load*

$F_m = P_m - F_i$ *Resultant Load on Members*

Static Failure Analysis

$\sigma_b = \dfrac{F_b}{A_t}$ *Tensile Stress in the Bolt*

$S_p = 120000$ [psi] *Rated Proof Load of Bolt*

$n_p = \dfrac{S_p}{\sigma_b}$ *Yielding Factor of Safety*

$n_l = \dfrac{S_p \cdot A_t - F_i}{C \cdot P}$ *Load Factor*

$n_o = \dfrac{F_i}{P \cdot [1 - C]}$ *Load Factor Guarding Against Joint Separation*

Shear Failure Analysis

$F = 104$ [lbf] *Shearing Force*

$I = \dfrac{\pi}{64} \cdot d^4$ *Moment of Inertia*

$A_s = \pi \cdot \left[\dfrac{d}{2}\right]^2$ *Cross Sectional Area of Bolt*

$\sigma_{ba} = F \cdot d \cdot \dfrac{w}{4 \cdot I}$

Failure by Bending of the Bolt (Typically compensated for with increased factor of safety

$\tau = \dfrac{F}{A_s}$ *Failure by Pure Shear*

$\sigma' = [\sigma_b^2 + 3 \cdot \tau^2]^{0.5}$ *Von Mises*

$$\sigma' = \frac{120000}{n_{total}}$$

$$\sigma_{plate} = \frac{F}{2.5^2 - 4 \cdot A_s}$$ *Rupture of Connected Members*

$$\sigma_{crushing} = \frac{-F}{t_1 \cdot d}$$ *Failure by Crushing of the Bolt of Plate*

SOLUTION

Unit Settings: SI C kPa kJ mass deg

α = 30 [Degrees]	A_d = 0.02835	A_s = 0.02835
A_t = 0.0175 [in^2]	C = 0.1796	d = 0.19 [in]
E_1 = 2.970E+07	F = 104 [lbf]	F_b = 1594
F_i = 1575	F_m = -1490	H = 0
I = 0.00006397	k_b = 1.316E+06	k_m = 6.010E+06
L = 0.75 [in]	l_1 = 0.395	l_d = 0
L_t = 0.75 [in]	$l_{t,1}$ = 0.395	N = 1 [-]
n_l = 28.11	n_o = 18.46	$\boxed{n_p = 1.318}$
$\boxed{n_{total} = 1.315}$	P = 104	P_b = 18.68
P_m = 85.32	P_{total} = 104 [lbf]	σ_b = 91067
σ_{ba} = 23167	$\sigma_{crushing}$ = -1825	σ_{plate} = 16.95
σ' = 91289	S_p = 120000 [psi]	t = 0 [in]
$\boxed{\tau = 3668}$	t_1 = 0.3 [in]	t_2 = 0.45 [in]
w = 0.3 [in]		

16 potential unit problems were detected.

Shaft Adapter

$\sigma_a = 2813.5$ [lb_f/in^2] *Stress amplitude*

$\sigma_m = \sigma_a$ *Mean stress*

$\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{ut}} = \frac{1}{n}$ *Modified-Goodman safety factor*

$S_e = k_a \cdot k_b \cdot k_c \cdot k_d \cdot k_e \cdot k_f \cdot S'_e$ *Endurance limit*

$k_a = a \cdot \left[S_{ut} \cdot psitokpsi \right]^b$ *Surface condition modification factor for S_{ut} in psi*

$psitokpsi = \frac{1}{1000} \cdot 1$ [in^2/lb_f]

$a = 2.7$ *For machined or cold-drawn surfaces*

$b = -0.265$ *For machined or cold-drawn surfaces*

$k_b = 0.879 \cdot d^{-0.107}$ *Size modification factor*

$d = 0.625$ *Diameter in inches*

$k_c = 0.59$ *Load modification factor*

$k_d = 1$ *Temperature modification factor*

$k_e = 0.702$ *Reliability factor*

$k_f = 1$ *Miscellaneous-effects modification factor*

$S'_e = 0.5 \cdot S_{ut}$ *Only as long as S_{ut} <= 200 kpsi*

n = 2

Minimum desired safety factor

$S_{ut} = 105000$

SOLUTION

Unit Settings: SI C kPa kJ mass deg

a = 2.7	b = -0.265
d = 0.625	k_a = 0.7866
k_b = 0.9243	k_c = 0.59
k_d = 1	k_e = 0.702
k_f = 1	n = 4.884
psitokpsi = 0.001 [in^2/lb_f]	σ_a = 2814 [lb_f/in^2]
σ_m = 2814 [lb_f/in^2]	S_e = 15810 [lb_f/in^2]
S'_e = 52500 [lb_f/in^2]	S_{ut} = 105000 [lb_f/in^2]

No unit problems were detected.

Shaft Analysis (Hortizontal Guide Shaft)

Constants

Dimensions

$L = 48$ [in] *Length of Beam*

$d_o = 0.75$ [in] *Outer Diameter*

$d_i = 0$ [in]

$A_s = \pi \cdot \left[\frac{d_o}{2}\right]^2$ *Cross Sectional Area of Beam*

$I = \frac{\pi}{64} \cdot d_o^4$ *Moment of Inertia*

$y = \frac{d_o}{2}$ *Perpendicular Distance to the Neutral Axis*

$\bar{y} = \frac{d_o^3 - d_i^3}{3 \cdot [d_o^2 - d_i^2]}$ *Centroid of top half, used for Q calculation*

Material Properties

$E = 29.6 \cdot 10^6$ *Elastic Modulus*

$\rho = 0.282$ [lbf/in^3] *Density of beam material*

$\sigma_{yield} = 150000$ [psi] *Yeild Strength*

$s_{ut} = 167953$ [psi]

Applied Forces

$P = 150$ [lbf] *Applied Point Load at End*

$\omega = \rho \cdot A_s$ *Distributed Load from Beams Mass*

Stress Concentration Constants

$K_{fb} = 1$ *Bendisng Stress Concentration Factor*

$K_{fs} = 1$ *Tortional Stress Concentration Factor*

Reaction Forces

$L_y = \frac{P}{2}$ *Reaction Force at Left Support*

$R_y = \frac{P}{2}$ *Reaction Force at Right Support*

Deflection Analysis

Deflection at End

$dx_p = \frac{P \cdot L^3}{48 \cdot E \cdot I}$ Deflection from Point Load

$dx_d = \frac{5 \cdot \omega \cdot L^4}{384 \cdot E \cdot I}$ Deflection from Distributed Load

$dx_{max} = dx_p + dx_d$ Total Maximum Deflection at Free End

Stresses

$M_{max} = P \cdot \frac{L}{4} + 1 / 8 \cdot \omega \cdot L^2$

Bending Stresses

$\sigma_x = M_{max} \cdot \frac{y}{I}$ Maximum Static Beam Bending Stress in the x-direction

$\sigma_y = 0$

$\sigma_z = 0$

Shear Stress

$\tau_{xy} = \frac{4 \cdot P}{3 \cdot A_s}$ Maximum Transverse Shear Stress in the xy plane

$\tau_{yz} = 0$

$\tau_{zx} = 0$

Von Mises

$\sigma_{max} = \frac{1}{2^{0.5}} \cdot \left[\left(\sigma_x - \sigma_y \right)^2 + \left(\sigma_y - \sigma_z \right)^2 + \left(\sigma_z - \sigma_x \right)^2 + 6 \cdot \left(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2 \right) \right]^{0.5}$ Von Mises Stress

$\sigma_{min} = \tau_{xy}$ Assume no bending stress at minimum stress

Failure Theories

Endurance Limit

$S_{e'} = 0.5 \cdot s_{ut}$

Surface Factor Ka

$a = 14.4$ [kpsi] Hot-Rolled Surface Finish

$b = -0.718$

$k_a = a \cdot \left[\frac{s_{ut}}{1000} \right]^b$ *Surface Factor*

Size Factor Kb

$d_e = 0.37 \cdot d_o$ *Equivalent Diameter of a nonrotating hollow round*

$k_b = 0.879 \cdot d_e^{-0.107}$ *Size Factor Assumeing 0.11<= d_e <= 2 in*

Loading Factor Kc

$K_c = 1$ *Loading is Bending*

Temperature Factor Kd

$K_d = 1$ *Assume Room Temperature Operation*

Reliability Facter Ke

$k_e = 0.814$ *Assume 99% Reliability*

Miscellaneous Effects Factor Kf

$k_f = 1$ *No Miscellaneous Effects*

$S_e = k_a \cdot k_b \cdot K_c \cdot K_d \cdot k_e \cdot k_f \cdot S_e'$ *Endurance Limit*

Loads and Failure Criteria

$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}$ *Alternating Stress*

$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2}$ *Midrange Stress*

$\frac{\sigma_a}{S_e} + \frac{\sigma_m}{s_{ut}} = \frac{1}{n_{modGoodman}}$ *Modified Goodman Fatigue Failure Theory*

$\sigma_a + \sigma_m = \frac{\sigma_{yield}}{n_{yield}}$

SOLUTION

Unit Settings: SI C kPa kJ mass deg

a = 14.4 [kpsi]	A_s = 0.4418 [in]
b = -0.718	dx_d = 0.01873 [in]
$\boxed{dx_{max} = 0.7705 \text{ [in]}}$	dx_p = 0.7517 [in]
d_e = 0.2775 [in]	d_i = 0 [in]
d_o = 0.75 [in]	E = 2.960E+07 [psi]
I = 0.01553 [in^4]	k_a = 0.3636 [-]
k_b = 1.008 [-]	K_c = 1 [-]
K_d = 1 [-]	k_e = 0.814 [-]
k_f = 1 [-]	K_{fb} = 1 [-]
K_{fs} = 1 [-]	L = 48 [in]

L_y = 75 [lbf]	M_{max} = 1836 [lbf*in]
$\boxed{n_{modGoodman} = 0.9913 \ [-]}$	$\boxed{n_{yield} = 3.383 \ [-]}$
ω = 0.1246 [lbf/in]	P = 150 [lbf]
ρ = 0.282 [lbf/in^3]	R_y = 75 [lbf]
σ_a = 21940 [psi]	σ_m = 22393 [psi]
σ_{max} = 44333 [psi]	σ_{min} = 452.7 [psi]
σ_x = 44326 [psi]	σ_y = 0 [psi]
σ_{yield} = 150000 [psi]	σ_z = 0 [psi]
S_e = 25062 [psi]	$\overline{S}_{e'}$ = 83977 [psi]
s_{ut} = 167953 [psi]	τ_{xy} = 452.7 [psi]
τ_{yz} = 0 [psi]	τ_{zx} = 0 [psi]
y = 0.375 [in]	$\overline{y}$ = 0.25 [in]

6 potential unit problems were detected.

Lead Screw Analysis

Define Variables

$F = 100$ [lbf] *Axial Compressive Force*

$n = 1$ [-] *Single or Double Threads (n=1 for single, n=2 for double)*

$p = 0.2$ [in] *Pitch*

$d_p = d - \frac{p}{4}$ *Pitch Diameter*

$d = 1$ [in] *Major Diameter*

$d_m = d - \frac{p}{4}$ *Mean Diameter*

$d_r = d - \frac{d_p}{2}$ *Minor Diameter*

$f_r = 0.2$ [-] *Coefficient of Friction (For the screw)*

$f_{rc} = 0.2$ [-] *Coefficient of Friction (For the collar)*

$d_c = 0.75$ [in] *Diameter of Collar*

$T_{Angle} = 29$ *Thread Angle*

$2 \cdot \alpha = T_{Angle}$ *Solves for alpha*

$n_t = 3$ *Number of Teeth Engaged*

$L = n \cdot p$

$length = 56$ [in] *Length of Lead Screw*

$E = 29.6 \cdot 10^6$ *Elastic Modulus for Steel*

$n_b = 4$ [-] *End Condition Factor for Both Ends Fixed*

$I = \frac{\pi}{64} \cdot d_r^4$ *Area Moment of Inertia using Minor Diameter*

Other (ACME) Threads

Load Raising Torque (Including Collar)

$$T_R = F \cdot \frac{d_m}{2} \cdot \left[\frac{L + \pi \cdot f_r \cdot d_m \cdot \frac{1}{\cos(\alpha)}}{\pi \cdot d_m - f_r \cdot L \cdot \frac{1}{\cos(\alpha)}} \right] + \frac{F \cdot f_{rc} \cdot d_c}{2}$$

Load Lowering Torque (Including Collar)

$$T_L = F \cdot \frac{d_m}{2} \cdot \left[\frac{\pi \cdot f_r \cdot d_m \cdot \frac{1}{\cos(\alpha)} - L}{\pi \cdot d_m + f_r \cdot L \cdot \frac{1}{\cos(\alpha)}} \right] + \frac{F \cdot f_{rc} \cdot d_c}{2}$$

Self Locking Check

$S_L = \pi \cdot f_r \cdot d_m$ *If S_L is greater than L, the screw is self-locking*

Efficiency

$$e_f = \frac{F \cdot L}{2 \cdot \pi \cdot T_R}$$

Stresses (Shigleys Figure 8.8 Axis)

Bending Stress

$\sigma_x = \frac{6 \cdot F}{\pi \cdot d_r \cdot n_t \cdot p}$ *Bending Stress in the x-direction*

$\sigma_y = \frac{-4 \cdot F}{\pi \cdot d_r^2}$ *Bending Stress in the y-direction*

$\sigma_z = 0$ *Bending Stress in the z-direction*

Shear Stress

$$\tau_{xy} = 0$$

$$\tau_{yz} = \frac{16 \cdot T_R}{\pi \cdot d_r^3}$$

$$\tau_{zx} = 0$$

Von Mises Stress

$$\sigma_{VM} = \frac{1}{2^{0.5}} \cdot \left[(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6 \cdot (\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2) \right]^{0.5}$$

Buckling

$F_b = \frac{n \cdot \pi^2 \cdot E \cdot I}{length^2}$ *Calculates Force Required for Buckling to Occur*

SOLUTION

Unit Settings: Eng F psia mass deg

α = 14.5 [degrees]	d = 1 [in]	d_c = 0.75 [in]
d_m = 0.95 [in]	d_p = 0.95 [in]	d_r = 0.525 [in]
E = 2.960E+07 [psi]	e_f = 0.1539 [-]	F = 100 [lbf]
F_b = 347.4 [lbf]	f_r = 0.2 [-]	f_{rc} = 0.2 [-]

I = 0.003729 [in^4]	L = 0.2 [in]	length = 56 [in]
n = 1 [-]	n_b = 4 [-]	n_t = 3 [-]
p = 0.2 [in]	$\boxed{\sigma_{VM} = 1565 \text{ [psi]}}$	σ_x = 606.3 [psi]
σ_y = -461.9 [psi]	σ_z = 0 [psi]	S_L = 0.5969 [in]
τ_{xy} = 0 [psi]	τ_{yz} = 727.8 [psi]	τ_{zx} = 0 [psi]
T_{Angle} = 29 [degrees]	T_L = 14.04 [lbf*in]	$\boxed{T_R = 20.68 \text{ [lbf*in]}}$

No unit problems were detected.

Threaded Faster Analysis

Constants

$d = 0.25$ [in] *Nominal Diameter of Screw*

$L_t = 2 \cdot d + 0.25$ [in] *Threaded Length*

$t = 0.006$ [in] *Washer Thickness (Shigley's A-32)*

$H = \frac{17}{64}$ *Nut Thickness (Shigley's A-31)*

$L = 2$ [in] *Fastener Length*

$w = 1.5$ [in] *Width of Material Being Fastened*

$l_1 = H + w$ *Grip Length*

$l_d = L - L_t$ *Length of Unthreaded Portion in Grip*

$l_{t,1} = L - l_d$ *Length of Threaded Portion in Grip*

$A_d = \pi \cdot \frac{d^2}{4}$ *Area of Unthreaded Portion*

$A_t = 0.0318$ [in^2] *Area of Threaded Portion (Shigley's Table 8-1,8-2)*

$E_1 = 10.7 \cdot 10^6$ *Material Elastic Modulus (Steel)*

$E_2 = 10^7$ *Material Elastic Modulus (Aluminum)*

$\alpha = 30$ [Degrees] *Half-Apex Angle (Always use 30 degrees)*

Stiffness

$t_1 = 0.25$ [in] *Thickness of 1st Material*

$t_2 = 1$ [in] *Thickness of 2nd Material*

$D_1 = 0.281$ [in] *Diameter of Bolt Head*

$D_2 = 0.538675$ [in] *Frustum Diameter at Material Break*

$k_b = \frac{A_d \cdot A_t \cdot E_1}{A_d \cdot l_{t,1} + A_t \cdot l_d}$ *Fastener Stiffness*

$k_1 = \frac{0.5774 \cdot \pi \cdot E_1 \cdot d}{\ln\left[\frac{(1.155 \cdot t_1 + D_1 - d) \cdot (D_1 + d)}{(1.155 \cdot t_1 + D_1 + d) \cdot (D_1 - d)}\right]}$ *1st Member Stiffness*

$k_2 = \frac{0.5774 \cdot \pi \cdot E_1 \cdot d}{\ln\left[\frac{(1.155 \cdot t_2 + D_2 - d) \cdot (D_2 + d)}{(1.155 \cdot t_2 + D_2 + d) \cdot (D_2 - d)}\right]}$ *2nd Member Stiffness*

$$k_3 = \frac{0.5774 \cdot \pi \cdot E_1 \cdot d}{\ln\left[\frac{(1.155 \cdot t_1 + D_1 - d)\cdot(D_1 + d)}{(1.155 \cdot t_1 + D_1 + d)\cdot(D_1 - d)}\right]}$$ *3rd Member Stiffness*

$$\frac{1}{k_m} = \frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3}$$ *Member Stiffness*

Tension Joints

$F_i = 0.75 \cdot S_p \cdot A_t$ *Preload Force*

$P_{total} = 200$ [lbf] *Total External Tensile Force*

$N = 2$ [-] *Number of Bolts*

$$P = \frac{P_{total}}{N}$$ *External Tensile Load per Bolt*

$$C = \frac{k_b}{k_b + k_m}$$ *Stiffness Constant of the Joint*

$P_b = C \cdot P$ *Portion of Force Taken by Bolt*

$P_m = [1 - C] \cdot P$ *Portion of Force Taken by Members*

$F_b = P_b + F_i$ *Resultant Bolt Load*

$F_m = P_m - F_i$ *Resultant Load on Members*

Static Failure Analysis

$$\sigma_b = \frac{F_b}{A_t}$$ *Tensile Stress in the Bolt*

$S_p = 180000$ [psi] *Rated Proof Load of Bolt*

$$n_p = \frac{S_p}{\sigma_b}$$ *Yielding Factor of Safety*

$$n_l = \frac{S_p \cdot A_t - F_i}{C \cdot P}$$ *Load Factor*

$$n_o = \frac{F_i}{P \cdot [1 - C]}$$ *Load Factor Guarding Against Joint Separation*

Shear Failure Analysis

$F = 100$ [lbf] *Shearing Force*

$$I = \frac{\pi}{64} \cdot d^4$$ *Moment of Inertia*

$$A_s = \pi \cdot \left[\frac{d}{2}\right]^2$$ *Cross Sectional Area of Bolt*

$$\sigma_{ba} = F \cdot d \cdot \frac{w}{4 \cdot I}$$

Failure by Bending of the Bolt (Typically compensated for with increased factor of safety

$$\tau = \frac{F}{4 \cdot A_s}$$ *Failure by Pure Shear*

$$\sigma_{plate} = \frac{F}{2.5 \ [in]^2 - 4 \cdot A_s}$$ *Rupture of Connected Members*

$$\sigma_{crushing} = \frac{-F}{t_1 \cdot d}$$ *Failure by Crushing of the Bolt or Plate*

SOLUTION

Unit Settings: SI C kPa kJ mass deg

α = 30 [Degrees]	A_d = 0.04909 [in^2]
A_s = 0.04909 [in^2]	A_t = 0.0318 [in^2]
C = 0.1685 [-]	d = 0.25 [in]
D_1 = 0.281 [in]	D_2 = 0.5387 [in]
E_1 = 1.070E+07 [psi]	E_2 = 1.000E+07 [psi]
F = 100 [lbf]	F_b = 4310 [lbf]
F_i = 4293 [lbf]	F_m = -4210 [lbf]
H = 0.2656 [in]	I = 0.0001917 [in^4]
k_1 = 2.555E+06 [lbf/in]	k_2 = 6.857E+06 [lbf/in]
k_3 = 2.555E+06 [lbf/in]	k_b = 218146 [lbf/in]
k_m = 1.077E+06 [lbf/in]	L = 2 [in]
l_1 = 1.766 [in]	l_d = 1.25 [in]
L_t = 0.75 [in]	$l_{t,1}$ = 0.75 [in]
N = 2 [-]	n_l = 84.95 [-]
n_o = 51.63 [-]	n_p = 1.328 [-]
P = 100 [lbf]	P_b = 16.85 [lbf]
P_m = 83.15 [lbf]	P_{total} = 200 [lbf]
σ_b = 135530 [psi]	σ_{ba} = 48892 [psi]
$\sigma_{crushing}$ = -1600 [psi]	σ_{plate} = 16.52 [psi]
S_p = 180000 [psi]	t = 0.006 [in]
τ = 509.3 [psi]	t_1 = 0.25 [in]
t_2 = 1 [in]	w = 1.5 [in]

No unit problems were detected.

Weld Analysis

Constants

$F = 100$ [lbf] *Applied force*

$b = 1.75$ [in] *Bar Length*

$d = 1$ [in] *Bar Height*

$h = 1 / 4$ *Weld Height*

$c = \frac{d}{2}$ *Distance from weld to neutral axis*

$L = 24$ [in] *Distance to applied load*

$A = 1.414 \cdot h \cdot [b + d]$ *Weld Throat Area*

$I_u = \frac{d^2}{6} \cdot [3 \cdot d + d]$ *Unit Second Moment of Area*

$I = 0.707 \cdot h \cdot I_u$ *Moment of Inertia*

$M = F \cdot L$ *Applied Moment*

$S_{ut} = 62000$ [psi]

Allowable Tensile Strength for AWS Electrode Number E60xx

Shear Stresses

$\tau' = \frac{F}{A}$ *Primary Shear Stress*

$\tau'' = M \cdot \frac{c}{I}$ *Nominal Throat Shear Stress*

$\tau = [\tau'^2 + \tau''^2]^{0.5}$ *Shear Stress*

$\tau_{allowable} = 0.3 \cdot S_{ut}$ *Allowable Shear Stresses*

SOLUTION

Unit Settings: Eng F psia mass deg

$A = 0.9721$ [in^2]
$b = 1.75$ [in]
$c = 0.5$ [in]
$d = 1$ [in]
$F = 100$ [lbf]
$h = 0.25$ [in]
$I = 0.1178$ [in^4]
$I_u = 0.6667$ [in^3]

$L = 24$ [in]
$M = 2400$ [lbf*in]
$S_{ut} = 62000$ [psi]
$\boxed{\tau = 10184 \text{ [psi]}}$
$\boxed{\tau_{allowable} = 18600 \text{ [psi]}}$
$\tau'' = 10184$ [psi]
$\tau' = 102.9$ [psi]

No unit problems were detected.

Vertical Guide Rails

Constants

Dimensions

$d = 0.5$ [in] *Shaft Diameter*

$L = 30$ [in] *Shaft Length*

$I = \frac{\pi}{64} \cdot d^4$ *Shaft Area Moment of Inertia*

$n = 4$ *End Conditions Factor (4 for both ends fixed)*

$A = \frac{\pi}{4} \cdot d^2$

Material Properties

$E = 29.6 \cdot 10^6$ *Modulus of Elasticity*

$S_y = 36000$ [psi] *Yield Strength of Steel*

$S_{ut} = 60000$ [psi] *Ultimate Strength of Steel*

Buckling Analysis

$F = \frac{n \cdot \pi^2 \cdot E \cdot I}{L^2}$ *Compression Force to Cause Buckling*

Failure Analysis

$\sigma' = \frac{F}{A}$ *Axial Compression Force*

Distortion Energy - Ductile Material

$n_D = \frac{S_y}{\sigma'}$ *Safety Factor for Max Compression Force*

Mohr Hypothesis - Brittle Material

$n_B = \frac{S_{ut}}{\sigma'}$ *Safety Factor for Max Compression Force*

SOLUTION

Unit Settings: SI C kPa kJ mass deg

$A = 0.1963$ [in^2]	$d = 0.5$ [in]
$E = 2.960E+07$ [psi]	$F = 3983$ [lbf]
$I = 0.003068$ [in^4]	$L = 30$ [in]
$n = 4$ [-]	$n_B = 2.957$ [-]
$n_D = 1.774$ [-]	$\sigma' = 20288$ [psi]

S_{ut} = 60000 [psi]

S_y = 36000 [psi]

No unit problems were detected.

Threaded Faster Analysis

Constants

$d = 0.31$ [in] *Nominal Diameter of Screw*

$L_t = 1.1$ [in] *Threaded Length*

$t = 0$ [in] *Washer Thickness (Shigley's A-32)*

$L = 1.38$ [in] *Fastener Length*

$d_w = 0.75$ [in] *Width of Material Being Fastened*

$$l_{grip} = d_w + \frac{d}{2}$$ *Grip Length*

$$l_d = L - L_t$$ *Length of Unthreaded Portion in Grip*

$$l_{tgrip} = l_{grip} - l_d$$ *Length of Threaded Portion in Grip*

$$A_d = \pi \cdot \frac{d^2}{4}$$ *Area of Unthreaded Portion*

$A_t = 0.0567$ [in^2] *Area of Threaded Portion (Shigley's Table 8-1,8-2)*

$$E_1 = 29.7 \cdot 10^6$$ *Material Elastic Modulus (Steel)*

Material Elastic Modulus (Steel)

$\alpha = 30$ [Degrees] *Half-Apex Angle (Always use 30 degrees)*

Stiffness

$t_1 = 0.75$ [in] *Thickness of 1st Material*

$t_2 = 0.63$ [in] *Thickness of 2nd Material*

Frustum Diameter at Material Break

$$k_b = \frac{A_d \cdot A_t \cdot E_1}{A_d \cdot l_{tgrip} + A_t \cdot l_d}$$ *Fastener Stiffness*

$$\frac{k_m}{E_1 \cdot d} = 0.78715 \cdot \mathbf{exp}\left[0.62873 \cdot \frac{d}{l_{grip}}\right]$$ *Equation only works if entire joint is made of the same material.*

Tension Joints

$$F_i = 0.75 \cdot S_p \cdot A_t$$ *Preload Force*

$P_{total} = 40$ [lbf] *Total External Tensile Force*

$N = 4$ [-] *Number of Bolts*

$P = \dfrac{P_{total}}{N}$ *External Tensile Load per Bolt*

$C = \dfrac{k_b}{k_b + k_m}$ *Stiffness Constant of the Joint*

$P_b = C \cdot P$ *Portion of Force Taken by Bolt*

$P_m = [1 - C] \cdot P$ *Portion of Force Taken by Members*

$F_b = P_b + F_i$ *Resultant Bolt Load*

$F_m = P_m - F_i$ *Resultant Load on Members*

Static Failure Analysis

$\sigma_b = \dfrac{F_b}{A_t}$ *Tensile Stress in the Bolt*

$S_p = 120000$ [psi] *Rated Proof Load of Bolt*

$n_p = \dfrac{S_p}{\sigma_b}$ *Yielding Factor of Safety*

$n_l = \dfrac{S_p \cdot A_t - F_i}{C \cdot P}$ *Load Factor*

$n_o = \dfrac{F_i}{P \cdot [1 - C]}$ *Load Factor Guarding Against Joint Separation*

Shear Failure Analysis

$F = 40$ [lbf] *Shearing Force*

$I = \dfrac{\pi}{64} \cdot d^4$ *Moment of Inertia*

$A_s = \pi \cdot \left[\dfrac{d}{2}\right]^2$ *Cross Sectional Area of Bolt*

$\sigma_{ba} = F \cdot d \cdot \dfrac{d_w}{4 \cdot I}$ *Failure by Bending of Bolt (compensated by increased safety factor)*

$P_w = 3.5$ *Plate width*

$P_h = 6$ *Plate height*

$\tau = \dfrac{F}{A_s}$ *Failure by Pure Shear*

$\sigma' = \left[\sigma_b^{\ 2} + 3 \cdot \tau^2\right]^{0.5}$ *Von Mises*

$\sigma' = \dfrac{S_p}{n_{total}}$

$\sigma_{plate} = \dfrac{F}{P_w \cdot P_h - 4 \cdot A_s}$ *Rupture of Connected Members*

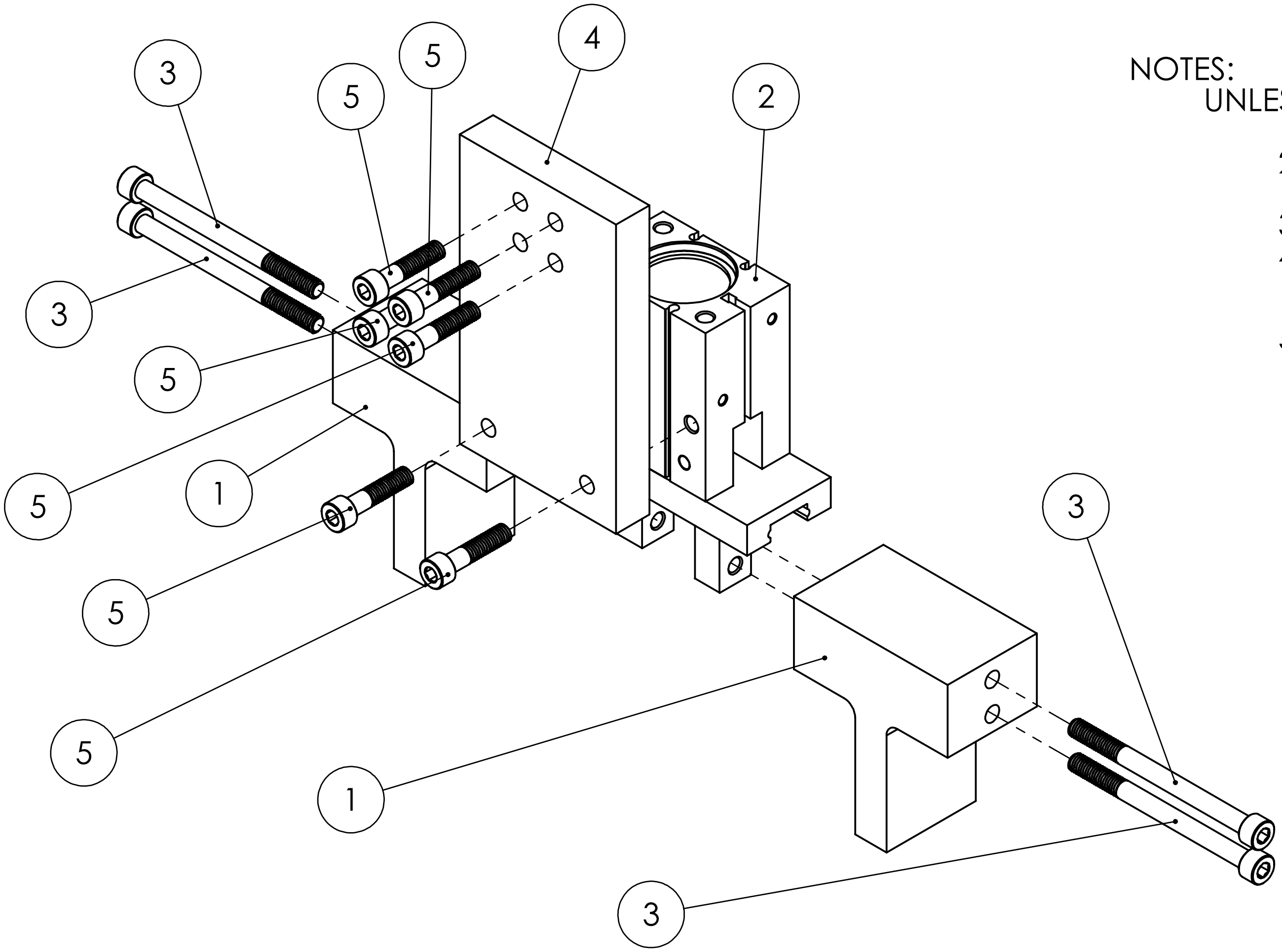
$\sigma_{crushing} = \dfrac{-F}{t_1 \cdot d}$ *Failure by Crushing of the Bolt of Plate*

SOLUTION

Unit Settings: SI C kPa kJ mass deg

α = 30 [Degrees]
A_d = 0.07548 [in^2]
A_s = 0.07548 [in^2]
A_t = 0.0567 [in^2]
C = 0.1832
d = 0.31 [in]
d_w = 0.75 [in]
E_1 = 2.970E+07 [psi]
F = 40 [lbf]
F_b = 5105 [lbf]
F_i = 5103 [lbf]
F_m = -5095 [lbf]
I = 0.0004533 [in^4]
k_b = 2.016E+06 [lbf/in]
k_m = 8.989E+06 [lbf/in]
L = 1.38 [in]
l_d = 0.28 [in]
l_{grip} = 0.905 [in]
L_t = 1.1 [in]
l_{tgrip} = 0.625 [in]
N = 4 [-]
n_l = 928.6
n_o = 624.7
$\boxed{n_p = 1.333}$
$\boxed{n_{total} = 1.333}$
P = 10 [lbf]
P_b = 1.832 [lbf]
P_h = 6 [in]
P_m = 8.168 [lbf]
P_{total} = 40 [lbf]
P_w = 3.5 [in]
σ_b = 90032 [psi]
σ_{ba} = 5129 [psi]
$\sigma_{crushing}$ = -172 [psi]
σ_{plate} = 1.933 [psi]
σ' = 90037 [psi]
S_p = 120000 [psi]
t = 0 [in]
$\boxed{\tau = 530 \text{ [psi]}}$
t_1 = 0.75 [in]
t_2 = 0.63 [in]

No unit problems were detected.

