Hexapod Linear Actuators

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Statement of Disclaimer

Table of Contents

List of Figures

List of Tables

List of Nomenclature

Actuator	A mechanism that puts something into automatic motion
Alt-Azimuth	A kinematic system, traditionally used for pointing telescopes
Angular Encoder	A sensor that counts the amount of revolutions an object has achieved
Backlash	Recoil arising between parts of a mechanism, a source of inaccuracy
Ball Joint	A three degree-of-freedom joint consisting of a ball in a socket
Ball Screws	A screw design that uses ogival grooves and a nut that balls are cycled through to translate the rotational motion of the screw into linear motion of the nut
EMLA	Electro Mechanical Linear Actuator: an actuator which uses an electric motor and some sort of screw design to drive a mechanical linear actuator
Hexapod	A 6 DOF positioning system that uses 6 actuators in parallel to achieve this
Inverse Dynamics	Is a method back solving for the forces and moments of an object based off of the kinematics of the system
Kinematic System	A positioning system whose degrees of freedom directly depend on each other and therefore their inaccuracies stack
Ogival Thread	A type of threads in which the geometry which is defined to consist of two arcs with equal radii and offset centers.
Parallel System	a positioning systems who's degrees of freedom are independent of each other
Piezoelectric	the property of certain elements to expand when induced with electricity
Pillow Block	A steel block used for supporting a bearing, in our case it supports the bearing for our screw.
Prismatic Joint	A joint whose purpose is for linear extension and retraction
Sheave	The portion of our actuator that encases the screw and provides the rigidity for our prismatic joint
Slewing Velocity	The maximum change in velocity
Stewart Platform	See hexapod
Stiction	Static friction. The threshold of force which must be overcome in order to move two surfaces against each other
U-Joint	A two degree of freedom joint that uses two yokes connected with pins to achieve this
U-U Joint	A Universal-Universal joint created by this team to simulate the three degrees of freedom of a ball joint while still providing a 180^{o} range of motion.

Executive Summary

This report details the design and manufacture of a single large actuator for eventual integration into a telescope positioning system. The positioning system is designed as a hexapod, which will include six actuators identical to the one detailed here. The single manufactured actuator functions as a working prototype and test piece. Testing indicates that it could function in a hexapod without further modification. This actuator meets or addresses initial specifications provided by project sponsors, and an evolved set of specifications has been defined throughout the development phase through direct communication with these sponsors.

Chapter 1 – Introduction

Sponsor Background and Needs

The purpose of this project is to design and develop an actuation system for large telescopes used in university research. These telescopes are quite heavy and large—120 kg and 1.5 meters in diameter, making them difficult to maneuver. Current mount systems for moving these telescopes are likewise very heavy, expensive, and immobile, thus making them financially and physically impractical for most small universities.

There are currently 85 colleges and universities in the U.S. with astronomy programs. [(1)] By creating an inexpensive, lightweight, and readily available method for moving and pointing large telescopes, we hope to open a new market catering to the needs of university astronomy programs. Making large-scale telescopes available to university astronomy programs will allow students to take part in cutting-edge research that would otherwise be unavailable to them due to budgetary constraints. Because this need is relatively new, and no simple solution currently exists, this endeavor may also be very lucrative.

Formal Problem Definition

Current actuators required for the needed hexapod system are too expensive for most astronomy programs.

Objective/Specification Development

Actual Precision has researched and designed a hexapod system that will point the telescopes through the use of linear actuators. The idea of using linear actuators in the form of a hexapod for pointing telescopes has recently gained favor with the astronomical community. However, acquiring actuators that are large enough and strong enough to point these systems is very expensive, as most actuators used in industry have too small of a stroke for use in these applications. Our hexapod system design is based on linear actuators that will accommodate large telescopes and still maintain the degree of accuracy provided by state-of-the-art observatory equipment. Because our system will be used by universities and private researchers, it has been constructed relatively inexpensively. We believe that our low-cost pointing system is competitive with current astronomical standards, and we hope to increase the availability and popularity of astronomical research through realization of this project.

Although we have designed an entire hexapod system, our focus has been on constructing a portion of the system. Rather than building all six actuators, we have built and tested only one, proving that actuators of the necessary size, stroke, and accuracy can be built for a reasonable price. Construction of the remaining five will be pursued in a future endeavor. Our test plan for the single actuator involves speed, accuracy, and efficiency, which are requirements for successful integration into the full hexapod system.

The requirements detailed in Table 1 are based on a series of conversations our team had with Dr. Ridgely, our project advisor. Some requirements are direct translations from numerical requirements, such as resolution and accuracy; many values are based on the customer’s expressed desires. Our goal for this process was to quantify those desires, and to create a set of measureable parameters to which we could design.

Table 1. Engineering Specifications

Spec #	Parameter Description	Requirement (units)	Tolerance	Risk	Compliance (Analysis, Test, Similarity, Inspection)
1	Cost/Actuator	$2000 per actuator	%100	H	S
2	Supported Load	60 kN	Minimum	M	A,T,I
3	Weight	Low	---	H	A,T
4	Base Radius	1.5 m	Given	H	A,T,I
5	Accuracy	.112 mm	Minimum	M	A,T
6	Resolution	.004 mm	Minimum	H	A,T
7	Range of Motion	1.0 m	Minimum	M	A,T,I
8	Sensors Digitizing Actuator Length	Yes	---	M	T,I
9	Extension Speed	5.56 cm/s	Minimum	L	T, I
10	Retraction Speed	5.56 cm/s	Minimum	L	T,I
11	Load/Actuator	1177 N	Minimum	M	A,T
12	Power Req.	2000 Watts	Maximum	M	A,T
13	Design Life	20,000 cycles	5,000 cycles	M	A,T,S
14	Max. Surface Temp.	50 °C	5 °C	M	T
15	System Resonant Frequency	15 Hz	Maximum	H	A,T
16	No closure areas less than	2 cm	Minimum	L	T,I

Project Management

We have learned from personal past experience and from professors that without a good management plan, even the best teams can and will fail. During the first three months of the project two of our group members were studying in Stockholm, Sweden. To overcome this distance and time difference challenge, we worked hard to communicate frequently and utilized a few key software technologies to succeed at this. We are shared all of our files, research, ideation, and any other pertinent information on Dropbox®, a web-based file sharing program. We had group meetings via Google Hangout® and recorded the proceedings on a Google® document to keep meetings on track and make review of each meeting possible.

For the past two quarters, all teammates have been present and attending Cal Poly, which has made final design and manufacturing a relatively straightforward process.

We have been cycling through the role as group manager or "Lead Engineer". During fall quarter, Vince Truman acted as Lead Engineer. Winter quarter Brett Hartt acted as Lead Engineer, during the spring Brian Gilchrist took on the responsibility. Documentation of progress has been performed by Brett Hartt, manufacturing considerations were undertaken by Vince Truman, and Brian Gilchrist took responsibility over testing procedures. Information gathering and prototype fabrication was a collaborative effort and each quarter's Lead Engineer delegated tasks and kept the progress of these items on track. Table 2, below, is a summarization the distribution of tasks, and Table 3 lists pertinent dates which we observed throughout the process. For further planning on how we will design, build, and test our actuator pair we have prepared a Gantt chart, which helped our team stay on schedule. This chart, provided in Appendix F, details the time at which each portion of the process was planned for completion.

Table 2. Summarization of Distribution of Tasks

Individual	Tasks	Lead Engineer
Brian Gilchrist	Test Planning and Design Sub-System TBD	Spring
Brett Hartt	Documentation and Design Sub-System TBD	Winter
Vince Truman	Manufacturing Considerations and Design Sub-System TBD	Fall
Collaboration	Info gathering and Prototype Manufacturing	

Table 3. Pertinent dates

Date	Item
2/26-2/2	Project Update Memo
3/26-3/30	Hardware Demo
5/7-5/11	Senior Project Design Expo XII
5/31	Final Reports (Hardcopy and PDF)

Chapter 2 – Background

Characteristics of Hexapods

Traditionally, serial kinematic systems have been the mechanically automated method for pointing telescopes, where each degree of freedom (DOF) is dependent on the others. The most common example of a serial kinematic system is the “alt-azimuth” system in with there are two perpendicular axes about which the telescope can rotate, one being vertical and the other horizontal. This system takes its name from the two degrees of rotational freedom that it provides: altitude (elevation) and azimuth (compass direction). It is rather straightforward to design, construct, and control such a system, but there are a few key drawbacks that should be addressed. Positional error accumulates with each additional degree of freedom, and difficulties in direct tracking of celestial bodies are the two largest drawbacks. Also, when used to move they very large, very heavy mirrors used in deep-space astronomy, these alt-azimuth systems become extremely heavy, expensive, and immobile.

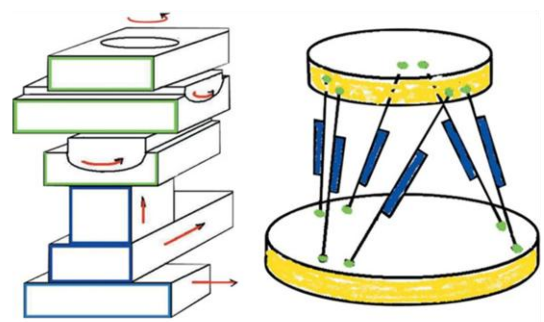


Figure 1. Serial Kinematic System (left) vs. Parallel Kinematic System (right) (16)

In contrast to the serial kinematics of alt-azimuth systems, a hexapod is a parallel kinematic system, where each degree of freedom is independent of the others. Advantages of this type of system include zero accumulation of positional error, direct tracking, and a lower inertia (meaning the acceleration and slewing velocity are increased). The added difficulty is the control system, which must simultaneously and separately extend 6 different actuators that can compete and interfere with each other. [2] See Figure 1 for a depicted difference between serial and parallel kinematic systems.

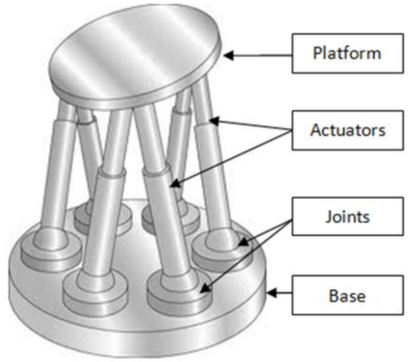


Figure 2. Main Components of a Hexapod (15)

Hexapods have many uses, from flight simulators to precision surgical robots, and there are many advantages to using hexapods over traditional mechanisms. A hexapod design is ideal for a telescope apparatus because the hexapod’s six degrees of freedom allow it to position the platform in any direction. [3]Hexapods also have a very high stiffness-to-weight ratio, making them stiff enough for astronomical research, and simultaneously light enough to be moved about with relative ease.

Because the hexapod is moved by six independent linear actuators, it is capable of extremely precise movements: each actuator supports $1/6^{th}$ of the load, meaning no single mechanical motion is responsible for all of the accuracy of the telescope. Compared to alt-azimuth systems, in which that accuracy of the telescope is dependent on only two degrees of freedom, hexapods have a distinct advantage.

There are a few disadvantages to hexapods that we have taken into account for our actuator design: friction, length of struts, accuracy, and thermal expansion. The primary concern with regards to friction, for us, is the connection between the actuators and the base of the telescope, as any stiction here can disrupt the smooth motion of the telescope. The length of struts (actuators) is a very important factor, because as the actuator extends, the risk of bending increases and the accuracy of the system can be compromised. Thermal expansion of the struts can cause inaccuracy in directional pointing. We do not believe this will be an issue in this particular application.

Previous Hexapod Analysis

A hexapod consists of a raised platform supported by a frame of extending and retracting actuators. Figure 4 depicts a working version of such a device. The first mathematical model of a hexapod system was proposed by Augustine Luis Cauchy in the 1800s, although development on physical systems did not start until the introduction of computers, which make possible the high frequency of positional sensor readings required to accurately point one. In 1965 D. Stewart, a British engineer, published his paper "6-DOF", which led to increased attention in hexapods. For this reason, hexapods are sometimes referred to as Stewart Platforms. [4]

Since D. Stewart's paper on hexapods, research has come a long way and has culminated in the publication of the paper "An Improved Solution to the inverse Dynamics of the General Stewart Platform". This paper was published by mechanical engineering professors at the University of Tabriz in Iran, and contains a closed form solution to the inverse dynamics of the Stewart Platform. [5] This solution was later confirmed by students in the University's engineering program. This was accomplished by comparing the closed form solution to a direct dynamic solution of the hexapod using a simulation program called ADAMS. [6] Specific assumptions were made in this analysis about the weight profile and design of the actuators, which makes using this paper's force calculations difficult. However, the geometrical analysis of the actuator is useful to us for determining the needed stroke of our actuator and the geometrical comparison of different designs, which we have done through the construction of a simulation program in MatLab.

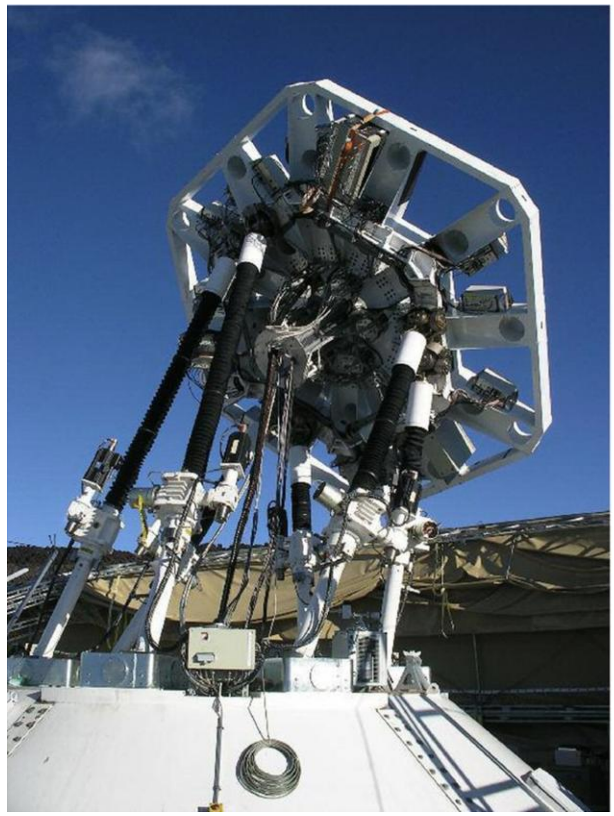


Figure 4. AMiBA Hexapod Telescope (2)

Figure 3. Electro Mechanical Linear Actuator by Machine Design (18)

Actuator Overview

There are several types of actuators available, including hydraulic, pneumatic, piezoelectric, and electromechanical. Hydraulic and pneumatic are similar, as each use fluid pressure to control the length of the actuator. The main difference is the working fluid; Hydraulic oil is used in hydraulics and air in pneumatics. Piezoelectric actuators use the effect of inducing electricity directly to metal to cause a linear expansion. These actuators can achieve very accurate extensions, but are limited by the stroke they can achieve. Considering the average stroke of piezoelectric actuators is around 20 mm, this category of actuators is easy to rule out as a project possibility. Electromechanical Linear Actuators, or EMLA, are a relatively new type of actuator that is rapidly becoming the standard for many applications. EMLAs are defined to consist of an electric motor and a transmission that converts rotary motion into linear motion. Current EMLA designs involve some type of screw. Screw designs are of interest to us as they can simultaneously have a high motion ratio, accommodate large loads, and exhibit a long design life. [7]

Current Actuator and Hexapod System Designs

Many patents have been granted for actuators. [8] Those of interest include US patent 2683379 [9], which uses a set of planetary gears to drive a central screw in a linear fashion. US patent 3,406,584 [10], Differential Roller Nut, is a patent on a mechanism that uses an external, internally threaded, member in conjunction with a threaded roller to transform rotary motion into linear motion.

Currently there are two high-profile telescope-hexapod systems in operation; the HPT and the AMiBA (Figure 4). The HPT, or Hexapod Telescope, was funded by the German Ministry of Economics, Trade, and Technology (NRW) with 8 million DM in 1989. It has a top plate hexapod diameter of 1.5 m and a designed payload of 120 kg. The tracking accuracy is about .05 arc seconds, achieved through use of lasers and gyroscopes. [11] It was completed and tested in 2006 in Bochum, Germany and later moved to the Atacama Desert in Chile. [12] The AMiBA is the largest operating astronomical hexapod mount. It was built in Duisburg, Germany and moved to Hilo, Hawaii in 2005. It has a 6 meter platform made from carbon fiber-reinforced plastic. The lower limit of elevation is 30 degrees from horizontal with 0.4 arcmin rms optical pointing error. The maximum slewing velocity is 0.67°/s, and it can handle a payload of 500 kg. The actuators used are "Jack Screws" comprised of a tubular ball screw with an integrated low backlash worm gear with a 10.67:1 transmission ratio. The worm gear is connected to an electric motor via a zero-backlash bevel gear set with a 4:1 gear ratio. The actuators vary in length 2.8 meters to 6.2 meters with an extension rate of 0 – 20 mm/s. [2]

Figure 5. Cut section of a nut traveling along a ball screw

Ball Screws

Unlike traditional lead screws, in which the teeth of the screw and the nut slide against each other to translate between rotary and linear motion, ball screws transfer motion to the nut through a set of ball bearings. The entire system consists of a screw with ogival-shaped threads, an example of which is depicted in Figure 6, as well as a nut with complimentary-shaped threads. Bearing balls cycle through the nut as it to climb or descend the screw. Figure 5 depicts a ball screw apparatus. Because rolling resistance is so much less than sliding friction, ball screws have an advantage over traditional lead screws. Since there is considerably less friction in a ball screw, the necessary torque to move a load is often 1/3 or less than a tradition screw, little to no static friction. [13]

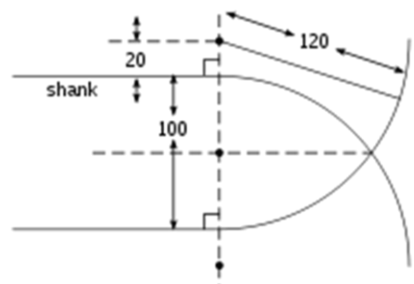


Figure 6. An example of ogival geometry, (25)

Safety Considerations

For the designed use of our actuator there are no ANSI or ASME standards of which we are aware. This made us careful and conscientious when designing our actuator, as we have no code or standard to make us aware of certain complications or safety issues. Of course, safety is always paramount when designing any system and our actuator is no exception.

The two major concerns with our actuator are pinching points and actuator failure. As operators work with the hexapod system, we have designed our hexapod as safe as possible by removing these potential hazards. We have eliminated the majority of pinching points by internalizing the majority of our moving parts with the rod and sheave. The actuator failure has been accounted for by designing the system appropriately.

Chapter 3 – Design Development

Discussion of Conceptual Designs

After building a list of requirements and extensive background research, our team went through a series of ideation steps designed to help isolate a set of top design concepts. We utilized a set of tools and diagrams to help prioritize requirements, and built decision matrices to weigh each design feature we came up with. The results of our first few rounds of ideation are presented in the following section.

Control Design Concept

The hexapod design concept against which we compared the other design concepts is a traditional, Stewart Platform hexapod. This system is used extensively in industrial and medical settings and has proved its usefulness in a variety of applications.

Design Concept #1

The first top design concept is comprised of the standard components of a hexapod: platform, base, actuators, U-joints at the base and ball joints at the platform. In addition to these standard components is the center pillar, see Figure 7. The pillar is rigidly fixed at the base and the platform is supported on the pillar by a ball and socket joint. The reason primary purpose of the center pillar is to increase the stability to the hexapod and reduce the load on each actuator. This allows for smaller, lighter and cheaper actuators. Further discussion of our top design is provided in Appendix B.

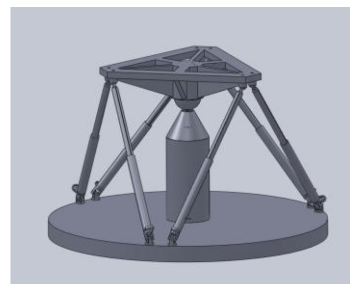


Figure 7. Sketch of Design Concept #1

Design Concept #2

The second top design concept is similar to the first as it has a platform, base, and a center pillar. The only difference is that instead of using actuators, cables are used instead. This is possible due to the center pillar. The cables can act in tension at all times and pivot the platform about the center pillar.

Design Concept #3

The third top design concept is most similar to the Design Concept #2. It uses a base, platform, center pillar, and cables for positioning. In addition to this Design Concept #3 implements an airbag system. The airbag is designed to provide additional support for the system.

Concept Selection

With further consideration of our design requirements we decided to utilize the control design concept, realizing that the complications arising from the addition of a seventh support do not outweigh the benefits. The cost savings achieved through reducing actuator size does not outweigh the cost of adding a center pillar, as the largest contributor cost is in the manufacture of the components and not the material itself. Increasing load capacity of an actuator of this scale is relatively inexpensive. The increase in stiffness was not proportional to the added cost. We made the decision to stay focused on designing an actuator that, when installed as a set of six in a hexapod arrangement, would fully support a telescope without any more engineering required. Simplicity is the driving force for this design.

Supporting Preliminary Analysis

To define the parameters of our actuator we started with geometrical analysis. As many different combinations of stroke, base plate radius, minimum, and maximum lengths will satisfy our requirements, we generated a model that would output required stroke and retracted length based on different conditions. We made the assumption that the center of the telescope mirror would not move, essentially a fixed center point, in order to narrow the set of possible geometries. Our maximum length, then, would occur when one "Λ" frame is fully extended, and minimum length would occur if the same "Λ" is fully retracted. See Appendix E for the Matlab code.

A motion study was also done based off of the work of Pedrammehr, S. et al. See Appendix E for more information.

The most important parameter for our accuracy requirement is strain under worst-case full compression loading. It is imperative that deflection be kept near encoder accuracy so that length can be read accurately. We performed a set of stress-strain calculations assuming axial loading, as our actuator is approximated to be a two-force member. Based off of our calculations, strain under maximum loading conditions is 30 μm. Because of our small tolerance for strain, our calculations for Euler buckling indicate we have a safety factor of approximately 30 for buckling, including the assumption that both ends of our actuator are fixed, a worst-case scenario.

The torque and power to raise the ball screw were calculated to be 2.50 Nm and 150 Watts respectively.

Proof of Concept Testing

In order to ensure that the system that we were building would meet all of our geometrical requirements, we constructed several small-scale models. Both of these models were constructed from PVC piping. In order to ensure that we could achieve the necessary stroke and performance from each of the three "Λ" frames, we constructed a scale model.

In order to ensure that the hexapod could achieve the necessary range of motion, we constructed a model of the upper joints made out of PVC and ball bearings. The U-U Joint is explained in greater detail in Chapter 4 – Description of the Final Design.

Figure 8. Proof-of-concept Λ frame and U-U Joint Models

Chapter 4 – Description of the Final Design

Hexapod Design without Central Pillar

After considering the feasibility of using a center pillar to support the weight of the top plate—and therefore reduce the size and cost of the supporting actuators—we decided that the inherent stiffness and strength of the Stewart platform would be sufficient to support the weight of the telescopes in our size range. Additionally, constraining the motion of the top plate would greatly increase the possibility of over-torquing the top plate, adding unacceptable amounts of force to any telescope mounted to the top platform.

Removing the central pillar greatly simplifies our design, but requires that all six actuator be robust enough to handle the static and dynamic loads. For our analysis and design, we prepared for the entire weight of the mirror and top platform to be supported by a single actuator. This situation should never occur, but over-engineering the assembly will ensure that compounding strain from our subsystems does not detract from our accuracy.

Our final design for the hexapod consists of three main subsystems: The platforms, the actuators, and the joints. For the current scope of our project, we are focusing only on the actuators and the joints; these will be the most expensive sub-systems of the hexapod system, and they will require the most analysis and design.

For full details on design analysis, please see Appendix E.

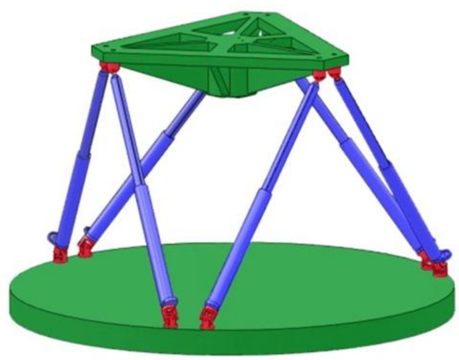


Figure 9. Subsystems of our hexapod

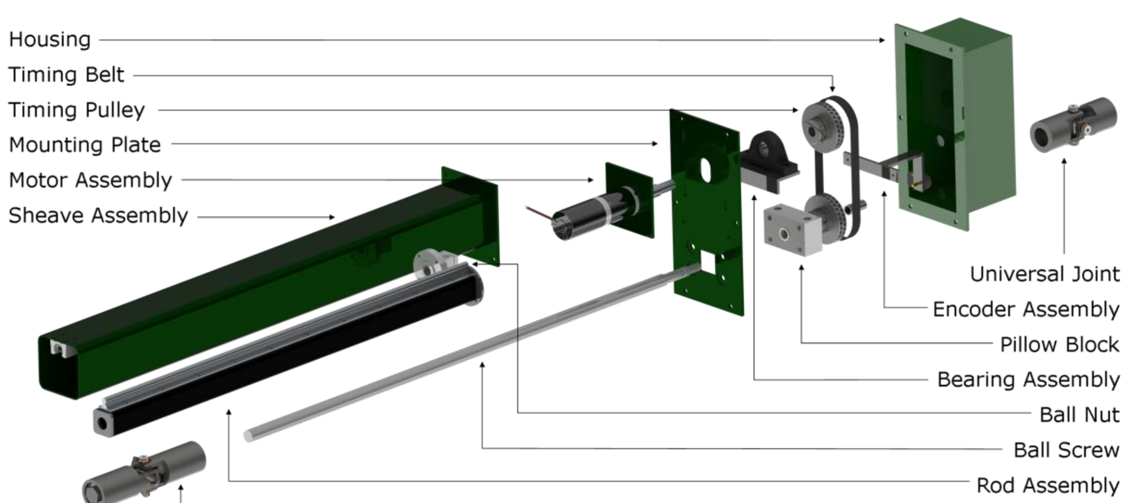


Figure 10. Exploded and labeled view of our actuator design

Sub System: Joints

Traditional models of steward platforms and many of the hexapods used in precision industries use UPS joints to connect the actuators to the platforms. UPS joints consist of a universal joint at the bottom, a prismatic joint in the middle, and a spherical joint at the top.

Universal Joint

The universal joint connecting the bottom of the actuator to the bottom platform is responsible for allowing the actuator to rotate about two perpendicular axes while constraining all other degrees of freedom. In order to achieve the range of motion that we need for the top platform, our universal joint needs to allow for 30 degrees of rotation about both the x and y axis.

Figure 11. Joints in a typical actuator: ball, prismatic, and universal

For our design of the universal joint, we gained inspiration from the universal joints used in automotive applications. Universal joints are commonly used to couple shafts that are not collinear, and therefore they are commonly available in a variety of sizes. However, these are not rated for axial loading, and are usually designed to couple two shafts together spinning at high velocity with small angular differences between them. Because we do not have a high velocity application and we will have high axial loads (compared to radial loads or torque), we have decided to manufacture our own universal joints. This will also allow us to achieve the range of motion that we need for our system, as most automotive universal joints that have the precision that we need are limited to motions below 25 degrees. Full drawings for the components in universal joint are provided in Appendix B.

Figure 12. U-joint at the base of our actuator

Prismatic Joint

In an actuator assembly, the purpose of the prismatic joint is to allow the two parts of the acuator to slide past each other. This joint can provide, at most, two degrees of freedom: linear translation, and rotation about the axis parallel to the direction of translation. Because our system is driven by a mechanical screw, it is imporant that our prismatic joint allows only for translation and constrains all rotational degrees of freedom.

Our original plan was to use a simple key/keyway, however this plan was quickly discarded due to errors that would arise from manufacturing such a long key or keyway. Also considered was friction and backlash that would arise from a key/keyway system. It is very important that there is no backlash in the rotation constraint. Even a small amount of backlash in the constraint could cause our actuator to extend or retract in a way that we cannot measure or account for in a control system.

To address this issue, we decided to use an open-sided sliding bearing and circular rod track, similar to the tracks commonly used in CNC macnines. The sliding bearing is easliy mounted at the top of the stationary part of the actuator, while the metal guide rod is affixed to the extending part of the actuator. We had originally planned on welding the guide rod to the extending rod, but we decided that using machine screws would not cause the warping that welding would. However, even withought the warping that may be caused by the welds, there is still the issue of non-linearity of the guide rod. To address this, we can take a high precision depth gauge and measure the profile of the rod. This profile can then be accounted for in the control system.

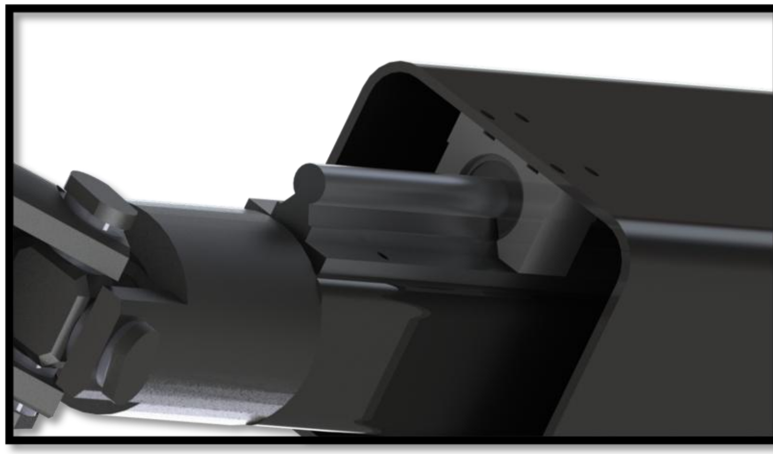


Figure 13. Linear Guide-way System

Ball Joint

At the top of the actuator, the ball joint is responsible for constraining only translation at the joint, and allowing for rotation about all three axes. It is important that there is one more rotation degree of freedom on the top joint that on the bottom joint; as the top platform moves through its entire range of motion, the actuators will need to rotate axially relative to the top platform. If the actuators were to be completely constrained from rotating about their longitudinal axis, the range of motion available to the hexapod would be greatly reduced, and it would likely be unable to move at all.

Figure 14. The Universal-Universal-Joint

Figure 15. Typical Ball Joint (26)

Acquiring a ball joint for the top of the actuator proved to be much more difficult than expected. In order to allow the hexapod to point to all point thirty degrees above the horizon, the range of motion needed by the ball joints is very close to 170 degrees. Similar to the universal joint discussed above, ball joints are commonly used in automotive applications, and there are even very high precision ball joints available. However, all commercially available ball joints that we researched were limited to a maximum travel far below what we needed for our actuators (in the 15 to 50 degree range).

In order to allow the actuator to rotate relative to the top platform, we considered using a combination of universal joint and thrust bearing. This would provide all three rotational degrees of freedom, and would enable the actuator to move to

any position. However, this design comes with one major caveat. Universal joints allow for two rotational degrees of freedom, but they are also capable of moving to positions where they completely “lock out” and are unable to move to the necessary position (this is partially why universal joints are not commonly used when the angle between the two shafts is greater than twenty degrees). To address the lock-up issue, we decided to add one more degree of freedom, and place a thrust bearing on both sides of the universal joint. This will allow for the universal joint to self-align when it approaches a typical locking angle.

Figure 16. U-U joint attached at the top of the actuator

We are calling this type of joint the Universal-Universal joint, and we believe that its self-aligning capabilities will satisfactorily mimic the behavior of a 180^o ball joint. One more consideration was whether or not the supports in the “U” configuration could support the load of the actuator in the worst case scenarios. In the worst case, the U-U Joint is at 90^o and all the weight of the actuator is on one support. We found the stress and strain sustained by one of the joints to be well below the critical stress and strain for our project.

Sub System: Actuator

The most important and the most complex subsystem of any hexapod is the actuator array. The collaborative motion of the actuators is responsible for all of the motion in the hexapod, so it is of utmost importance that each actuator is, on its own, exceptional. Our actuator design is comprised of six main parts: the drive mechanism, the sheave, the housing, the motor, the transmission, and the measuring system.

Drive Mechanism

While we had once considered using a hydraulic cylinder as the drive mechanism for our actuator, we have moved forward with a mechanical screw to drive the linear motion necessary. While hydraulic systems would give us the stroke and strength that we need, they have a lot of inherent stiction, greatly reducing their accuracy.

For our design, we are using a ball screw and ball nut combination. Ball screws are similar to lead screws, but they use a complex ball-bearing feed system to “roll” the nut along the screw. Because there is only rolling resistance, which is much less than the sliding friction in lead screws, there is almost no stiction, and the torque to move a load is reduced by 60-70%. However, because the ball screw has such low friction, the self-locking properties of lead screws are lost. We decided to address this by using the self-locking properties of the transmission, and if necessary, using the motor to keep the screw in place.

By purchasing a combination ball screw, ball nut, and mount block, we will be able to achieve very high precision motions with our actuator. The ball nut that we will be using is not pre-loaded onto ball screw, but backlash still will not be an issue. There are several ways to mount ball screws and ball nuts. Our configuration is such that the screw rotates in place and drives the ball screw up and down. It is for this reason that we need to constraint the rotation of the ball joint with the prismatic joint described previously—if the ball nut is not constrained, it will not translate at all and will simply rotate with the ball screw.

With a diameter of 1 inch, buckling is not a concern, as we have a factor of safety of approximately 30. Because the universal joint and the Universal-Universal joint will align the actuator so that all loads act longitudinally, we do not need to worry about high bending moments or excessively eccentric loads. Therefore, we can fix the ball screw only at one end using a manufactured mount block.

Our design requirements include both very high precision (a resolution of 1 arc second on the top platform translates to ~4 μm of linear motion in the lead screw) and high speed (5 $\frac{deg}{s}$ translates to ~6 $\frac{cm}{s}$ of linear speed). To address both of these criteria, we had considered using a series actuator system, where a high-precision actuator was mounted to the top of a coarser yet faster actuator. This solution turned out to not only be overly complicated, but also unnecessary. By using a very precise measuring device, we will be able to achieve both the accuracy and speed of actuation that we need for our system.

Rod

In our actuator design, the rod is one of two weight-bearing structures that actually moves, the other being the ball screw. However, while the ball screw rotates but does not translate, the rod translates but does not rotate. Although it is a critical weight-bearing member of the actuator system, the rod is a relatively simple part. Our rod design is simple square metal tubing stock with an appropriate wall thickness to support the top platform load. The inner area of the rod also needs to be large enough to accommodate the thread diameter of the ball screw. At the bottom of the rod, we will weld a flange to which we can mount the ball screw. This way, the rod and the ball nut will act as a single body, translating together up and down the screw. In order to keep the rod from rotating, we will use screws to attach the linear track to the rod's side. At the top of the rod we will weld a plate to which we can attach the Universal-Universal Joint.

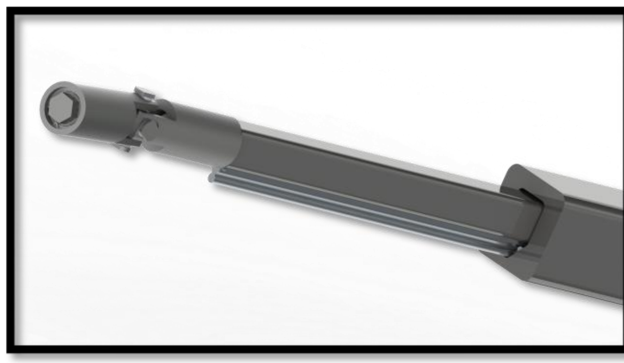


Figure 17. Extending portion of the actuator

Figure 18. Sheave

Sheave

In order to protect the screw and to constrain the motion of the rod, we will mount a metal sheave that encloses the drive system. The sheave is a very simple part, and since it is not weight-bearing, it does not need to be as thick or robust as any of the other parts of the system. However, sliding bearing that constrains the rotation of the extending rod is mounted to the sheave, so the sheave must be able to sustain low levels of torsion. The sheave will be welded to a flange which will be bolted to the mounting plate.

Motor

Figure 19. Anaheim Automation motor (27)

The motor we specified for this design was not chosen for any specific performance characteristics; rather, it was chosen because we were able to obtain it for free. Our analysis suggests that, to achieve the 5 degrees per second of telescope rotation requested by the sponsor, that each actuator must be capable of extension and retraction at slightly less than 6 cm/s. a motor that can achieve this must be able to output 300 Watts with 300 rpm at output shaft. Also, we need an internal gear ratio to keep the motor running efficiently. We were able to narrow our selection down to a few options, all of which cost more than our budget permitted for a motor. At Dr. Ridgely's suggestion, we instead picked a motor out of a set of donated motors in a storage room, settling on one that makes plenty of torque, but only runs at around 17 rpm at full power. Our concern, primarily, was that there was no way to meet the large speed requirement with this motor, but Ridgely assured our team that the telescope would still quite useful with a much lower slewing velocity, essentially removing the speed requirement. This opened the door for us to design a screw that could still handle the initial power throughput and provide for a possible motor switch at some point in the life of the actuator. Our motor, by Anaheim Automation, would typically cost around $200, and we were fortunate enough to obtain one for free.