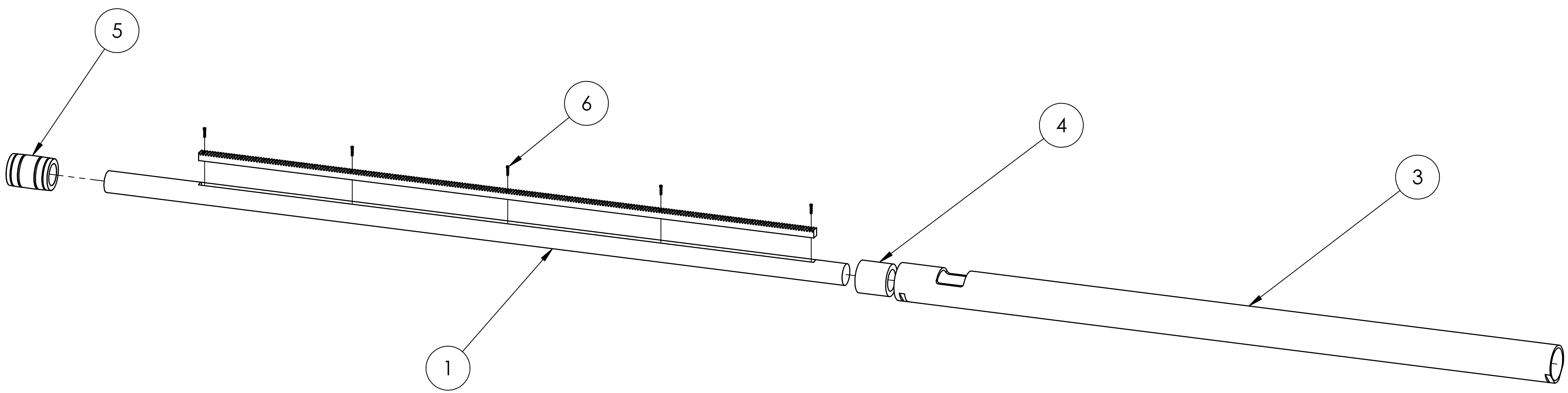


NOTES:
UNLESS OTHERWISE SPECIFIED:
1. ALL DIMENSIONS IN INCHES
2. TOLERANCES
x.xx=±.01
3. BREAK SHARP EDGES .03 MAX
4. INSIDE TOOL RADIUS .01 MAX
5. 63

ITEM NO.	PART NUMBER	DESCRIPTION	Exploded View/QTY.
1	6in Part Finger		2
2	Gripper		1
3	91290A468		4
4	Spacer Plate		1
5	91502A199		6

Cal Poly Mechanical Engineering	Lab Section: 02	Assignment #	Title: Gripper Assembly	Drwn. By: Samuel Adler
ME 429 - SPR 2016	Dwg. #: G4	Nxt Asb:	Date: 5/2/2016	Scale: 1:3	Chkd. By: ME STAFF

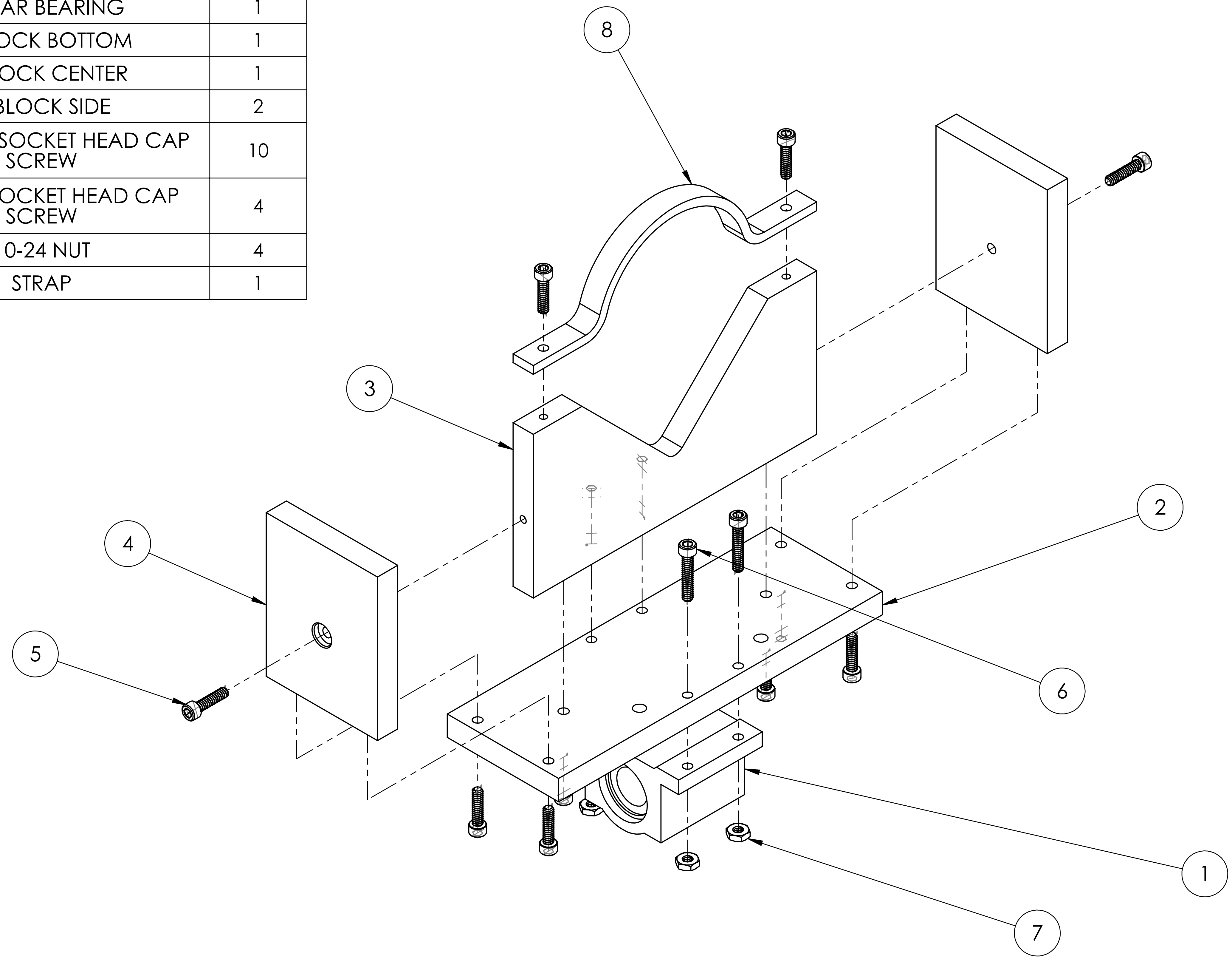
ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	R22-TGP	TELESCOPING ARM	1
2	6295K143	RACK	1
3	T231 2250	BASE TUBE	1
4	TNYLNSM	NYLATRON NSM BEARING	1
5	9533T9	LINEAR SLEEVE BEARING	1
6	91251A151	6-32 SOCKET HEAD CAP SCREW	5



Cal Poly Mechanical Engineering	Lab Section: 02	SENIOR PROJECT	Title: TELESCOPING SUBASSEMBLY		Drwn. By: LOUIS ROSEGUO
ME 429 - SPRING 2016	Dwg. #: T-SCOPE	Nxt Asb: TS-COM	Date: 5/3/2016	Scale: 1:10	Chkd. By: ME STAFF

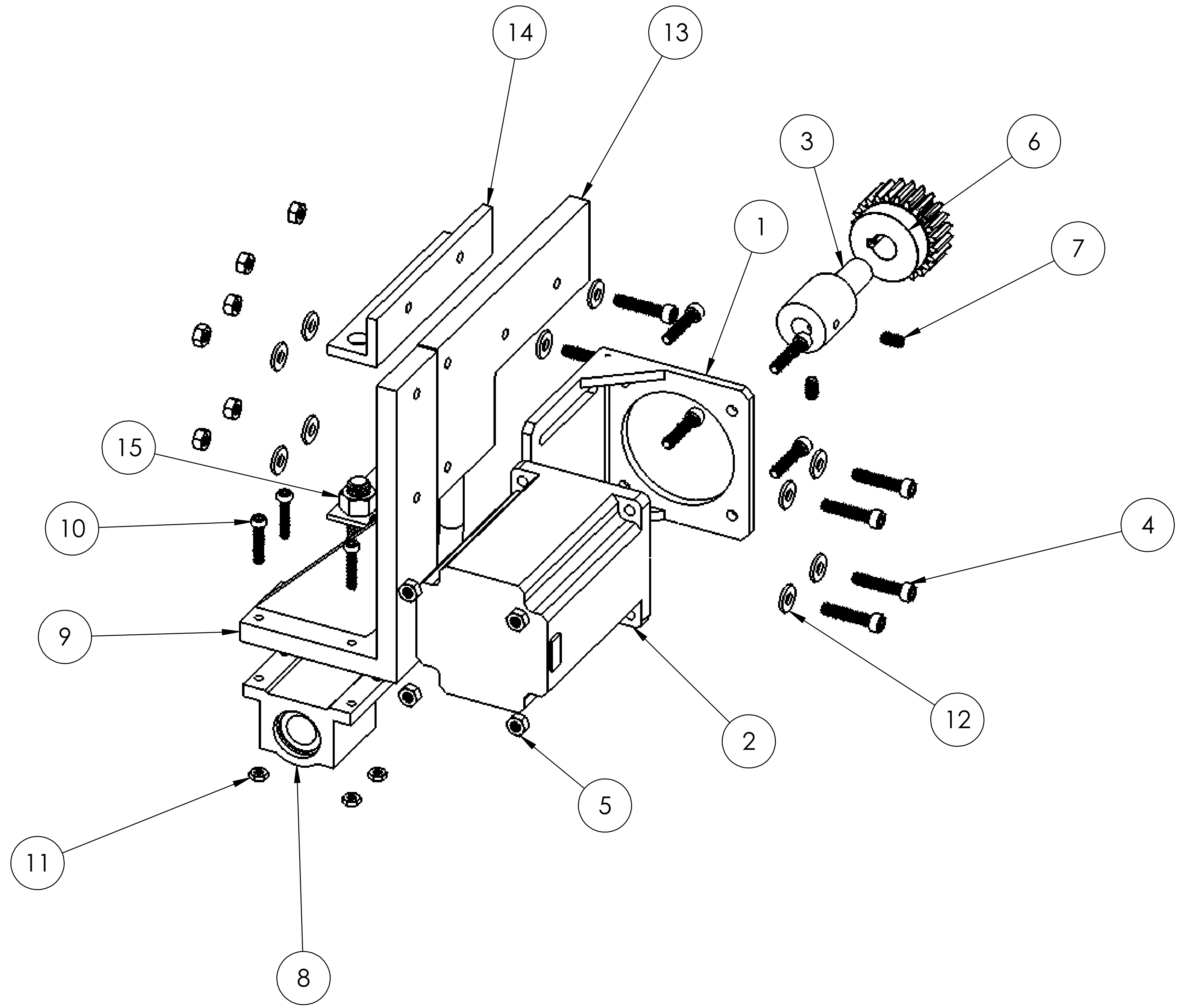
M3

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	6374K132	LINEAR BEARING	1
2	8910K702	V BLOCK BOTTOM	1
3	9517K531	V BLOCK CENTER	1
4	8910K21	V BLOCK SIDE	2
5	91251A245	10-24 3/4" SOCKET HEAD CAP SCREW	10
6	91251A247	10-24 1" SOCKET HEAD CAP SCREW	4
7	90480A011	10-24 NUT	4
8	9018K11	STRAP	1



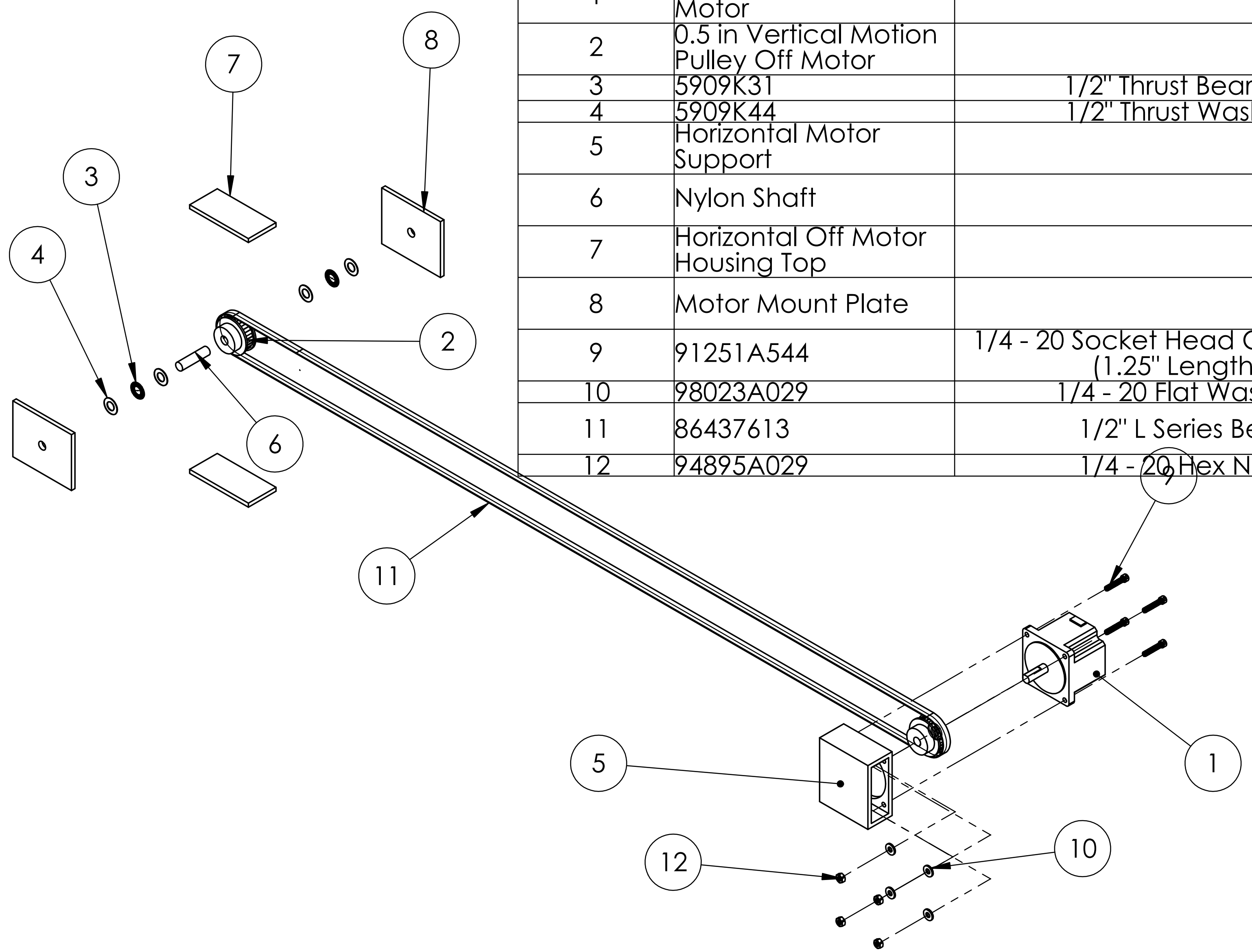
Cal Poly Mechanical Engineering	Lab Section: 02	SENIOR PROJECT	Title: V BLOCK AND BEARING	Drwn. By: LOUIS ROSEGUO
ME 429 - SPRING 2016	Dwg. #: VS01	Nxt Asb: H-MOTR	Date: 5/3/2016 / Scale:	Chkd. By: ME STAFF

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	ST-M7	NEMA 34 MOTOR MOUNT	1
2	HT34-506	HT34-506 STEPPER MOTOR	1
3	6776T26	SHAFT ADAPTER	1
4	91251A544	1/4-20 X 1 1/4" SOCKET HEAD CAP SCREW	10
5	94895A029	1/4"-20 HEX NUT	10
6	6867K61	PINION	1
7	92158A432	SET SCREW	2
8	6374K132	LINEAR BEARING	1
9	8910K949	MOTOR BASE BOTTOM	1
10	91251A247	10-24 1" SOCKET HEAD CAP SCREW	4
11	90480A011	10-24 NUT	4
12	98023A029	1/4" WASHER	10
13	MH01	MOTOR BASE HANGER	1
14	9017K76	ANGLE IRON	1
15	3043T89	U-BOLT	1



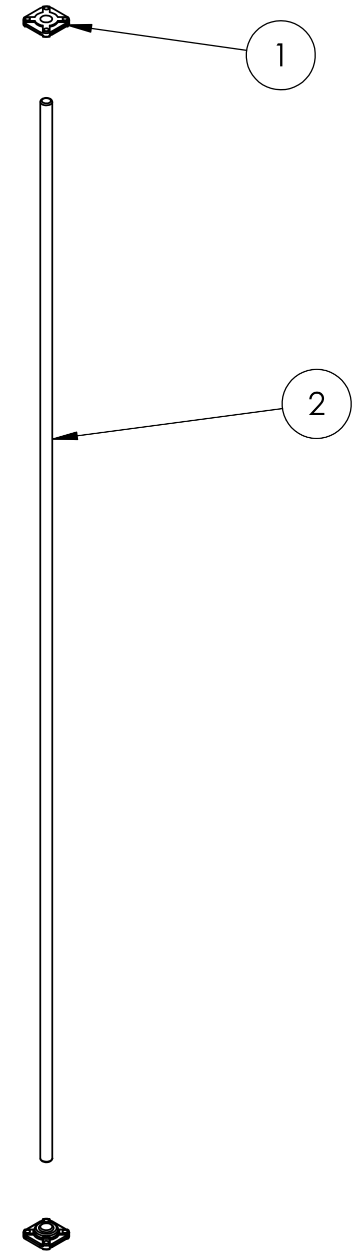
Cal Poly Mechanical Engineering	Lab Section: 02	SENIOR PROJECT	Title: MOTOR MOUNT ASSEMBLY	Drwn. By: LOUIS ROSEGUO
ME 429 - SPRING 2016	Dwg. #: MM01	Nxt Asb: T-SCOPE	Date: 5/3/2016 Scale: 1:3	Chkd. By: ME STAFF

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	OMHT34 504 Stepper Motor		1
2	0.5 in Vertical Motion Pulley Off Motor		2
3	5909K31	1/2" Thrust Bearing	3
4	5909K44	1/2" Thrust Washer	6
5	Horizontal Motor Support		1
6	Nylon Shaft		1
7	Horizontal Off Motor Housing Top		2
8	Motor Mount Plate		2
9	91251A544	1/4 - 20 Socket Head Cap Screw (1.25" Length)	4
10	98023A029	1/4 - 20 Flat Washer	4
11	86437613	1/2" L Series Belt	1
12	94895A029	1/4 - 20 Hex Nut	4



Cal Poly Mechanical Engineering ME 429 - SPRING 2016	Lab Section:02	SENIOR PROJECT	Title:HORIZONTAL MOTION ASSEMBLY	Drwn. By:RYAN CANFIELD
	Dwg. #:	Nxt Asb:	Date:5/2/2016 / Scale:1:8	Chkd. By: ME STAFF

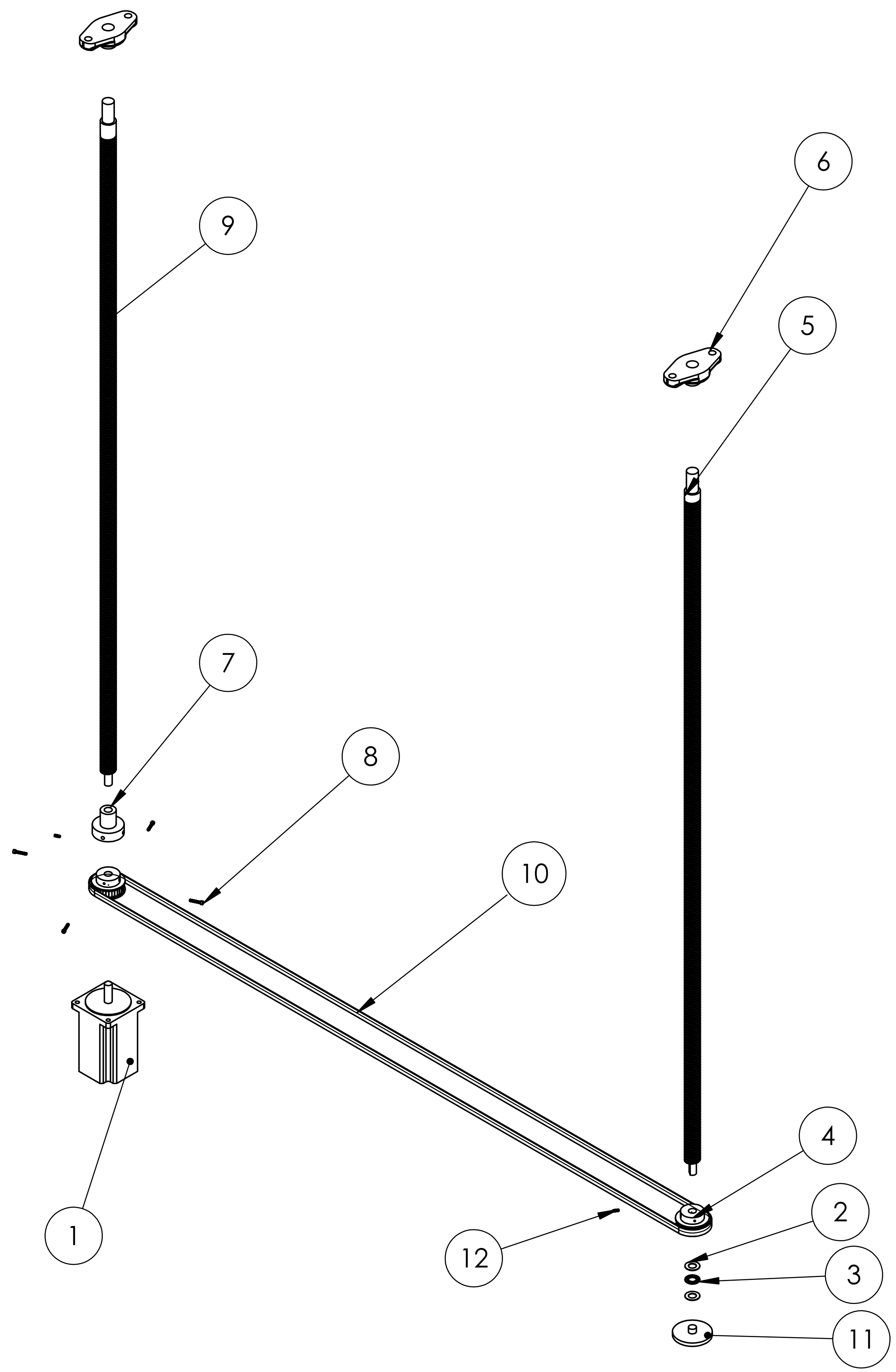
ITEM NO.	PART NUMBER	DESCRIPTION	Exploded View/QTY.
1	EFSI - 08	Flange Vertical Shaft Mount	2
2	SN 4 X 28	Vertical Linear Guide Shafts	1



Cal Poly Mechanical Engineering	Lab Section:02	SENIOR PROJECT	Title:ACME NUT HOUSING	Drwn. By:RYAN CANFIELD
ME 429 - SPRING 2016	Dwg. #:	Nxt Asb:	Date:5/2/2016	Scale:1:10	Chkd. By: ME STAFF

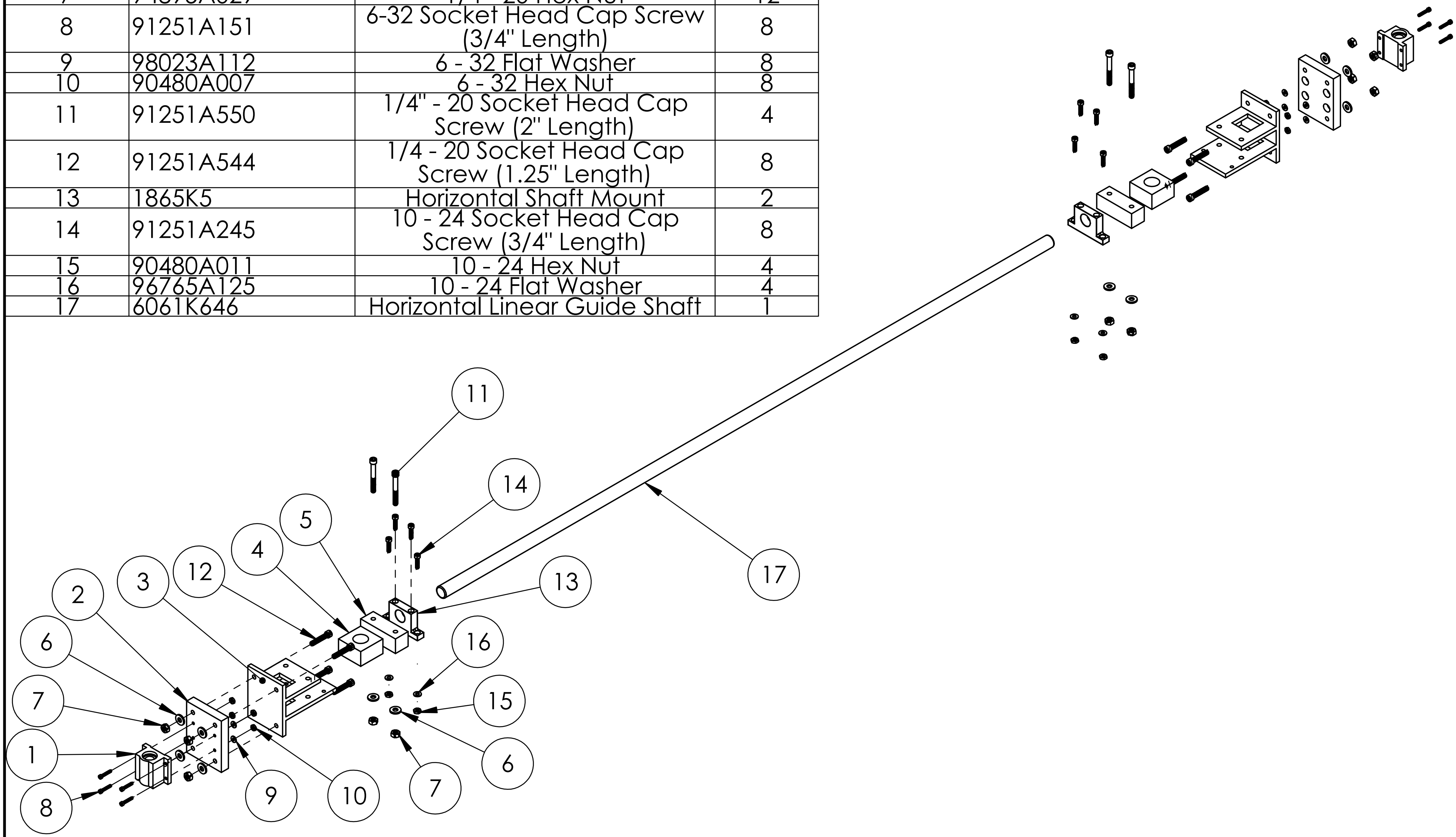
M7

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	OMHT 34 - 506	VERTICAL STEPPER MOTOR	1
2	5909K44	1/2" Thrust Washer	4
3	5909K31	1/2" Thrust Bearing	2
4	86437613	L Series Timing Belt Pulley	2
5	59286	ACME 1" - 5 Lead Screw (54" Length Off Motor Side)	1
6	UCF204-12	Lead Screw End support	2
7	8927K58	Lead Screw Motor Coupler	1
8	91251A151	6-32 Socket Head Cap Screw (3/4" Length)	4
9	59286	ACME 1" - 5 Lead Screw (52.5" Length Motor Side)	1
10	6484K167	L Series Timing Belt (90" Length)	1
11	8927K58	Vertical Shaft Base	1
12	92158A171	6 - 32 Set Screw (3/8" Length)	2

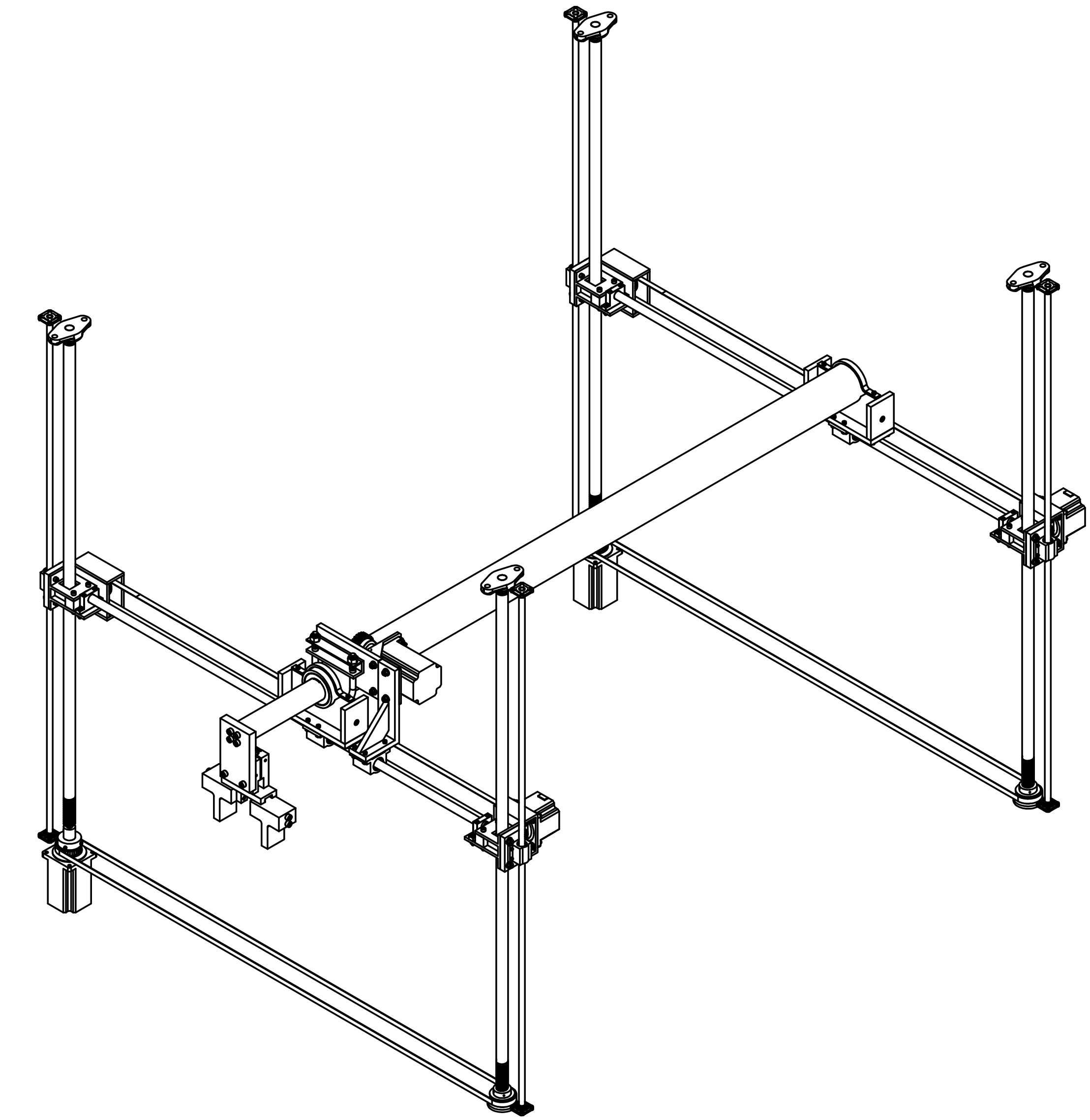


Cal Poly Mechanical Engineering	Lab Section: 02	SENIOR PROJECT	Title: VERTICAL ACTUATOR ASSEM.	Drwn. By:RYAN CANFIELD
ME ### - QTR YEAR	Dwg. #:	Nxt Asb:	Date: 5/3/16 / Scale:1:10	Chkd. By: ME STAFF

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	6374K125	Mounted Linear Sleeve Bearing	2
2	8975K226	Vertical Weld Plate	2
3	8975K87	Lead Screw Nut Housing	2
4	89422	Square Lead Screw Nut	2
5	9143K21	Spacer Block	2
6	98023A029	1/4 - 20 Flat Washer	12
7	94895A029	1/4 - 20 Hex Nut	12
8	91251A151	6-32 Socket Head Cap Screw (3/4" Length)	8
9	98023A112	6 - 32 Flat Washer	8
10	90480A007	6 - 32 Hex Nut	8
11	91251A550	1/4" - 20 Socket Head Cap Screw (2" Length)	4
12	91251A544	1/4 - 20 Socket Head Cap Screw (1.25" Length)	8
13	1865K5	Horizontal Shaft Mount	2
14	91251A245	10 - 24 Socket Head Cap Screw (3/4" Length)	8
15	90480A011	10 - 24 Hex Nut	4
16	96765A125	10 - 24 Flat Washer	4
17	6061K646	Horizontal Linear Guide Shaft	1

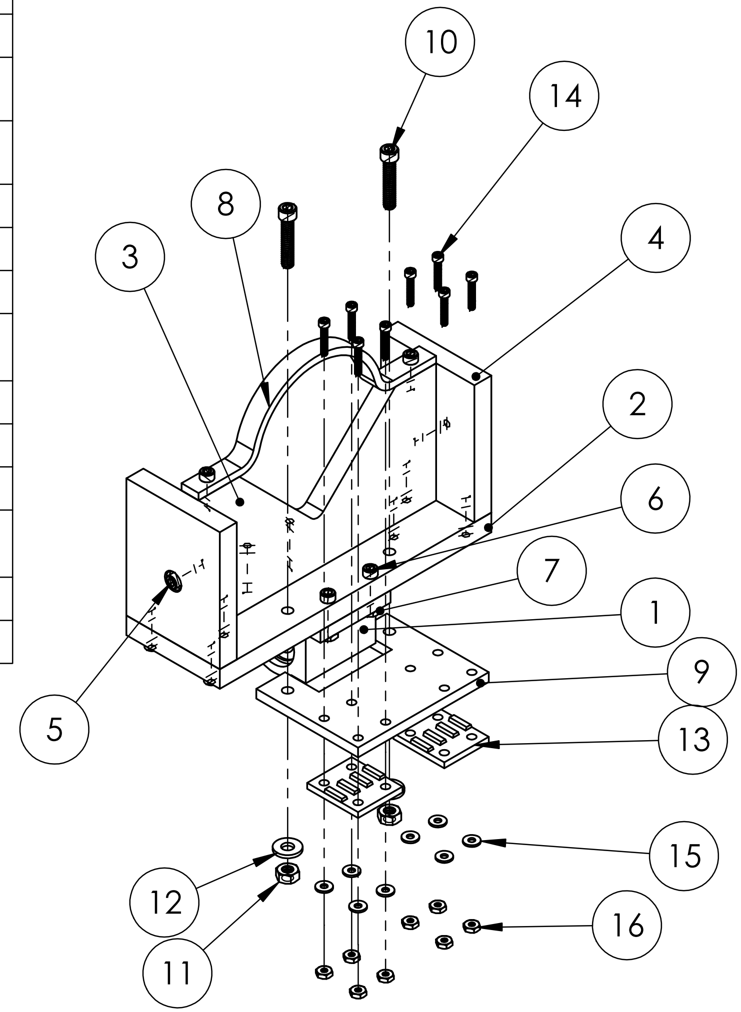


Cal Poly Mechanical Engineering	Lab Section:02	SENIOR PROJECT	Title:ACME NUT HOUSINGASSEMBLY		Drwn. By:RYAN CANFIELD
ME 429 - SPRING 2016	Dwg. #:	Nxt Asb:	Date:5/2/2016	Scale:1:2	Chkd. By: ME STAFF



Cal Poly Mechanical Engineering	Lab Section:02	Assignment #	Title:FULL ASSEMBLY DRAWING		Drwn. By:RYAN CANFIELD
ME 429 -SPRING 2016	Dwg. #:	FINAL ASSEMBLY	Date:5/4/16	Scale: 1:16	Chkd. By: ME STAFF

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	6374K132	V BLOCK LINEAR SLEEVE BEARING	1
2	V Block Bottom		1
3	V Block Center		1
4	V Block Side		2
5	91251A245	10 - 24 Socket Head Cap Screw (3/4" Length)	10
6	91251A247	10 - 24 Socket Head Cap Screw (1" Length)	4
7	90480A011	10 - 24 Hex Nut	4
8	Strap		1
9	Belt Adaptor		1
10	91251A544	1/4 - 20 Socket Head Cap Screw (1.25" Length)	2
11	94895A029	1/4 - 20 Hex Nut	2
12	98023A029	1/4 - 20 Flat Washer	2
13	Belt Holder		2
14	91251A151	6-32 Socket Head Cap Screw (3/4" Length)	8
15	98023A112	6 - 32 Flat Washer	8
16	90480A007	6 - 32 Hex Nut	8

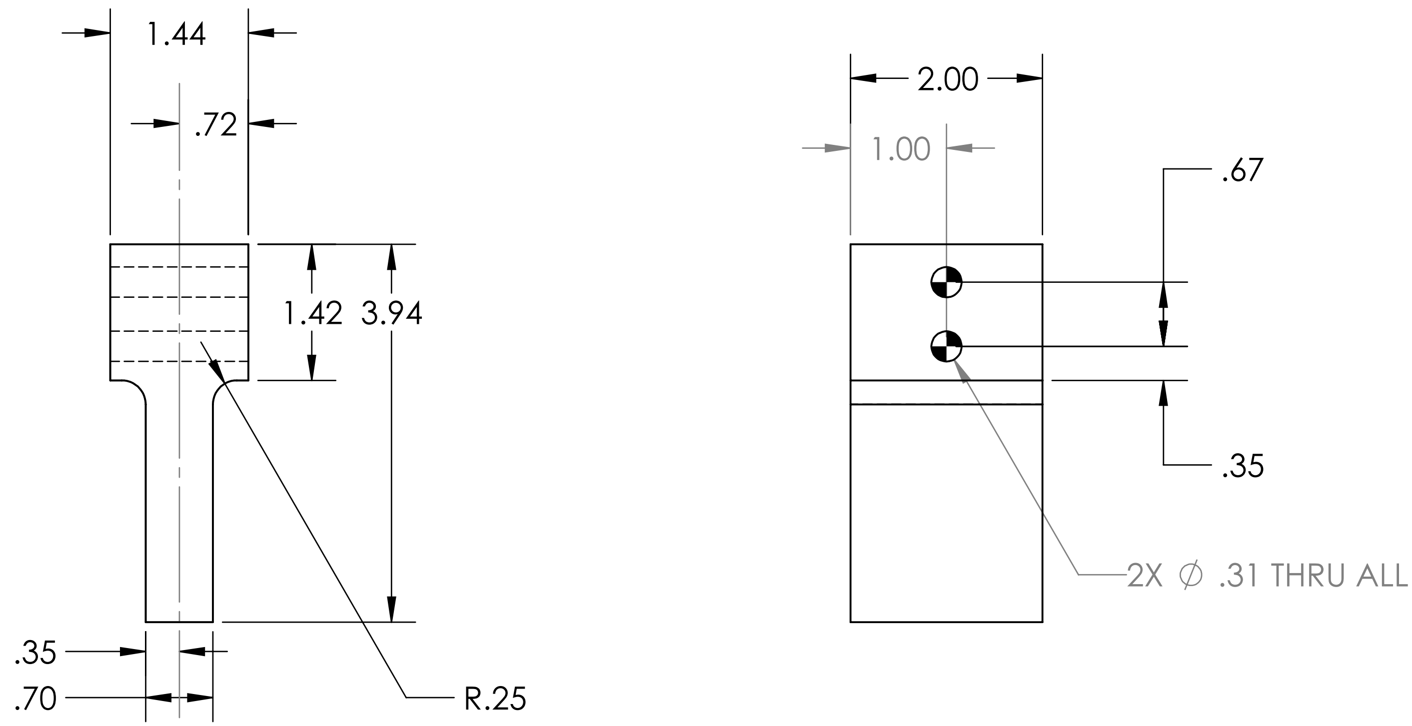


Cal Poly Mechanical Engineering	Lab Section:02	SENIOR PROJECT	Title:HORIZONTAL BELT ATTACHMENT	Drwn. By:RYAN CANFIELD
ME 429 - SPRING 2016	Dwg. #:	Nxt Asb:	Date:5/3/2016 Scale:	Chkd. By: ME STAFF

NOTES:

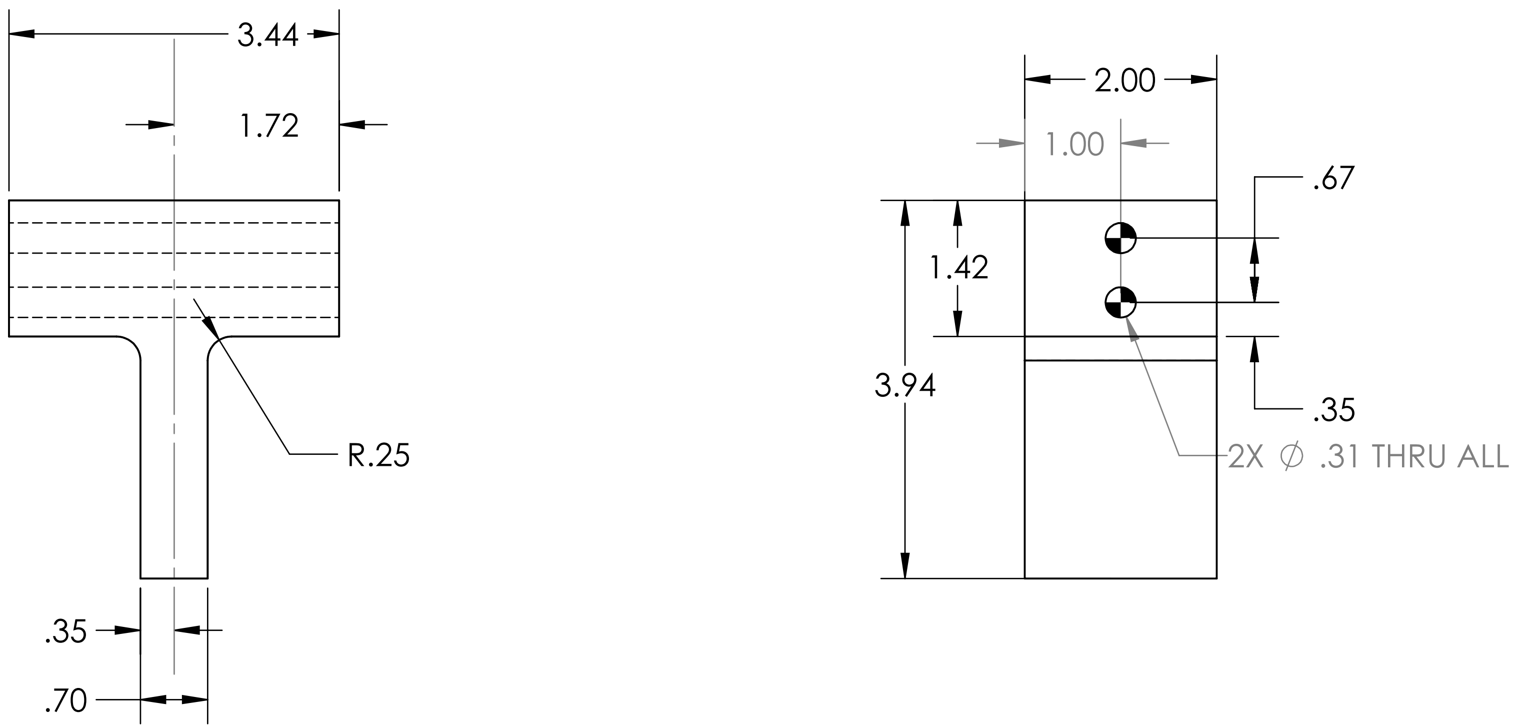
UNLESS OTHERWISE SPECIFIED:

1. ALL DIMENSIONS IN INCHES
2. TOLERANCES
   x.xx=±.01
3. BREAK SHARP EDGES .03 MAX
4. INSIDE TOOL RADIUS .01 MAX
5. $\sqrt{}$ 63



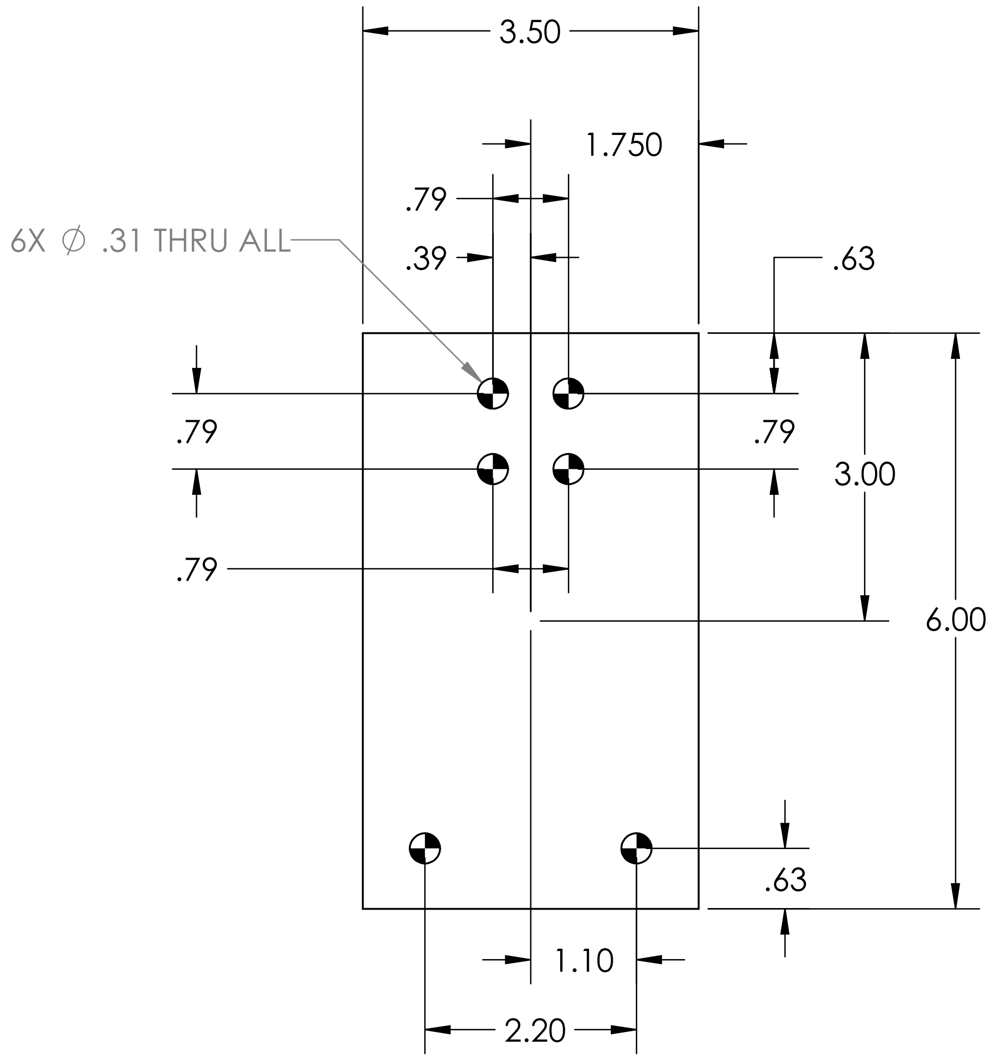
Cal Poly Mechanical Engineering	Lab Section: 02	Assignment #	Title: 4in Part Finger	Drwn. By: Samuel Adler
ME 429 - SPR 2016	Dwg. #: G2	Nxt Asb:	Date: 5/2/16 Scale: 1:2	Chkd. By: ME STAFF

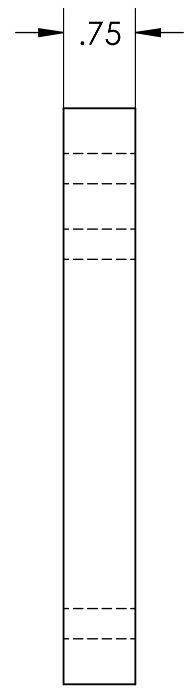
NOTES:
UNLESS OTHERWISE SPECIFIED:
1. ALL DIMENSIONS IN INCHES
2. TOLERANCES
x.xx=±.01
3. BREAK SHARP EDGES .03 MAX
4. INSIDE TOOL RADIUS .01 MAX
5. 63 √



Cal Poly Mechanical Engineering	Lab Section: 02	Assignment #	Title: 6in Part Finger	Drwn. By: Samuel Adler
ME 429 - SPR 2016	Dwg. #: G1	Nxt Asb:	Date: 5/2/16 / Scale: 1:2	Chkd. By: ME STAFF

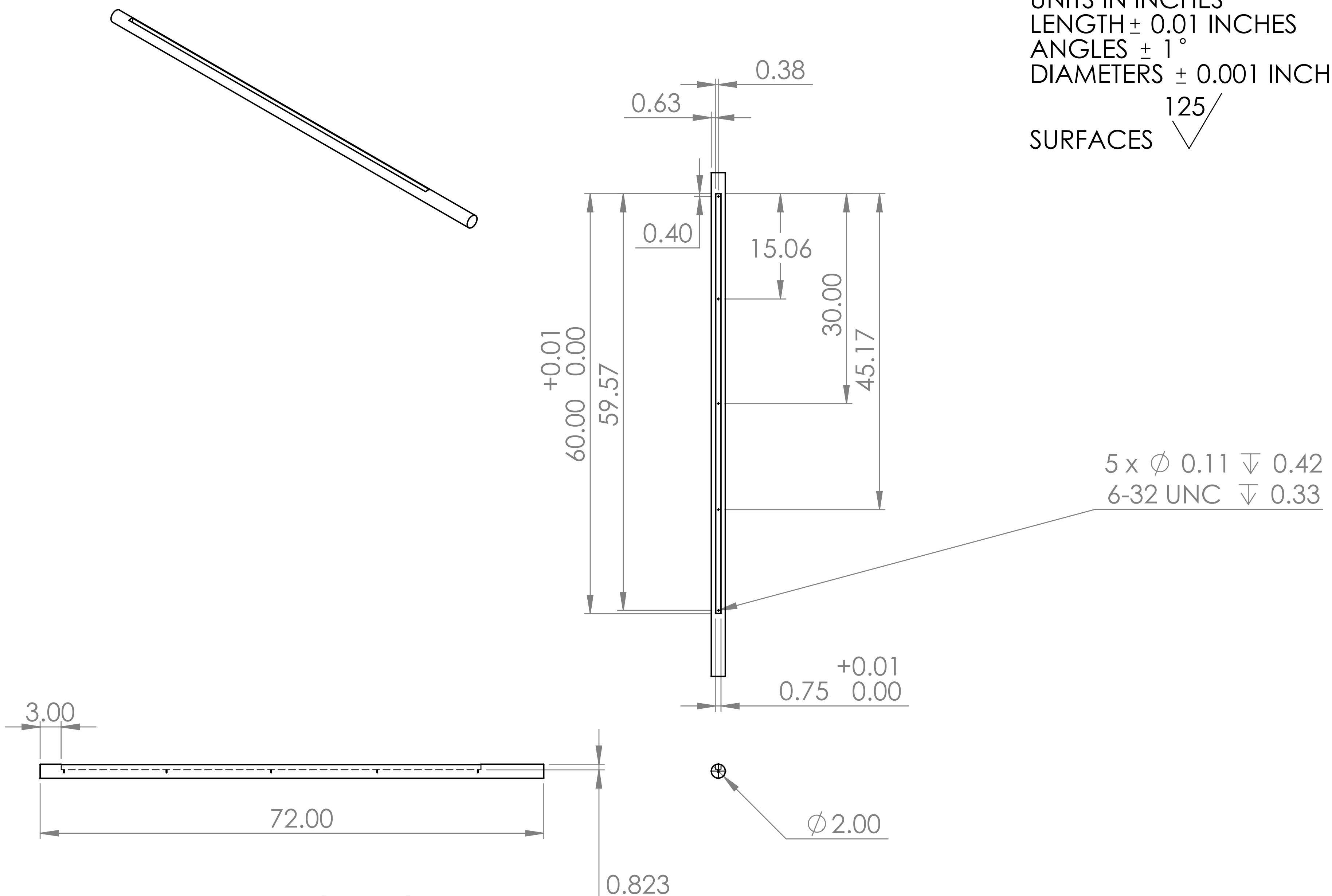
NOTES:
UNLESS OTHERWISE SPECIFIED:
1. ALL DIMENSIONS IN INCHES
2. TOLERANCES
x.xx=±.01
3. BREAK SHARP EDGES .03 MAX
4. INSIDE TOOL RADIUS .01 MAX
5. 63√





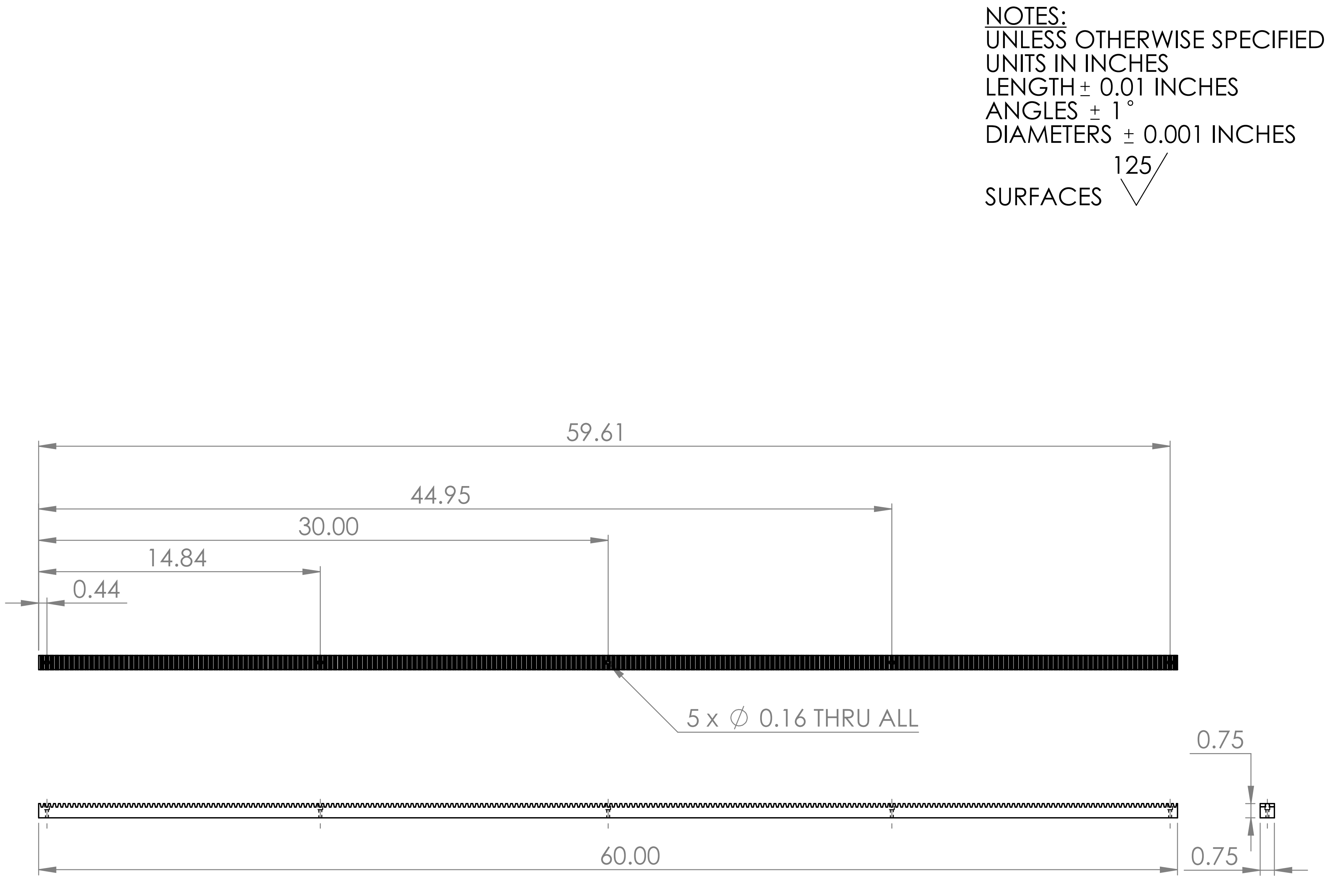
Cal Poly Mechanical Engineering	Lab Section: 02	Assignment #	Title: Spacer Plate	Drwn. By: Samuel Adler
ME 429 - SPR 2016	Dwg. #: G3	Nxt Asb:	Date: 5/2/16 · Scale: 1:2	Chkd. By: ME STAFF

NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH ± 0.01 INCHES
ANGLES ± 1°
DIAMETERS ± 0.001 INCHES
SURFACES 125



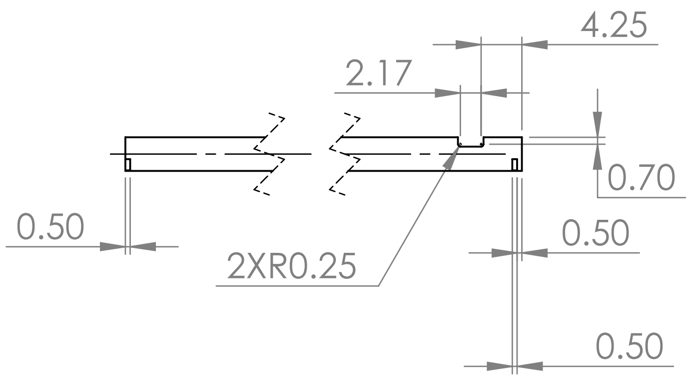
Cal Poly Mechanical Engineering	Lab Section: 02	SENIOR PROJECT	Title: TELESCOPING ARM	Drwn. By: LOUIS ROSEGUO
ME 429 - Spring 2016	Dwg. #: ARM1	Nxt Asb: ARM	Date: 5/3/2016 / Scale: 1:20	Chkd. By: ME STAFF

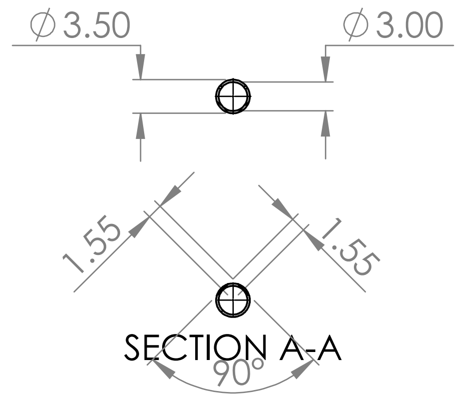
N5



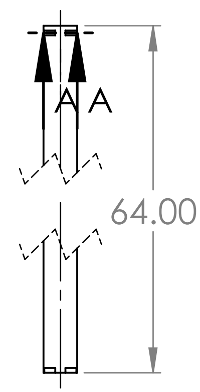
Cal Poly Mechanical Engineering	Lab Section: 02	SENIOR PROJECT	Title: RACK	Drwn. By: LOUIS ROSEGUO
ME 429 - Spring 2016	Dwg. #:629K5143	Nxt Asb:ARM	Date: 5/2/2016	Scale: 1:7	Chkd. By: ME STAFF

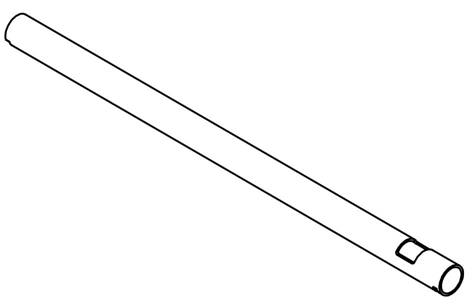
N6





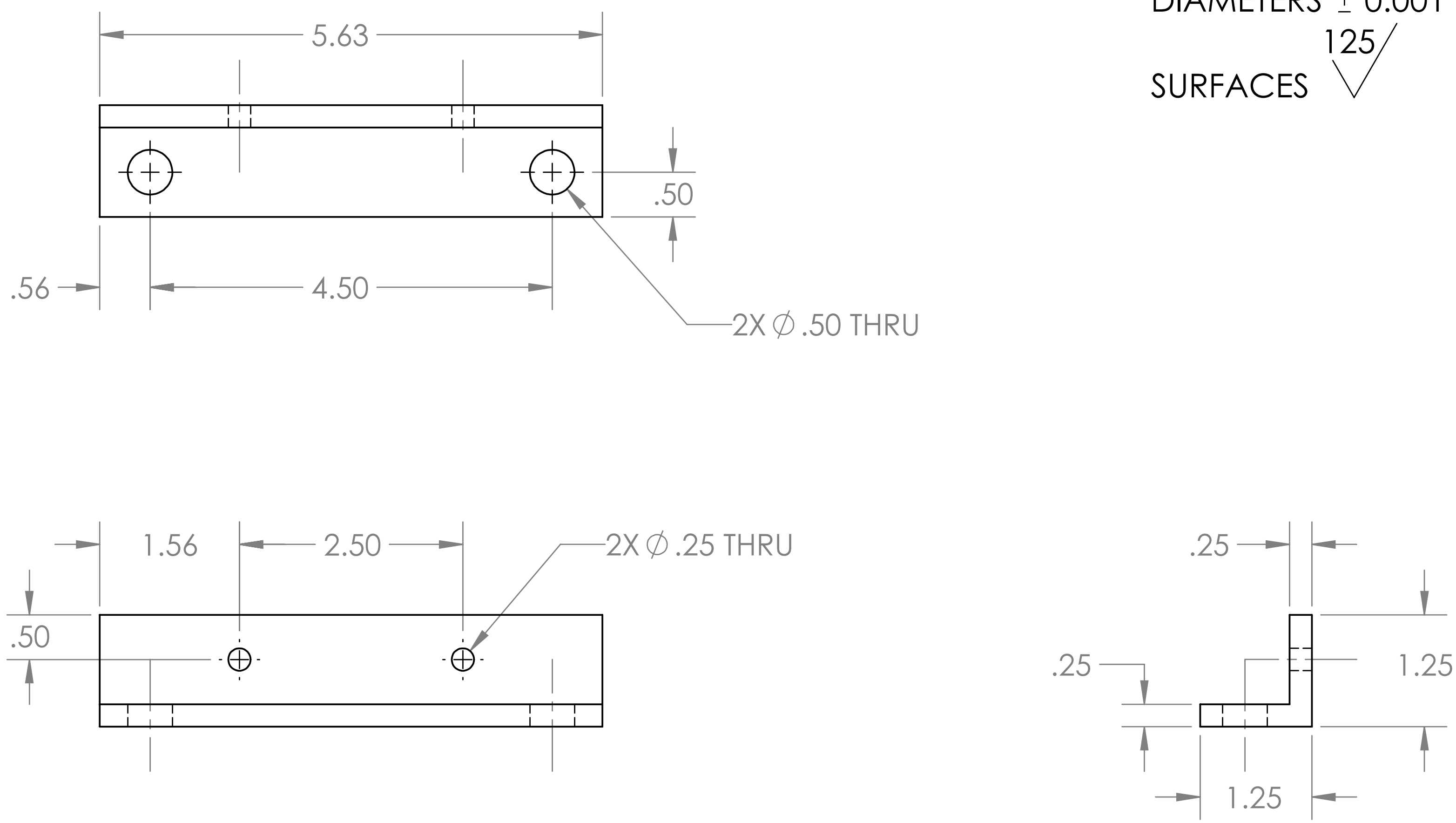
NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH ± 0.01 INCHES
ANGLES ± 1°
DIAMETERS ± 0.001 INCHES
125
SURFACES





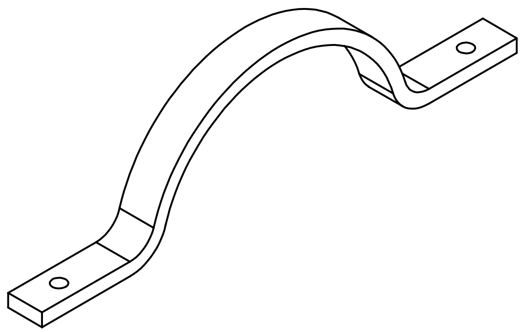
Cal Poly Mechanical Engineering	Lab Section: 02	SENIOR PROJECT	Title: BASE TUBE		Drwn. By: LOUIS ROSEGUO
ME 429 - Spring 2016	Dwg. #: T2312250	Nxt Asb: T-SCOPE	Date: 5/3/2016	Scale: 1:20	Chkd. By: ME STAFF

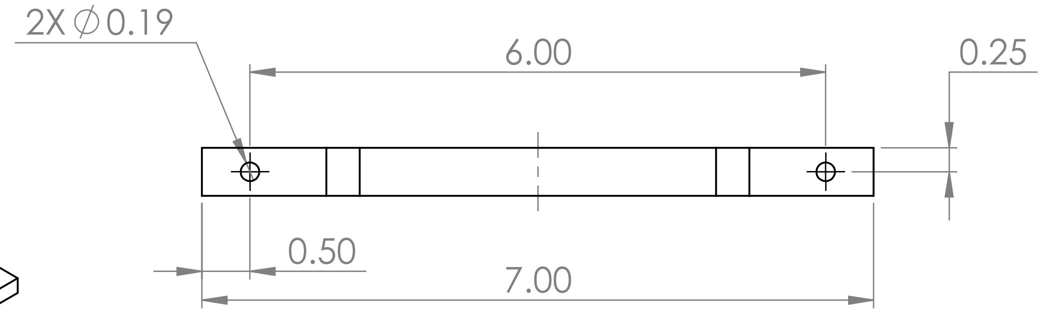
NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH ± 0.01 INCHES
ANGLES ± 1°
DIAMETERS ± 0.001 INCHES
125
SURFACES

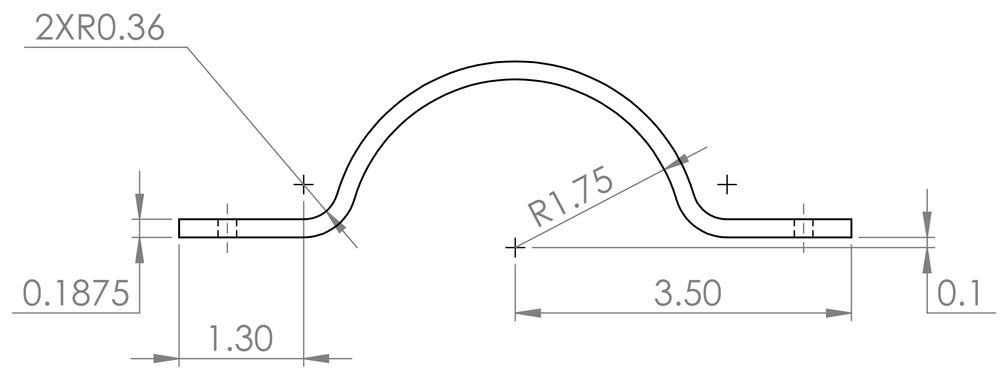


Cal Poly Mechanical Engineering	Lab Section: 02	SENIOR PROJECT	Title: ANGLE IRON		Drwn. By: LOUIS ROSEGUO
ME 429 - Spring 2016	Dwg. #: 9017K76	Nxt Asb: MOUNT	Date: 5/2/2016	Scale: 1:2	Chkd. By: ME STAFF

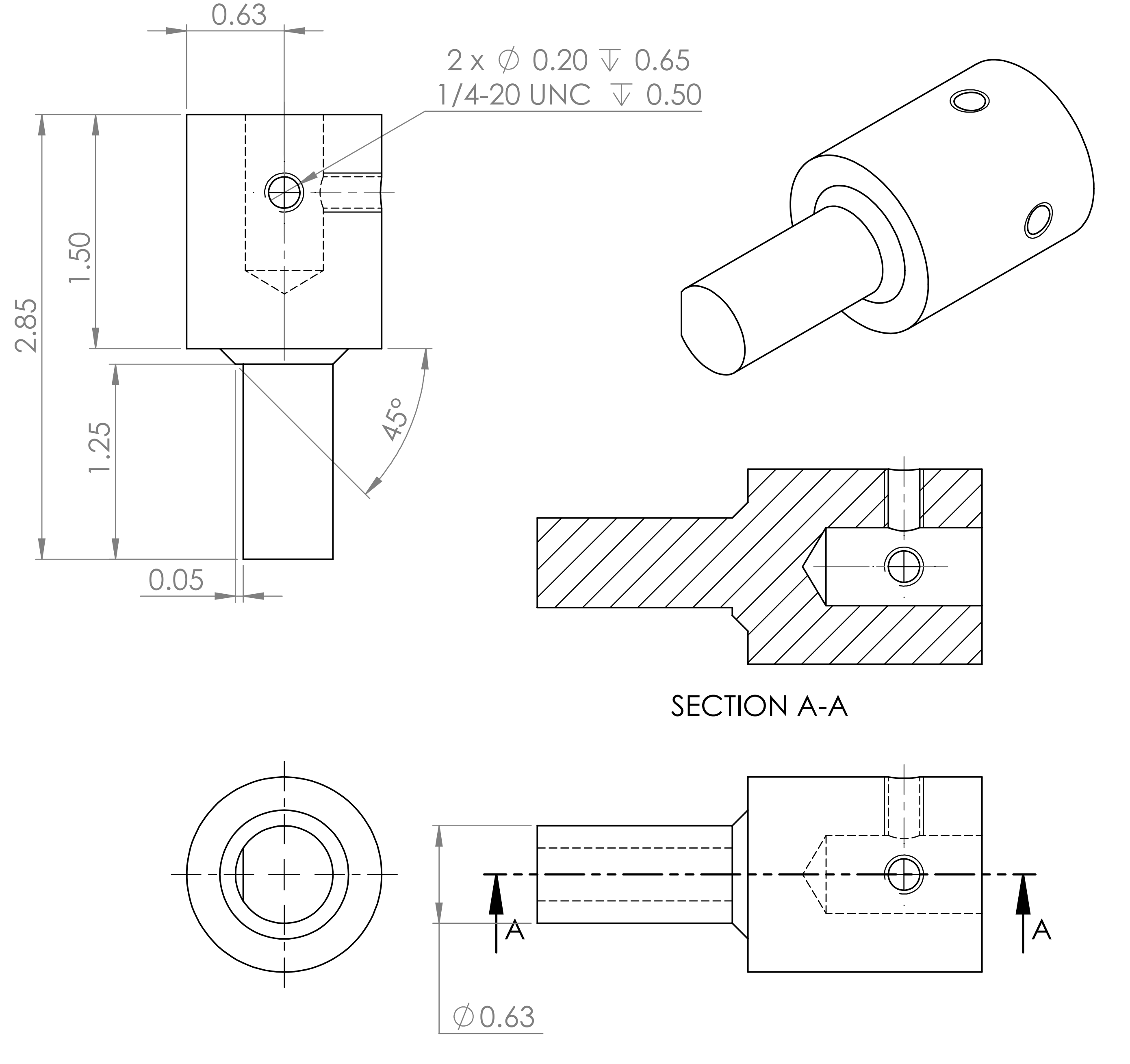
NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH ± 0.01 INCHES
ANGLES ± 1°
DIAMETERS ± 0.001 INCHES
SURFACES 125√



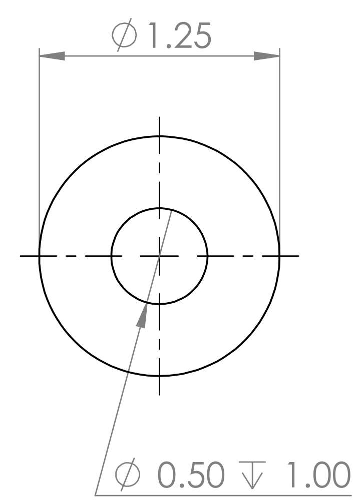




Cal Poly Mechanical Engineering Lab	Section: 02	SENIOR PROJECT	Title: STRAP		Drwn. By: LOUIS ROSEGUO
ME 429 - Spring 2016	Dwg. #: 9018K11	Nxt Asb:V-BLOCK	Date: 5/2/2016	Scale: 1:2	Chkd. By: ME STAFF

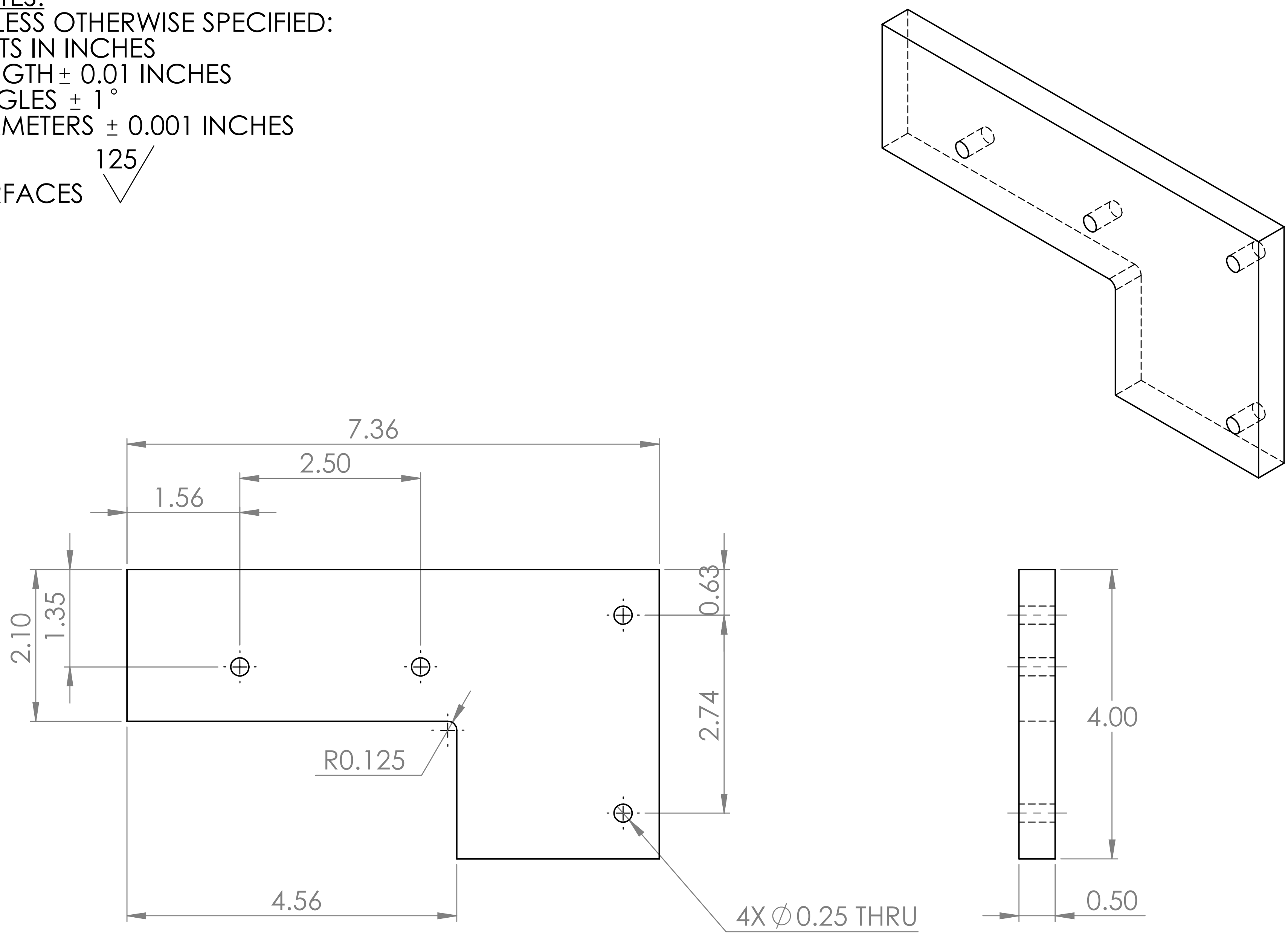


NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH ± 0.01 INCHES
ANGLES ± 1°
DIAMETERS ± 0.001 INCHES
SURFACES 125 √



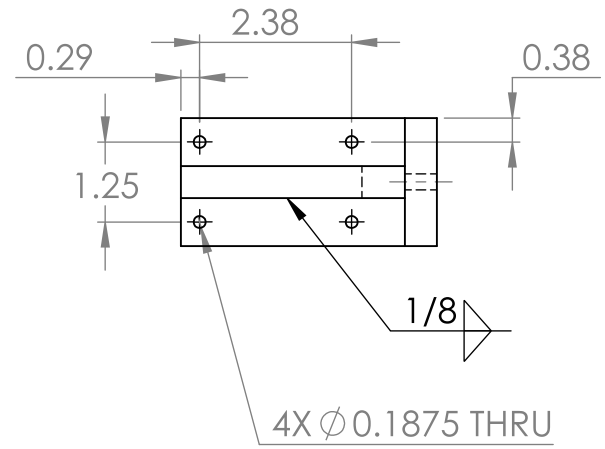
Cal Poly Mechanical Engineering ME 429 - Spring 2016	Lab Section: 02	SENIOR PROJECT	Title: SHAFT ADAPTER		Drwn. By: LOUIS ROSEGUO
	Dwg. #: 6776T26	Nxt Asb: MOUNT	Date: 5/3/2016	Scale: 1:1	Chkd. By: ME STAFF

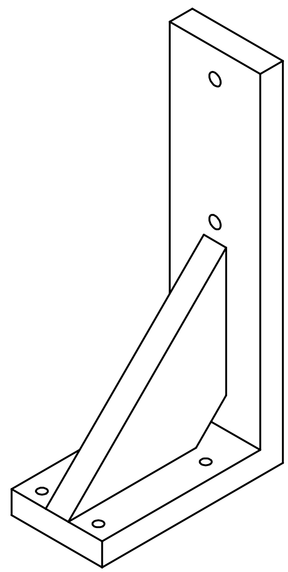
NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH ± 0.01 INCHES
ANGLES ± 1°
DIAMETERS ± 0.001 INCHES
SURFACES 125

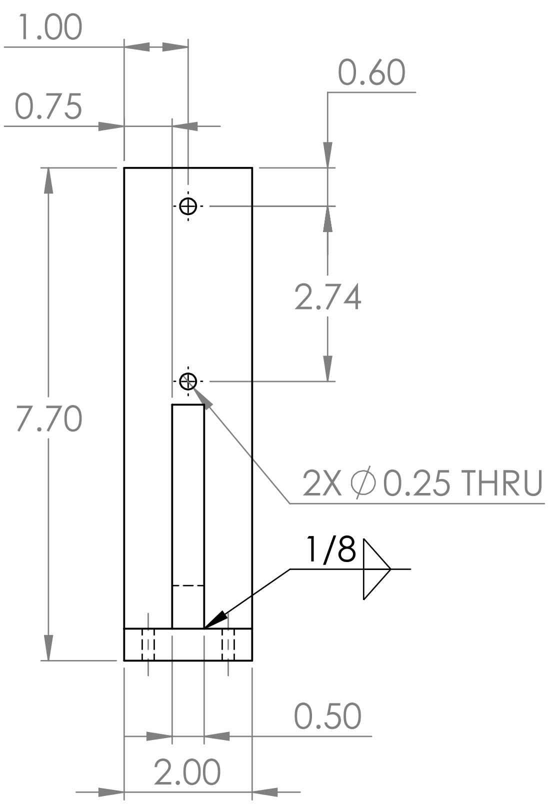


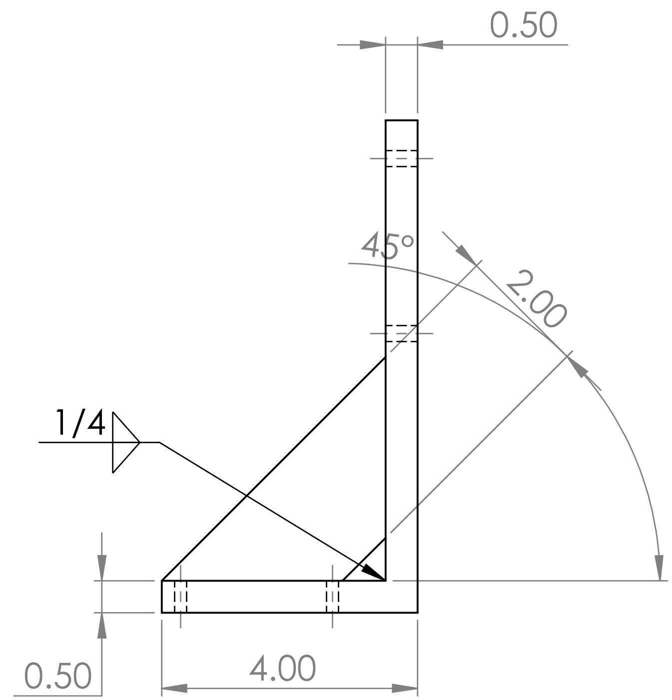
Cal Poly Mechanical Engineering	Lab Section: 02	SENIOR PROJECT	Title: MOTOR HANGER BASE		Drwn. By: LOUIS ROSEGUO
ME 429 - Spring 2016	Dwg. #: MB01	Nxt Asb: MOUNT	Date: 5/2/2016	Scale: 1:2	Chkd. By: ME STAFF

NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH ± 0.01 INCHES
ANGLES ± 1°
DIAMETERS ± 0.001 INCHES
125
SURFACES



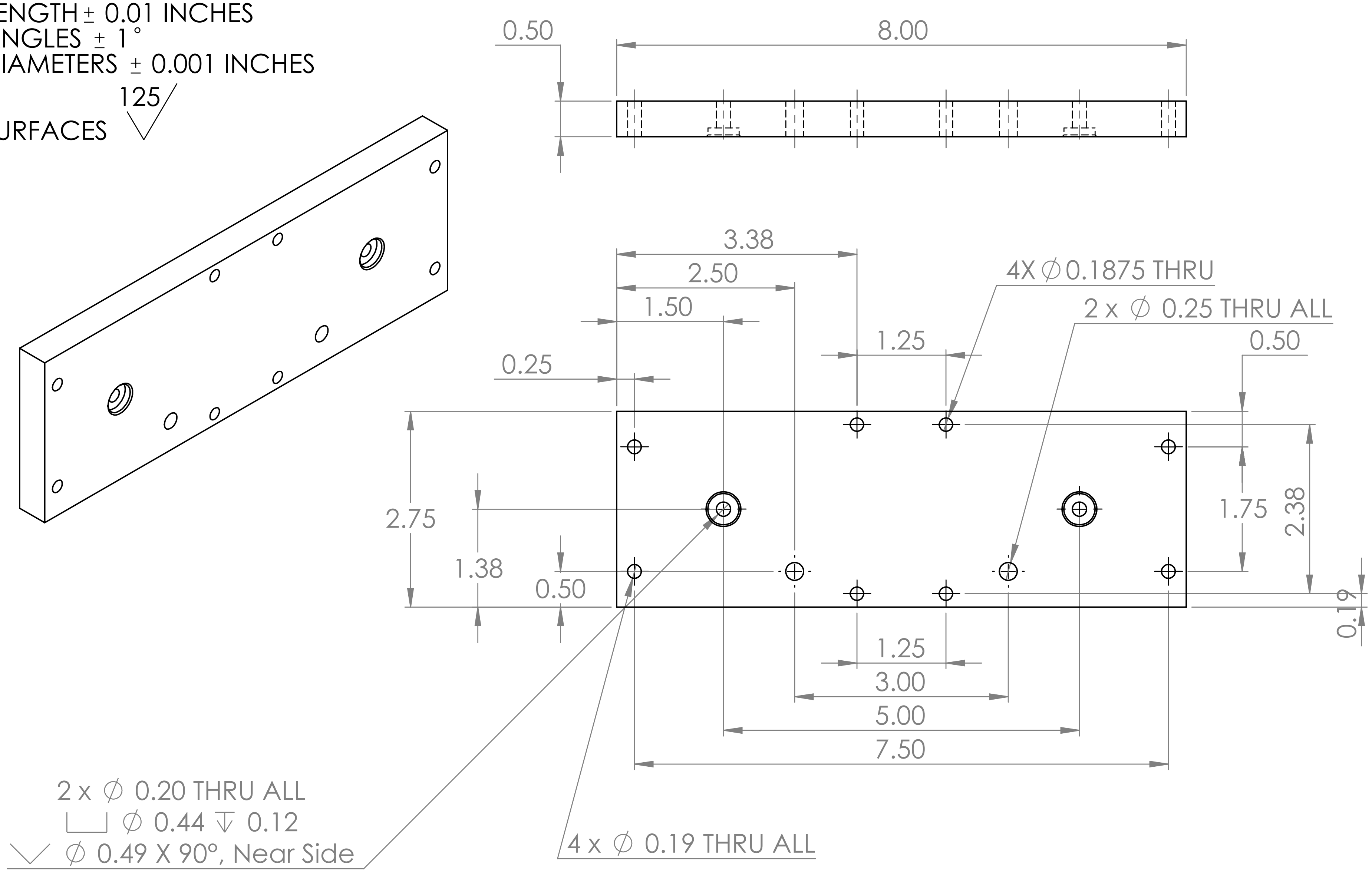






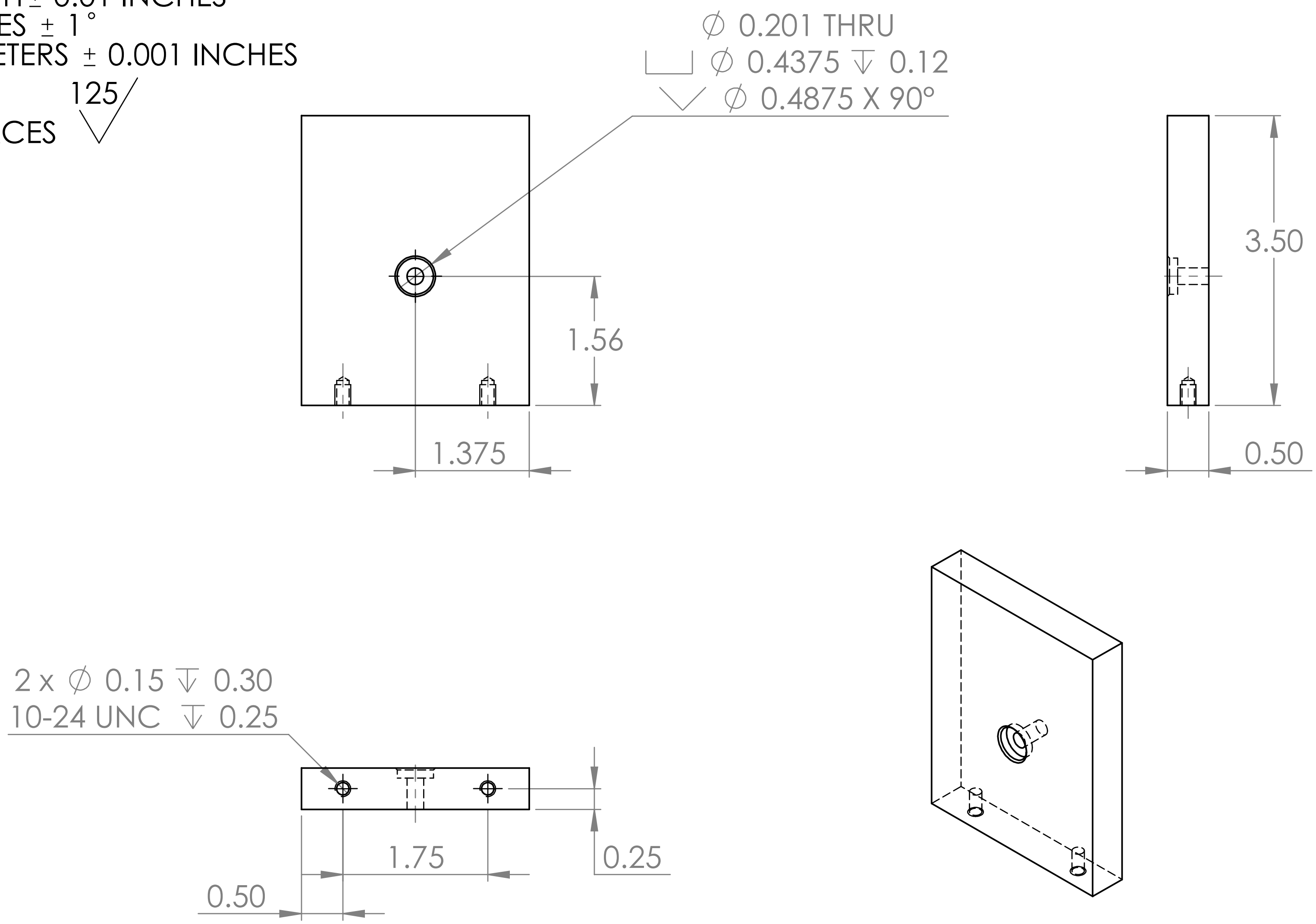
Cal Poly Mechanical Engineering	Lab Section: 02	SENIOR PROJECT	Title: MOTOR BASE BOTTOM	Drwn. By: LOUIS ROSEGUO
ME 429 - Spring 2016	Dwg. #:8910K949	Nxt Asb: MOUNT	Date: 5/2/2016 · Scale: 1:3	Chkd. By: ME STAFF

NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH ± 0.01 INCHES
ANGLES ± 1 °
DIAMETERS ± 0.001 INCHES
SURFACES 125

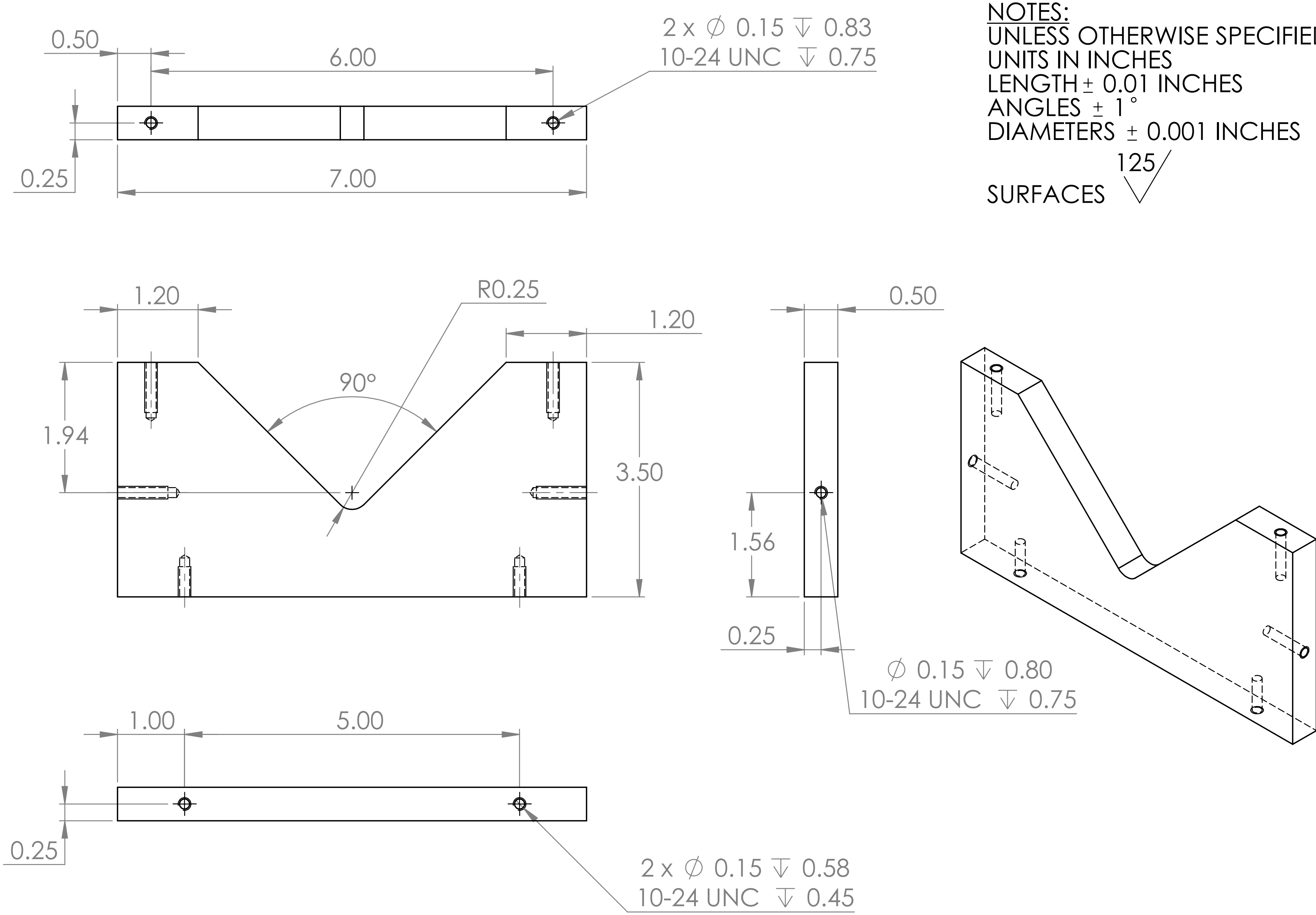


Cal Poly Mechanical Engineering Lab Section: 02		SENIOR PROJECT	Title: V-BLOCK BOTTOM	Drwn. By: LOUIS ROSEGUO
ME 429 - Spring 2016	Dwg. #: 8910K702	Nxt Asb: V-BLOCK	Date: 5/2/2016 / Scale: 1:2	Chkd. By: ME STAFF

NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH ± 0.01 INCHES
ANGLES ± 1°
DIAMETERS ± 0.001 INCHES
SURFACES 125



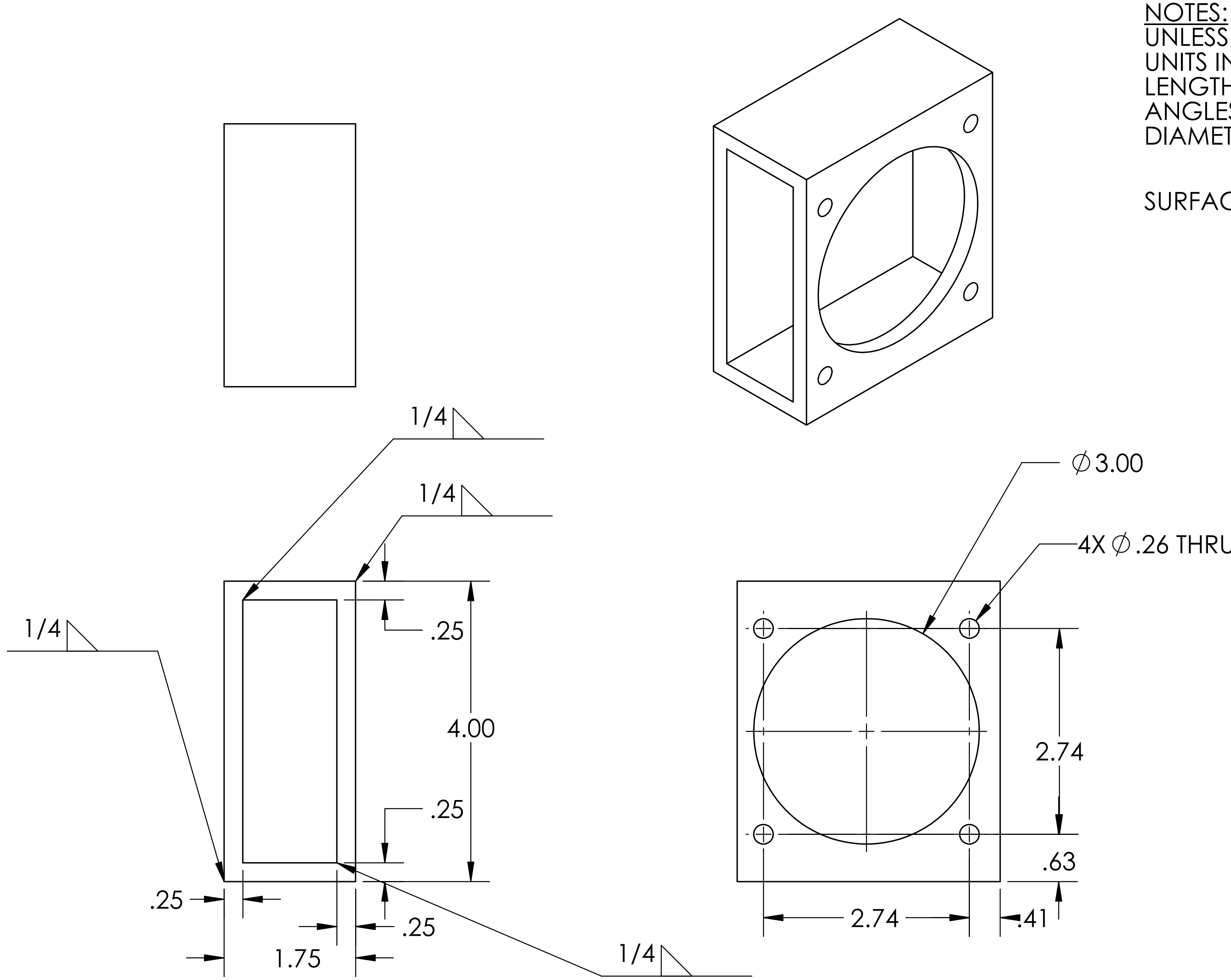
Cal Poly Mechanical Engineering Lab	Section: 02	SENIOR PROJECT	Title: V-BLOCK SIDE	Drwn. By: LOUIS ROSEGUO
ME 429 - Spring 2016	Dwg. #: 8910K21	Nxt Asb: V-BLOCK	Date: 5/2/2016 / Scale: 1:2	Chkd. By: ME STAFF



NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH ± 0.01 INCHES
ANGLES ± 1 °
DIAMETERS ± 0.001 INCHES
SURFACES 125

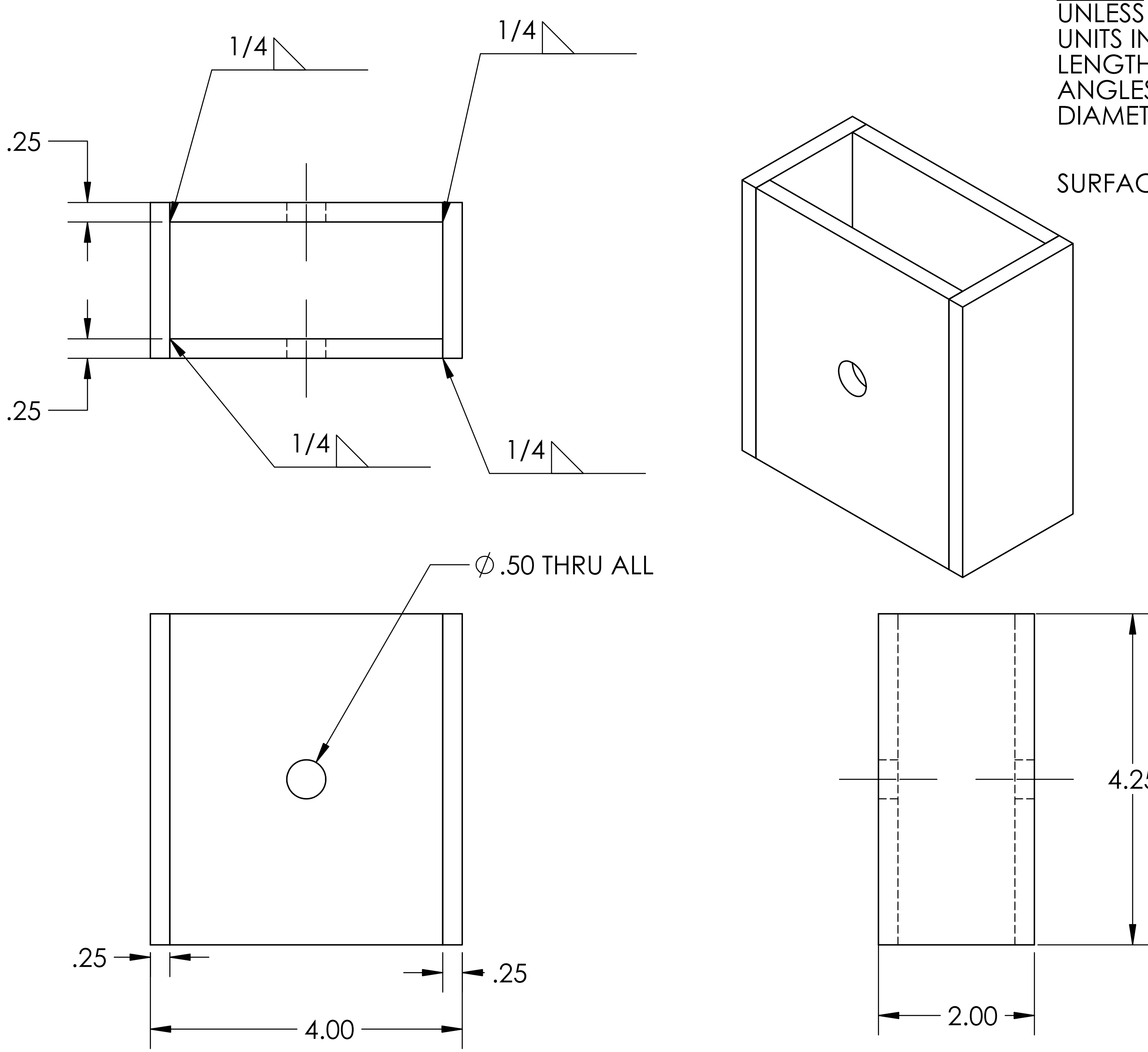
Cal Poly Mechanical Engineering	Lab Section: 02	SENIOR PROJECT	Title: V-BLOCK CENTER		Drwn. By: LOUIS ROSEGUO
ME 429 - Spring 2016	Dwg. #:9517K531	Nxt Asb: V-BLOCK	Date: 5/2/2016	Scale: 1:2	Chkd. By: ME STAFF

NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH: ± 0.01 INCHES
ANGLES: ± 1°
DIAMETERS: ± 0.001 INCHES
SURFACES: 125



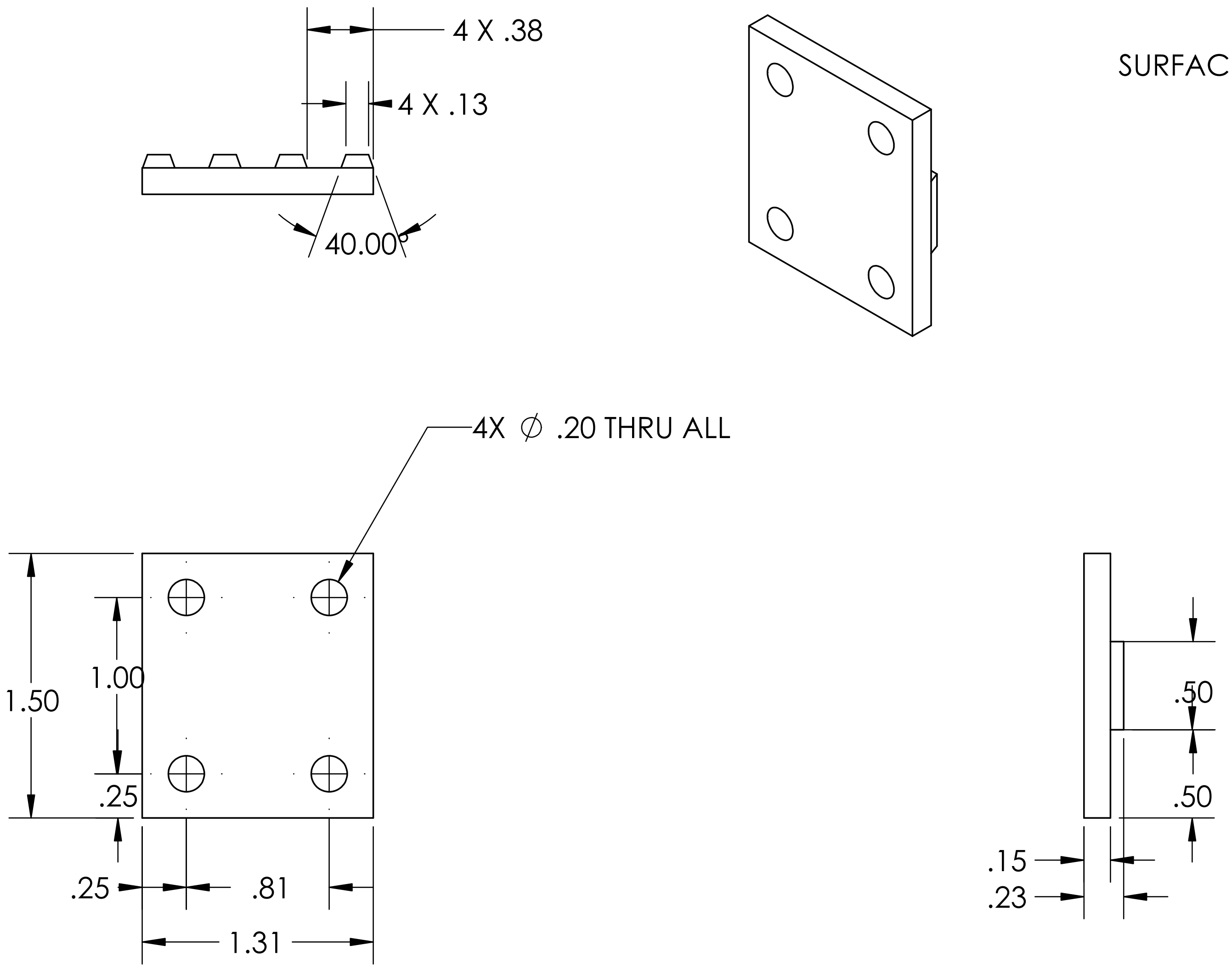
Cal Poly Mechanical Engineering	Lab Section:02	SENIOR PROJECT	Title:MOTOR HOUSING	Drwn. By:RYAN CANFIELD
ME 429 - SPRING 2016	Dwg. #: 8975K612	Nxt Asb:	Date:5/2/2016 · Scale:1:2	Chkd. By: ME STAFF

NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH: ± 0.01 INCHES
ANGLES: ± 1°
DIAMETERS: ± 0.001 INCHES
SURFACES: 125



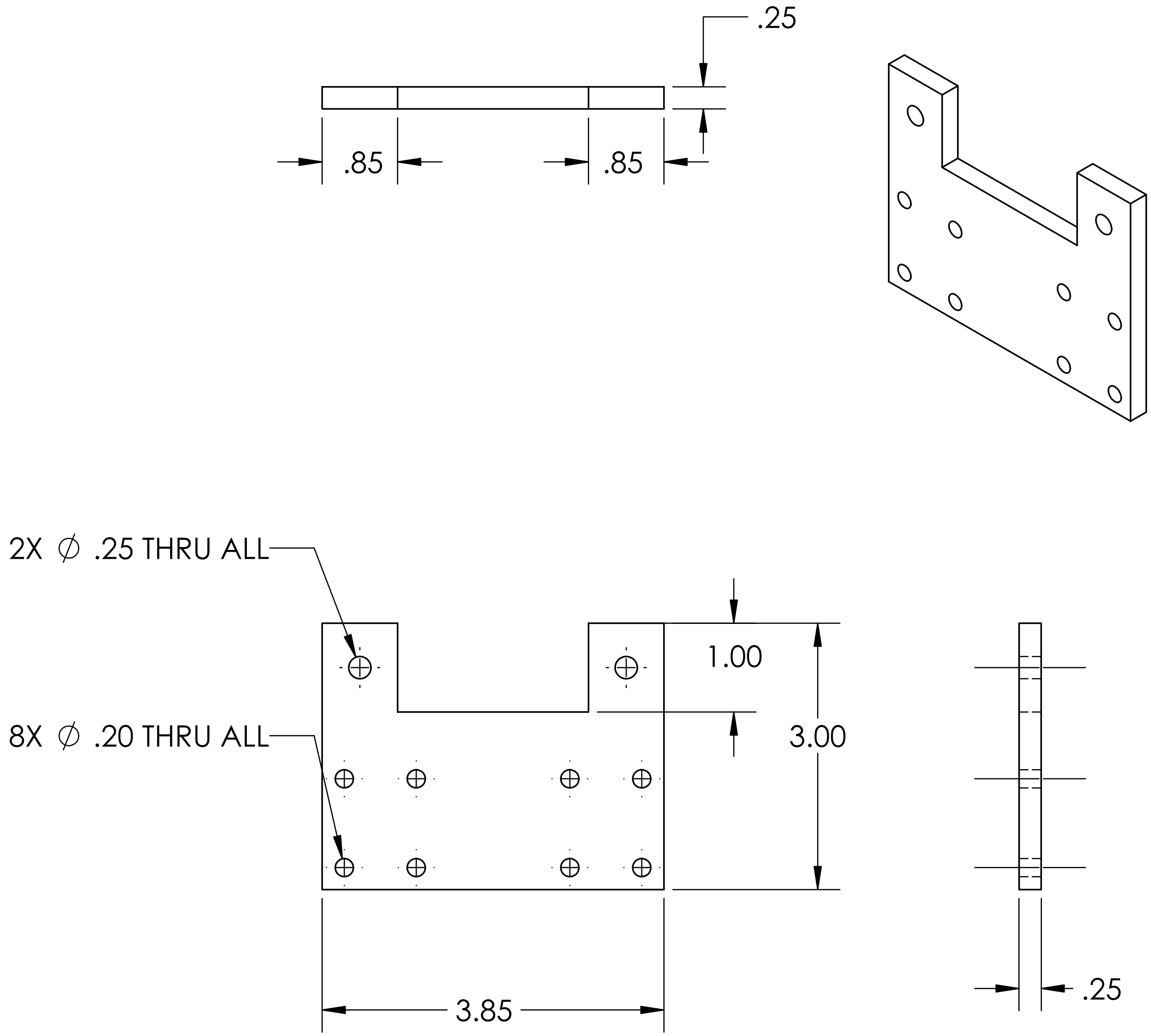
Cal Poly Mechanical Engineering	Lab Section:02	SENIOR PROJECT	Title:OFF MOTOR PULLEY MOUNT	Drwn. By:RYAN CANFIELD
ME 429 - SPRING 2016	Dwg. #: 8975K87	Nxt Asb:	Date:5/2/2016 / Scale:1:2	Chkd. By: ME STAFF

NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH: ± 0.01 INCHES
ANGLES: ± 1°
DIAMETERS: ± 0.001 INCHES
SURFACES: 125

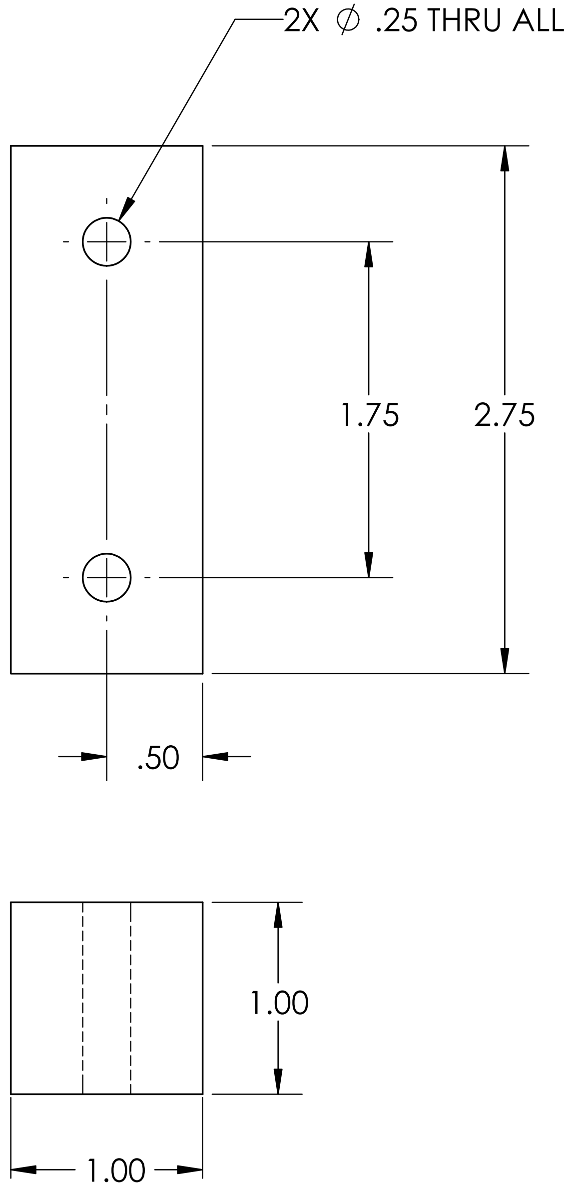


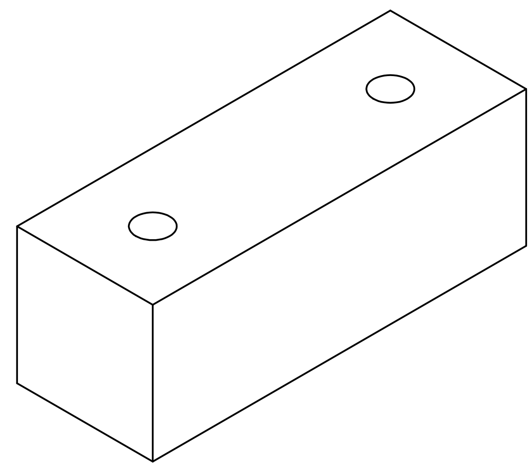
Cal Poly Mechanical Engineering	Lab Section:02	SENIOR PROJECT	Title:BELT HOLDER	Drwn. By:RYAN CANFIELD
ME 429 - SPRING 2016	Dwg. #:	Nxt Asb:	Date:5/2/2016 Scale:1:1	Chkd. By: ME STAFF

NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH: ± 0.01 INCHES
ANGLES: ± 1 °
DIAMETERS: ± 0.001 INCHES
SURFACES: 125 $\surd$



Cal Poly Mechanical Engineering	Lab Section:02	SENIOR PROJECT	Title:HORIZONTAL BELT ADAPTOR		Drwn. By:RYAN CANFIELD
ME 429 - SPRING 2016	Dwg. #: 8975K87	Nxt Asb:	Date:5/2/2016	Scale:1:2	Chkd. By: ME STAFF



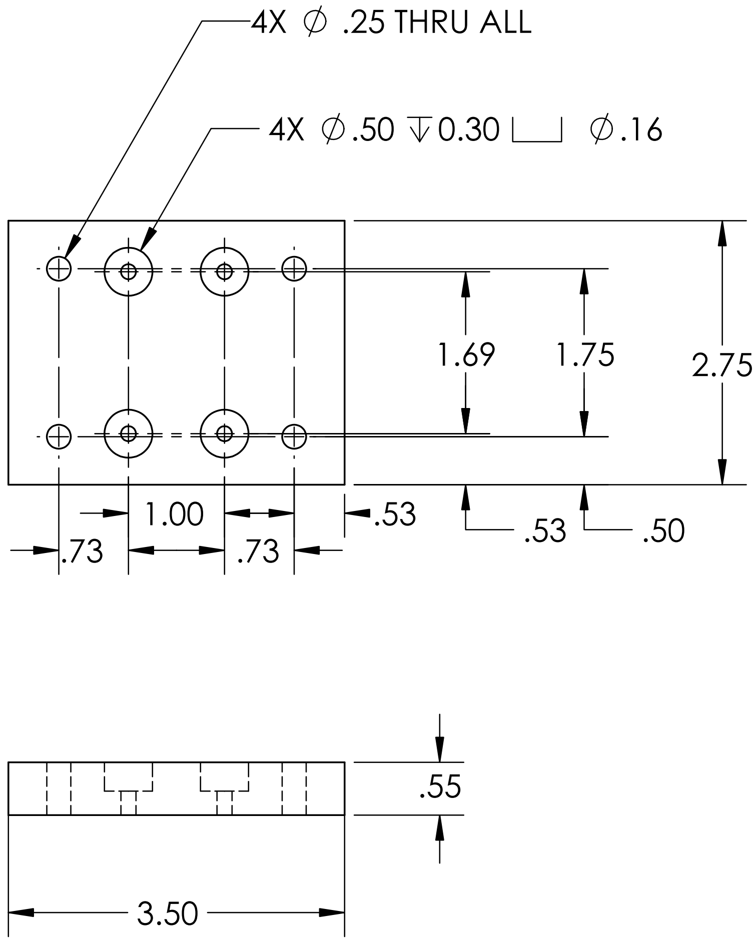


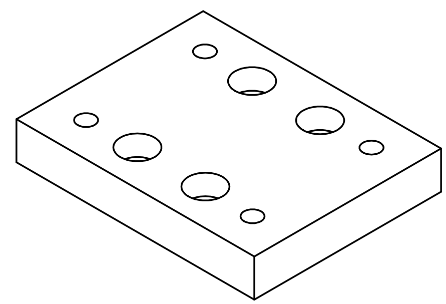
NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH: ± 0.01 INCHES
ANGLES: ± 1°
DIAMETERS: ± 0.001 INCHES
SURFACES: 125 √

Cal Poly Mechanical Engineering	Lab Section:02	SENIOR PROJECT	Title:SPACE BLOCK	Drwn. By:RYAN CANFIELD
ME 429 - SPRING 2016	Dwg. #: 9143K21	Nxt Asb:	Date:5/2/2016 Scale:1:1	Chkd. By: ME STAFF

N20

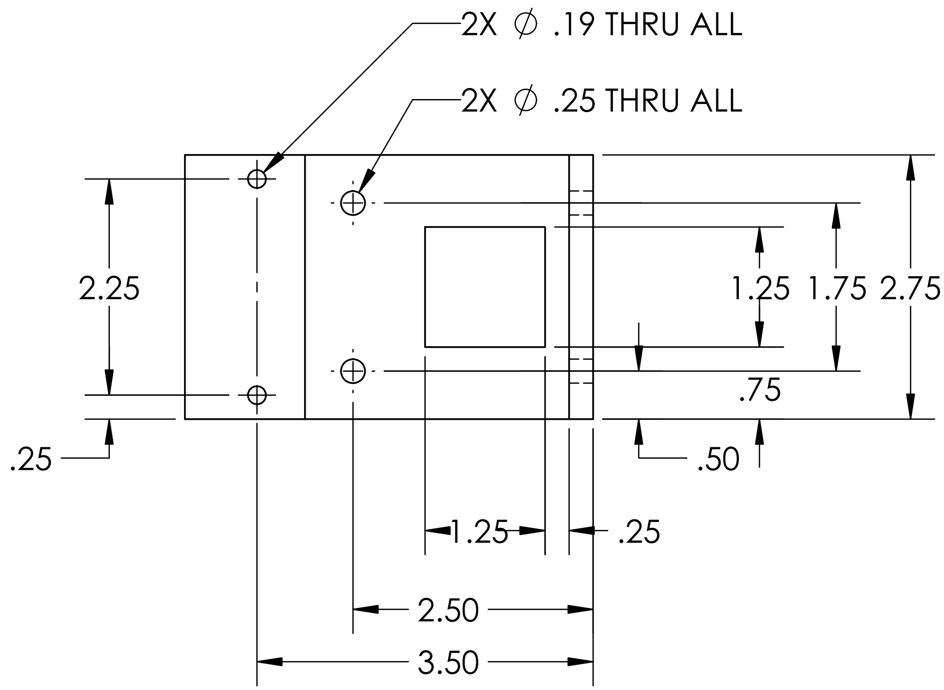
NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH: ± 0.01 INCHES
ANGLES: ± 1°
DIAMETERS: ± 0.001 INCHES
SURFACES: 125