Belt Drive

The selection of a belt drive was made after studying multiple power transfer options, including helical, spur, and worm gears, flat, timing, and v-belts, and even a direct drive to the screw itself. The most important facet here is backlash, or more specifically, the absence of it. It is desirable for our control system to have the most direct control over the screw as possible.

Gears initially seemed like a good option, especially helical gears, because they can be relatively backlash-free and can be designed for long lifetimes. Cost, once again, dissuaded our team from picking a helical gear set, and both worm and spur gears exhibit too much backlash when reversing drive direction. It does seem, after these considerations, that a direct drive would be optimal. It is difficult, however, to design a mounting system for multiple motor possibilities, a challenge highlighted in the previous section detailing our motor selection, and actuator length is a concern for us, as a small change in length can have dramatic effects on the position of the telescope. Our geometry was optimized in an earlier analysis to provide the most positional freedom for the least stoke and overall, retracted length of the actuator.

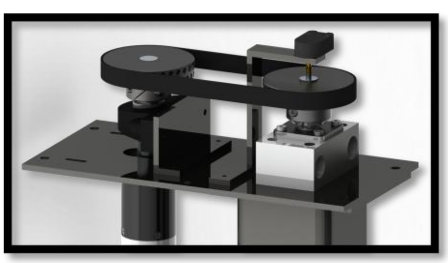


Figure 20. Pulley Transmission System

The options left over were all various forms of belts. While v-belts and flat belts are much more inexpensive option than timing belts of a similar size, they tend to have unpredictable amounts of slip, a characteristic undesirable in this application. A timing belt will be less expensive than a helical gear set, provide quick response to a motor torque, and will not slip over the pulley. We decided on a set of timing pulleys and a 20-mm belt from SDP/SI, which met both our requirement for low slip and relatively low price.

Our design includes a drive pulley assembly that can act as a tensioner, as it is mounted to the housing using slots instead of holes. This will allow for the removal of slack, both in initial assembly and after use if the belt stretches. The whole belt system rounds out to around $200, and we are happy with this result, as this is a major portion of our system.

Currently, there is a simple 1:1 ratio between the motor output shaft and the input to the ball screw. This was done intentionally, expecting a new motor at some point in the life of the actuator. It will be easier for a designer to select a gear motor that meets all specifications and use it to drive a reduction-free system, as opposed to having to contend with any of the possible gear ratios we could have chosen for this design.

Optical Encoder

Deciding how to measure the position of this system proved to be a challenging endeavor. These measurements are a crucial part of a control system, as an accurate sensor will quickly and reliably provide a signal that will be read as an error signal by a controller and corrected. Modern controllers have the ability to correct mistakes much faster than a human ever could, and with much more precision. These controllers, however, are useless without being fed accurate information. Our sponsor requires an instrument resolution of 1 arc second, which, with respect to our geometry, corresponds to a 4μm length change being readable and communicable by some type of encoder.

Figure 21. Example of a US Digital angular encoder (28)

There are many options on the market that could possibly handle this, and they seem to break down into three distinct categories. Many devices directly measure length, others measure rotations, and still others measure angles. The devices that perform each function are too numerous to list here, but we will highlight a few of the possibilities we considered. Our first inclination was to select a linear encoder, which would directly measure the length of our actuators. Our required stroke of 1 meter proved to be too long to measure with a reasonably-priced linear encoder. These devices tend to be incredibly accurate over small distances, but as the range of motion they are asked to measure increases, more error is introduced into their measuring devices, creating a lack of accuracy where it is most needed. An angle measurement device would not suffer from such a drawback, as many are designed to operate over a large range of angles, but it is difficult to mount one such that it will return the results we require, as no actuator rotates solely about one axis. Multiple angle measurement devices would be necessary to compute the length of even one actuator. This is definitely a viable option, but there is an easier way.

An angular encoder, one which reads shaft rotation angles, is an easy-to-implement solution, and they are also relatively inexpensive. Accuracy here is not a barrier, either. To achieve the aforementioned 4μm resolution, it is simply a matter of picking an encoder that can read enough divisions of a circle and multiplying by screw thread pitch. In our case, if an actuator can transmit 3000 positions of the screw per revolution, we will have achieved the required resolution. Yes, this is an indirect measurement of length, but it is one that can be calibrated out by measuring lead error along the entire screw, which can account for as much as 20μm/300mm.

Mounting one encoder on each actuator will allow us to indirectly but accurately measure their lengths, which is the value a control system needs to read in order to keep the telescope pointed correctly. We decided to use an S4 encoder from U.S. Digital on a recommendation from Dr. Ridgely, as they provide a high-quality product for a reasonable price. Mounting this encoder directly the screw will eliminate any possible discrepancies between screw position and motor position; we simply don't need to know them to be accurate in this case. We are left with a few unknowns, the largest being deflection and backlash in our universal joints, which contribute to the actuator length but are exceedingly difficult to take useful measurement of. Design of these will be critical to maintain overall accuracy, and the steps we have taken to ensure said accuracy are detailed in the universal joint section of this paper.

Housing

Our drive housing will be manufactured as purely a functional piece, and in this design is not intended to resemble or perform as a production unit would. It is fabricated from steel plate, and we added a mount for the bearing that will support the drive shaft as well as a sliding mount that will handle pulley tension. All forces experienced by the actuator will be passed through the housing, so thick steel plate will be used to transfer loads. The bottom of the housing will be used to attach the bottom universal joint to the rest of the actuator.

This housing is not designed to be waterproof or shockproof in our prototype. Further developments on this project will need to implement these features; as the actuators will potentially be introduced to these harsh environments. For our prototype the housing is simply a way to keep the internals from being unnecessarily abused.

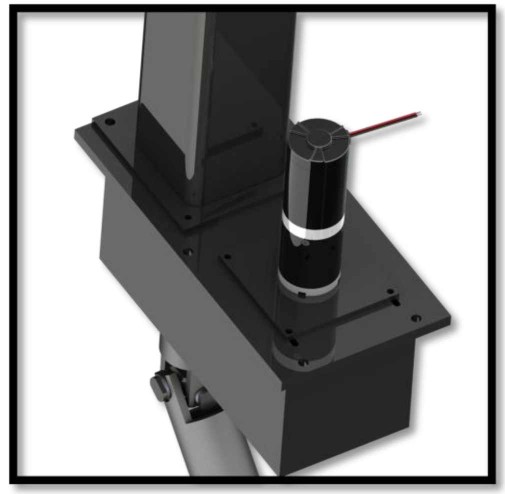


Figure 22. Housing for transmission system

Sub System: Controls

Because there are many pre-existing systems for controlling the position of linear actuators, the control system for our actuator was not of high priority to us. Knowing that we could purchase or borrow an existing control system allowed us to spend our time focusing on the design and development of the actuator itself.

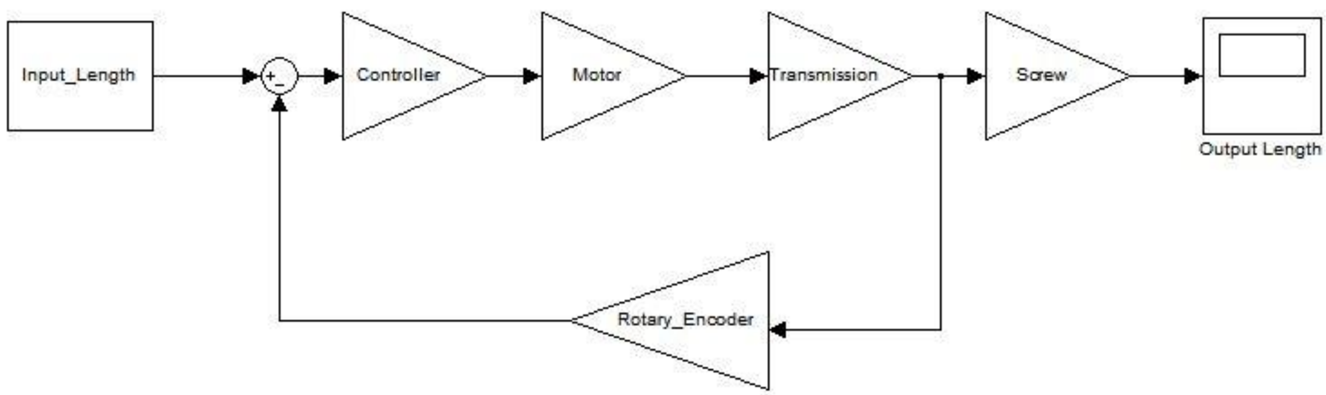


Figure 23. Control Schematic for the Actual Precision linear actuator

Cost Analysis (BOM)

Table 4. Summary of the cost for producing one actuator*

Part	Cost, USD
Motor	194.00
Pillow Mount Block	784.22
Ball Screw	
Ball Nut	
Rod	36.28
Optical Shaft Encoder	91.73
Sheave	131.85
Base Mount Bearing	44.82
Housing	25.00
Bottom Plate	10.00
Timing Belt Pulley (2x)	102.60
Drive Belt	100.85
Universal Joint	59.84
Universal-Universal Joint	177.76
Miscellaneous**	113.67
Total	1872.62

* Note that we have obtained a free motor and angular encoder for use in our prototype.
**Miscellaneous covers the cost of the bolts, paint, and other fixtures needed for assembling the actuator.

Safety Considerations

In its current state, the actuator has multiple pinch points. These will need to be covered before installation into a production hexapod, but are accessible for testing and observational purposes right now. Universal joints are obvious candidates for boots, and a bellows should be installed over the exposed rod area to contain grease and protect the tracks.

Operational heat is a non-issue with this motor, but once a motor that meets specifications is installed the system may experience significantly more heating due to losses.

Maintenance Considerations

The ball screw and universal joints require infrequent greasing. Lubrication is necessary to achieve the long design life of the ball nut and screw, and the universal joints require it to move smoothly throughout their entire range.

The drive belt may require tensioning over time, which will require the housing to be disassembled and the motor assembly to be slid further along the slots designed for this purpose. If no belt stretch is experienced, or if thermal cycling due to environmental conditions results in a change in relative position between the drive pulley and the screw pulley, the motor assembly's position can be adjusted by housing removal and screw re-tightening.

Chapter 5 – Product Realization

Manufacturing Processes

Rod (GHT100)

i. Machine the rod flange and the rod tip plate (GHT103, GHT104).

ii. Weld the flange and the tip plate to the rod.

iii. Drill the hole pattern into the side the of rod, and screw the guide rail (GHT102) to the side of the rod.

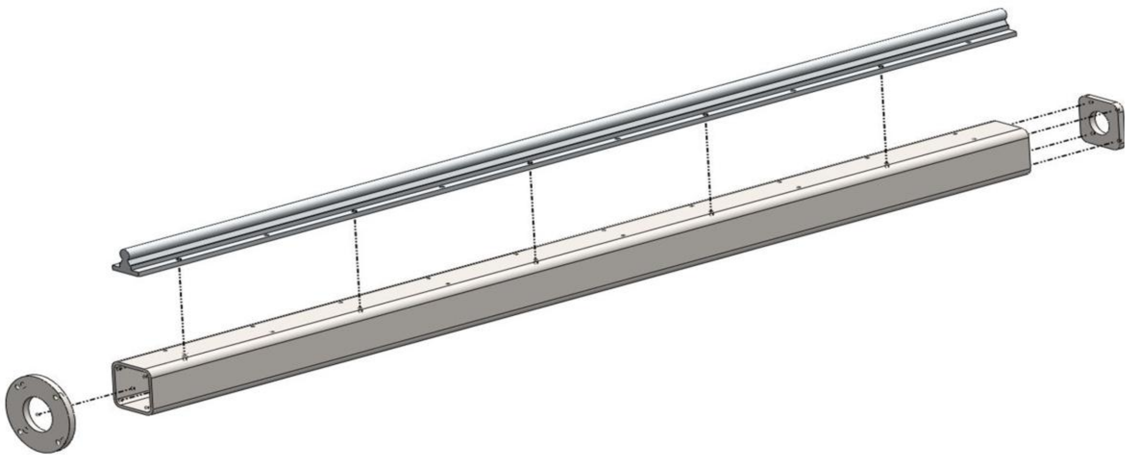


Figure 24. Exploded view of the rod assembly (GHT100)

Motor Mount (GHT200)

a. Plate (GHT202)

   i. Cut the plate to the correct size and sandblast it.

   ii. Drill holes for the mounting the motor to the plate, as well as holes for mounting the plate to the housing.

   iii. Attach the plate to the motor with screws.

b. Drive Shaft (GHT203)

   i. Turn the shaft to the correct size.

   ii. Drill the motor shaft hole.

   iii. Mill the set screw hole and counter bore; tap the hole.

   iv. Use a set screw to attach to the motor output shaft (GHT201).

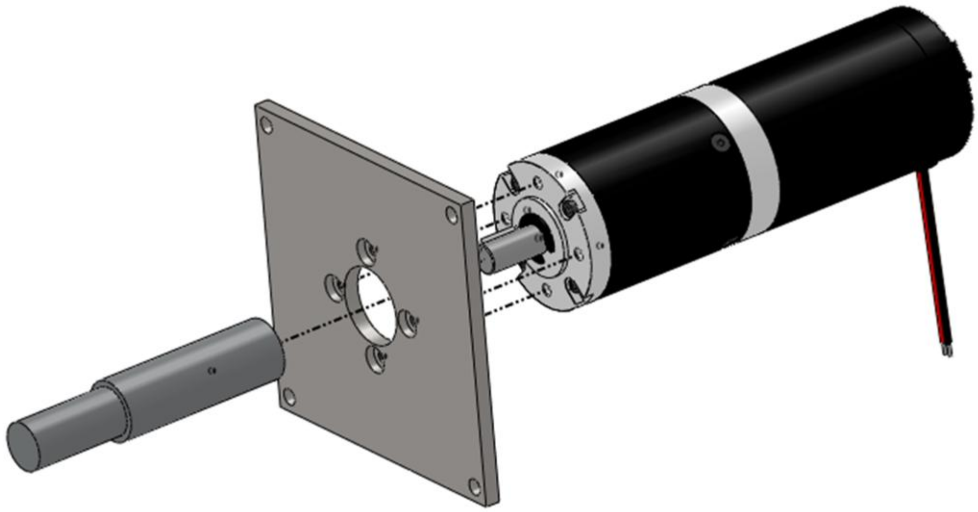


Figure 25. Exploded view of the motor mount assembly (GHT200)

Bearing Assembly (GHT300)

c. Plates (GHT302, GHT303)

   i. Cut the plates to the correct size and weld them together

   ii. Drill and tap the holes.

   iii. Screw the base mount bearing (GHT301) onto the plate.

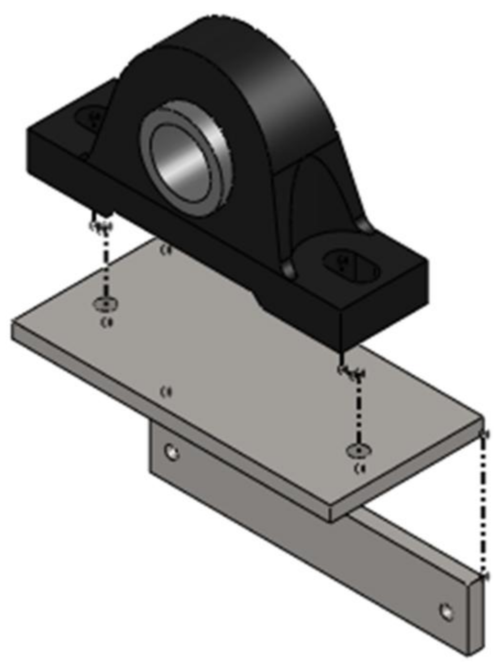


Figure 26. Exploded view of the bearing assembly

Sheave (GHT400)

d. Sheave (GHT401)

   i. Cut to the correct length.

   ii. Drill the hole pattern to attach to linear bearing block (GHT404).

e. Sheave Cap (GHT402)

   i. Machine the plate to get the proper geometry.

   ii. Weld the cap to the top of sheave after the bearing block is attached.

f. Flange (GHT403)

   i. Machine the flange for proper geometry.

   ii. Welded the flange to bottom of sheave.

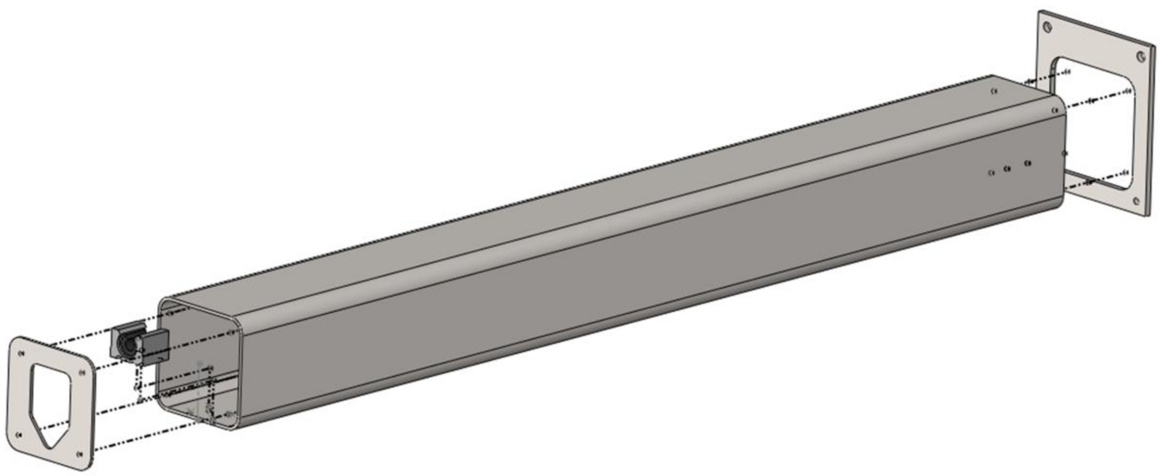


Figure 27. Exploded view of the sheave assembly

Encoder Assembly (GHT500)

g. Three flange plates (GHT502-GHT504)

   i. Cut the plates to the correct size.

   ii. Drill and tap holes in two of the plates.

   iii. Weld the three flanges together.

   iv. Screw the encoder (GHT501) into the bottom flange

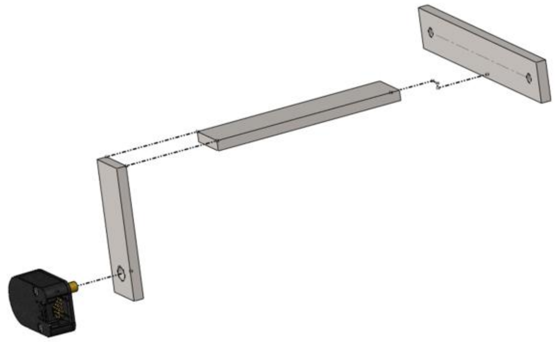


Figure 28. Exploded view of the encoder assembly

Housing (GHT600)

h. Sides, Flanges, and Bottom Plate (GHT601-GHT605)

   i. Cut all the plates to the correct size.

   ii. Sand blast all the pieces

   iii. Weld the pieces together in the correct formation, adding gussets for reinforcement.

   iv. Drill the holes on the flanges

   v. Re-sand blast the entire housing. Apply primer and paint.