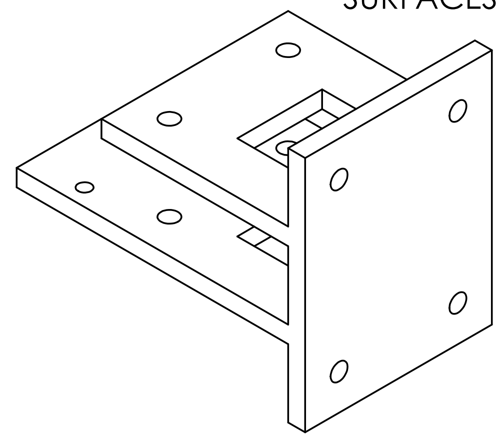


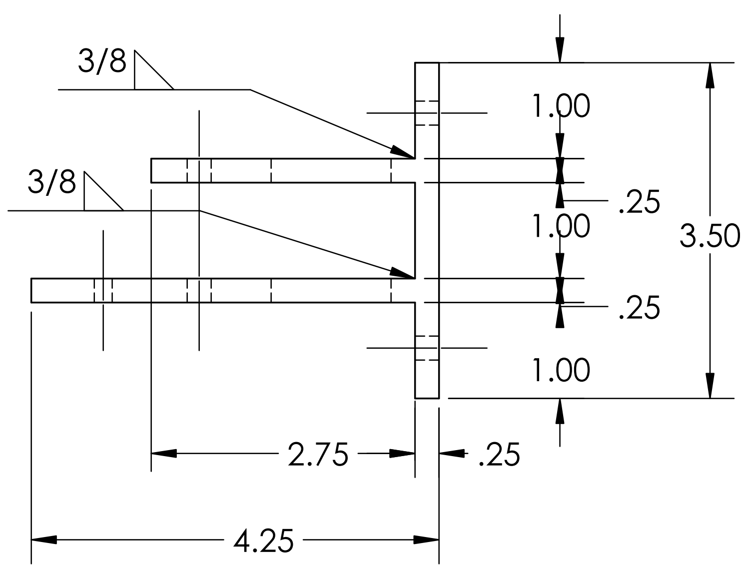


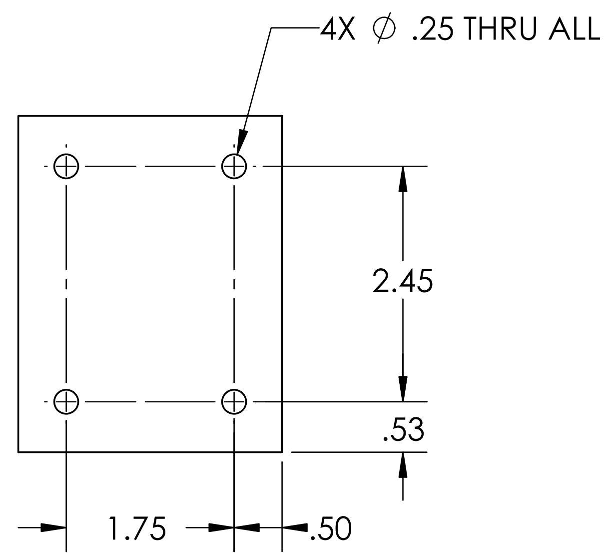
Cal Poly Mechanical Engineering	Lab Section:02	SENIOR PROJECT	Title:VERTICAL WELD PLATE	Drwn. By:RYAN CANFIELD
ME 429 - SPRING 2016	Dwg. #: 8975K226	Nxt Asb:	Date:5/2/2016 Scale:1:2	Chkd. By: ME STAFF

NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH: ± 0.01 INCHES
ANGLES: ± 1°
DIAMETERS: ± 0.001 INCHES
SURFACES: 125



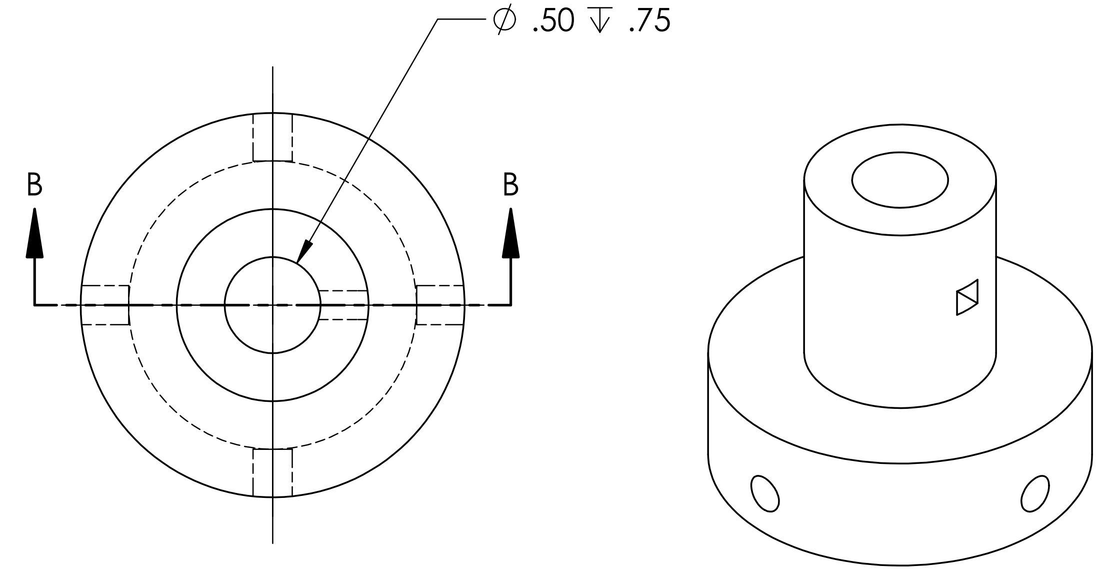


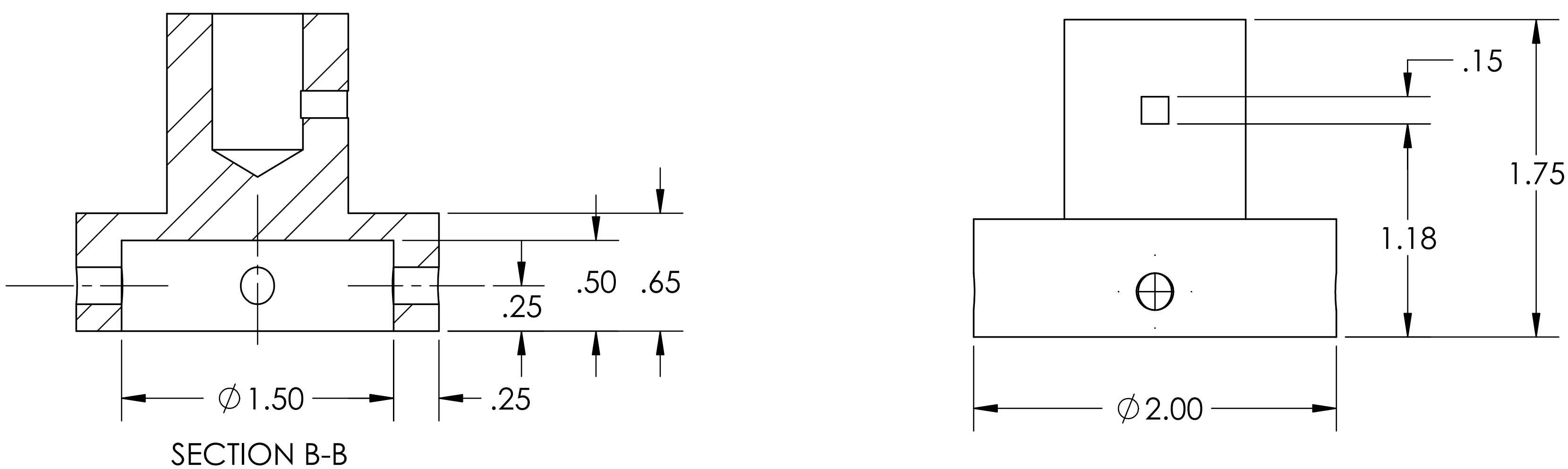




Cal Poly Mechanical Engineering	Lab Section:02	SENIOR PROJECT	Title:ACME NUT HOUSING	Drwn. By:RYAN CANFIELD
ME 429 - SPRING 2016	Dwg. #: 8975K87	Nxt Asb:	Date:5/2/2016 Scale:1:2	Chkd. By: ME STAFF

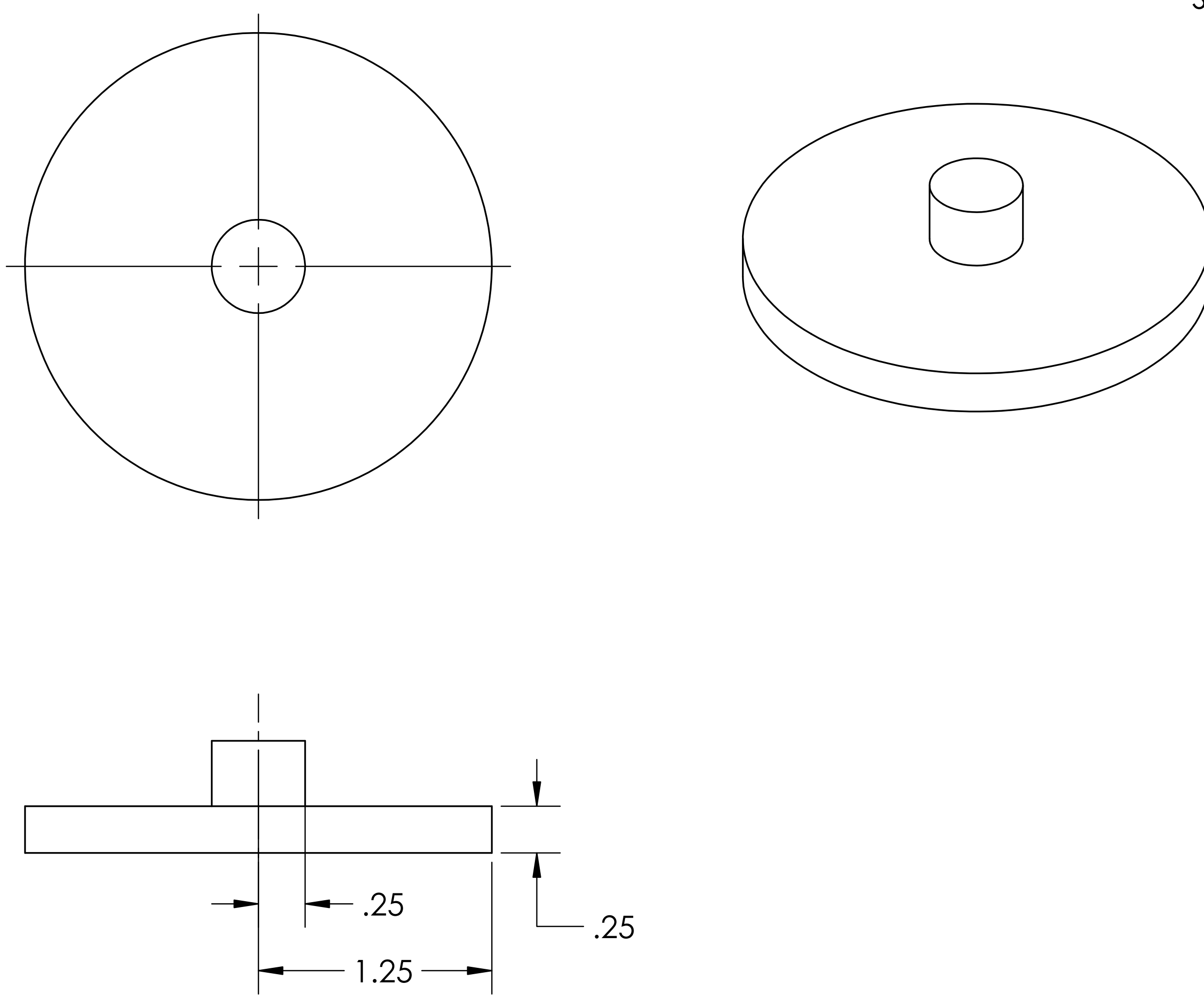
NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH: ± 0.01 INCHES
ANGLES: ± 1°
DIAMETERS: ± 0.001 INCHES
SURFACES: 125 (surface finish symbol)



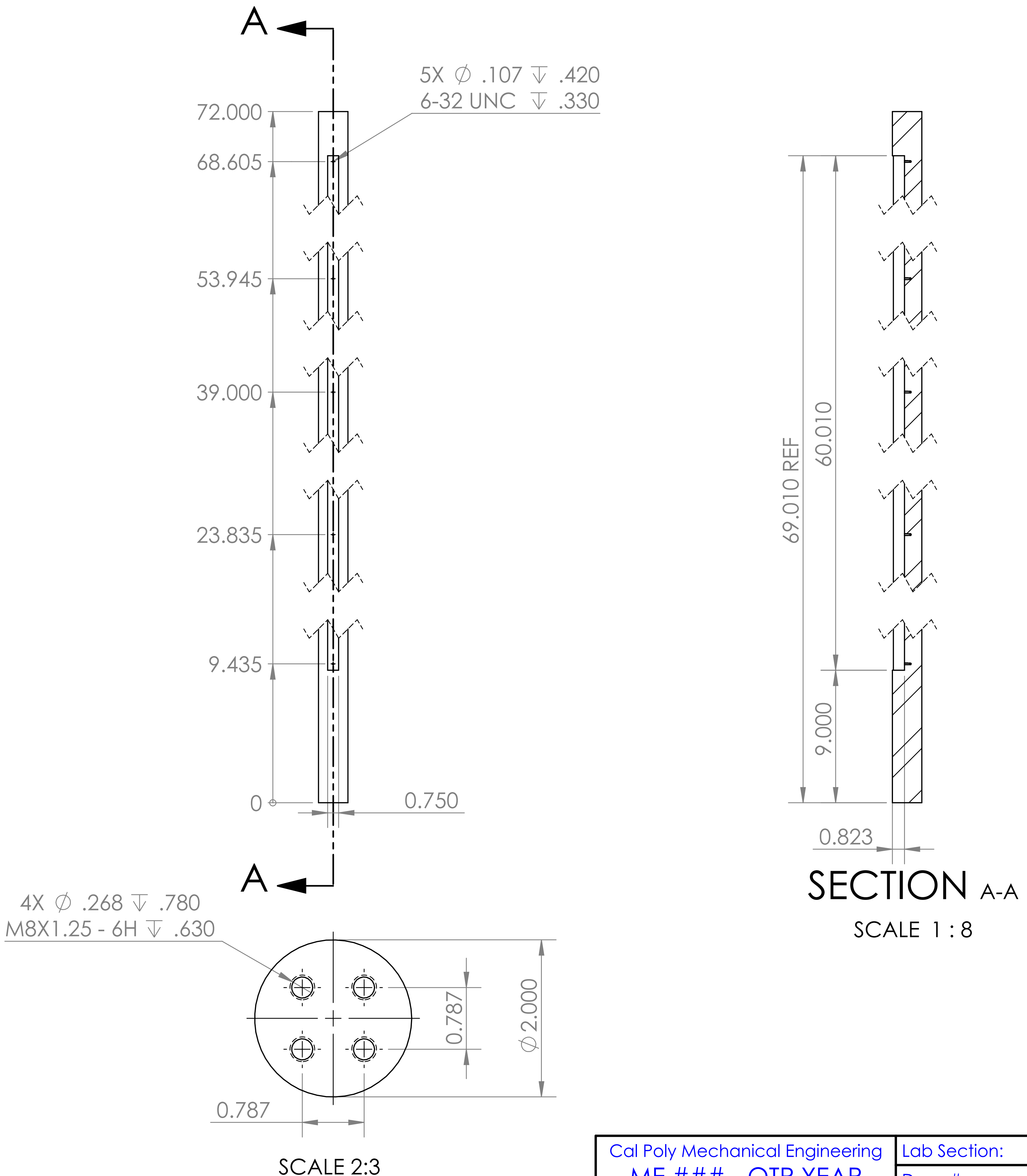


Cal Poly Mechanical Engineering	Lab Section:02	SENIOR PROJECT	Title:MOTOR-LEAD SCREW COUPLER	Drwn. By:RYAN CANFIELD
ME 429 - SPRING 2016	Dwg. #: 8927K58	Nxt Asb:	Date:5/2/2016 / Scale:1:2	Chkd. By: ME STAFF

NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH: ± 0.01 INCHES
ANGLES: ± 1°
DIAMETERS: ± 0.001 INCHES
SURFACES: 125 √

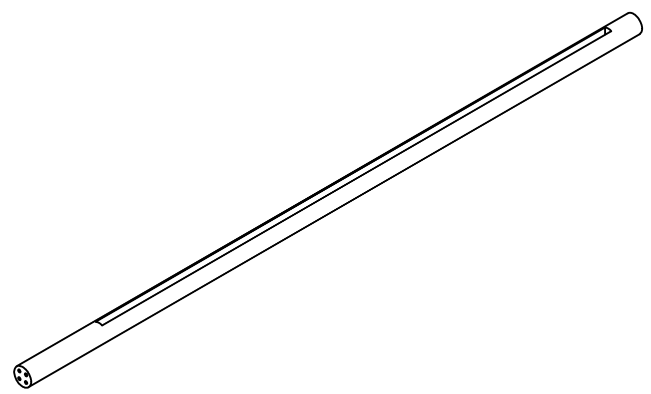


Cal Poly Mechanical Engineering	Lab Section:02	SENIOR PROJECT	Title:LEAD SCREW BASE	Drwn. By:RYAN CANFIELD
ME 429 - SPRING 2016	Dwg. #: 8927K58	Nxt Asb:	Date:5/2/2016 · Scale:1:1	Chkd. By: ME STAFF



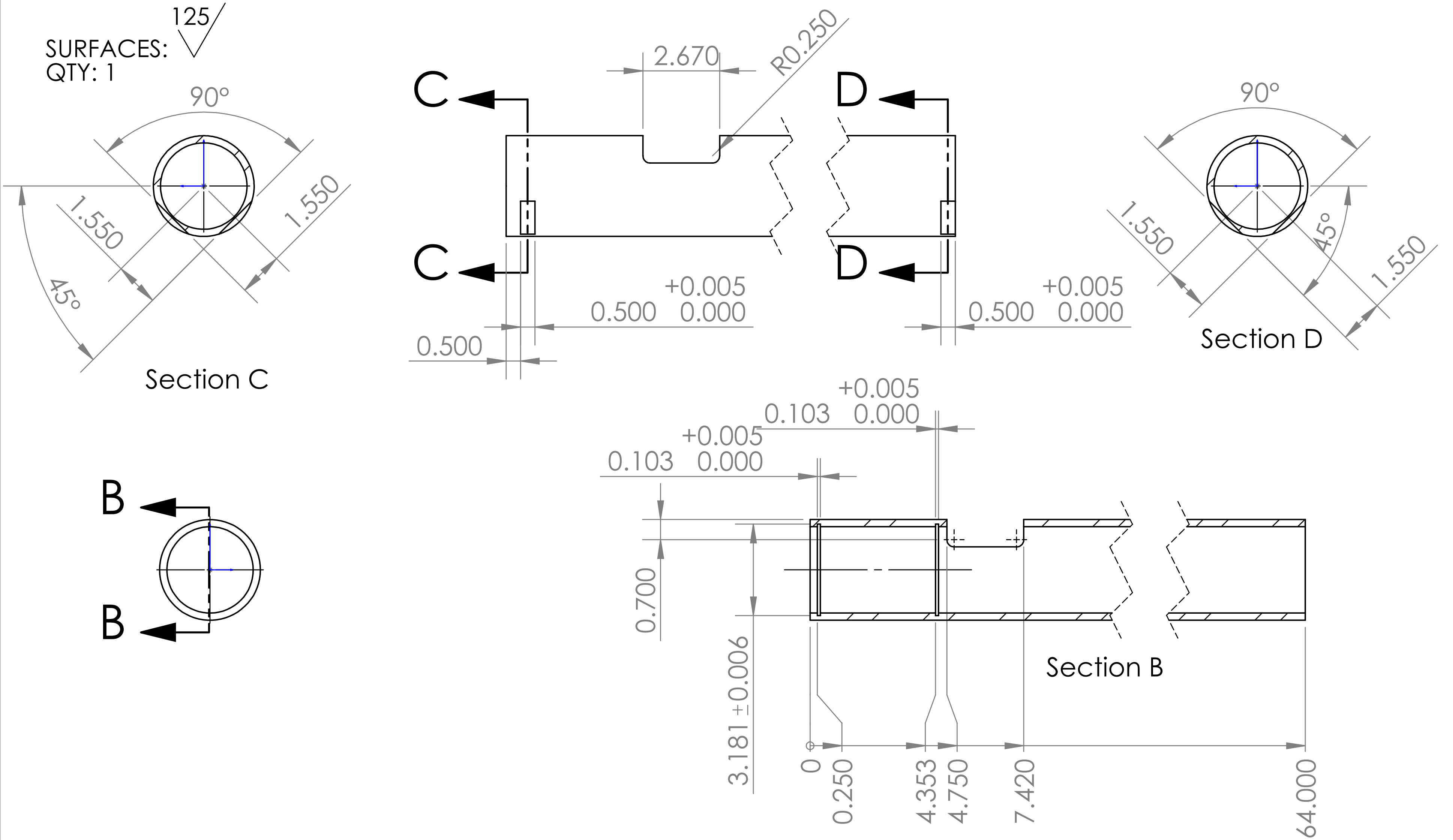
NOTES

UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
QUANTITY: 1
TOLERANCES:
LENGTH: ± 0.005
DIAMETER: ± 0.005



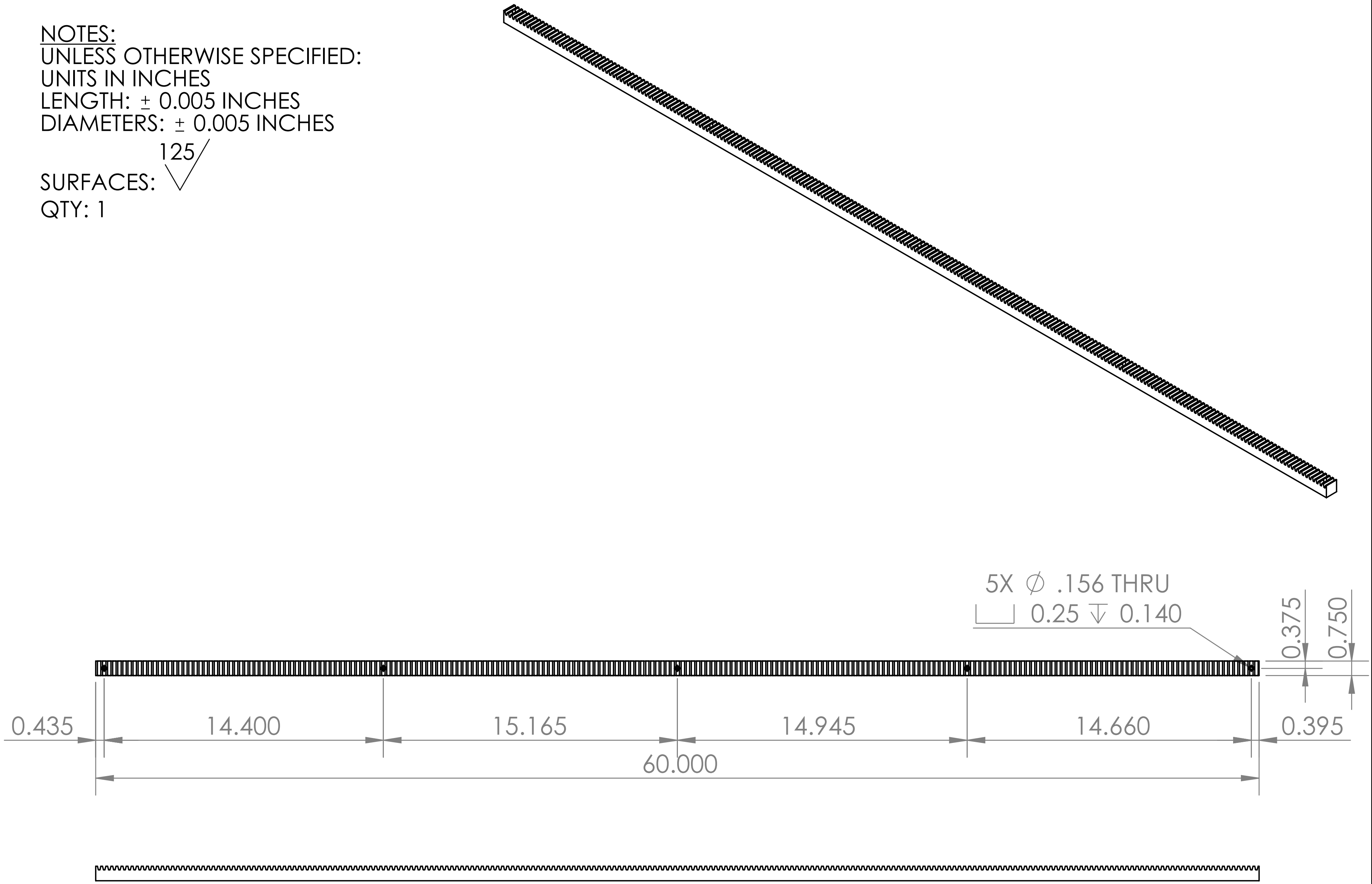
Cal Poly Mechanical Engineering ME ### - QTR YEAR	Lab Section:	Assignment #	Title: TELESCOPING ARM		Drwn. By: RYAN CANFIELD
	Dwg. #:	Nxt Asb:	Date:10/2/16	Scale: 1:8	Chkd. By: ME STAFF

NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH: ± 0.005 INCHES
ANGLES: ± 1 °
DIAMETERS: ± 0.005 INCHES
SURFACES: 125
QTY: 1



Cal Poly Mechanical Engineering	Lab Section:	SENIOR PROJECT	Title: BASE TUBE	Drwn. By: LOUIS ROSEGUO
ME ### - Qtr Year	Dwg. #:T2312250	Nxt Asb: T-SCOPE	Date:9/30/2016 Scale: 1:5	Chkd. By: ME STAFF

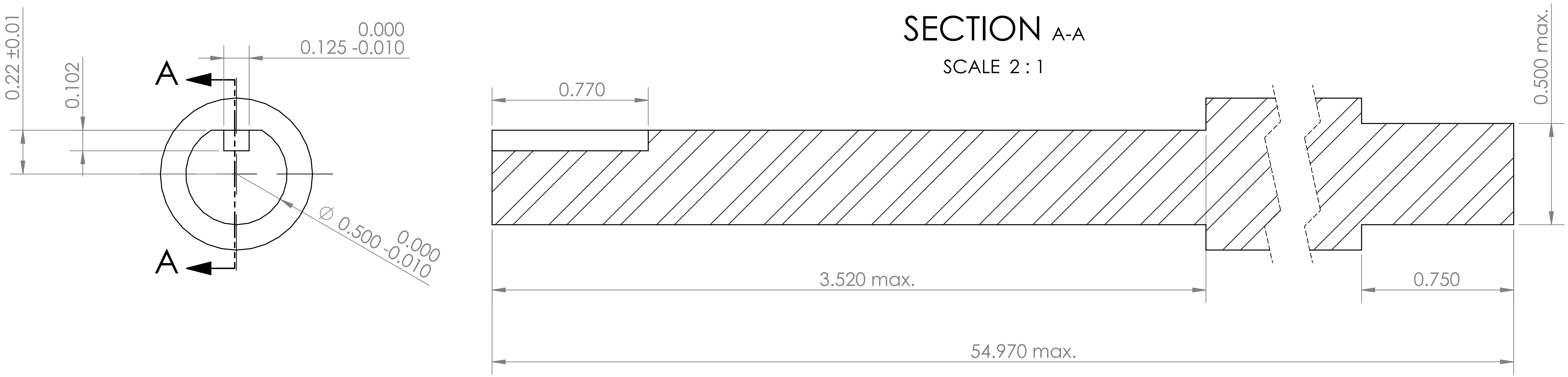
NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH: ± 0.005 INCHES
DIAMETERS: ± 0.005 INCHES
SURFACES: 125
QTY: 1



Cal Poly Mechanical Engineering	Lab Section:	Assignment #	Title:		Drwn. By:
ME 430 - Fall 2016	Dwg. #:	Nxt Asb:	Date:	Scale: 1:7	Chkd. By: ME STAFF

NOTES

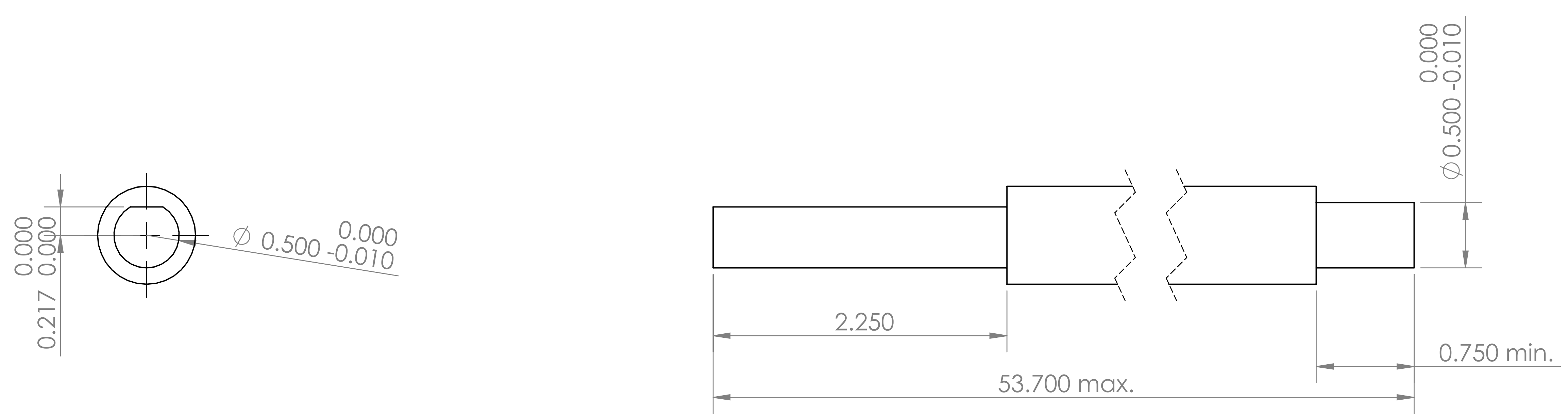
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
QUANTITY: 2
TOLERANCES:
LENGTH: ± 0.005
DIAMETER: ± 0.005



Cal Poly Mechanical Engineering ME ### - QTR YEAR	Lab Section:	Assignment #	Title: DRIVING LEAD SCREW	Drwn. By: RYAN CANFIELD
	Dwg. #:	Nxt Asb:	Date: 10/2/16 / Scale:2:1	Chkd. By: ME STAFF

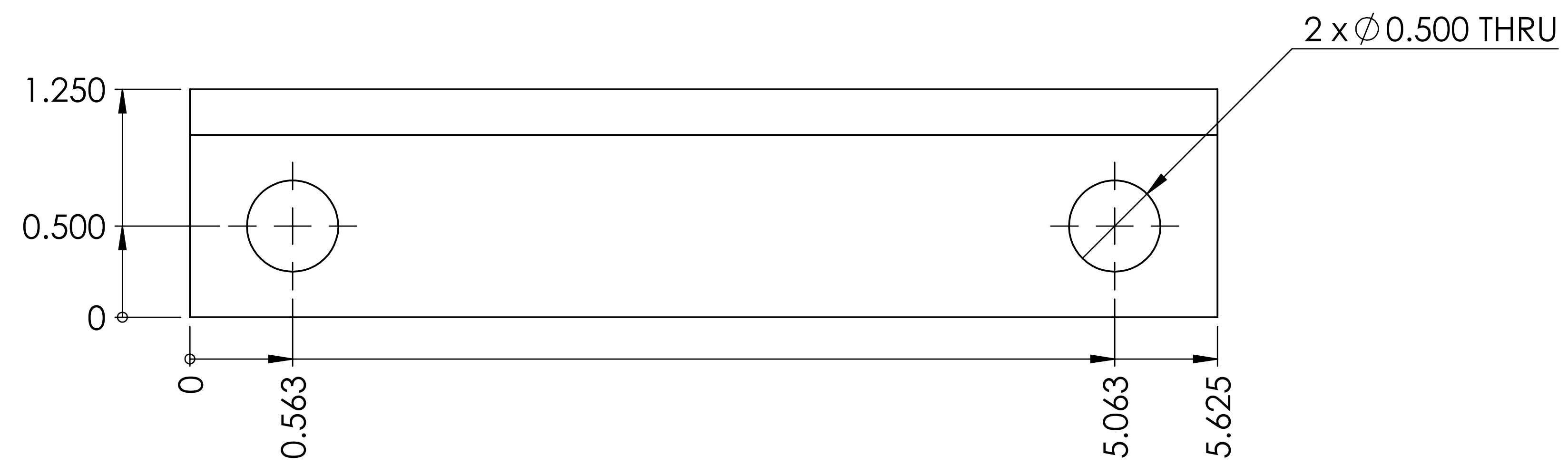
NOTES

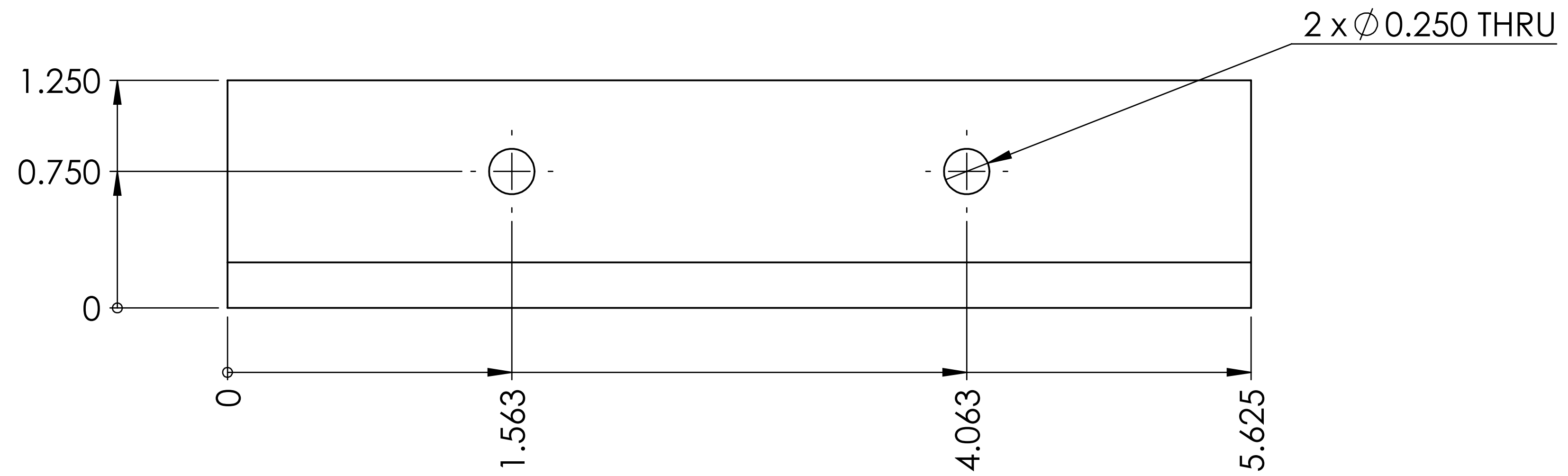
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
QUANTITY: 2
TOLERANCES:
LENGTH: ± 0.005
DIAMETER: ± 0.005



Cal Poly Mechanical Engineering	Lab Section:	Assignment #	Title: NON DRIVEN LEAD SCREW	Drwn. By: RYAN CANFIELD
ME ### - QTR YEAR	Dwg. #:	Nxt Asb:	Date: 10/2/16 Scale: 1:1	Chkd. By: ME STAFF

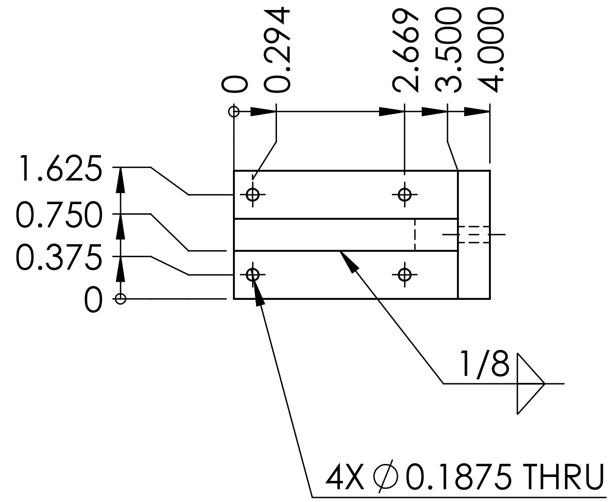
NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH ± 0.005 INCHES
ANGLES ± 1 °
DIAMETERS ± 0.001 INCHES
STEEL
QTY: 1

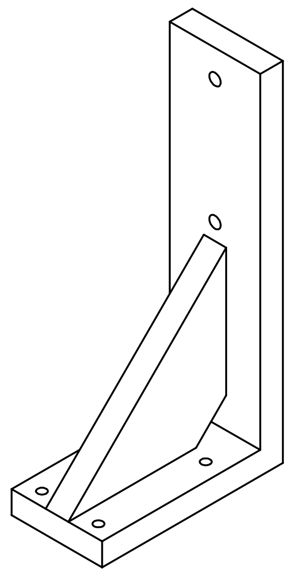


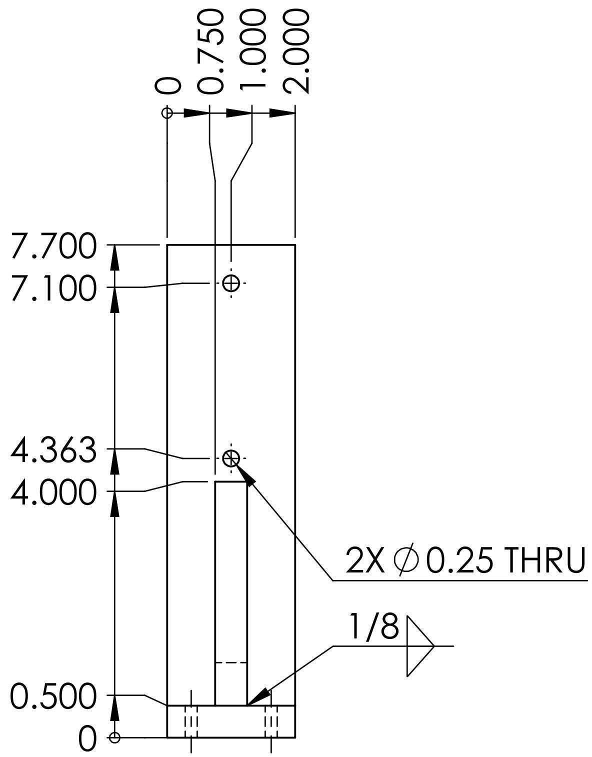


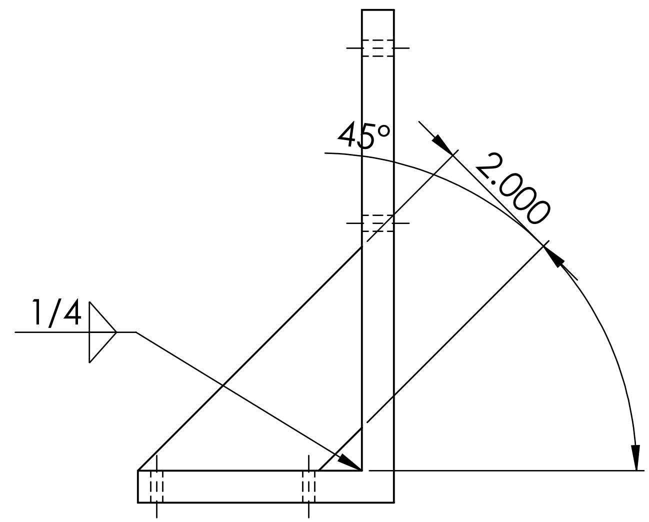
Cal Poly Mechanical Engineering	Lab Section:	Assignment #	Title: ANGLE IRON	Drwn. By:
ME ### - QTR YEAR	Dwg. #:	Nxt Asb:	Date: / Scale:	Chkd. By: ME STAFF

NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH ± 0.005 INCHES
ANGLES ± 1 °
DIAMETERS ± 0.001 INCHES
STEEL
QTY: 1



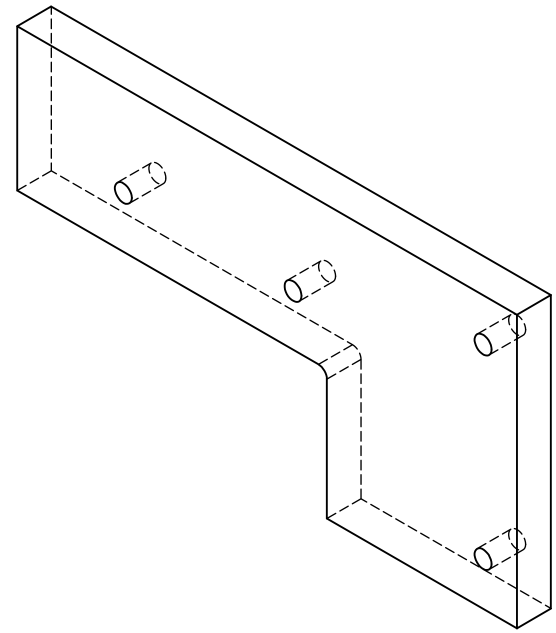


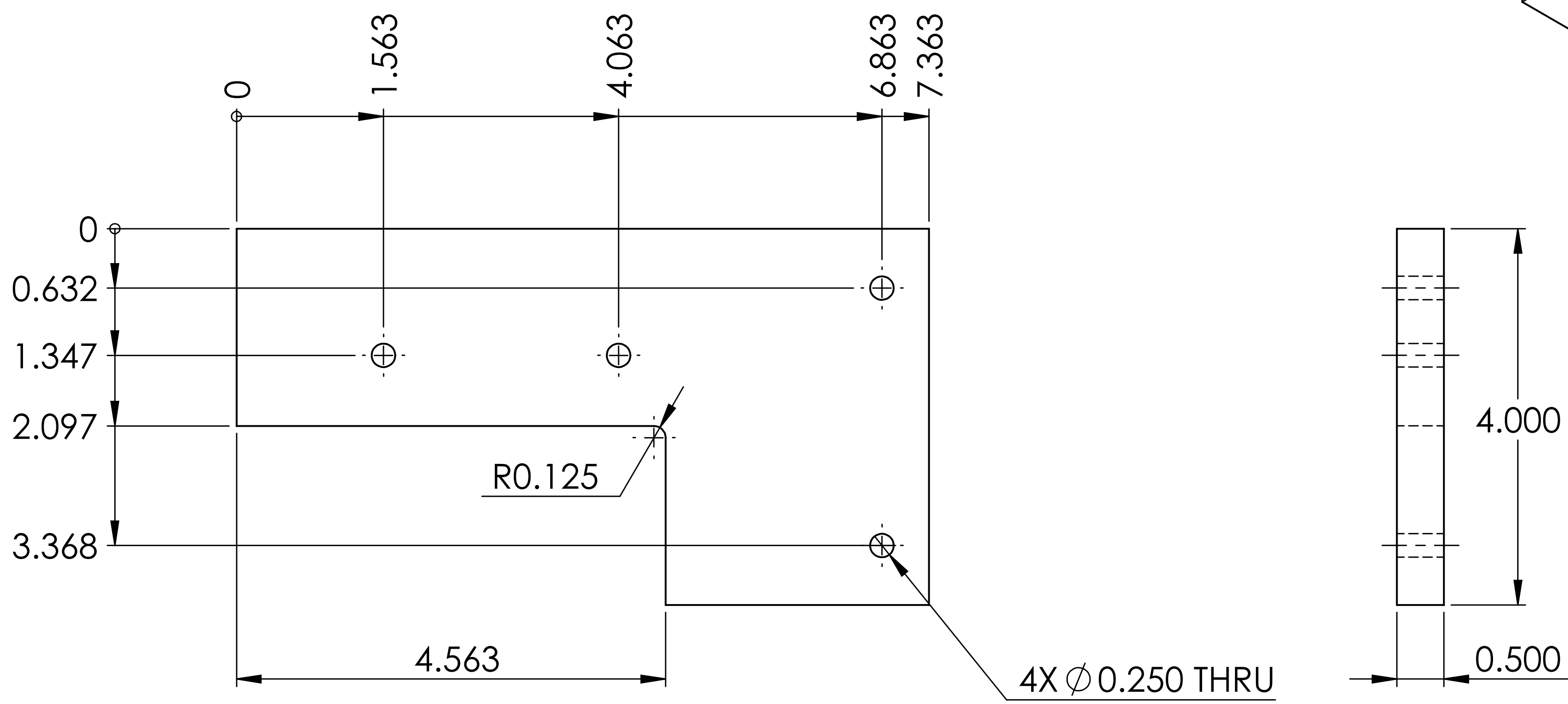




Cal Poly Mechanical Engineering Lab Section: 02		SENIOR PROJECT	Title: MOTOR HANGER		Drwn. By: LOUIS ROSEGUO
ME 429 - Spring 2016	Dwg. #:8910K949	Nxt Asb: MOUNT	Date: 5/2/2016	Scale: 1:3	Chkd. By: ME STAFF

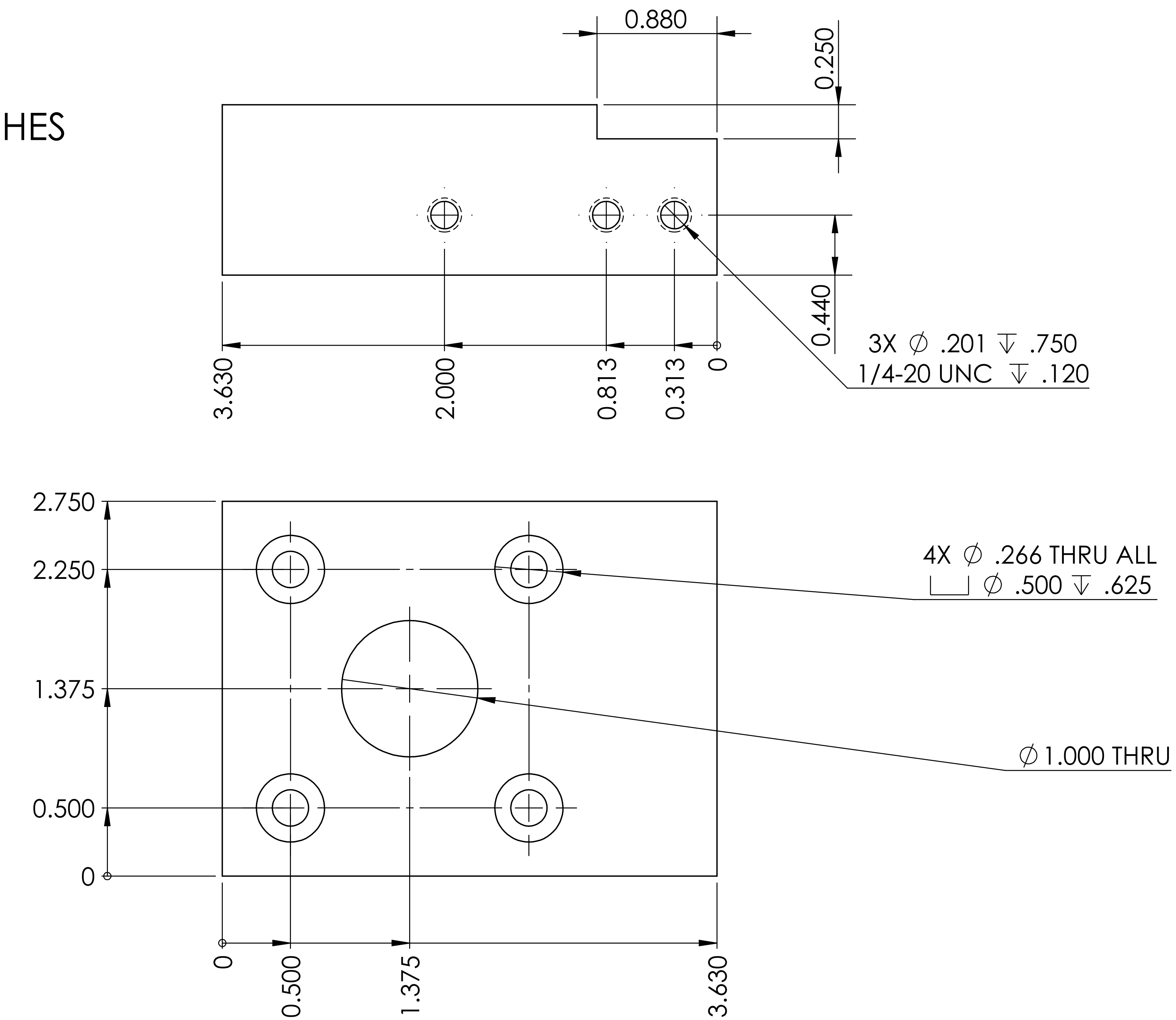
NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH ± 0.005 INCHES
ANGLES ± 1°
DIAMETERS ± 0.001 INCHES
STEEL
QTY: 1





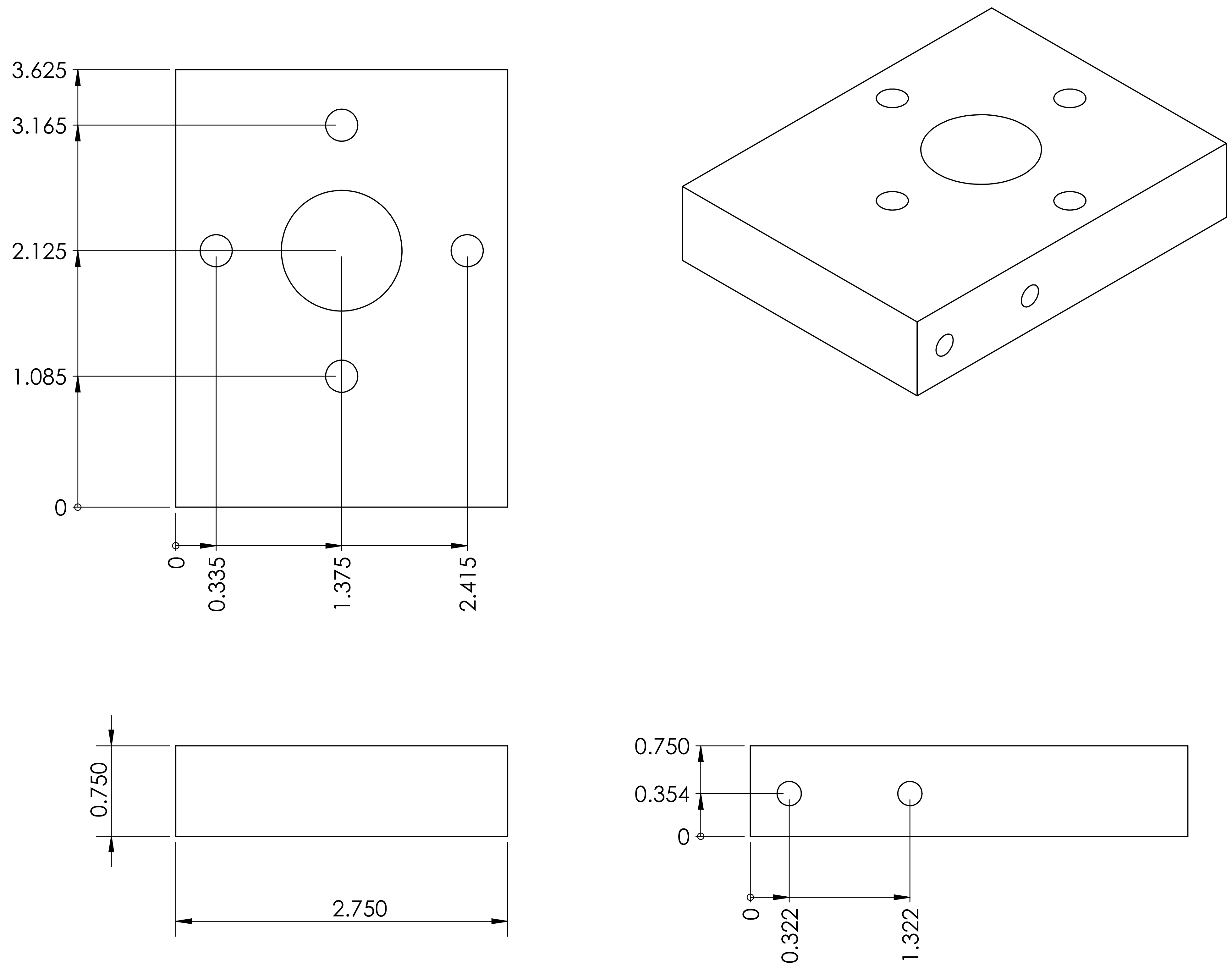
Cal Poly Mechanical Engineering - Lab Section: 02	SENIOR PROJECT	Title: MOTOR HANGER BASE		Drwn. By: LOUIS ROSEGUO
ME 429 - Spring 2016	Dwg. #: MB01	Nxt Asb: MOUNT	Date: 5/2/2016	Scale: 1:2	Chkd. By: ME STAFF

NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH ± 0.005 INCHES
ANGLES ± 1 °
DIAMETERS ± 0.001 INCHES
ALUMINUM
QTY: 2



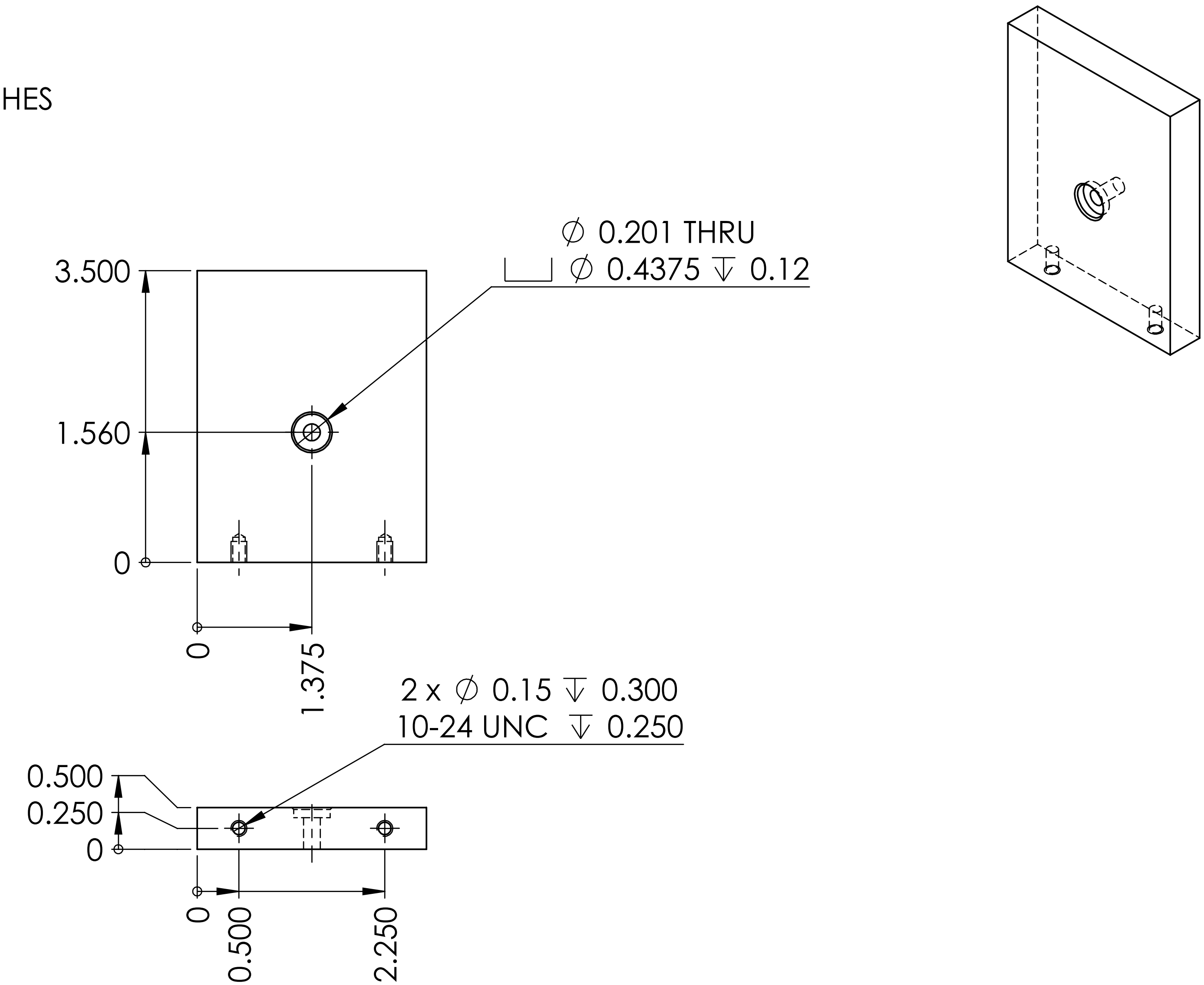
Cal Poly Mechanical Engineering	Lab Section:	Assignment #	Title: MOTOR SUPPORT BOTTOM	Drwn. By: RYAN CANFIELD
ME ### - QTR YEAR	Dwg. #:	Nxt Asb:	Date: Scale:	Chkd. By: ME STAFF

NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH ± 0.005 INCHES
ANGLES ± 1 °
DIAMETERS ± 0.001 INCHES
ALUMINUM
QTY: 2



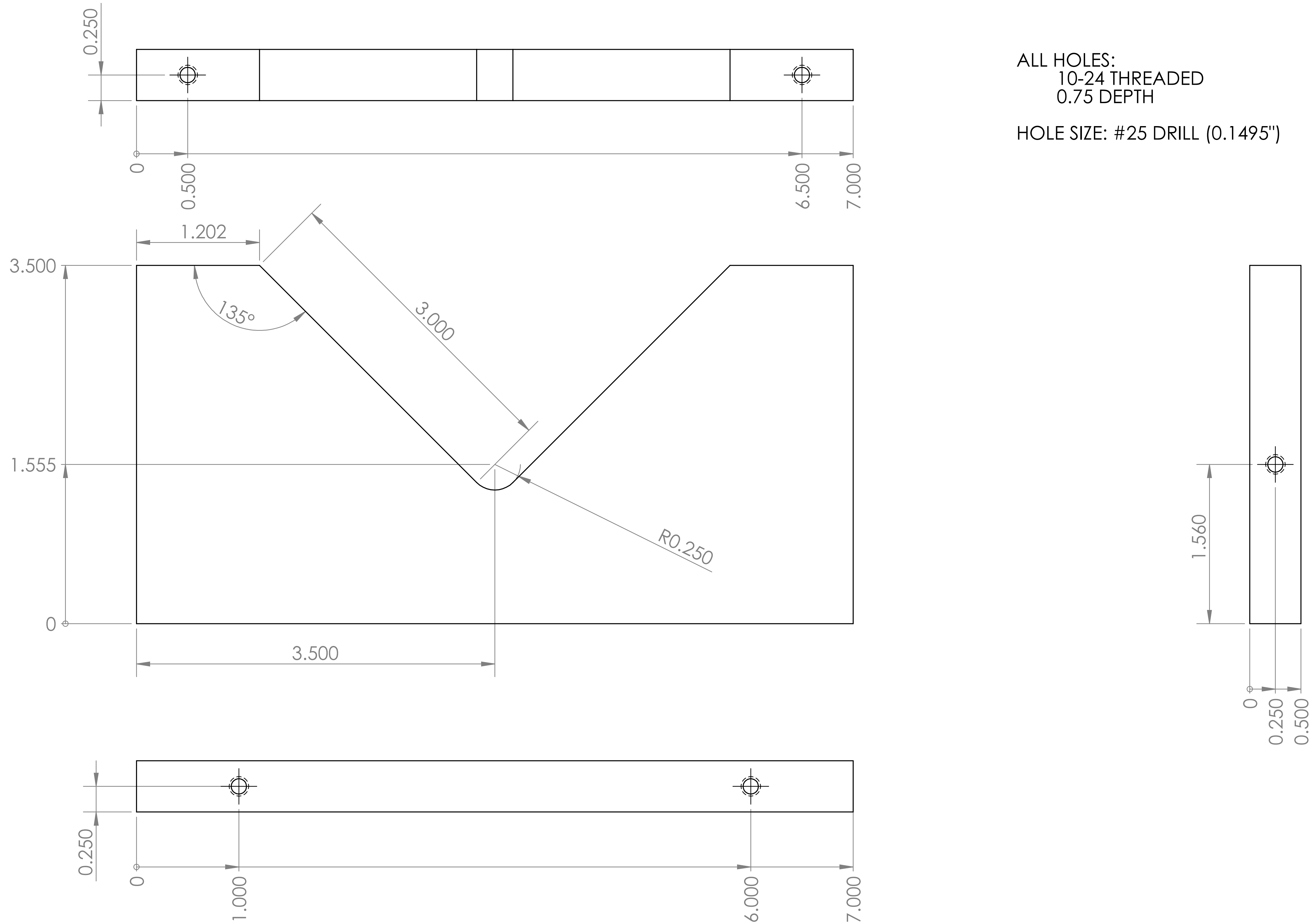
Cal Poly Mechanical Engineering	Lab Section:	Assignment #	Title: MOTOR SUPPORT TOP		Drwn. By:
ME ### - QTR YEAR	Dwg. #:	Nxt Asb:	Date:	Scale:	Chkd. By: ME STAFF

NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH ± 0.005 INCHES
ANGLES ± 1 °
DIAMETERS ± 0.001 INCHES
STEEL
QTY: 4



Cal Poly Mechanical Engineering, Lab Section: 02		SENIOR PROJECT	Title: V-BLOCK SIDE		Drwn. By: LOUIS ROSEGUO
ME 429 - Spring 2016	Dwg. #: 8910K21	Nxt Asb: V-BLOCK	Date: 5/2/2016	Scale: 1:2	Chkd. By: ME STAFF

O12

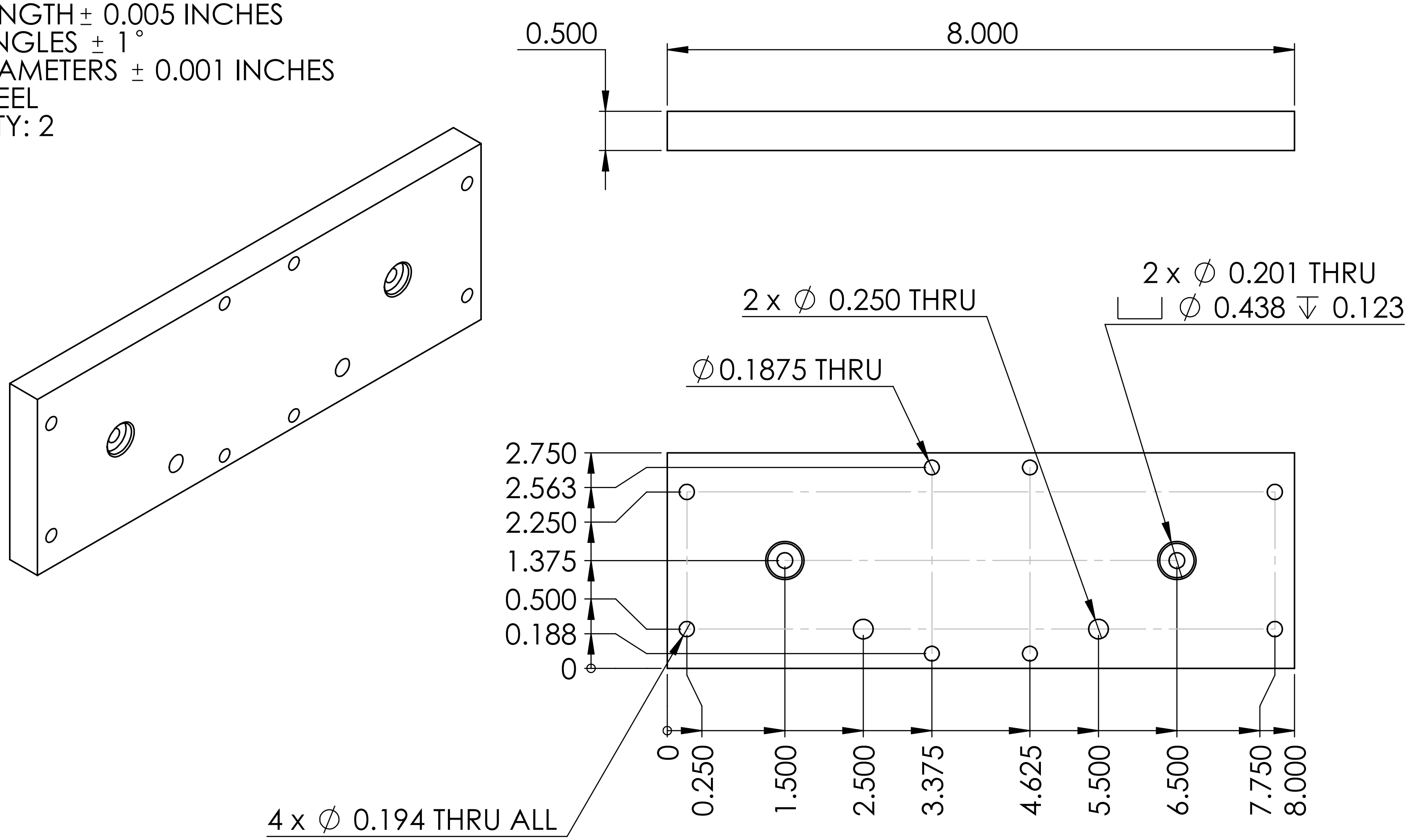


ALL HOLES:
10-24 THREADED
0.75 DEPTH

HOLE SIZE: #25 DRILL (0.1495")

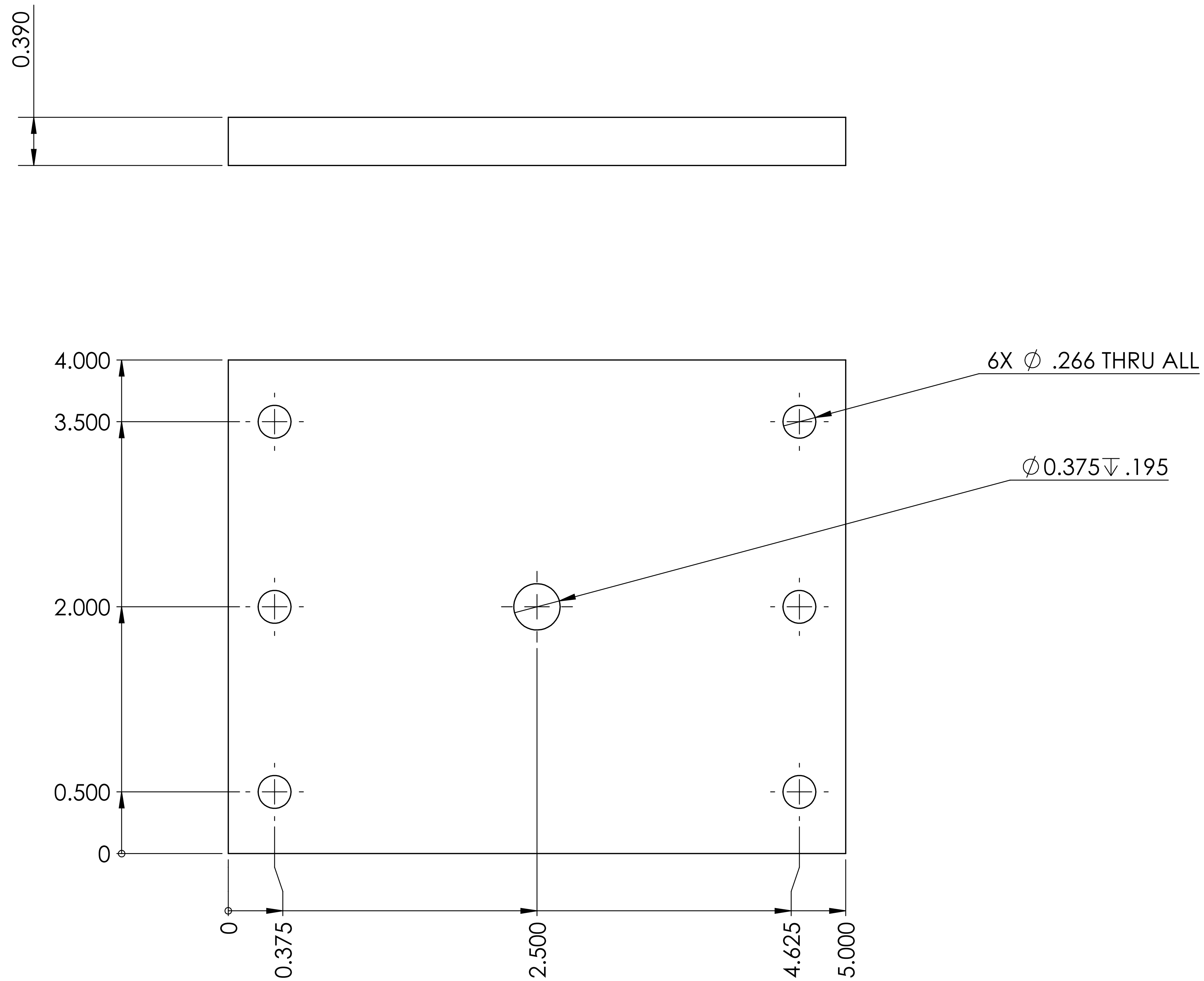
Cal Poly Mechanical Engineering ME ### - QTR YEAR	Lab Section:	Assignment #	Title: V BLOCK CENTER		Drwn. By:
	Dwg. #:	Nxt Asb:	Date:	Scale:	Chkd. By: ME STAFF

NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH ± 0.005 INCHES
ANGLES ± 1 °
DIAMETERS ± 0.001 INCHES
STEEL
QTY: 2



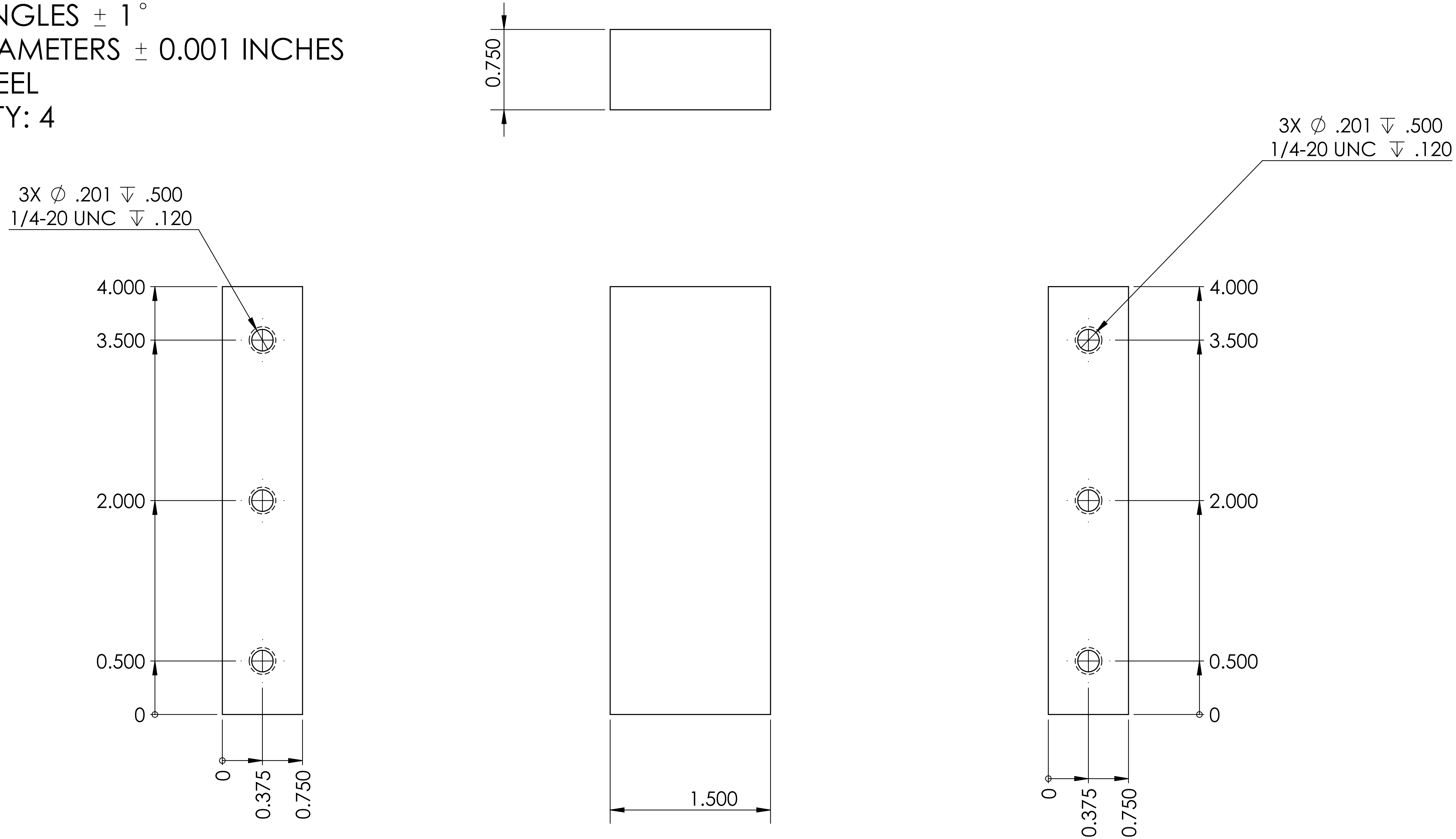
Cal Poly Mechanical Engineering Lab Section: 02	SENIOR PROJECT	Title: V-BLOCK BOTTOM		Drwn. By: LOUIS ROSEGUO
ME 429 - Spring 2016	Dwg. #: 8910K702	Nxt Asb: V-BLOCK	Date: 5/2/2016	Scale: 1:2	Chkd. By: ME STAFF

NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH ± 0.005 INCHES
ANGLES ± 1 °
DIAMETERS ± 0.001 INCHES
STEEL
QTY: 4



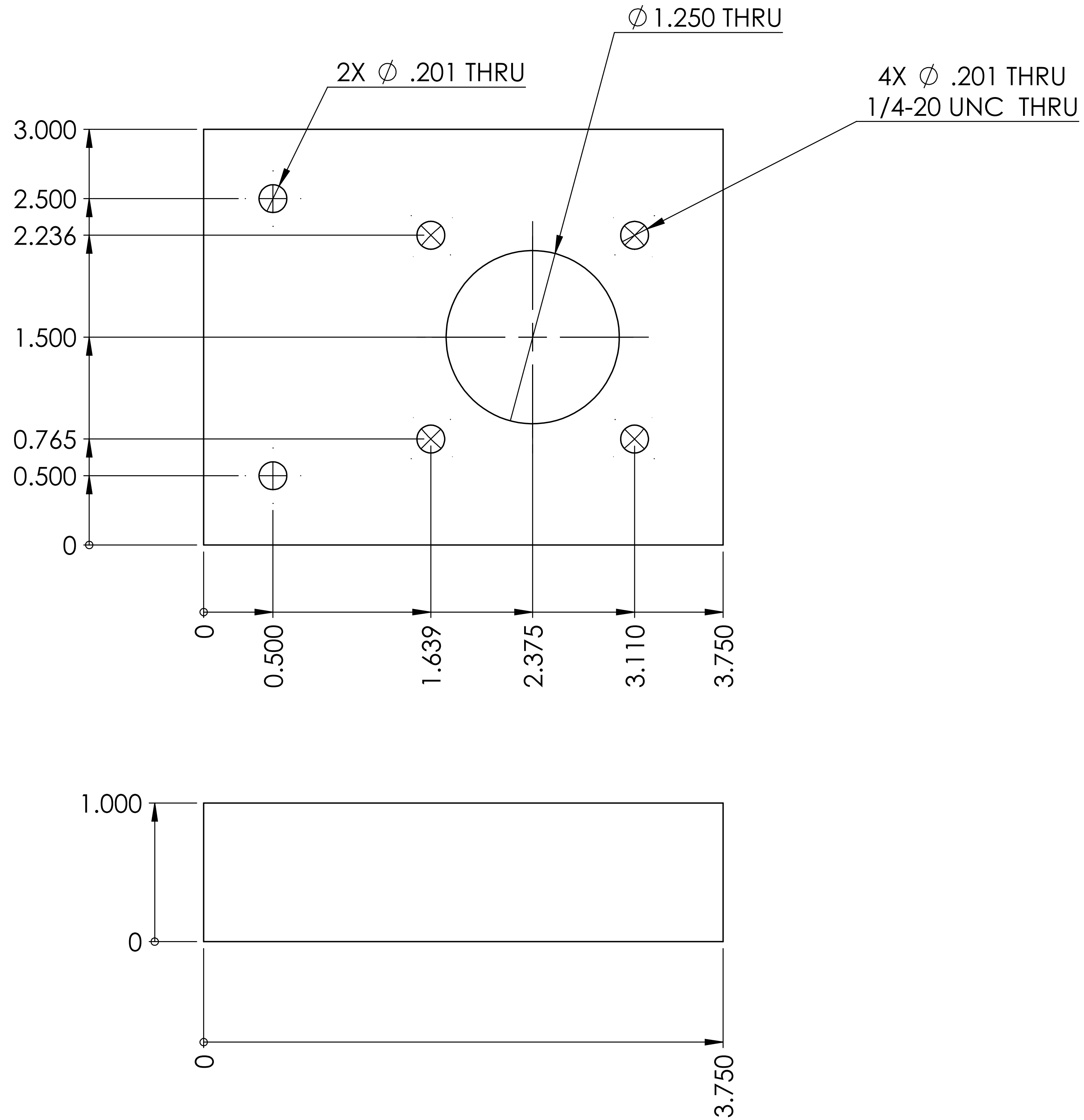
Cal Poly Mechanical Engineering	Lab Section:	Assignment #	Title: PULLEY HOUSING SIDE	Drwn. By: RYAN CANFIELD
ME ### - QTR YEAR	Dwg. #:	Nxt Asb:	Date: / Scale:	Chkd. By: ME STAFF

NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH ± 0.005 INCHES
ANGLES ± 1 °
DIAMETERS ± 0.001 INCHES
STEEL
QTY: 4



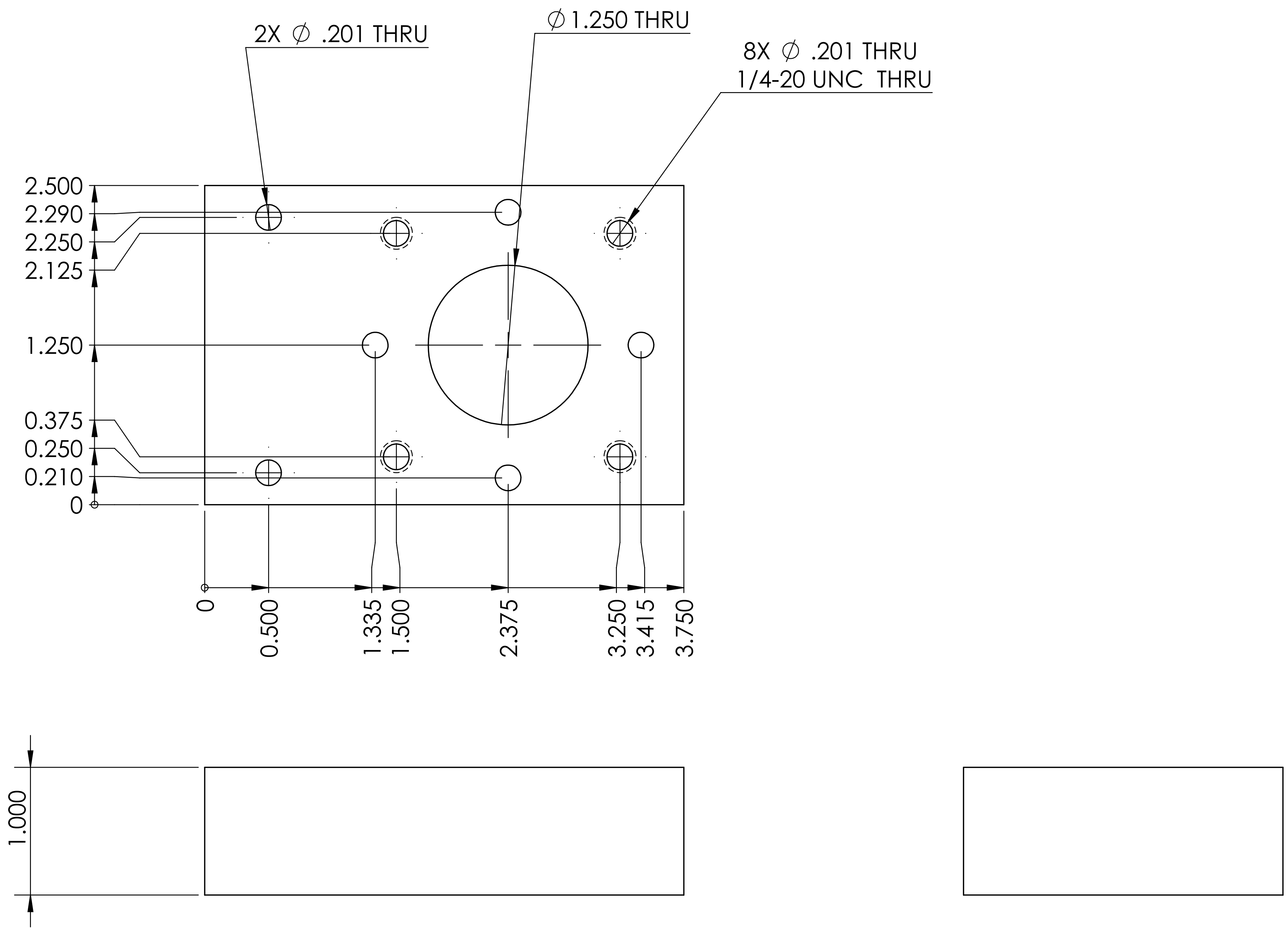
Cal Poly Mechanical Engineering	Lab Section:	Assignment #	Title: PULLEY HOUSING TOP		Drwn. By: RYAN CANFIELD
ME ### - QTR YEAR	Dwg. #:	Nxt Asb:	Date:	Scale:	Chkd. By: ME STAFF

NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH ± 0.005 INCHES
ANGLES ± 1 °
DIAMETERS ± 0.001 INCHES
STEEL
QTY: 2



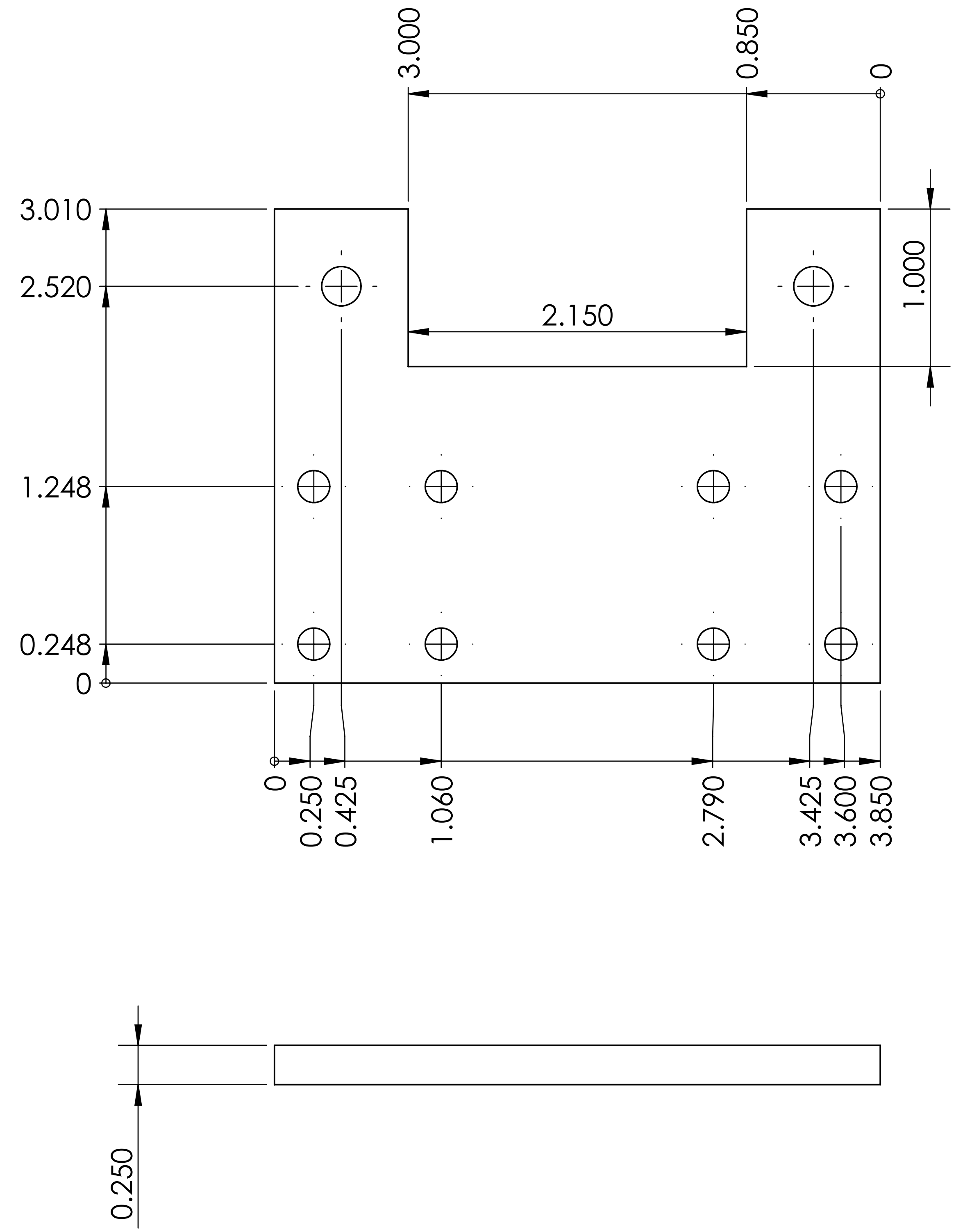
Cal Poly Mechanical Engineering ME ### - QTR YEAR	Lab Section:	Assignment #	Title: IDLER HOR. SUPPORT	Drwn. By: RYAN CANFIELD
	Dwg. #:	Nxt Asb:	Date: / Scale:	Chkd. By: ME STAFF

NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH ± 0.005 INCHES
ANGLES ± 1 °
DIAMETERS ± 0.001 INCHES
STEEL
QTY: 2



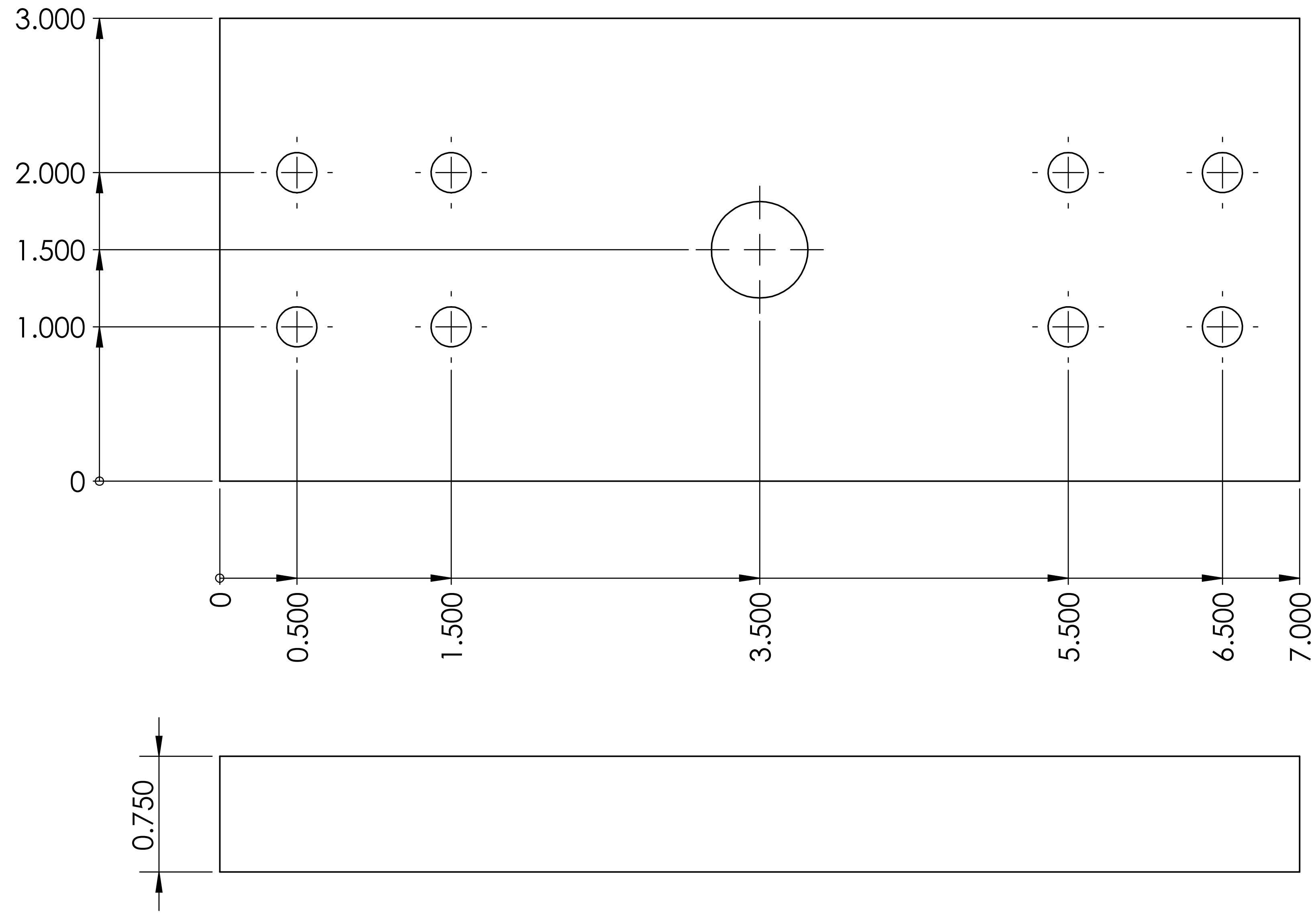
Cal Poly Mechanical Engineering	Lab Section:	Assignment #	Title: MOTOR SIDE HOR. SUPPORT		Drwn. By: RYAN CANFIELD
ME ### - QTR YEAR	Dwg. #:	Nxt Asb:	Date:	Scale:	Chkd. By: ME STAFF

NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH ± 0.005 INCHES
ANGLES ± 1 °
DIAMETERS ± 0.001 INCHES
ALUMINUM
QTY: 2



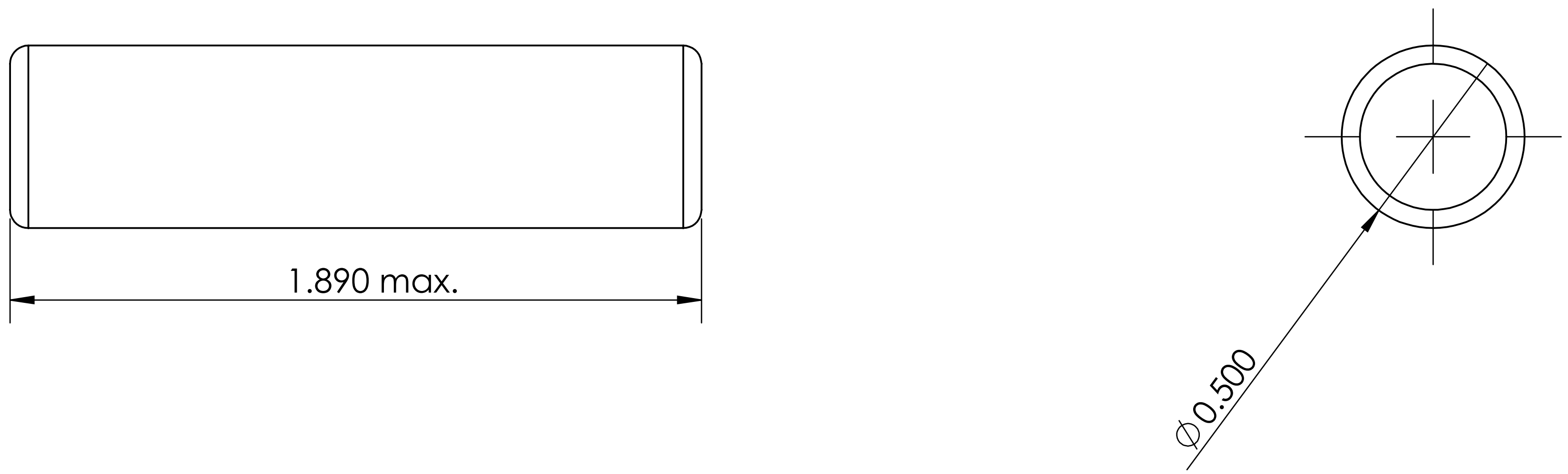
Cal Poly Mechanical Engineering	Lab Section:	Assignment #	Title: BELT ADAPTER	Drwn. By: RYAN CANFIELD
ME ### - QTR YEAR	Dwg. #:	Nxt Asb:	Date: / Scale:	Chkd. By: ME STAFF

NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH ± 0.005 INCHES
ANGLES ± 1 °
DIAMETERS ± 0.001 INCHES
STEEL
QTY: 4



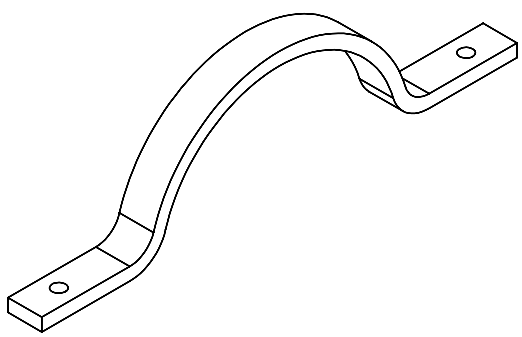
Cal Poly Mechanical Engineering	Lab Section:	Assignment #	Title: LEAD SCREW SUPPORT	Drwn. By: RYAN CANFIELD
ME ### - QTR YEAR	Dwg. #:	Nxt Asb:	Date: Scale:	Chkd. By: ME STAFF

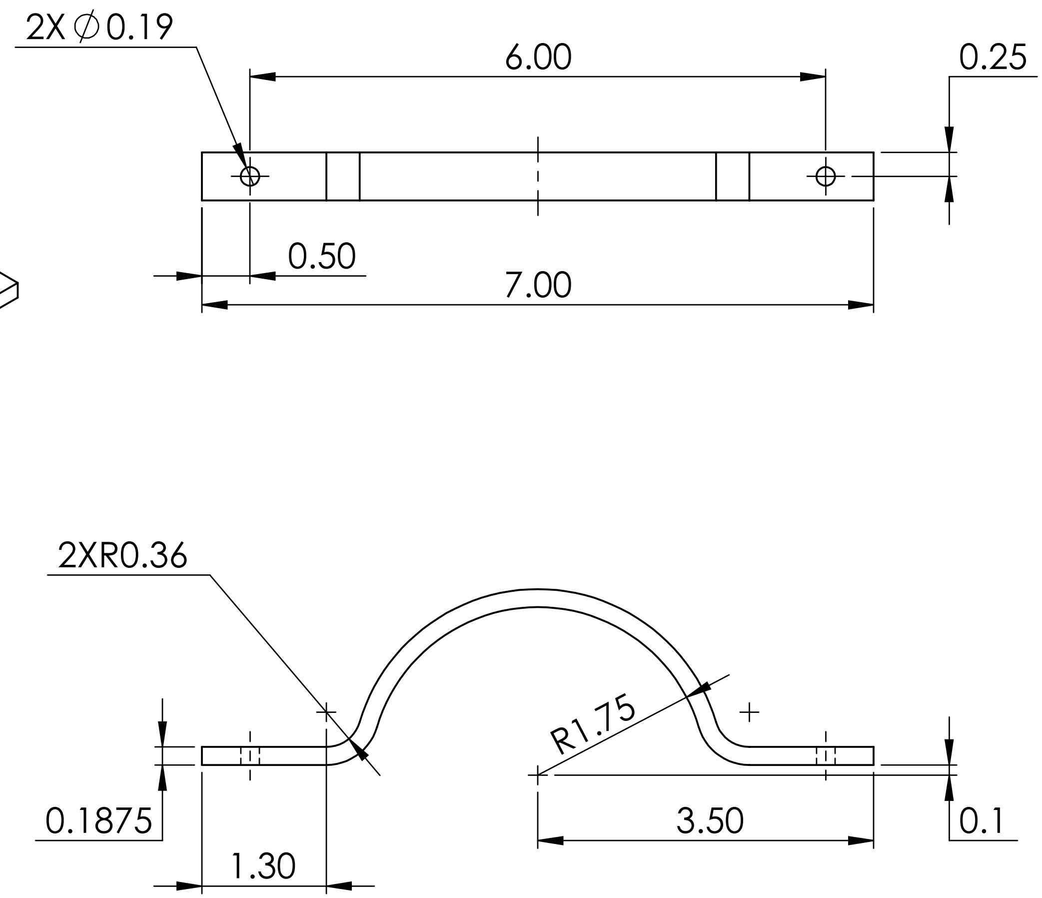
NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH $\pm$ 0.005 INCHES
ANGLES $\pm$ 1 °
DIAMETERS $\pm$ 0.001 INCHES
STAINLESS STEEL
QTY: 2



Cal Poly Mechanical Engineering	Lab Section:	Assignment #	Title: PULLEY SHAFT	Drwn. By: RYAN CANFIELD
ME ### - QTR YEAR	Dwg. #:	Nxt Asb:	Date: / Scale:	Chkd. By: ME STAFF

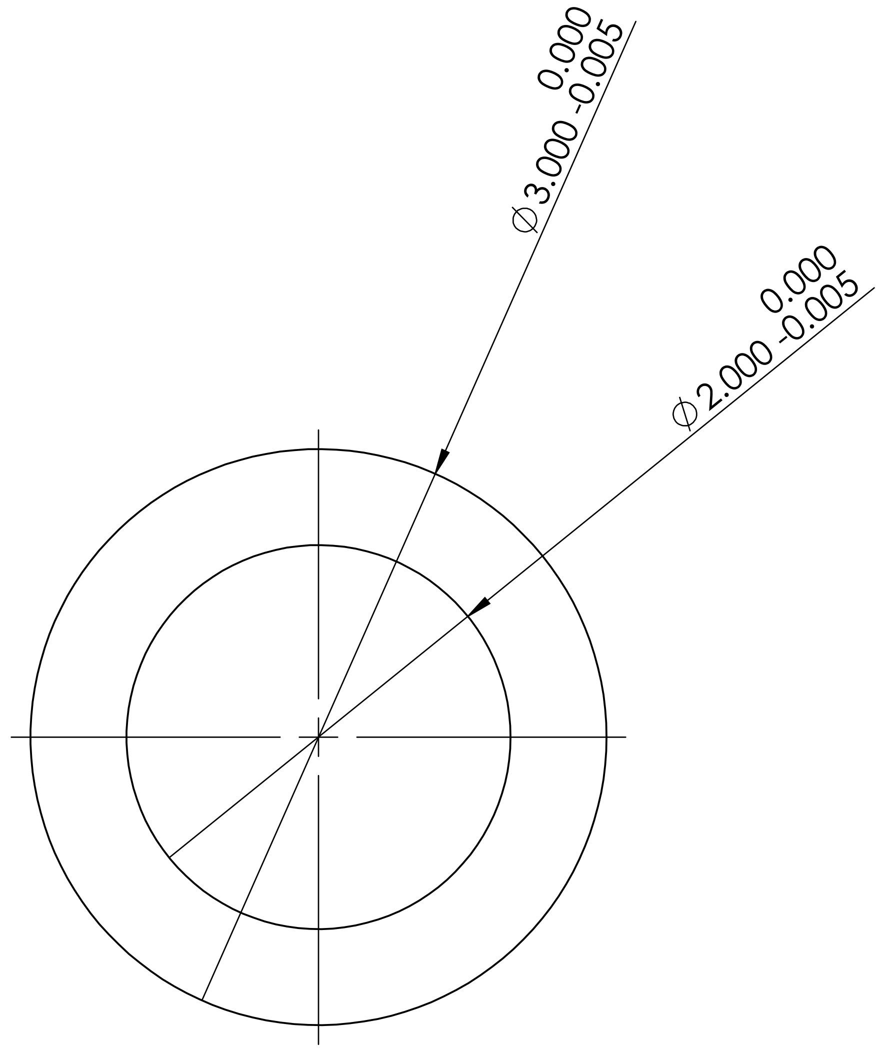
NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH ± 0.005 INCHES
ANGLES ± 1°
DIAMETERS ± 0.001 INCHES
STEEL
QTY: 2

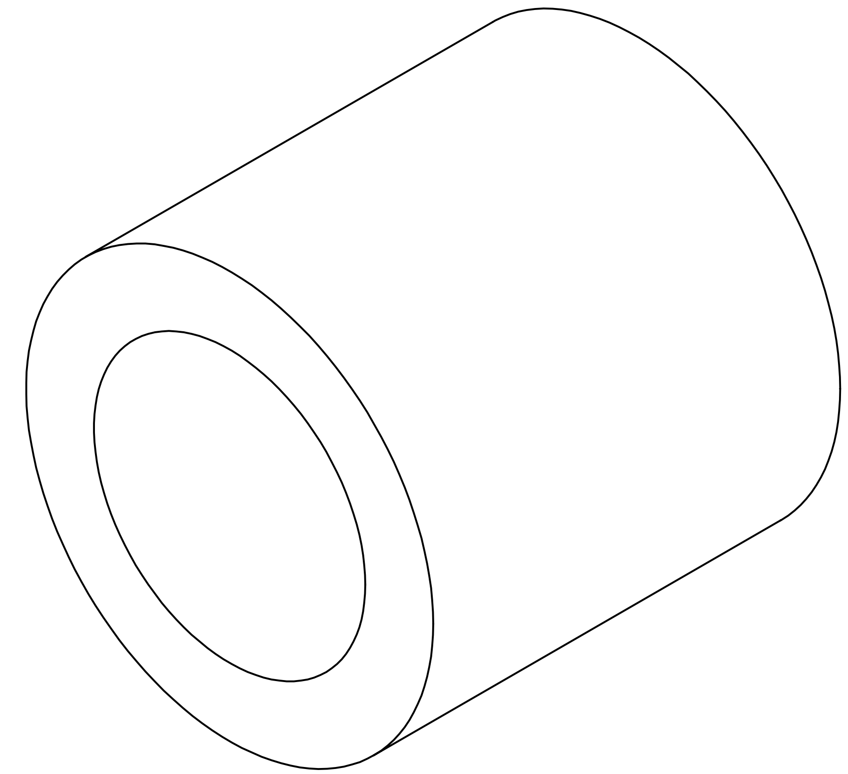


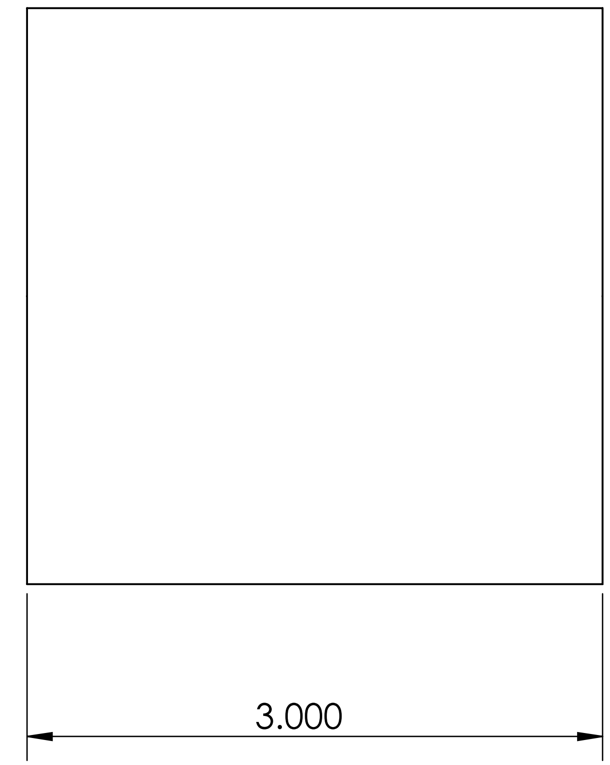


Cal Poly Mechanical Engineering: Lab Section: 02	SENIOR PROJECT	Title: STRAP		Drwn. By: LOUIS ROSEGUO
ME 429 - Spring 2016	Dwg. #: 9018K11	Nxt Asb:V-BLOCK	Date: 5/2/2016	Scale: 1:2	Chkd. By: ME STAFF

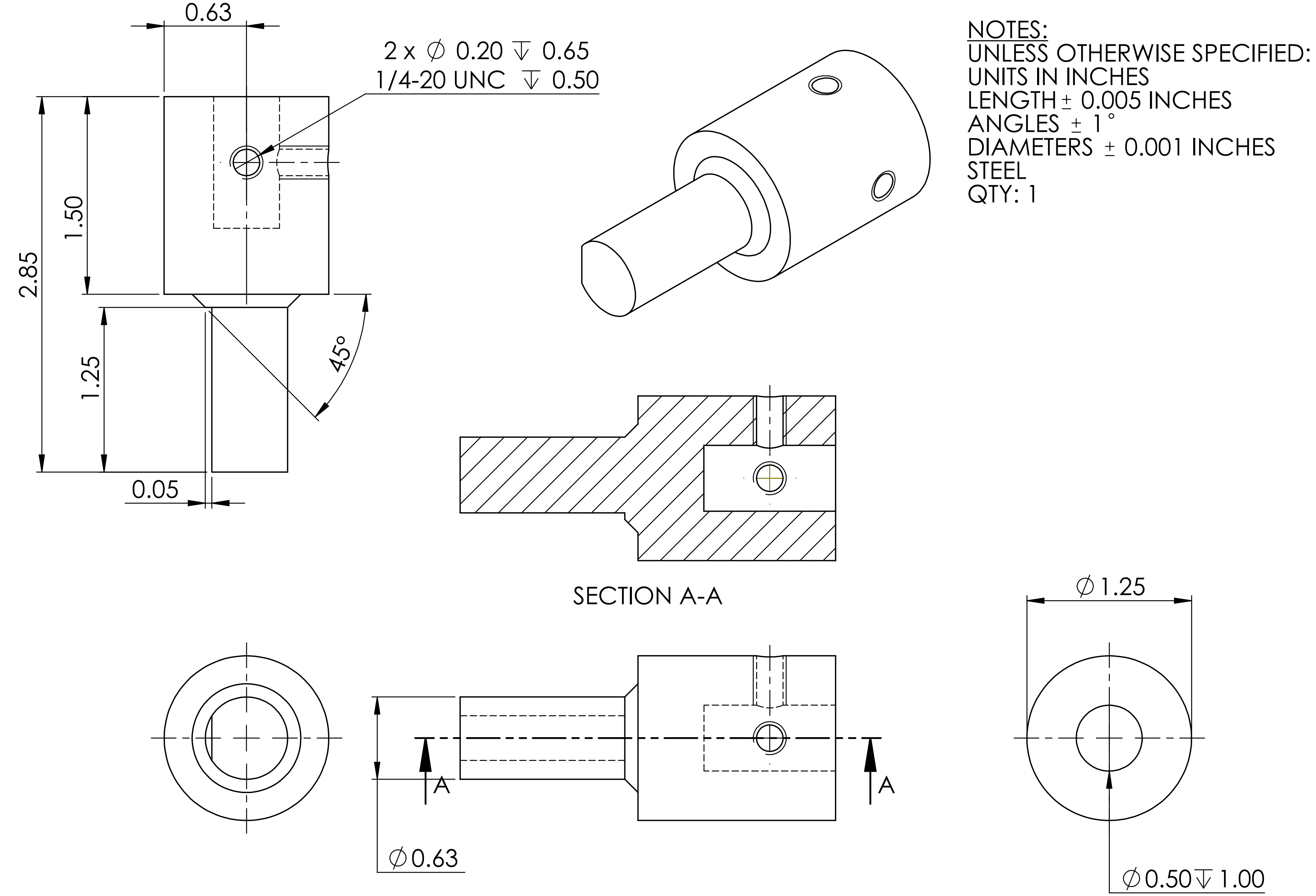
NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH ± 0.005 INCHES
ANGLES ± 1 °
DIAMETERS ± 0.001 INCHES
NYLON
QTY: 1







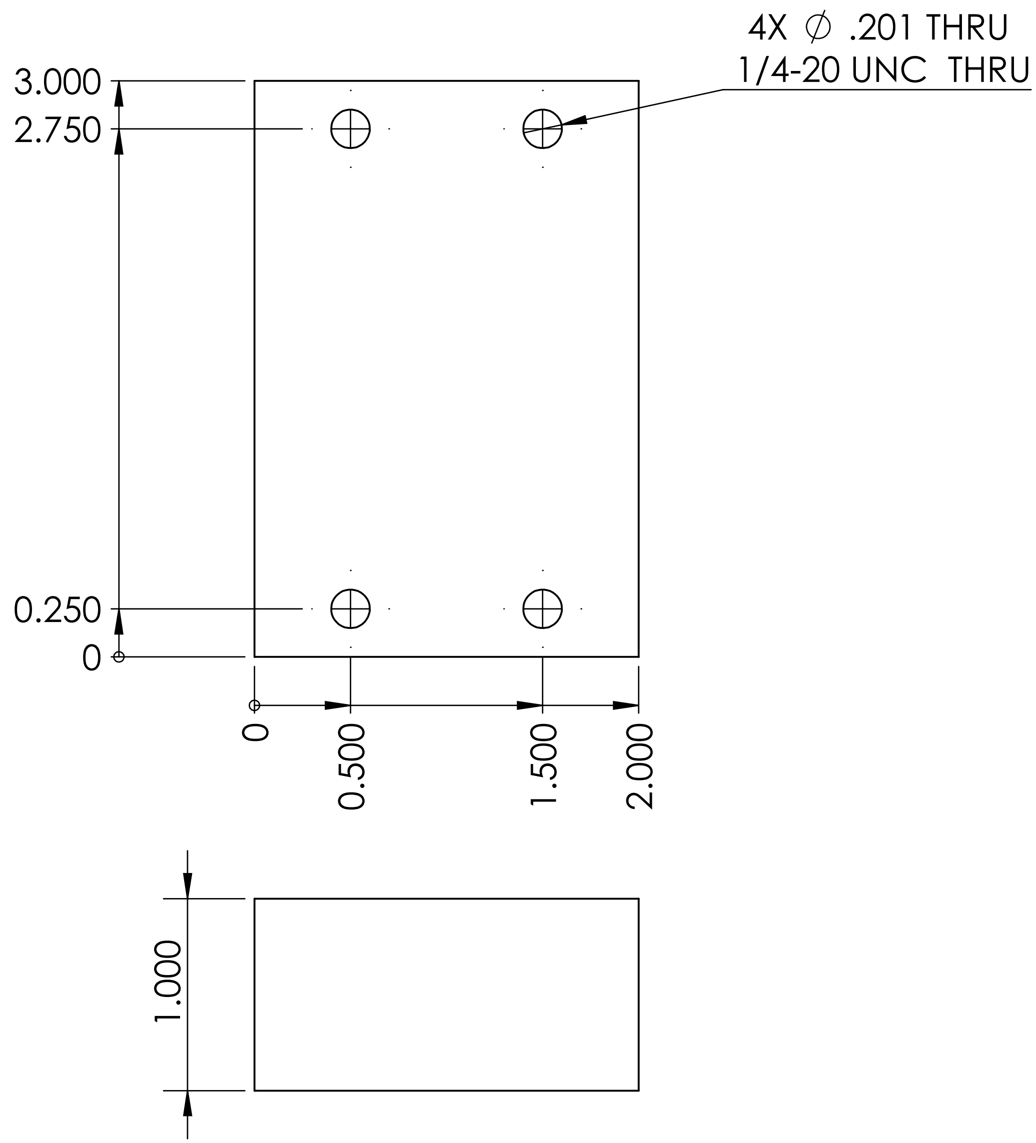
Cal Poly Mechanical Engineering	Lab Section:	Assignment #	Title: NYLON BLOCK	Drwn. By: RYAN CANFIELD
ME ### - QTR YEAR	Dwg. #:	Nxt Asb: TELE	Date: Scale:	Chkd. By: ME STAFF



NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH ± 0.005 INCHES
ANGLES ± 1°
DIAMETERS ± 0.001 INCHES
STEEL
QTY: 1

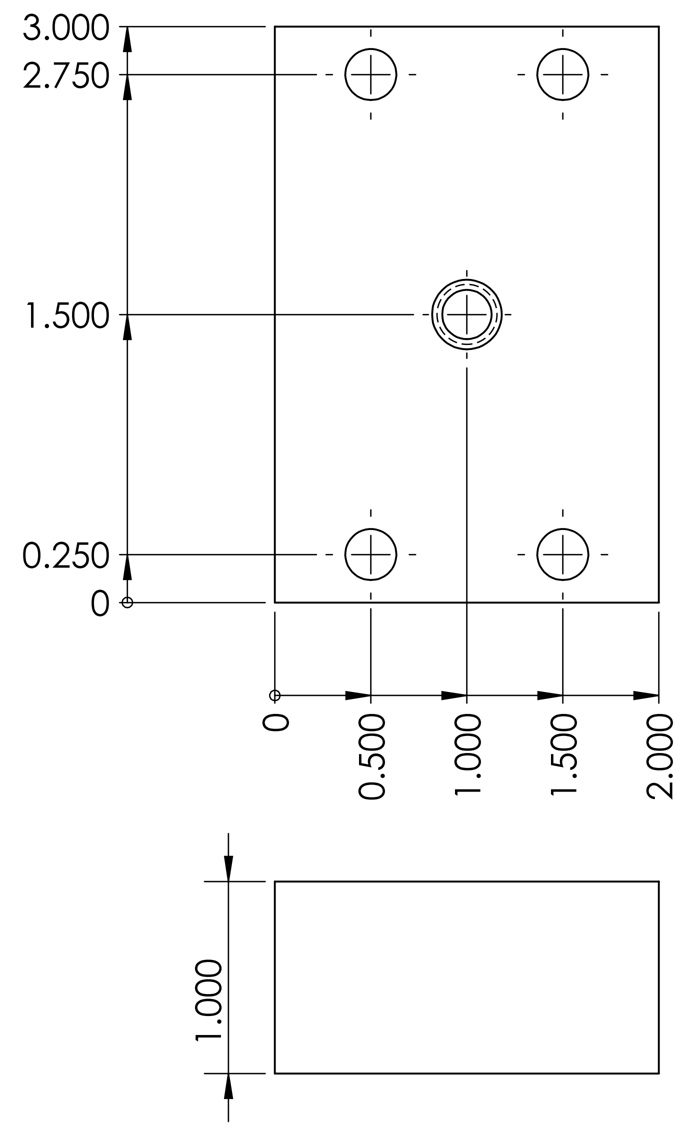
Cal Poly Mechanical Engineering: Lab Section: 02	SENIOR PROJECT	Title: SHAFT ADAPTER		Drwn. By: LOUIS ROSEGUO
ME 429 - Spring 2016	Dwg. #: 6776T26	Nxt Asb: MOUNT	Date: 5/3/2016	Scale: 1:1	Chkd. By: ME STAFF

NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH ± 0.005 INCHES
ANGLES ± 1 °
DIAMETERS ± 0.001 INCHES
STEEL
QTY: 2



Cal Poly Mechanical Engineering Lab Section: 02	SENIOR PROJECT	Title: IDLER BOTTOM		Drwn. By: RYAN CANFIELD
ME 429 - Spring 2016	Dwg. #:	Nxt Asb:	Date:	Scale:	Chkd. By: ME STAFF