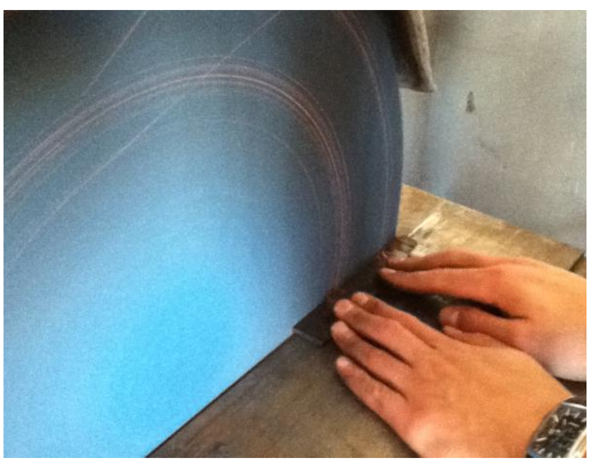


Figure 29. Brian grinding a flange piece for the housing

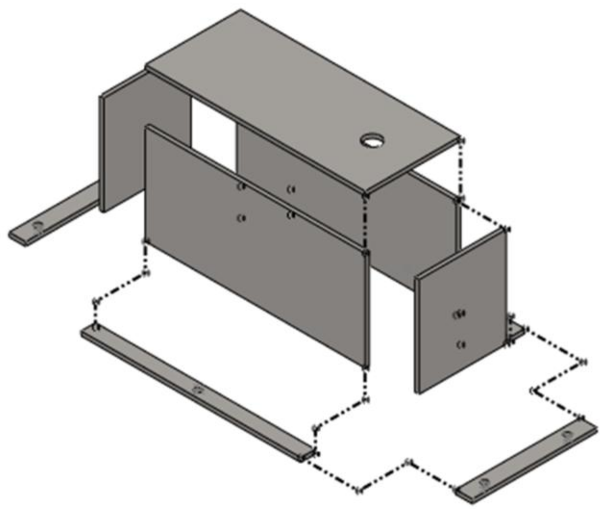


Figure 30. Exploded view of the housing assembly

Universal-Universal Joint (GHT700)

i. Yoke (GHT701)
   i. Mill one yoke and test the strength.
   ii. If the yoke passes all the tests, manufacture another yoke.

j. Shaft, Spacer, and Bearings (GHT702, GHT703, GHT704)
   i. Turn the shaft to the correct diameters. Tap the end of the shaft.
   ii. Cut the spacing pipe to the correct length.
   iii. Press-fit the bearings and spacer into the yoke.
   iv. Insert the shaft and thread the bearing collar (GHT705) onto the end of the shaft.

Figure 31. Brett milling a yoke for one of the universal joints

k. Cross Block and Pegs (GHT751, GHT753)
   i. Cut the Cross Block to appropriate dimensions and chamfer all the edges.
   ii. Drill and tap the holes in the Cross Block.
   iii. Lathe the pegs to the correct diameter and thread the ends.

l. Assemble all the components including the bushings (GHT752)

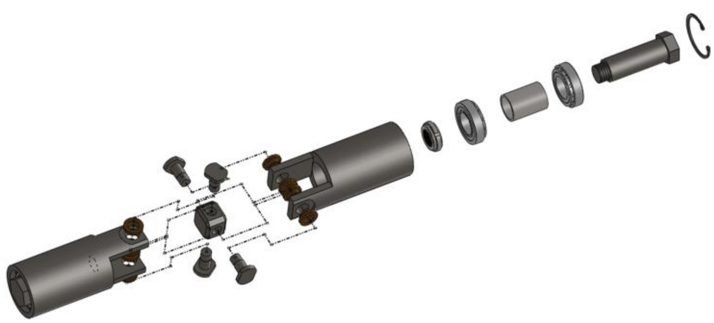


Figure 32. Exploded view of the Universal-Universal Joint

Figure 33. Manufactured Universal-Universal Joint

Universal Joint (GHT800)

m. Yoke (GHT801)

   i. Mill both of the yokes.

   ii. Weld end caps onto the yokes.

n. Mill and lath the cross block and pegs as before.

o. Assemble all of the components.

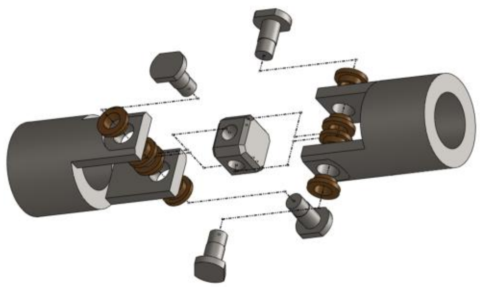


Figure 35. Exploded view of the universal joint assembly

Figure 34. Manufactured Universal Joint

Final Assembly (GHT000)

p. Pillow Block (GHT001), and Ball Screw (GHT002)

   i. Press-fit the ball screw to the pillow block.

   ii. Bolt the mount plate (GHT005) to the pillow block.

   iii. Bolt the rod flange (GHT100) onto ball nut flange (GHT003).

   iv. Bolt the sheave (GHT400) to the mount plate.

   v. Attach the output shaft (GHT006) to the ball screw and tighten the set screw.

   vi. Clamp one timing pulley (GHT004) onto the output shaft.

   vii. Attach a flexible coupling to end of the output shaft.

q. Motor (GHT200) and Bearing (GHT300)

   i. Press-fit the drive shaft into the mounted bearing after inserting the shaft through the mount plate.

   ii. Screw the motor mount plate (GHT202) to the mount plate (GHT005) and to the encoder assembly (GHT500).

   iii. Place the second timing pulley (GHT004) on the end of the drive shaft and tighten it down with the clamp screw.

r. Install the belt. Tighten it by sliding the motor mount assembly along the slots in the mount plate.

s. Bolt the housing (GHT600) to the mount plate (GHT005)

t. Attach the U Joint (GHT700) and the U-U Joint (GHT800) to the bottom of housing (GHT600) and top of the rod (GHT100) respectively.

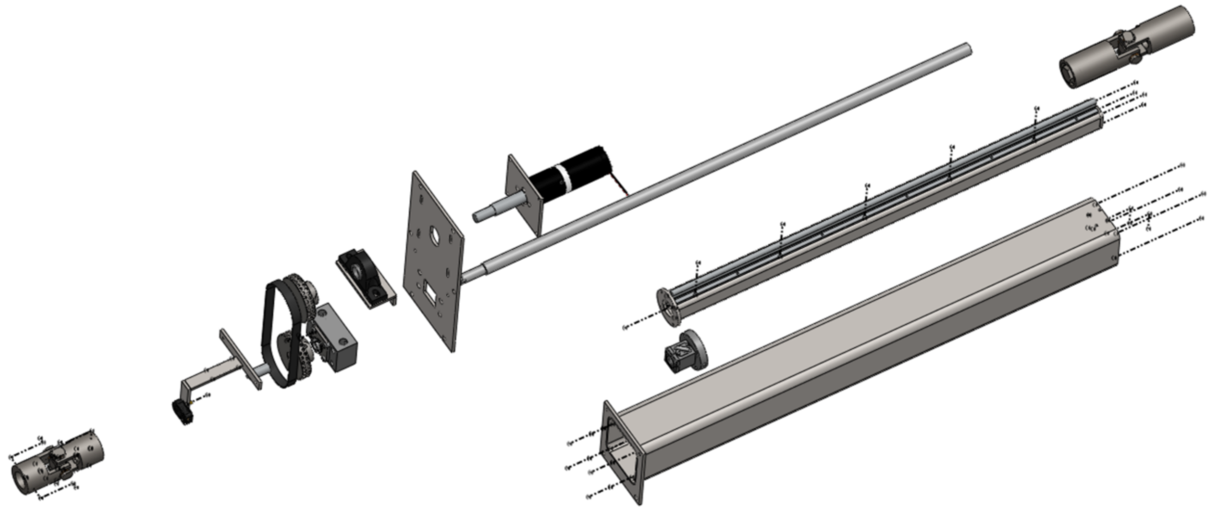


Figure 36. Exploded view of the final assembly

Figure 37. Manufactured Actuator, with bottom U-joint not attached

Differences between Prototype and Design

Housing and Housed Components

Our team made the mistake of constructing the housing out of 3/8” steel in most places instead of ¼” like had been suggested. This and some warping resulting from the welding process left the housing too small to accommodate all components as they were designed. Our drive-side bearing was too wide, so it had to be ground down to fit in, and the flange to which it was mounted required the same grinding treatment. A set of two bolt holes had to be re-located as well, as the designed holes did not clear the housing walls on the inside.

Bearing and Mount Plate

The ends of the bearing housing and mount plate had to be ground down to fit inside the housing. No functionality modifications were made here.

Encoder Flange

Holes had to be relocated closer to the center of the plate that bolts the sheave, encoder, and mounting plate together, as the housing did not fit correctly with the bolts as designed. No functionality was lost in this process.

U-U Joint

While dimensionally similar to the design part, the yokes are slightly out of spec making rotation difficult at more extreme angles. Shoulder bolt holes for the cross block bolts are not perfectly aligned, leaving shoulder bolts with a tendency to walk out of their holes. Thread Lock was used to address this problem temporarily.

Recommendations for Future Manufacturing

The consensus among our group is that our parts were all designed well, but our manufacturing skills are still lacking. For instance, the yokes of our U-U joint are very close to working ideally, but we could not hold the tight tolerances required for smooth operation. If another group were able to machine these items and others with a CNC milling machine, the tight tolerances held would yield a much more useful result. Any other parts which could be made on a CNC machine would undoubtedly speed up the process, as many parts took hours to complete. In the Cal Poly shops, there is a large line for the traditional knee mills, but since CNC milling machines require a high level of certification in order to use, they usually sit idle.

Welding caused us few problems, but the ones it did cause were difficult to ignore, such as the warping in the housing. Welding is doable by students, but we would recommend more practice time be taken and more time allowed for finishing the welding process, as it cannot be rushed. When welding the rod to its flange, they must be perfectly perpendicular. Extra care should be taken when welding these pieces, as a rod that is not perpendicular to the flange puts unnecessary stresses in the bearing block, shortening its life and possibly destroying it. We did not see this issue and have no reason to believe it would affect our actuator, but we were not immune from the types of mistakes that could have caused it.

It became apparent that threaded holes in steel plate were only useful for small screws in low-load conditions. When designing these parts, maximizing free space seemed like a high priority, especially when we weren’t sure everything would fit together. Looking back, it would have been much easier to design each part with clearance holes and simply buy nuts and washers instead of tapping holes, such as on the encoder flange and the bearing mount flange.

Chapter 6 – Design Verification Plan (Testing)

Test Descriptions with List of Necessary Equipment

We plan to start testing with one actuator and verify it meets our performance requirements. We have designed tests specifically for one actuator. Based on the specifications for the actuator we will need to verify supported load, accuracy, resolution, range of motion, extension and retraction speed, surface temperature, and resonant frequency. See Table 7 for our design validation plan.

Tests for Actuator without Control System

Supported Load

To test this specification, we placed the actuator on the floor and put 150lb of weights atop the rod, and measure the strain. The stress-strain relationship is approximately linear in steel while operating in the elastic realm, so it will be possible to scale the resulting strain value to our maximum design load.

Range of Motion

To verify we have obtained our range of motion requirement we will hang or lay the actuator its side, starting at a fully retracted position we will extend it to its full capacity and measure the length of these two positions. We will just need a tape measure for this test.

Extension and Retraction Speed

Due to budget constraints we are building our actuator with a motor that doesn't meet the requirements of extension speed. However, we will be building the actuator such that the motor can be swapped out for an appropriate motor. To validate that the actuator can be extended at the desired rate we will need to find the efficiency of the power transmission and ball screw. This will be accomplished by calculating power required to raise a load. We will relate power required to raise the weight and power consumed while rising to calculate efficiency. This efficiency can then be used to confirm that an appropriate motor could be used to attain the speed requirement.

Surface Temperature

To measure the surface temperature we will reciprocate the actuator from full extension to full compression at maximum speed for three hours and measure the surface temperature of the actuator. We will need a meat thermometer.

Tests which require a Control System

Accuracy

As we do not have the precision measuring tools needed to directly measure the accuracy of our actuator, we will need to measure it indirectly. We will position the actuator such that one end is touching the ground and the other a plate with a laser on it such that when the actuator is extended that directly relates to a change in angle of the plate and change in angle of the pointed laser. We will then use said change in laser angle at a measureable length to determine the accuracy. We will need a rigid plate, a laser, and a caliper.

Resolution

We will use a method similar to that described for accuracy validation. We will instruct our control system to extend the actuator the equivalent of a number of encoder pulses, then measure the resulting length change. We will perform this test for many different screw lengths to account for possible errors in thread pitch.

Testing Results

Successful Tests with Results

Extension, Retraction Speed and Efficiency

Our first test involved calculating power requirements and efficiency values for our actuator. We performed these tests by stacking a set of 25lb weights on top of the actuator rod and raising the load. We set the power supply at a constant voltage of 24V and recorded supplied current. Comparing the resulting power consumed values with the power required to raise the load yielded efficiency values. We did not have enough weights to approach the maximum design loading on the actuator, but we assume that with more weight our system will become more efficient, as driveline losses become a lower percentage of the overall energy and power consumed. The overall trend in both the power and efficiency values is increasing, and it can be safely extrapolated that with the maximum design load, the motor would stay safely under its maximum amperage rating. We used the average time from the four tests in Table 5 to calculate power input, which was then used to calculate efficiency.

Table 5. Actuator extension speed

	Distance (in)	Time (s)	Speed (in/s)
1	7.87	105.40	7.47E-02
2	7.87	106.10	7.42E-02
3	7.87	105.20	7.48E-02
4	7.87	105.80	7.44E-02
Average	7.87	105.63	7.45E-02

With the current motor, we are not expecting speed to match initial requirements. This recorded speed is for future calculations and is not intended to be a validation of our actuator's performance goals.

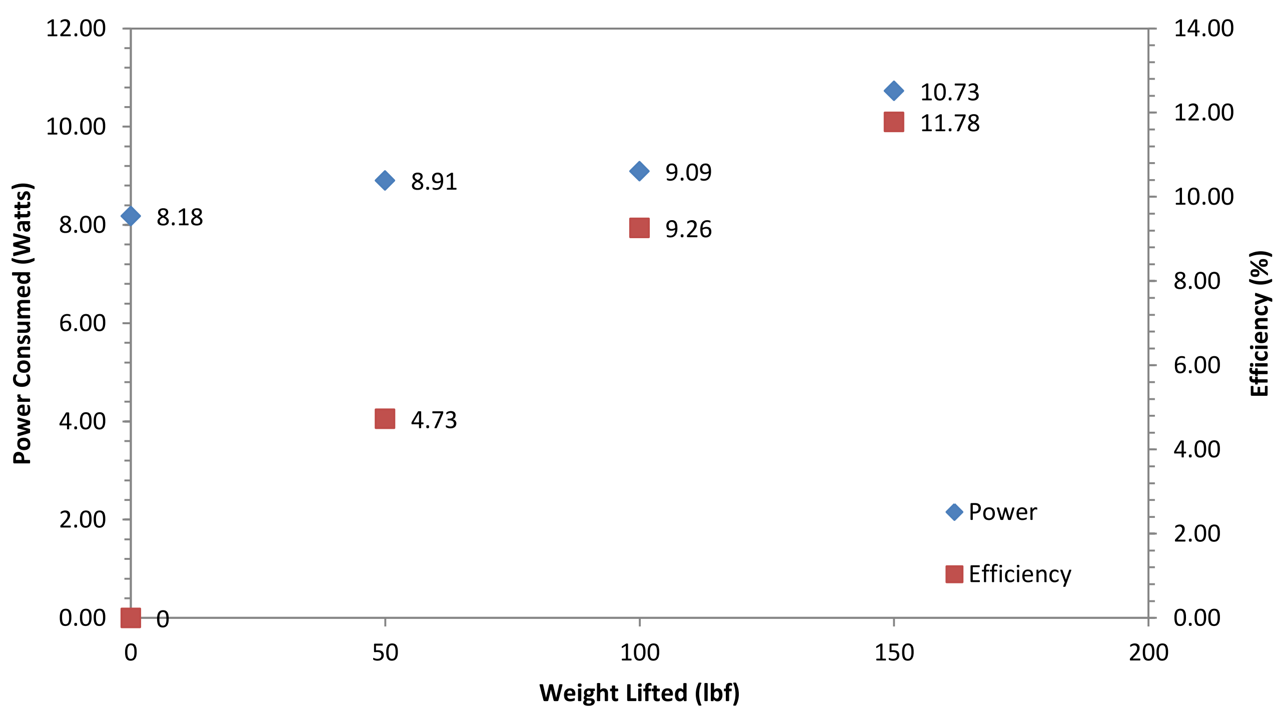


Figure 38. Results of power and efficiency testing

Surface Temperature

We ran our actuator up and down for three hours, and registered no appreciable temperature increase on the motor's surface. This is due to the extremely light loading conditions the motor is being subjected to. It is designed to lift large loads slowly, and we are asking it to lift a small load equally slowly. We are assuming that the drive train losses are distributed across a wide enough area to be negligible in terms of temperature rise, or are dissipated in the ball nut where temperatures cannot be measured. Power dissipated in the system is shown in Table 6.

Figure 39. Brian installing weights for load testing

Table 6. Power Consumed, Efficiency, and Dissipated Power

Weight Raised (lb)	Average Current (A)	Voltage (V)	Power Consumption (Watts)	Power Input to Load (Watts)	Efficiency (%)	Lost Power (Watts)
0	0.34	24	8.18	0.00	0.00	8.18
50	0.37	24	8.91	0.42	4.73	8.49
100	0.38	24	9.09	0.84	9.26	8.25
150	0.45	24	10.73	1.26	11.78	9.46

Unsuccessful Tests

Supported Load

While we can assume based on our efficiency tests and observations that our actuator could hold enough weight, there is some slop in the ball nut that makes it impossible to take accurate strain values, as there is some induced shear on the rod which would not be present in the production model. This issue is addressed in the conclusions and recommendations section.

Range of Motion

Unfortunately, there is some interference between the rod's flange and the sheave at the top of the stroke, making a full range of motion test impossible. Currently, there is approximately 80-85 cm of interference-free stroke, and it is difficult to tell exactly where in the stroke the interference begins. To avoid damage to the system caused by an extremely strong motor and poor alignment, we decided to put this test on hold until this issue could be addressed.

Accuracy

This test requires a control system for positioning, as the item of interest being observed here is variation between desired position and actual position. Until a system is installed, this test cannot be completed.

Resolution

Similarly to the accuracy test, it is not possible to perform this test without a controller. We can assume based on our installed encoder and specifications that adequate resolution exists, but we cannot currently test whether or not our system can respond correctly to a small length change.

Chapter 7 – Conclusions and Recommendations

The construction of our actuator took much more time than we anticipated, but that was not surprising. Most processes, even if they went without issue, take time to do right. We had very few issues in the manufacturing process, and the ones that did come up were caught early or were easily addressed.

One overall design issue rests with the constraints on the extending rod. Our team believed we had addressed the issue of rod rotation by adding a linear track to the side, but there is enough slop between the ball nut and linear block mount so that, at small extensions, the rod can rotate about the linear track's axis. This could be addressed by adding a second track to any of the three bare sides of the rod (and accompanying it with a larger sheave to contain it and linear bearing block to guide it), which would eliminate the possibility of rotation. Another symptom of imperfect constraints is interference between the rod's flange and the sheave walls, making it impossible to extend the actuator completely. Better constrains should fix this issue as well.

The yokes of our U-U joints collide sooner than desired due to simple design clearances. By narrowing each of the flanges on the yokes, the joint will be able to rotate much more freely. Also, because we used shoulder bolts to hold the yokes together, the tolerance between the shoulder bolts and the bushings in which they rotate are not very good. We had planned to manufacture our own form of shoulder bolt with the necessary interference fit to prevent backlash, but opted shoulder bolts in order to save time.

Table 7. DVP sheet

Actual Precision DVP									
Report Date:		Sponsor:	Component/Assembly:		REPORTING ENGINEER:				
TEST PLAN									
Item No	Specification or Clause Reference	Test Description	Acceptance Criteria	Test Responsibility	Test Stage	SAMPLES TESTED		TIMING	
						Quantity	Type	Start date	Finish date
1	Stiffness	Weight loading and strain measurement		Brian	DV	1	P	N/A	N/A
2	Accuracy	Angle measurement test apparatus	.112 mm	Brett	DV	20	P	N/A	N/A
3	Resolution	Angle measurement test apparatus	.004 mm	Vince	DV	20	P	N/A	N/A
4	Range of Motion	Validation through full extension and retraction of actuator	1.2 m stroke	Brian	DV	5	P	N/A	N/A
5	Extension and Retraction Speed	Efficiency testing, to appropriately spec a motor to be implemented once funding permits	5.56 cm/s	Brett	DV	4	P	6/2/2012	6/2/2012
6	Surface Temperature	Extend and retract actuator and ensure the temperature doesn't exceed a certain value	50 °C	Vince	DV	3	P	6/2/2012	6/2/2012
7	Cost	Keep receipts	$2,000	All	PV	1	P	4/2/2012	6/2/2012
8	Power	Measure power used to extend actuator	2000 Watts	All	PV	4	P	6/2/2012	6/2/2012

Appendix A - QFD, Decision Matrices etc.

Support/Motion Decision Matrix

	Weight	Price	Manufacturability	Precision	Steadines s	Failsafe	Standalone	Vibration Isolation	Superfluous Range of motion	Speed of Motion	Totals
Weighting factor	0.25	0.75	1.25	2.5	2	1	0.1	1.25	0.4	0.5	10
Actuator - Actuator	0	0	0	0	0	0	0	0	0	0	0
Air Bag - Actuator	1	1	0	-1	-1	-1	-1	-1	-1	0	-6.25
Pillar - Actuator	0	0	0	0	1	1	0	1	-1	0	3.85
Piston - Actuator	0	0	-1	0	-1	0	-1	0	-1	0	-3.75
Airbag - Cable	1	1	1	-1	-1	-1	-1	-1	-1	1	-4.5
Pillar - Cable	1	1	1	0	0	1	0	0	-1	1	3.35
Piston - Cable	1	1	1	-1	-1	0	-1	0	-1	1	-2.25
AirbagPillar - Cable	1	1	1	0	0	1	-1	0	-1	1	3.25
AribagPillar-Actuator	1	0	-1	0	0	1	-1	0	-1	1	0

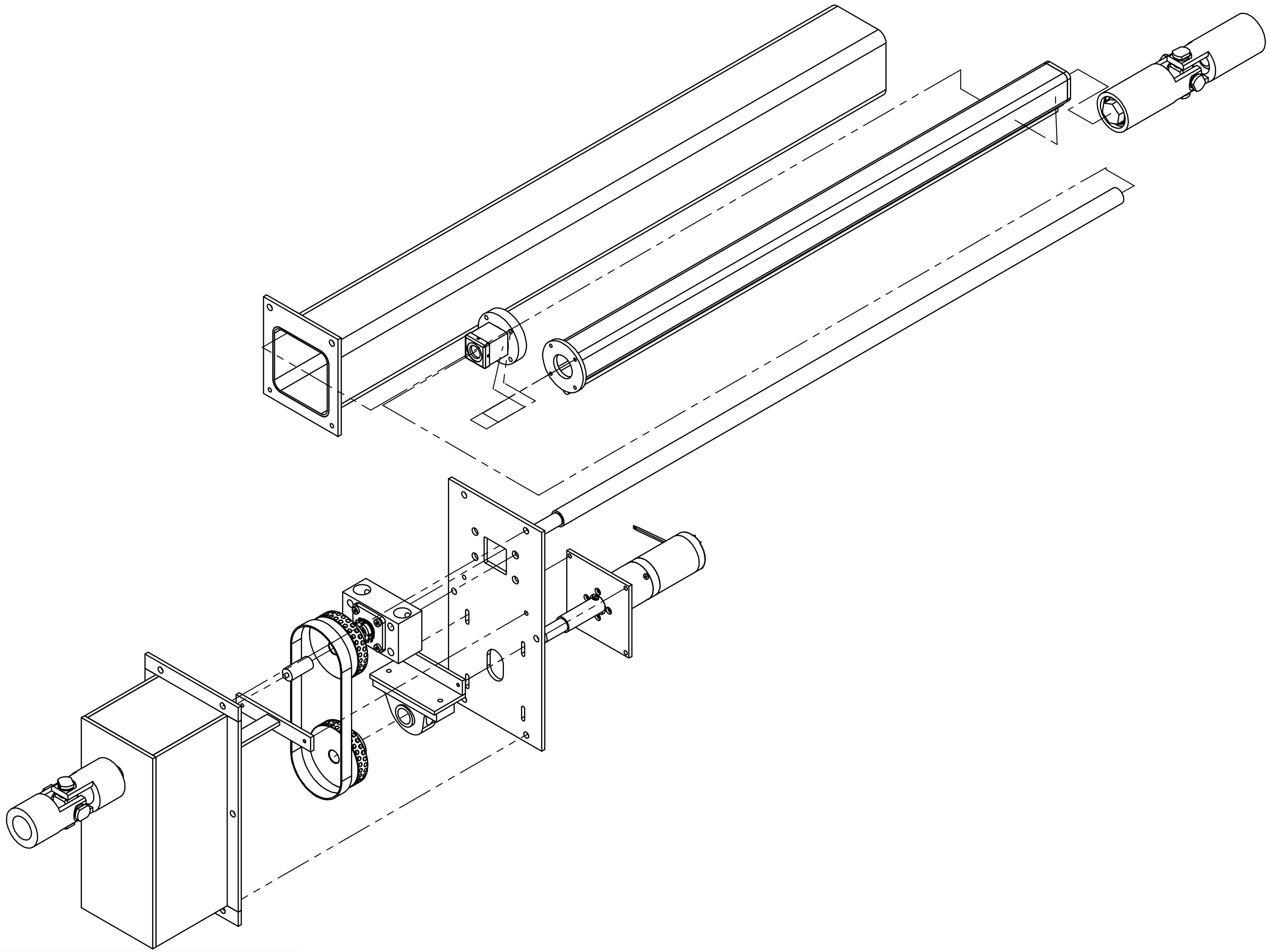
Table 2. House of Quality for Hexapod Actuators

● = 9 Strong Correlation
○ = 3 Medium Correlation
Δ = 1 Small Correlation
Blank = No Correlation

Telescope Hexapod Actuators		Must-Have (*)	Weighting (Total 100)	Cost/Actuator ($)	Supported Load (kN)	Weight (kg)	Base radius (cm)	Accuracy (mm)	Resolution (mm)	Range of Motion (cm)	Sensors digitizing actuator length	Extension speed (cm/s)	Retraction Speed (cm/s)	Load/Actuator (N)	Power Req. (Watts)	Design life (years) (nights/year) (cycles/night)	Max. Surface Temp. (°C)	Resonant frequency > (Hz)	No closure areas less than (cm)	Great	Good	Average	Poor	Zero
Customer Requirements	Low Cost	*	10	●																			●	
	Moderately Portable		4			●																●		
	Support Mass of telescope tube assembly 120 kg	*	10		●	●								●							●			
	1.5 Meter Telescope Base	*	10		Δ	Δ	●			Δ											●			
	Pointing accuracy worst case 20 arcseconds		7					●	○					Δ						●				
	Pointing resolution worst case 1 arcsecond (0.00022°)	*	10					○	●					○						●				
	Telescope can point at all of sky 30° and up		7							●										●				
	Control with PC and stock control system		7								●									●				
	Optimal speed of motion 5 degrees/sec		5			Δ		Δ	Δ	Δ		●	●							●				
	Must look sturdy and safe		6			○								●	○					●				
	Surfaces cannot get too hot		4												●	Δ	●				●			
	Expected to last 10 years		7													●	Δ				●			
	No vibrations below 15 Hz	*	9			Δ				Δ		Δ	Δ					●			●			
	No pinch hazards		4							Δ		Δ	Δ						●			●		
			100																					
	Units			$	kN	kg	cm	mm	mm	cm	—	cm/s	cm/s	kN	Watts	cycles	°C	Hz	cm					
	Targets			Low	10	Low	150	0.112	0.0056	120	yes	5	5	10	2000	20000	50	15	2					
	AMiBA Hexapod Telescope Mount			High	100	1000	—	—	0.0002	340	yes	2	2	—	—	—	—	—	—					
	HPT			High	—	200	150	—	0.011	—	yes	—	—	—	1850	—	—	—	—					

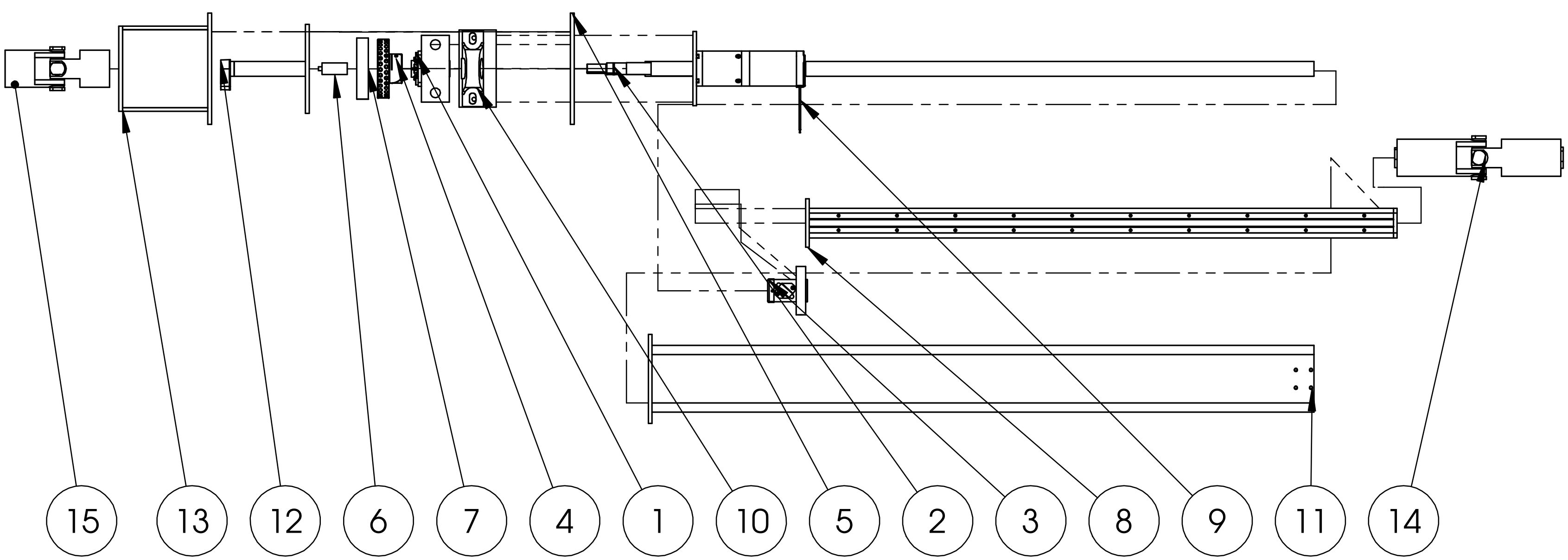
This house of quality facilitates prioritization of requirements, and visually conveys the goals of our design team. In the 'Weighted Total' column, we assign values to each customer desire based on apparent importance to the project. The 'Engineering Requirements' section is a quantification of customer requirements, allowing us to assign values that can later be tested verified. We benchmarked two competitors, the AMiBA [15] and the HPT [11], and compared them both qualitatively to our customer desires and quantitatively to our engineering requirements. This process will help our team direct energy toward areas that need improvement and toward areas that our competitors do not currently address, with a final product that fills a market gap not currently addressed.

Appendix B Drawing Packet



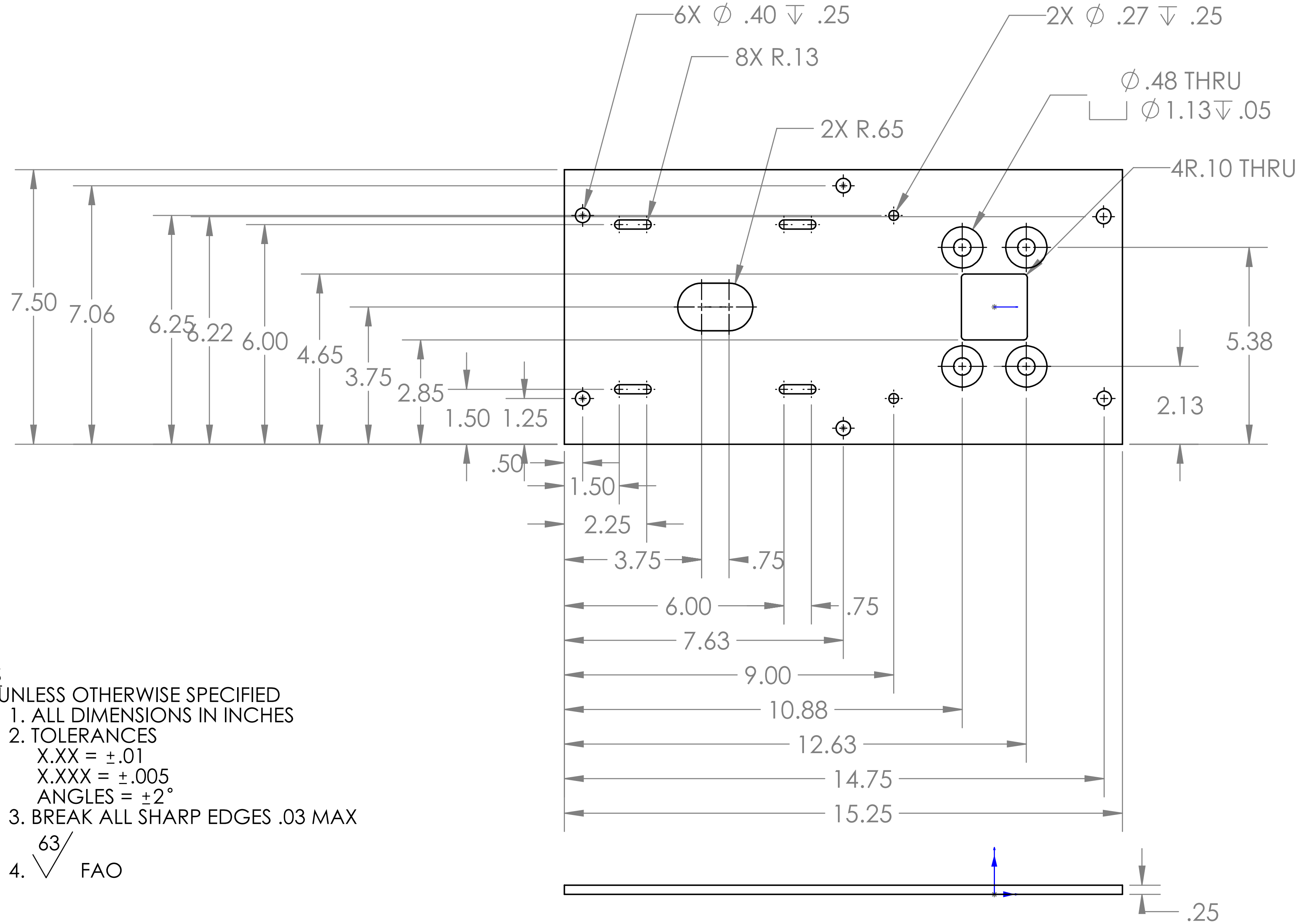
Mechanical Engineering

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.	Vendor	VendorNo
1	GHT001	Pillow Block	1	Thomson Linear	7824157
2	GHT002	Ball Screw	1	Thomson Linear	7820426
3	GHT003	Ball Nut & Flange	1	Thomson Linear	5966K47
4	GHT004	Timing Belt Pulley	2	SDI	A 6A14M32D20
5	GHT005	Mount Plate	1		
6	GHT006	Output Shaft	1		
7	GHT007	Drive Belt	1		
8	GHT100	Rod Assembly	1		
9	GHT200	Motor Assembly	1		
10	GHT300	Bearing Mount Assembly	1		
11	GHT400	Sheave Assembly	1		
12	GHT500	Encoder Assembly	1		
13	GHT600	Housing Assembly	1		
14	GHT750	Universal Universal Joint	1		
15	GHT800	Universal Joint	1		