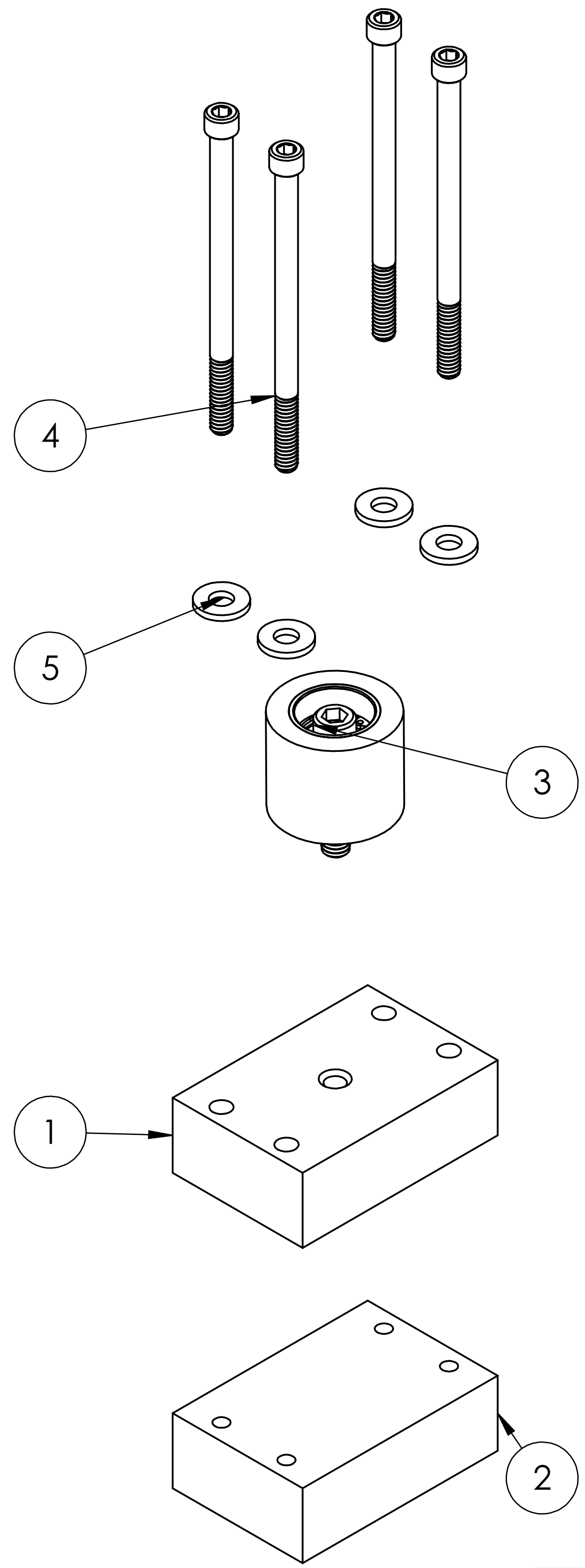
NOTES:
UNLESS OTHERWISE SPECIFIED:
UNITS IN INCHES
LENGTH ± 0.005 INCHES
ANGLES ± 1 °
DIAMETERS ± 0.001 INCHES
STEEL
QTY: 2



Cal Poly Mechanical Engineering Lab Section: 02	SENIOR PROJECT	Title: IDLER TOP		Drwn. By: RYAN CANFIELD
ME 429 - Spring 2016	Dwg. #:	Nxt Asb:	Date:	Scale:	Chkd. By: ME STAFF

P1

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	-	Idler Top	1
2	-	Idler Bottom	1
3	22875T62	Threaded Idler	1
4	91251A550	1/4" - 20 Socket Head Cap Screw (2" Length)	4
5	98023A029	1/4 - 20 Flat Washer	4



NEXT ASSY | USED ON

APPLICATION

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN		
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

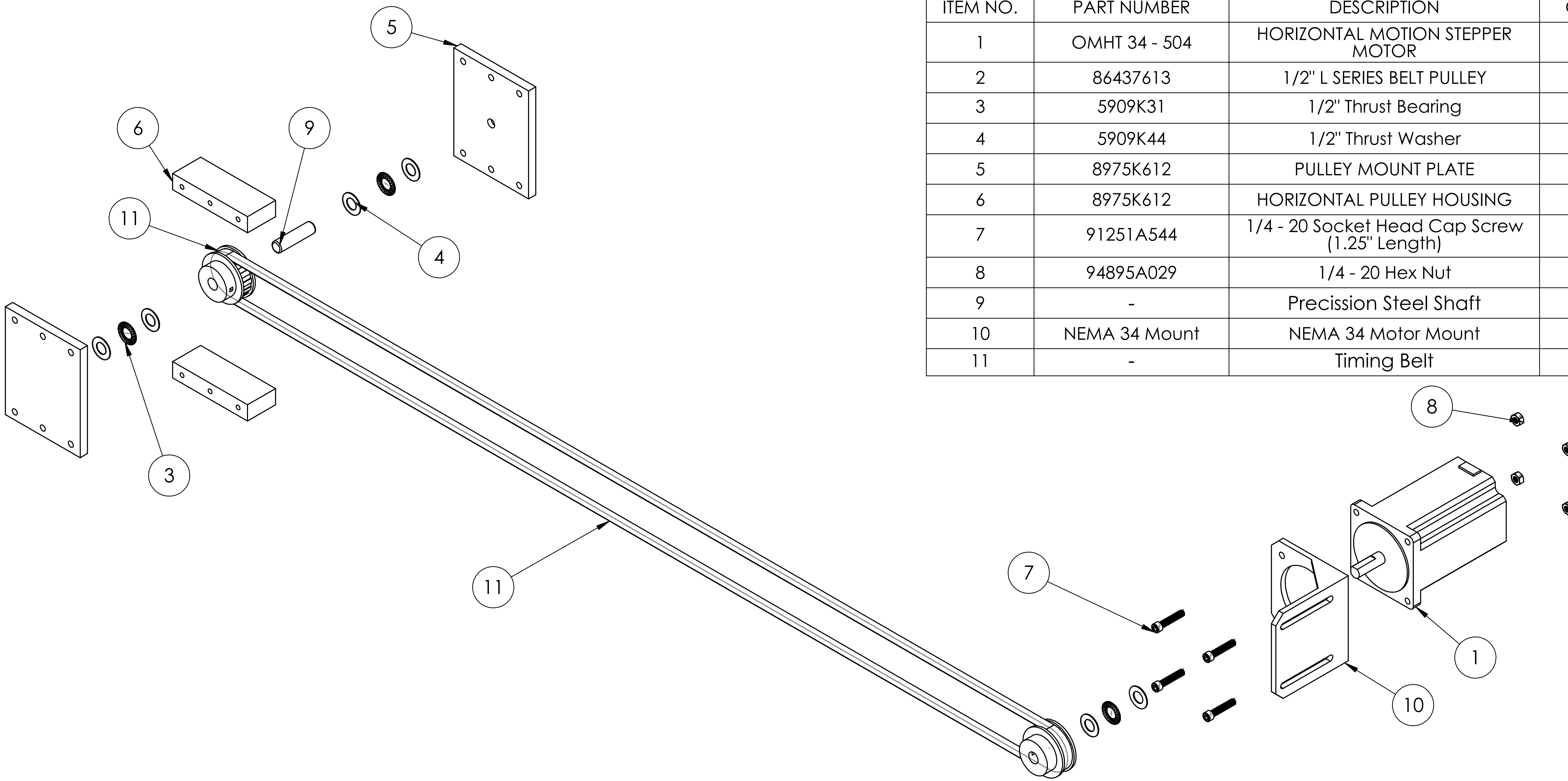
TITLE:

SIZE: B | DWG. NO.: Idler Assembly Exploded View | REV

SCALE: 1:4 | WEIGHT: | SHEET 1 OF 1

P2

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	OMHT 34 - 504	HORIZONTAL MOTION STEPPER MOTOR	1
2	86437613	1/2" L SERIES BELT PULLEY	2
3	5909K31	1/2" Thrust Bearing	3
4	5909K44	1/2" Thrust Washer	6
5	8975K612	PULLEY MOUNT PLATE	2
6	8975K612	HORIZONTAL PULLEY HOUSING	2
7	91251A544	1/4 - 20 Socket Head Cap Screw (1.25" Length)	4
8	94895A029	1/4 - 20 Hex Nut	4
9	-	Precission Steel Shaft	1
10	NEMA 34 Mount	NEMA 34 Motor Mount	1
11	-	Timing Belt	1



SOLIDWORKS Educational Product. For Instructional Use Only

NEXT ASSY | USED ON | APPLICATION

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
FINISH
DO NOT SCALE DRAWING

	NAME	DATE
DRAWN		
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		
COMMENTS:		

TITLE:

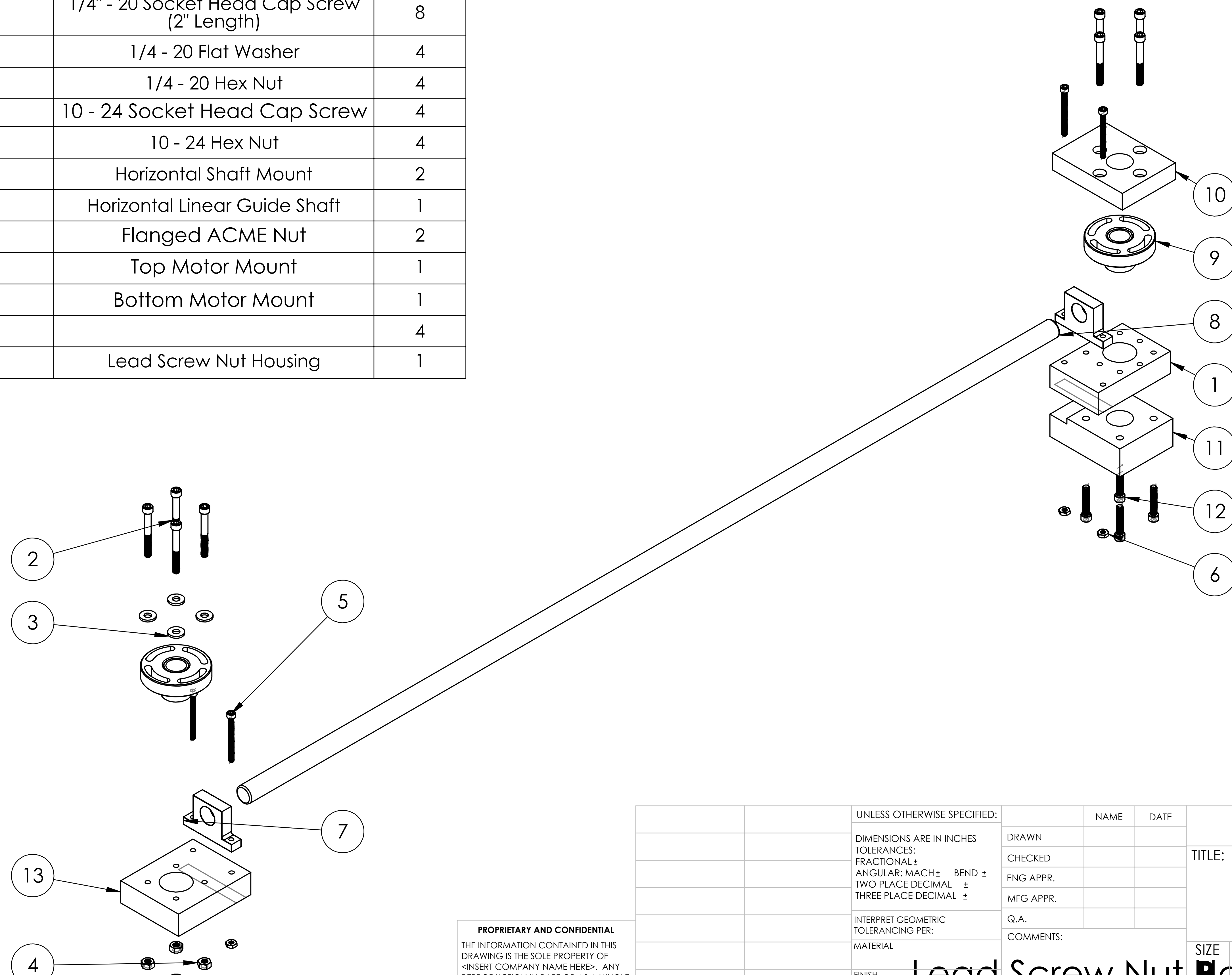
Horizontal Actuator Assembly Exploded View

SIZE B | DWG. NO. | REV

SCALE: 1:12 | WEIGHT: | SHEET 1 OF 1

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	8975K87	Lead Screw Nut Housing	1
2	91251A550	1/4" - 20 Socket Head Cap Screw (2" Length)	8
3	98023A029	1/4 - 20 Flat Washer	4
4	94895A029	1/4 - 20 Hex Nut	4
5	-	10 - 24 Socket Head Cap Screw	4
6	90480A011	10 - 24 Hex Nut	4
7	1865K5	Horizontal Shaft Mount	2
8	6061K646	Horizontal Linear Guide Shaft	1
9	-	Flanged ACME Nut	2
10	-	Top Motor Mount	1
11	-	Bottom Motor Mount	1
12	91251A544		4
13	8975K87	Lead Screw Nut Housing	1

P3



NEXT ASSY | USED ON
APPLICATION

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
FINISH
DO NOT SCALE DRAWING

	NAME	DATE
DRAWN		
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

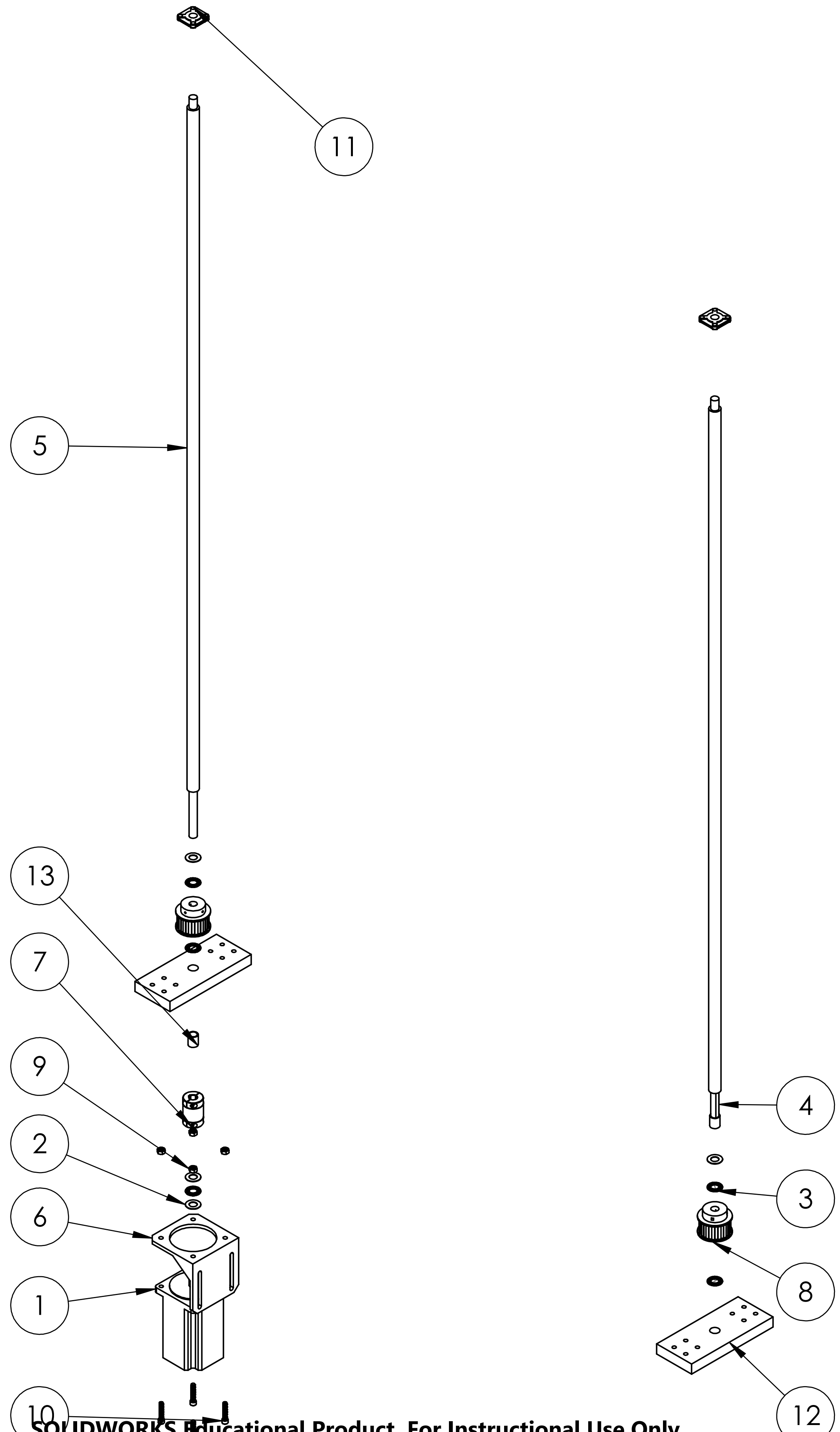
TITLE:

SIZE B DWG. NO. Lead Screw Nut Holder Assembly Exp REV

SCALE: 1:16 WEIGHT: SHEET 1 OF 1

P4

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	OMHT 34 - 506	VERTICAL STEPPER MOTOR	1
2	5909K44	1/2" Thrust Washer	4
3	5909K31	1/2" Thrust Bearing	5
4	59286	ACME 1" - 5 Lead Screw (54" Length Off Motor Side)	1
5	59286	ACME 1" - 5 Lead Screw (52.5" Length Motor Side)	1
6		NEMA 34 Motor Mount	1
7	9800T1		1
8	86437613	L Series Timing Belt Pulley	2
9	94895A029	1/4 - 20 Hex Nut	4
10	91251A544	1/4 - 20 Socket Head Cap Screw (1.25" Length)	4
11	EFSI - 08	Flange Vertical Shaft Mount	2
12	-	Lead Screw Support	2
13	6381K47		2



NEXT ASSY | USED ON
APPLICATION

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
FINISH
DO NOT SCALE DRAWING

	NAME	DATE
DRAWN		
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

TITLE:
VERTICAL ACTUATOR ASSEMBLY

SIZE B | DWG. NO. | REV

WEIGHT: | SHEET 1 OF 1

Part #	Description	Price/Part	Quantity	Cost	Store Purchase Link
629K5143	Rack, phi = 14.5, pitch = 12, F = 0.75"	$74.95	1	$74.95	McMaster
6867K61	Pinion, phi = 14.5, pitch = 12, F = 0.75"	$49.40	1	$49.40	McMaster
9533T9	Linear Sleeve Bearing for Telescoping	$111.35	1	$111.35	McMaster
9968K31	Retaining Ring for Bearing	$14.16	2	$28.32	McMaster
TNYLNSM	Nylon Sleeve Bearing (2X3X26)	$100.00	0.0769	$7.69	Professional Plastics
HT34-506	Stepper Motor for Rack and	$185.40	1	$185.40	Walker Industrial
92158A432	Set Screws for Shaft Adapter	$5.97	0.2	$1.19	McMaster
6776T26	Steel Rod for Custom Shaft Adapter	$27.30	1	$27.30	McMaster
6374K132	Linear Sleeve Bearing for V-	$65.18	3	$195.54	McMaster
9018K11	Metal Sheet for Custom Straps	$20.57	1	$20.57	McMaster
9017K76	Angle Iron for Motor Hanger	$5.81	1	$5.81	McMaster
3043T89	U-Bolt for Motor Hanger	$4.99	1	$4.99	McMaster
8910K21	.5"x4"x1' Steel for V-block	$25.65	1	$25.65	McMaster
9517K531	.5"x4"x2' Precision Steel for V-	$72.03	1	$72.03	McMaster
8910K702	.5"x3"x2' Steel for V-block	$36.28	1	$36.28	McMaster
90480A011	10 - 24 Nut	$1.84	0.08	$0.15	McMaster
8910K949	.5"x2"x2' Steel for Motor Mount	$24.61	1	$24.61	McMaster
91251A247	10 - 24 Socket Head Screw, 1"	$12.80	0.08	$1.02	McMaster
91251A245	10 - 24 Socket Head Screw,	$11.04	0.1	$1.10	McMaster
ST-M7	NEMA 34 Motor Mount	$10.87	1	$10.87	Stepper Online
91251A544	1/4 - 20 Socket Head Cap Screw (1 1/4 in)	$11.42	0.2	$2.28	McMaster
94895A029	Grade 8 Hex Nut 1/4" - 20 Thread Size	$3.22	0.1	$0.32	McMaster
98023A029	Washer for 1/4" - 20 Bolt	$6.36	0.1	$0.64	McMaster
91251A151	6-32 Socket Head Cap Screw	$8.63	0.05	$0.43	McMaster
dom3.5x.25	3.5" OD 3" ID (Steel, 64" Length	$207.36	1	$207.36	Speedy Metals
45gr2-72	2" Solid Pipe (72" Length)	$155.50	1	$155.50	Speedy Metals
5909K36	Thrust Needle Roller Bearing	$3.17	4	$12.68	McMaster
304070	Lead Screw for Vertical	$21.00	4	$84.00	IMService
HT34-506	Stepper Motor for Lead	$185.40	2	$370.80	Walker Industrial
SN 4 X 28	Vertical Linear Guide Shafts	$12.47	4	$0.00	LinTech
9143K21	Spacer Block Steel	$14.59	1	$14.59	McMaster
89422	ACME Lead Screw Nut	$5.91	4	$23.64	Roton
8975K87	Aluminum Lead Screw Nut Housing	$13.65	2	$27.30	McMaster
6374K125	Mounted Linear Sleeve Bearings (Vertical)	$47.84	4	$0.00	Amazon
EFSI - 08	Vertical Shaft Spherical Flange End Bearings	$8.15	8	$65.20	igus
16L050-6FA6	L Belt Pulley (1/2 in Belt Width)	$15.44	8	$123.52	B&B Manufacturing
7959K26	Horizontal MotionLead Screw Driving Timing Belt (1/2"	$42.12	4	$168.48	McMaster
5909K31	Thrust Bearing (1/2" ID)	$3.11	8	$24.88	McMaster
5909K44	Thrust Washer	$1.02	8	$8.16	McMaster
8975K226	Vertical Weld Plate Aluminum	$20.02	1	$0.00	McMaster
8927K58	Steel Rod for Shaft Coupling	$36.22	1	$36.22	McMaster
UCF204-12	Top Lead Screw Bearing (3/4 in)	$6.90	4	$27.60	The Big Bearing Store

91102A730	6 - 32 Steel Flat Washer	$6.29	0.16	$1.01	McMaster
90480A007	6 - 32 Steel Hex Nut	$1.24	0.16	$0.20	McMaster
91251A151	6-32 Socket Head Cap Screw	$8.63	0.16	$1.38	McMaster
98023A029	Washer for 1/4" - 20 Bolt	$6.36	0.24	$1.53	McMaster
94895A029	Grade 8 Hex Nut 1/4" - 20 Thread Size	$3.22	0.24	$0.77	McMaster
91251A550	1/4 - 20 Socket Head Cap Screw (2 in)	$6.32	0.32	$2.02	McMaster
91251A544	1/4 - 20 Socket Head Cap Screw (1 1/4 in)	$11.42	0.4	$4.57	McMaster
91251A245	10 - 24 Socket Head Cap Screw (3/4)	$11.04	0.16	$1.77	McMaster
96765A125	10 - 24 Flat Washer	$4.09	0.12	$0.49	McMaster
90480A011	10 - 24 Low Strength Steel	$1.84	0.12	$0.22	McMaster
92158A171	6 - 32 Set Screw (3/8" Length)	$4.65	0.4	$1.86	McMaster
8554K1	Self Lubricating Nylon Rod	$6.50	0.067	$0.43	McMaster
6061K646	Precision Steel Shaft 3/4" Diameter, 42" Length	$38.10	2	$76.20	McMaster
1865K5	3/4" Shaft Support	$20.27	4	$81.08	McMaster
HT34-504	Horizontal Motion Stepper	$97.20	2	$194.40	Walker Industrial
8975K612	Horizontal Motor Aluminum Housing (2'x3.5"x0.25")	$14.08	2	$28.16	McMaster
HFZ40	Double Acting Pneumatic Air Gripper	$233.27	1	$233.27	Radwell International
4V130EM5B	Solenoid Valve (Double Acting, 5/3 way, exhaust center, 24V	$32.28	1	$32.28	Radwell International
a15090900ux0886	Tube Port Insert (M5x0.8 screw, 6mm tube diameter)	$26.20	0.2	$5.24	Amazon
US98A060040200MGE	Pneumatic Tubing (6mm OD, 4mm ID, 200 m)	$65.59	0.05	$3.28	Radwell International
91290A461	Fasteners (M8x1.25, 75 mm)	$9.46	0.16	$1.51	McMaster
8910K58	Pneumatic gripper finger steel (1" Thick, 4" Width, 1/2 ft long)	$25.46	1	$25.46	McMaster
1376N11	Adhesive rubber grip sheet (1" x 36", 1/64" Thick)	$8.06	1	$8.06	McMaster
91502A199	Fasteners (M8x1.25, 35 mm)	$6.38	0.24	$1.53	McMaster
8910K38	Spacer Metal Sheet (3/4" Thick, 3.5" Width, 1/2 ft long)	$19.17	1	$19.17	McMaster
7959K28	L-Series Timing Belt (1" Width) (Pitch Length > 120 in)	$108.79	2	$217.58	SDP/SI
SPB16L100BF-500	L-Series Timing Belt Pulley	$18.50	6	$111.00	Automation Direct
6381K453	Multipurpose Sleeve Bearing (3/4" Long)	$1.88	8	$15.04	McMaster-Carr
13T460	Multipurpose Sleeve Bearing (1.5" Long)	$16.16	1	$16.16	Grainger
8975K255	6061 Aluminum (1.5" x 3" x 2ft)	$64.44	1	$64.44	McMaster-Carr
4459T183	4130 Steel (0.5" Thick)	$30.69	1	$30.69	McMaster-Carr
6554K51	Steel (0.75" x 8" x 3ft)	$96.84	1	$96.84	McMaster-Carr
8975K45	6061-T6 Aluminum (0.75" x 1.5" x 2ft)	$19.17	2	$38.34	McMaster-Carr
383	NEMA 34 Stepper Motor	$10.87	4	$43.48	Stepper Online
9800T1	Rigid Shaft Coupling	$41.93	2	$83.86	McMaster-Carr
98491A101	Key Stock	$2.45	1	$2.45	McMaster-Carr

90044A316	Socket Head Cap Screw (1/4"-20 X 3.5" Long)	$5.30	3	$15.90	McMaster-Carr
91251A542	Socket Head Cap Screw (1/4"-20 X 1" Long)	$7.84	1	$7.84	McMaster-Carr
47065T107	Double Rod	$33.95	8	$271.60	McMaster-Carr
47065T501	Quad Rod	$59.37	5	$296.85	McMaster-Carr
47065T189	Double Rod Support	$18.49	12	$221.88	McMaster-Carr
2468T61	Frame Wheel	$107.20	4	$428.80	McMaster-Carr
91251A012	Socket Head Cap Screw (1/2"'-20 1" Long)	$5.35	4	$21.40	McMaster-Carr
92196A536	Socket Head Cap Screw (1/4"'-20 7/16" Long)	$6.50	1	$6.50	McMaster-Carr
95462A525	Hex Nut (1/2")	$14.76	1	$14.76	McMaster Carr
47065T226	Drop in Connector	$1.12	100	$112.00	McMaster-Carr
8364T5	Precision Steel Shaft	$7.09	1	$7.09	McMaster-Carr
47065T155	Concealed 80/20 Connector	$1.79	10	$17.90	McMaster-Carr
5909K36	Thrust Needle Roller Bearing	$3.17	13	$41.21	McMaster
22875T4	Threaded Idler	$15.68	8	$125.44	McMaster-Carr
5909K31	Thrust Bearing (1/2" ID)	$3.11	26	$80.86	McMaster
98409A294	Internal Retaining Ring for 3" Bore Diameter	$2.59	2	$5.18	McMaster-Carr
8961K16	DIN Rail	$8.67	1	$8.67	McMaster-Carr
73262	ACME Nut Flange	N/A	4	N/A	Nook Industries
STR8	Stepper Motor Driver	$217.00	3	$651.00	Applied Motion
	Power Supply	$0.00		$0.00	
D0-06DD1-D	DL06 PLC Base Unit	$251.00	1	$251.00	Automation Direct
H0-CTRIO2	PLC Motion Control Expansion Module	$214.00	2	$428.00	Automation Direct
D0-06LCD	PLC LCD Display (Optional)	$83.00	1	$83.00	Automation Direct
PC-DSOFT6	DirectSOFT 6 Programming and Documentation Software	$395.00	1	$395.00	Automation Direct
D2-DSCBL	Programming Cable	$14.00	1	$14.00	Automation Direct
GCX3131	22 mm Emergency Stop	$9.25	1	$9.25	Automation Direct
GCX3300	22 mm 2-Position Selector	$7.25	2	$14.50	Automation Direct
966-1442-ND	Limit Switch With Roller	$6.76	10	$67.60	Digi-Key
ECX3510	22 mm 2-Position Momentary Joystick	$20.50	1	$20.50	Automation Direct
ECX3520	22 mm 4-Position Momentary Joystick	$27.50	1	$27.50	Automation Direct
GCX3100	22 mm Black Momentary	$4.75	2	$9.50	Automation Direct
GCX3102	22 mm Green Momentary Pushbutton	$4.75	1	$4.75	Automation Direct
GCX3104	22 mm Blue Momentary	$4.75	1	$4.75	Automation Direct
ECX2051-24L	22 mm Red LED Indicator	$5.00	1	$5.00	Automation Direct
ECX2052-24L	22 mm Green LED Indicator	$5.00	1	$5.00	Automation Direct
ECX2053-24L	22 mm Yellow LED Indicator	$5.00	1	$5.00	Automation Direct
PBGX12	12-Hole 22 mm Pushbutton Enclosure	$101.00	1	$101.00	Automation Direct
1470-2223-ND	Power Supply	$425.01	1	$425.01	Digi-Key
FAZ-D0P5-1-NA-SP	0.5 A Circuit Breaker	$18.00	6	$108.00	Automation Direct
FAZ-C7-1-NA-SP	7 A Circuit Breaker	$18.00	3	$54.00	Automation Direct
FAZ-C1-1-NA-SP	1 A Circuit Breaker	$18.00	6	$108.00	Automation Direct
CF14JT10K0CT-ND	10 kOhm Resistor	$0.04	20	$0.80	Digi-Key
CF14JT1K00CT-ND	1 kOhm Resistor	$0.04	20	$0.80	Digi-Key
CF14JT330RCT-ND	330 Ohm Resistor	$0.04	10	$0.40	Digi-Key

CF14JT220RCT-ND	220 Ohm Resistor	$0.04	5	$0.20	Digi-Key

ME429 DVP&R Format

Report Date: 5/4/2016		Sponsor: Haas Automation, Inc.				Component/Assembly: Tender	REPORTING ENGINEER: Louis

TEST PLAN										TEST REPORT			
Item No	Specification or Clause Reference [1]	Test Description [2]	Acceptance Criteria [3]	Test Responsibility [4]	Test Stage [5]	SAMPLES TESTED		TIMING		TEST RESULTS			NOTES
						Quantity	Ty [6]	Start dat	Finish dat	Test Res [7]	Quantity Pass	Quantity Fail	
1	Loading Speed	Begin timer at door open, end timer when retracted safely from Vf-3	30 Second Max	All	DV	10	B	5/27/2016	6/7/2016				
2	Repeatability	Computer vision, or lasers report variances in position	0.025" TIR	All	DV	50	B	5/27/2016	6/7/2016				
3	Emergency Stop	Video, or accelerometer set to trigger a timer	0.5 Second Max	All	DV	10	B	5/27/2016	6/7/2016				
4	Payload	Grip part, move, observe for slip	5 lbs Minimum	All	DV	10	B	5/27/2016	6/7/2016				
5	Part Size	Measure	2"x4"x6"	Sam	CV	1	A	5/27/2016	6/7/2016				
6	Part Capacity	Calculate	40 Parts Minimum	All	DV	40	B	5/27/2016	6/7/2016				
7	24 hour MTTR	Time access to part, removal, reattachment	24 Hous Maximum	All	DV	1	B	5/27/2016	6/7/2016				
8	Operator change	Time Rotation	10 Minutes Maximum	All	DV	5	B	5/27/2016	6/7/2016				
9	Time to Reconfigure	Measure time spent inactive	20 Minutes Maximum	All	DV	5	B	5/27/2016	6/7/2016				
10	Reach	Measure	1.5 Meters Minimum	Louis	CV	1	A	5/27/2016	6/7/2016				
11	Communicate with VF-3	VF-3 Signals simulated with buttons; Tender signals simulated with lights	Absolute	All	DV	50	B	5/27/2016	6/7/2016				
12	Sense foreign objects in range of motion	Introduce foreign object, observe for change in behavior	Absolute	All	DV	20	B	5/27/2016	6/7/2016				

[1] Internal and / or Customer Specification Reference

[2] From Specification clause description of test being undertaken

[3] Pass / Fail targets and Pass / Fail criteria
e.g cycles, volts, minimum values, no fail etc.

[4] Student who takes responsibility to make sure test is completed

[5] Test Stage:
CV= Concept verification (hand made)
DV= Design verification (part tooled)
PV= Product and Process validation

[6] Sample type:
A = Concept verification
B = Design verification
C = Product validation

[7] Record actual value of result

S1

S2

S3

S4

S5

S6

```
// Name Telescoping Motion Pins
int TelPulse = 9;
int TelDir = 2;
int TelEna = 3;

// Name Horizontal Motion Pins
int HPulse = 10;
int HDir = 6;
int HEna = 7;

// Name Vertical Motion Pins
int VPulse = 11;
int VDir = 4;
int VEna = 5;

// Name Input Pins
int Estop = 19; // Analog pin A5
int Go = 18;  // Analog pin A4
int Telout = 17;  // Analog pin A3
int Telin = 16; // Analog pin A2
int Hleft = 15;  // Analog pin A1
int Hright = 14; // Analog pin A0
int Vertup = 13;
int Vertdown = 12;

// Declare Global Variables
boolean AllClear = true;  // When true, allows motors to move. Cleared by
Emergency Stop. Set by Toggling "Go."
boolean ToggleState = false;  // Must be set true, then false by toggling
"Go" to allow motion after E-Stop.

// Define Functions

// This function checks to see if the emergency stop has been pressed
void checkEstop()
{
  if (digitalRead(Estop))
  {
    AllClear = false;
    disablemotion();
  }
}

// This function should enable or re-enable motion when called
```

```
void enablemotion()
{
    // Turn on driver enable outputs
  digitalWrite(TelEna, HIGH);
  digitalWrite(HEna, HIGH);
  digitalWrite(VEna, HIGH);

  // Set Velocity to Zero
  analogWrite(TelPulse, 0);
  analogWrite(HPulse, 0);
  analogWrite(VPulse, 0);
}

// This function should be called when "AllClear" becomes false and should
not allow anything to happen until "Go" is toggled.
void waitforallclear(){
  if (!ToggleState) // If ToggleState is false
  {
    if (digitalRead(Go))  // If "Go" is switched on from off, i.e. this
should not work if "Go" was on to begin with
    {
      ToggleState = !ToggleState; // Toggle ToggleState
    }
  }
  else  // If ToggleState is true, i.e. "Go" was switched from off to on
  {
    if(!digitalRead(Go))  // If "Go" is switched back off
    {
      ToggleState = !ToggleState; // Toggle ToggleState again
      AllClear = true;  // Signal the All Clear and let the system run again
      enablemotion();
    }
  }
}

// This function should disable all motion when called, and should be called
when "AllClear" is false.
void disablemotion()
{
  // Shut off driver enable outputs
  digitalWrite(TelEna, LOW);
  digitalWrite(HEna, LOW);
  digitalWrite(VEna, LOW);
```

```
  // Set Velocity to Zero
  analogWrite(TelPulse, 0);
  analogWrite(HPulse, 0);
  analogWrite(VPulse, 0);
}

// This function dictates telescoping motion
void telescoping()
{
  if (digitalRead(Telout) && !digitalRead(Telin)) // If the machine receives
a signal to extend the arm
  {
    digitalWrite(TelDir, HIGH); // Set direction of motion to outward
(remember to invert if it's the wrong way)
    analogWrite(TelPulse, 128); // Send pulse to telescoping motor
  }
  else if (!digitalRead(Telout) && digitalRead(Telin))  // If the machine
receives a signal to retract the arm
  {
    digitalWrite(TelDir, LOW);  // Set direction of motion to inward
(remember to similarly invert if it's the wrong way)
    analogWrite(TelPulse, 128); // Send pulse to telescoping motor
  }
  else  // In all other cases
  {
    analogWrite(TelPulse, 0); // Stop motor
  }
}

// This function dictates horizontal motion
void horizontal()
{
    if (digitalRead(Hleft) && !digitalRead(Hright)) // If the machine
receives a signal to move to the left
  {
    digitalWrite(HDir, HIGH); // Set direction of motion to the left
(remember to invert if it's the wrong way)
    analogWrite(HPulse, 128); // Send pulse to horizontal motors
  }
  else if (!digitalRead(Hleft) && digitalRead(Hright))  // If the machine
receives a signal to move to the right
  {
    digitalWrite(HDir, LOW);  // Set direction of motion to the right
(remember to similarly invert if it's the wrong way)
```

```
    analogWrite(HPulse, 128); // Send pulse to horizontal motors
  }
  else  // In all other cases
  {
    analogWrite(HPulse, 0); // Stop motors
  }
}

// This function dictates vertical motion
void vertical()
{
    if (digitalRead(Vertup) && !digitalRead(Vertdown)) // If the machine
receives a signal to move upwards
  {
    digitalWrite(VDir, HIGH); // Set direction of motion to upward (remember
to invert if it's the wrong way)
    analogWrite(VPulse, 128); // Send pulse to vertical motors
  }
  else if (!digitalRead(Vertup) && digitalRead(Vertdown))  // If the machine
receives a signal to move downwards
  {
    digitalWrite(VDir, LOW);  // Set direction of motion to downward
(remember to similarly invert if it's the wrong way)
    analogWrite(VPulse, 128); // Send pulse to vertical motors
  }
  else  // In all other cases
  {
    analogWrite(VPulse, 0); // Stop motor
  }
}


void setup() {
  // Set Up Output Pins
  pinMode(TelPulse, OUTPUT);
  pinMode(TelDir, OUTPUT);
  pinMode(TelEna, OUTPUT);

  pinMode(HPulse, OUTPUT);
  pinMode(HDir, OUTPUT);
  pinMode(HEna, OUTPUT);

  pinMode(VPulse, OUTPUT);
  pinMode(VDir, OUTPUT);
```

```
  pinMode(VEna, OUTPUT);

  // Set Up Input Pins
  pinMode(Estop, INPUT);
  pinMode(Go, INPUT);
  pinMode(Telout, INPUT);
  pinMode(Telin, INPUT);
  pinMode(Hleft, INPUT);
  pinMode(Hright, INPUT);
  pinMode(Vertup, INPUT);
  pinMode(Vertdown, INPUT);

  enablemotion();


}

void loop() {

  checkEstop();

  if (AllClear)
  {
    telescoping();
    horizontal();
    vertical();
  }

  else
  {
    waitforallclear();
  }


}
```