Actual Precision	DATE: 3/7/12	SCALE: 1 : 12	TITLE: Final Assembly
Mechanical Engineering			DWG #: GHT000
	UNITS: INCHES	NEXT ASSY: N/A	SIGNATURE:

4 3 2 1



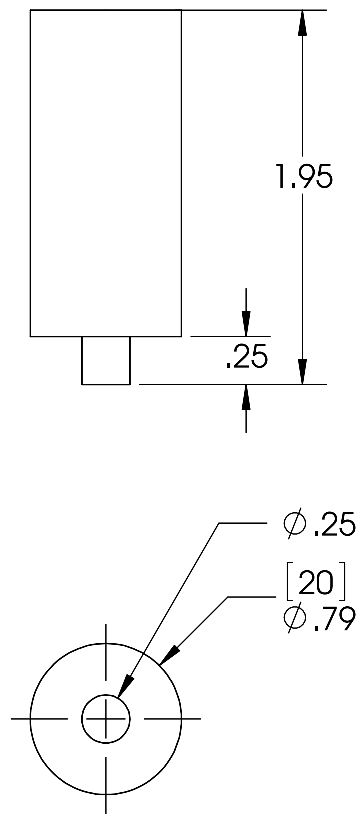
NOTES
UNLESS OTHERWISE SPECIFIED
1. ALL DIMENSIONS IN INCHES
2. TOLERANCES
X.XX = ±.01
X.XXX = ±.005
ANGLES = ±2°
3. BREAK ALL SHARP EDGES .03 MAX
4. $\sqrt{63}$ FAO

Actual Precision

Mechanical Engineering CAL POLY

DATE: 3-6-12	SCALE 1:4	MATERIAL: Carbon Steel
TOLERANCE: 0.01	NEXT ASSY: GHT000	DRAWING #: GHT005
UNITS: INCHES	TITLE: Mount Plate	SIGNATURE:

5 4 3 2 1



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2. (DIMENSIONS) ARE IN MM
3. TOLERANCES
X.XX = ±.01
X.XXX = ±.005
ANGLES = ±2°
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5. 63√ FAO

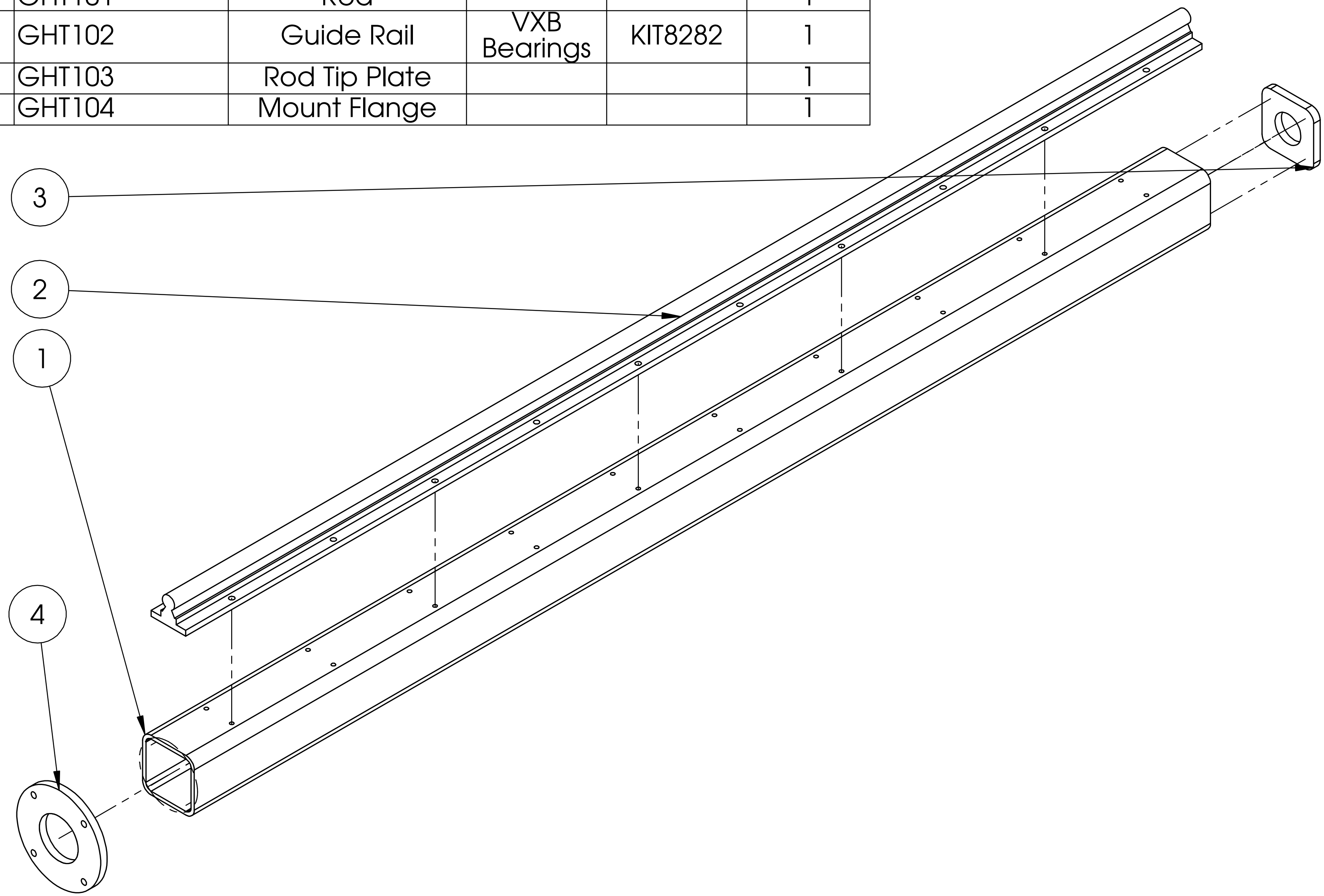
Actual Precision

Mechanical Engineering

DATE: 3-6-12	SCALE 1:1	MATERIAL: Carbon Steel
TOLERANCE: ±0.01	NEXT ASSY: GHT000	DRAWING #: GHT006
UNITS: INCHES	TITLE: Output Shaft	SIGNATURE:

4 3 2 1

ITEM NO.	PART NUMBER	DESCRIPTION	Vendor	VendorNo	QTY.
1	GHT101	Rod			1
2	GHT102	Guide Rail	VXB Bearings	KIT8282	1
3	GHT103	Rod Tip Plate			1
4	GHT104	Mount Flange			1

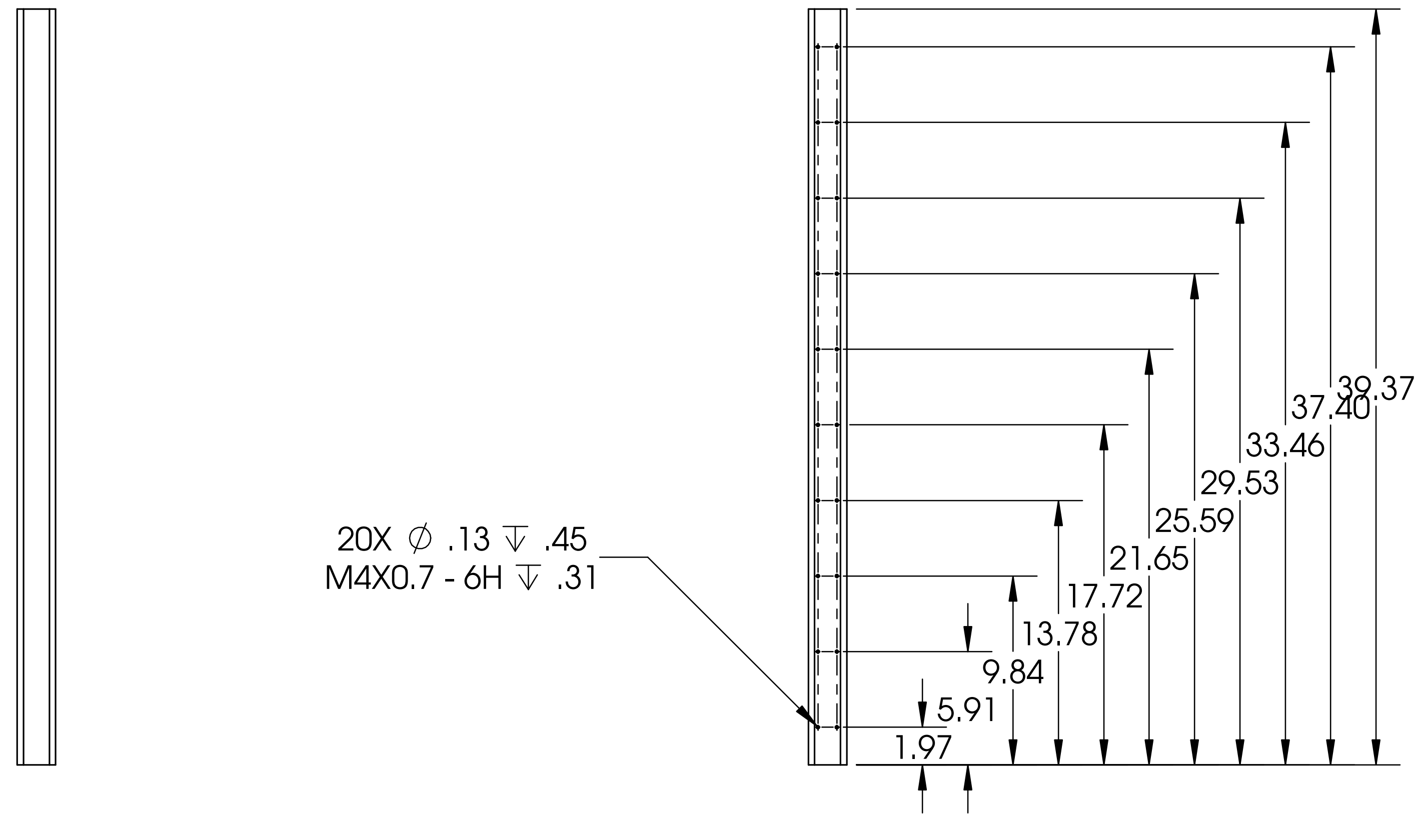


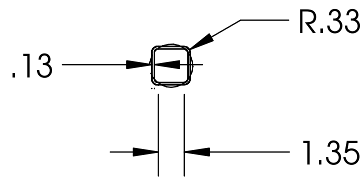
Actual Precision

Mechanical Engineering

DATE: 3/7/12	SCALE: 1 : 4	TITLE: Rod Assembly
		DWG #: GHT100
UNITS: INCHES	NEXT ASSY: GHT000	SIGNATURE:

4 3 2 1





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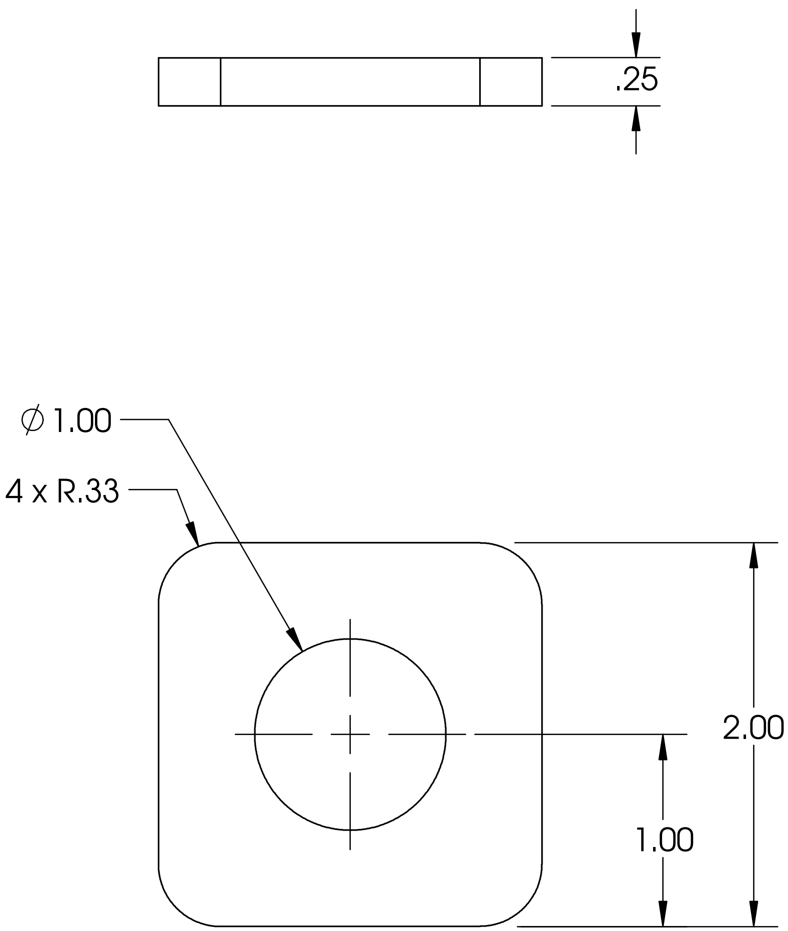
X.XX = ±.01
X.XXX = ±.005
ANGLES = ±2°
3. BREAK ALL SHARP EDGES .03 MAX
4. 63 √ FAO

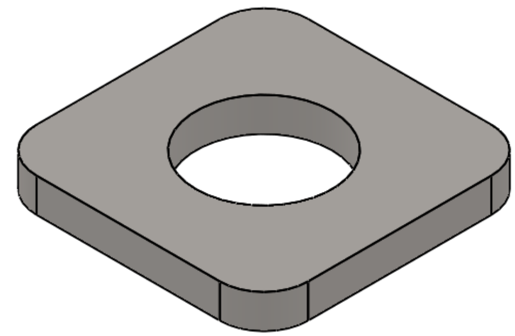
Actual Precision

Mechanical Engineering

DATE: 3-6-12	SCALE 1:10	MATERIAL: Carbon Steel
TOLERANCE: 0.01	NEXT ASSY: GHT100	DRAWING #: GHT101
UNITS: INCHES	TITLE: Rod	SIGNATURE:

4 3 2 1





NOTES

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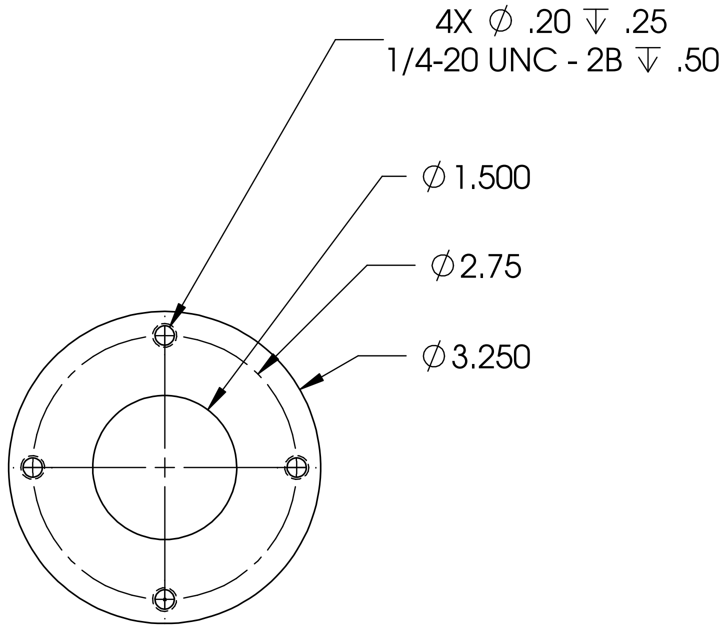
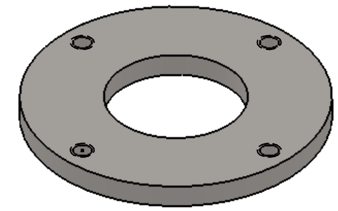
2. TOLERANCES
   X.XX = ±.01
   X.XXX = ±.005
   ANGLES = ±2°
3. BREAK ALL SHARP EDGES .03 MAX
4. √63 FAO

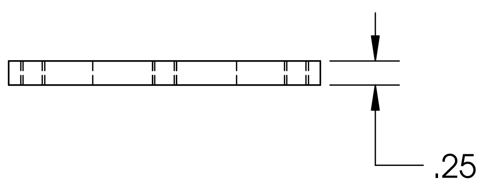
Actual Precision

Mechanical Engineering

DATE: 3-6-12	SCALE 1:1	MATERIAL: Carbon Steel
TOLERANCE: ±0.01	NEXT ASSY: GHT100	DRAWING #: GHT102
UNITS: INCHES	TITLE: Rod Tip Plate	SIGNATURE:







NOTES

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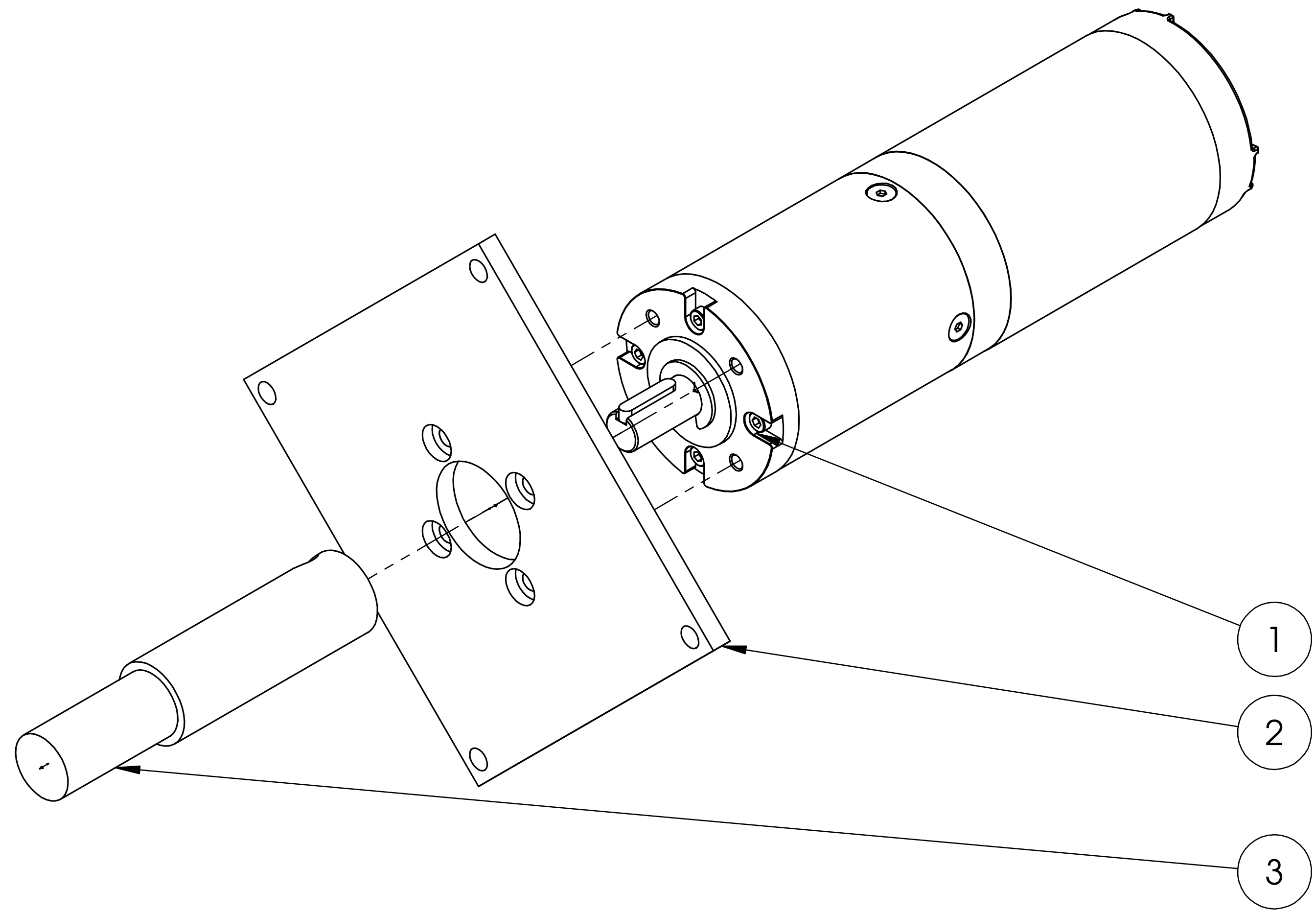
   X.XX = ±.01
   X.XXX = ±.005
   ANGLES = ±2°
3. BREAK ALL SHARP EDGES .03 MAX
4. 63√ FAO

Actual Precision	DATE: 3/7/12	SCALE: 1 : 2	TITLE: Rod Mounting Flange
Mechanical Engineering	TOLERANCE: ±0.01	MATERIAL: Carbon Steel	DWG #: GHT103
	UNITS: INCHES	NEXT ASSY: GHT100	SIGNATURE:

4 3 2 1

ITEM NO.	PART NUMBER	DESCRIPTION	Vendor	VendorNo	QTY.
1	GHT201	Motor	Anaheim Automation	BDPG-60-110-R168	1
2	GHT202	Motor Mount Plate			1
3	GHT203	Drive Shaft			1

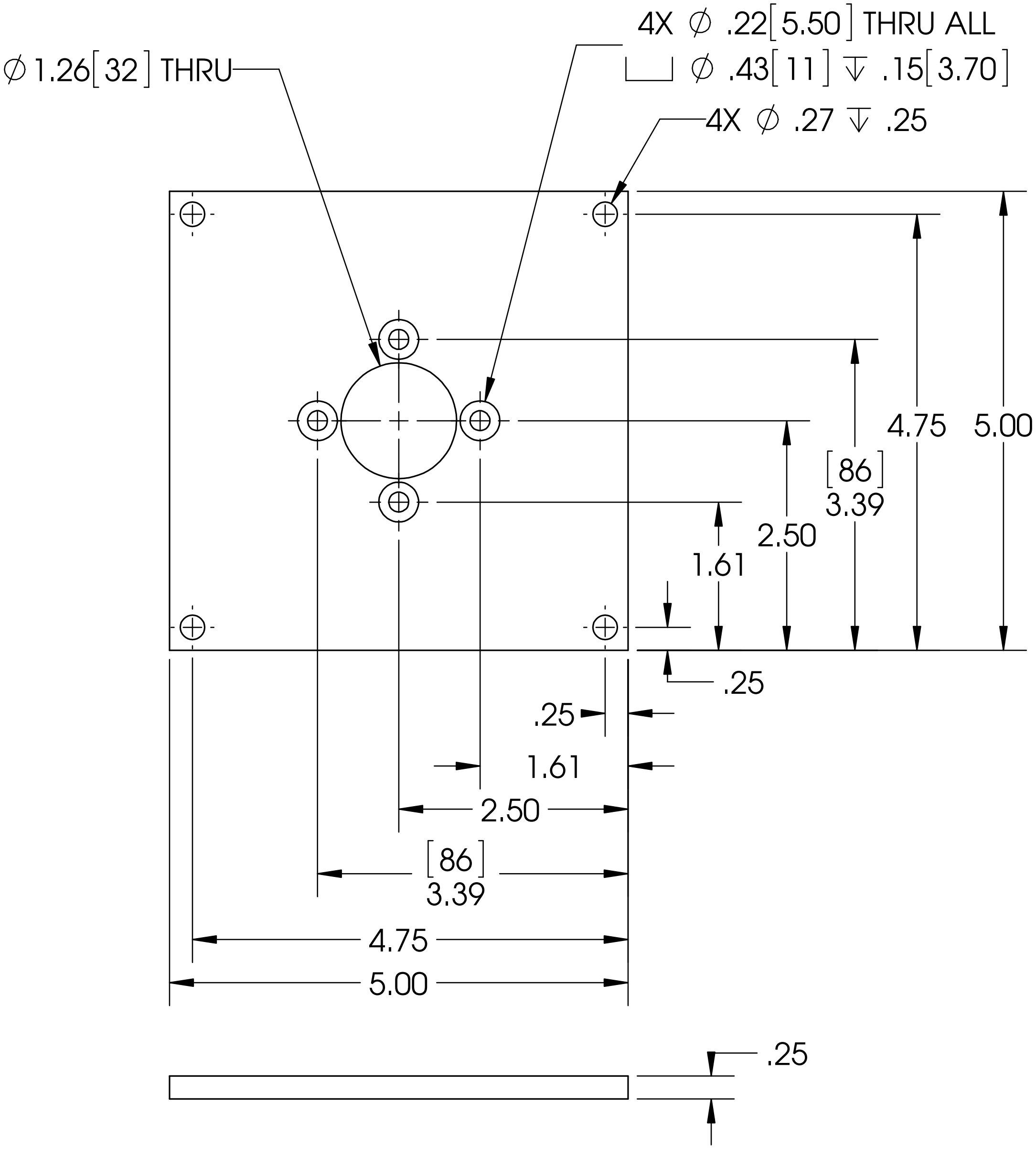


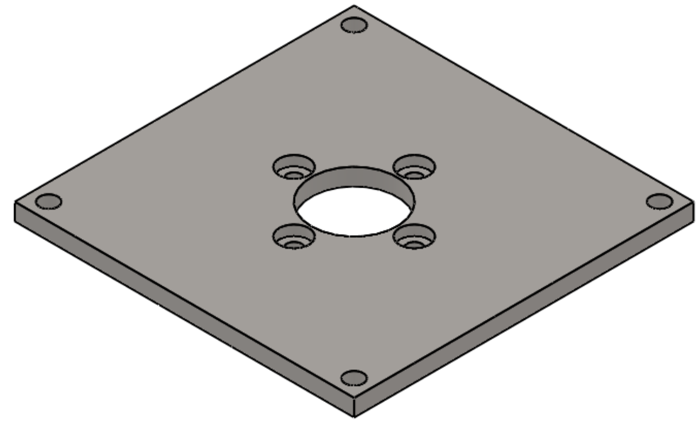
Actual Precision

Mechanical Engineering

DATE: 3/7/12	SCALE: 1 : 2	TITLE: Motor Drive Assembly
		DWG #: GHT200
UNITS: INCHES	NEXT ASSY: GHT000	SIGNATURE:

4 3 2 1





NOTES

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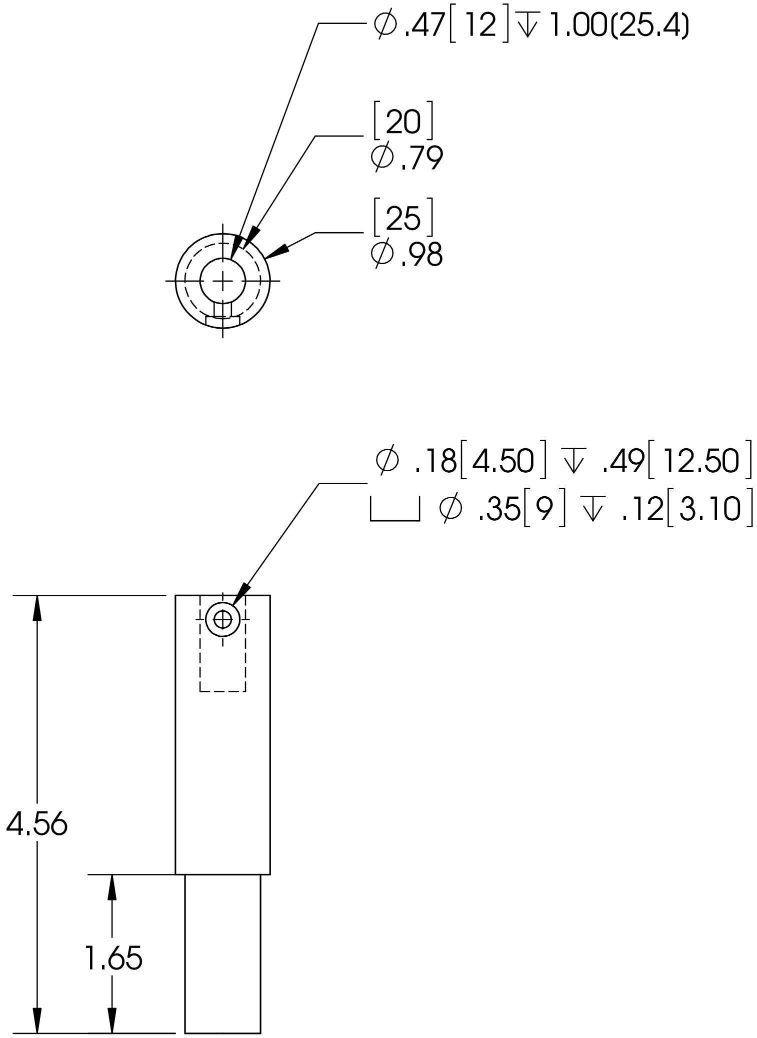
1. ALL DIMENSIONS IN INCHES
2. (DIMENSIONS) ARE IN MM
3. TOLERANCES
   X.XX = ±.01
   X.XXX = ±.005
   ANGLES = ±2°
4. BREAK ALL SHARP EDGES .03 MAX
5. 63√ FAO

Actual Precision

Mechanical Engineering

DATE: 3-6-12	SCALE 1:2	MATERIAL: Carbon Steel
TOLERANCE: ±0.01	NEXT ASSY: GHT200	DRAWING #: GHT202
UNITS: INCHES (mm)	TITLE: Motor Mount Plate	SIGNATURE:

4 3 2 1



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3. TOLERANCES
X.XX = ±.01
X.XXX = ±.005
ANGLES = ±2°
4. BREAK ALL SHARP EDGES .03 MAX
5. 63 √ FAO

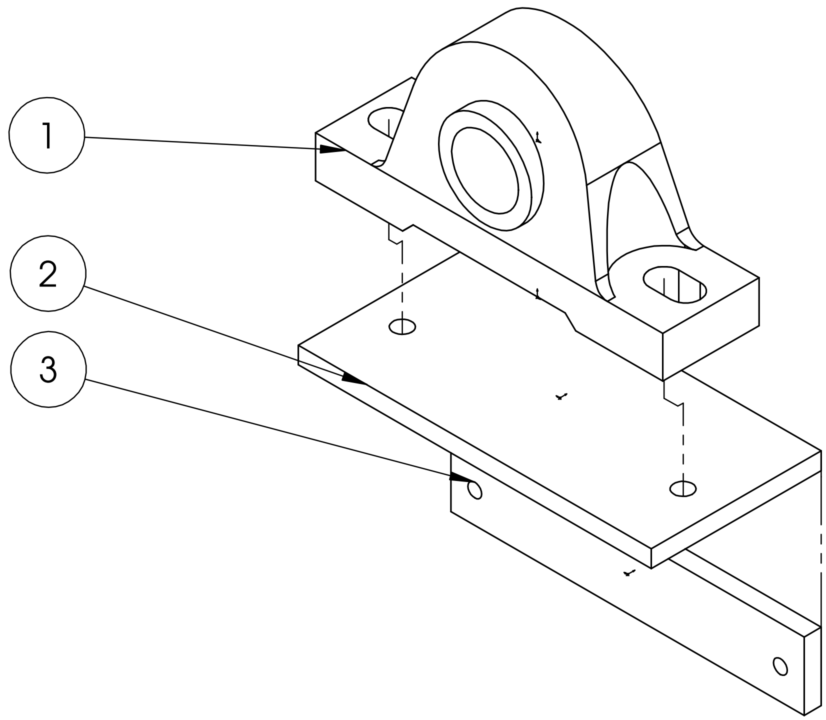
Actual Precision

Mechanical Engineering

DATE: 3-6-12	SCALE 1:2	MATERIAL: Aluminum
TOLERANCE: 0.01	NEXT ASSY: GHT200	DRAWING #: GHT203
UNITS: INCHES (mm)	TITLE: Drive Shaft	SIGNATURE:

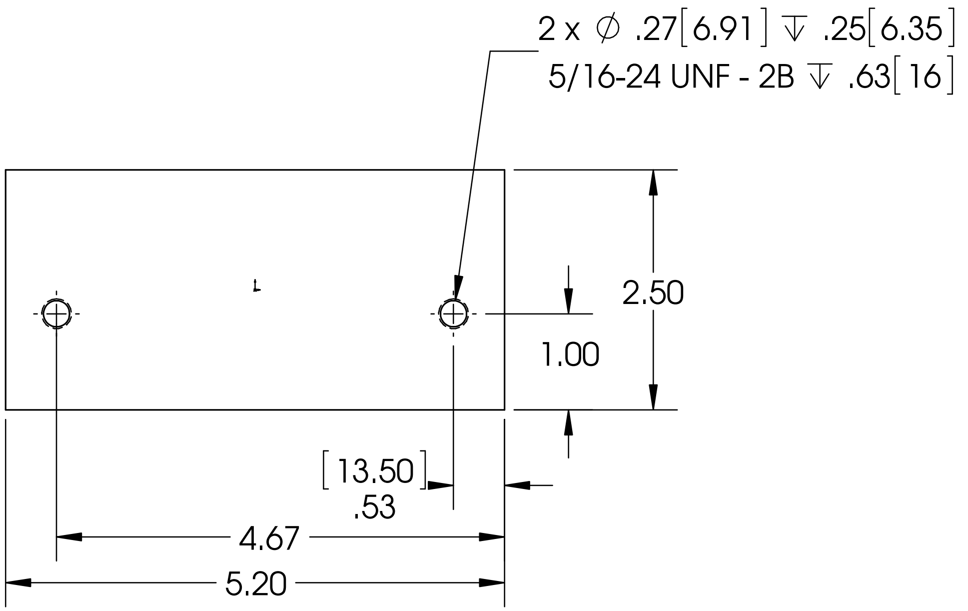
4 3 2 1

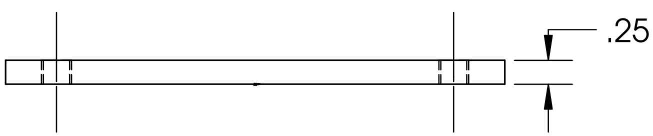
ITEM NO.	PART NUMBER	DESCRIPTION	Vendor	VendorNo	QTY.
1	GHT301	Base Mount Bearing	McMaster-Carr	57085K52	1
2	GHT302	Bearing Mount Plate			1
3	GHT303	Bearing Mount Flange			1

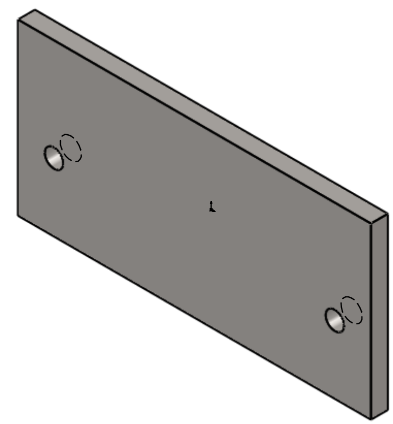


Actual Precision	DATE: 3/7/12	SCALE: 1 : 2	TITLE: Bearing Mount Assembly
Mechanical Engineering			DWG #: GHT300
	UNITS: INCHES	NEXT ASSY: GHT000	SIGNATURE:

4 3 2 1







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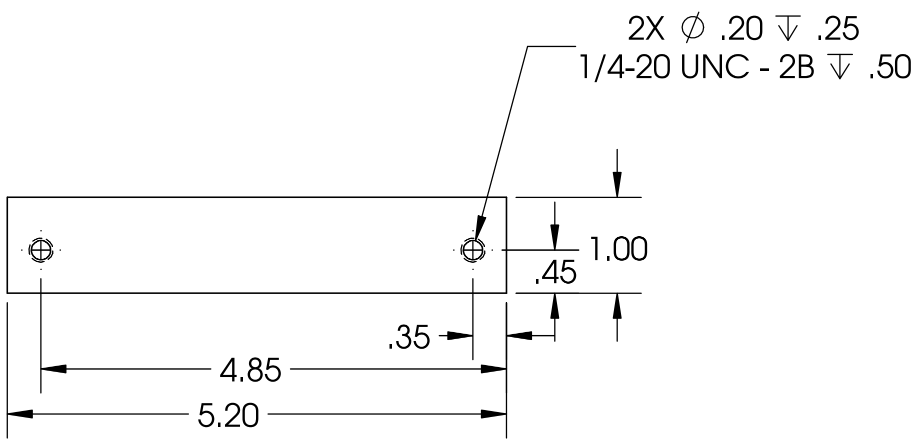
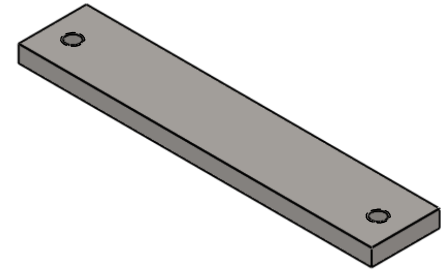
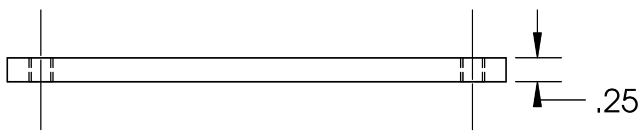
2. (DIMENSIONS) ARE IN MM
3. TOLERANCES
X.XX = ±.01
X.XXX = ±.005
ANGLES = ±2°
4. BREAK ALL SHARP EDGES .03 MAX
5. 63√ FAO

Actual Precision	DATE: 3-6-12	SCALE 1:2	MATERIAL: Carbon Steel
Mechanical Engineering	TOLERANCE: ±0.01	NEXT ASSY: GHT300	DRAWING #: GHT302
	UNITS: INCHES (mm)	TITLE: Bearing Mount Plate	SIGNATURE:

4 3 2 1







<u>NOTES</u>

UNLESS OTHERWISE SPECIFIED

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2. TOLERANCES
   X.XX = ±.01
   X.XXX = ±.005
   ANGLES = ±2°
3. BREAK ALL SHARP EDGES .03 MAX
4. $\sqrt{}$ 63 FAO

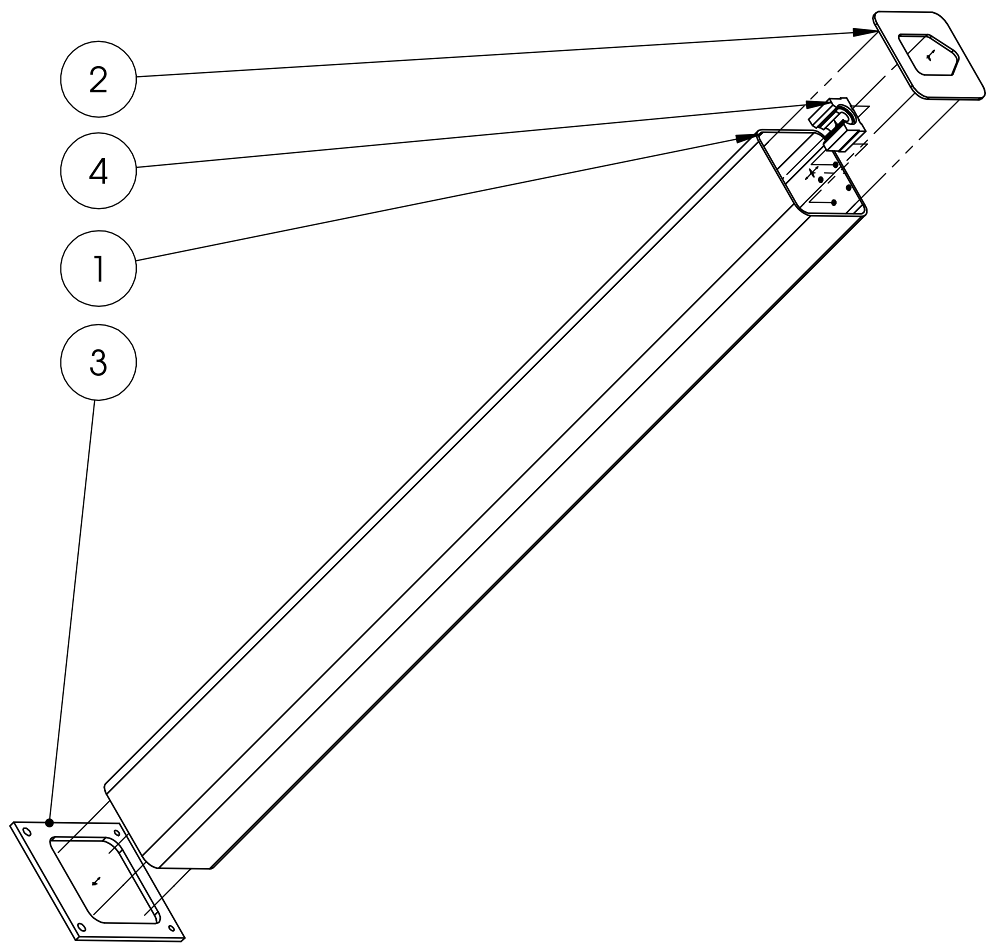
Actual Precision

Mechanical Engineering

DATE: 3-6-12	SCALE 1:2	MATERIAL: Carbon Steel
TOLERANCE: ±0.01	NEXT ASSY: GHT300	DRAWING #: GHT303
UNITS: INCHES	TITLE: Bearing Mount Flange	SIGNATURE:

4 3 2 1

ITEM NO.	PART NUMBER	DESCRIPTION	Vendor	VendorNo	QTY.
1	GHT401	Sheave			1
2	GHT402	Sheave Cap			1
3	GHT403	Sheave Flange			1
4	GHT404	Linear Bearing Block	VXB Bearings	KIT8512	1

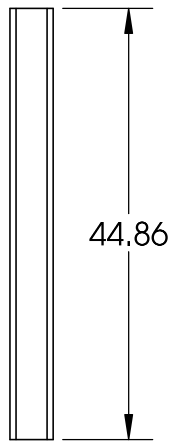


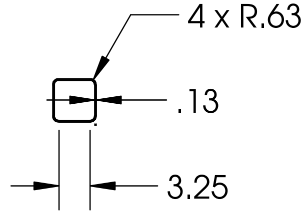
Actual Precision

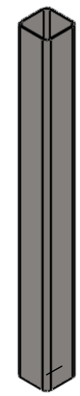
Mechanical Engineering

DATE: 3/7/12	SCALE: 1 : 8	TITLE: Sheave Assembly
		DWG #: GHT400
UNITS: INCHES	NEXT ASSY: GHT000	SIGNATURE:

4 3 2 1







NOTES
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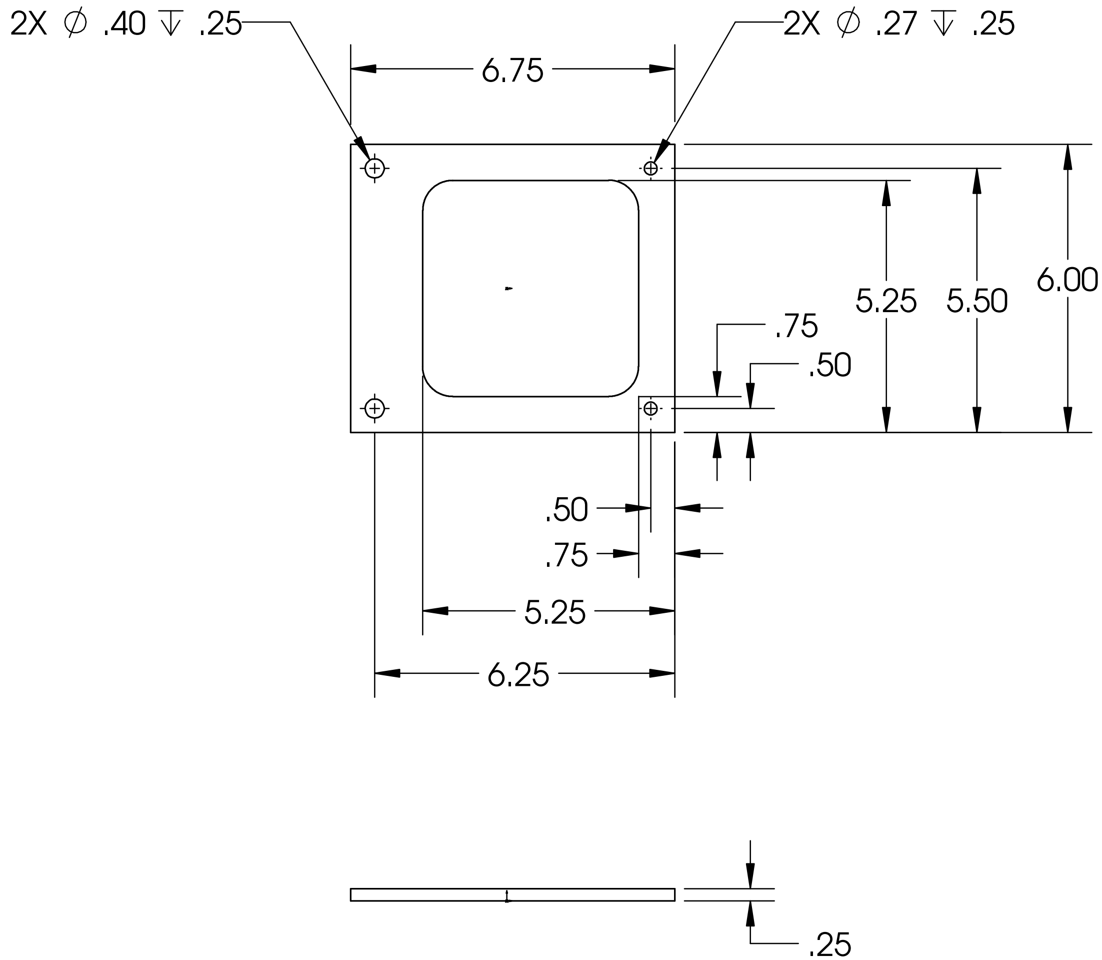
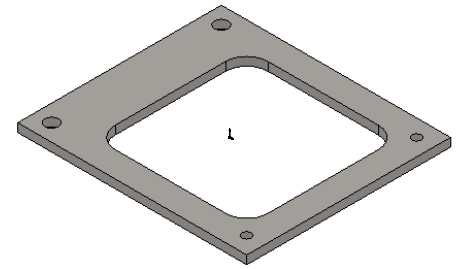
2. TOLERANCES
X.XX = ±.01
X.XXX = ±.005
ANGLES = ±2°
3. BREAK ALL SHARP EDGES .03 MAX
4. 63√ FAO

Actual Precision

Mechanical Engineering

DATE: 2-26-12	SCALE 1:20	MATERIAL: Carbon Steel
TOLERANCE: 0.01	NEXT ASSY: GHT400	DRAWING #: GHT401
UNITS: INCHES	TITLE: Sheave	SIGNATURE:





NOTES

UNLESS OTHERWISE SPECIFIED

1. ALL DIMENSIONS IN INCHES
2. TOLERANCES
   X.XX = ±.01
   X.XXX = ±.005
   ANGLES = ±2°
3. BREAK ALL SHARP EDGES .03 MAX
4. 63√ FAO

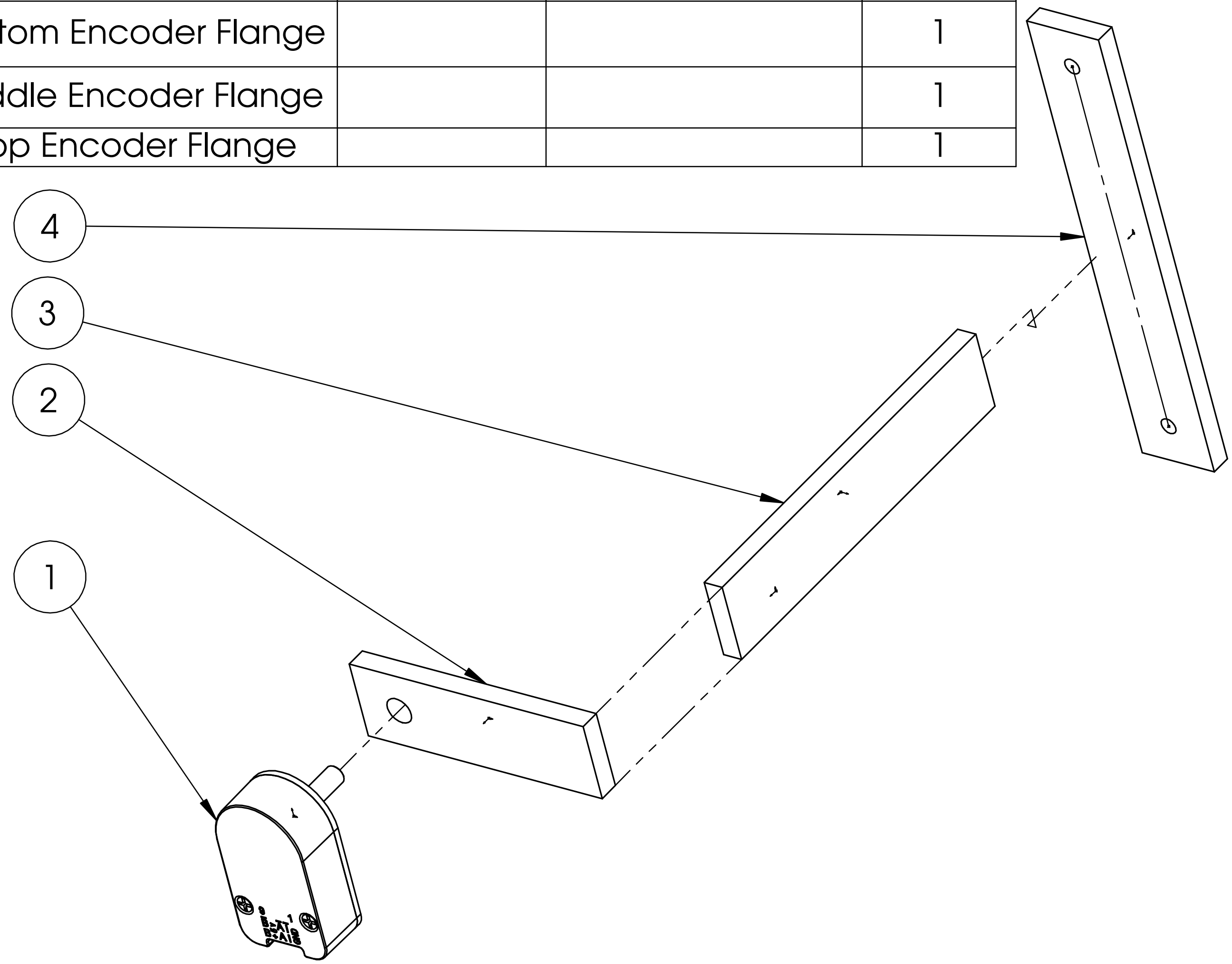
Actual Precision

Mechanical Engineering

DATE: 3-7-12	SCALE 1:4	MATERIAL: Carbon Steel
TOLERANCE: 0.01	NEXT ASSY: GHT400	DRAWING #: GHT403
UNITS: INCHES	TITLE: Sheave Flange	SIGNATURE:

4 3 2 1

ITEM NO.	PART NUMBER	DESCRIPTION	Vendor	VendorNo	QTY.
1	GHT501	Optical Shaft Encoder	U.S. Digital	S5-1000-236-NE-D-B	1
2	GHt502	Bottom Encoder Flange			1
3	GHT503	Middle Encoder Flange			1
4	GHT504	Top Encoder Flange			1

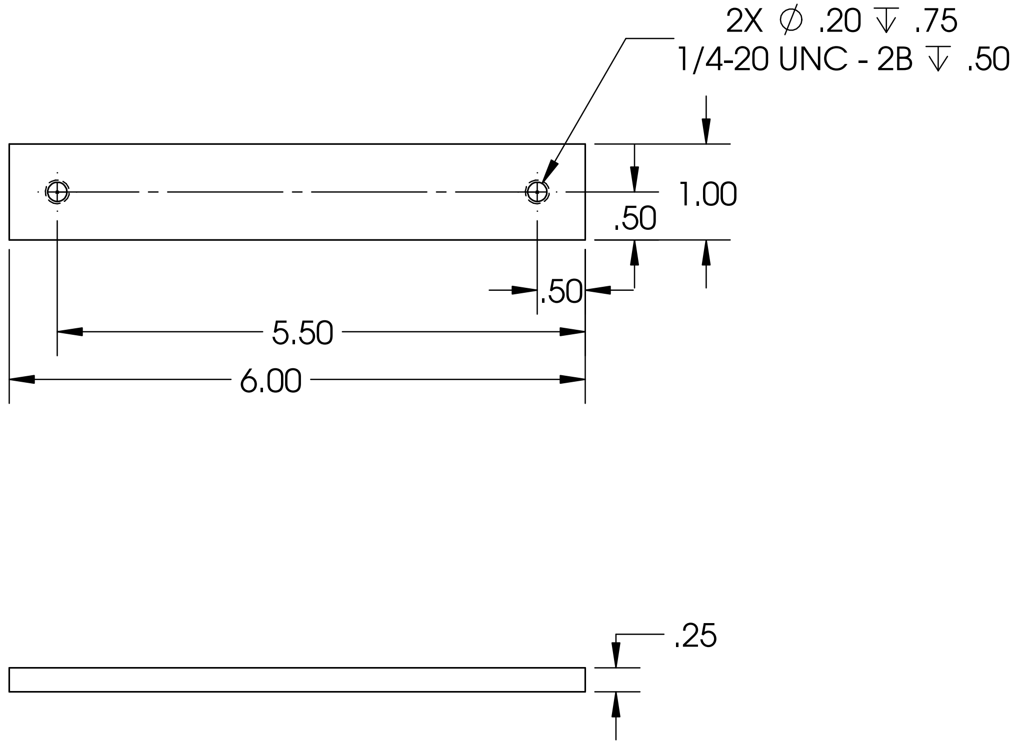


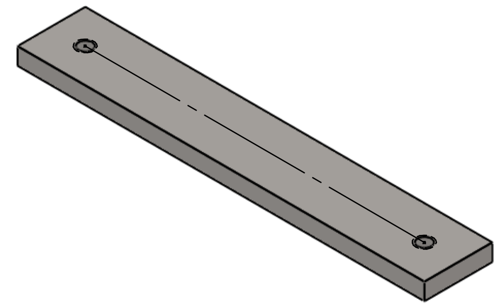
Actual Precision

Mechanical Engineering

DATE: 3/7/12	SCALE: 1 : 2	TITLE: Encoder Assembly
		DWG #: GHT500
UNITS: INCHES	NEXT ASSY: GHT000	SIGNATURE:

4 3 2 1





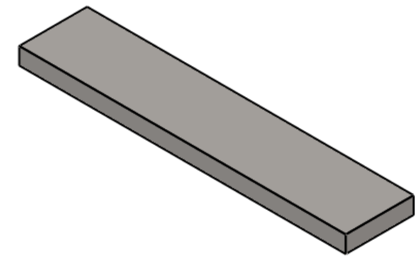
NOTES
UNLESS OTHERWISE SPECIFIED
1. ALL DIMENSIONS IN INCHES
2. TOLERANCES
X.XX = ±.01
X.XXX = ±.005
ANGLES = ±2°
3. BREAK ALL SHARP EDGES .03 MAX
4. 63√ FAO

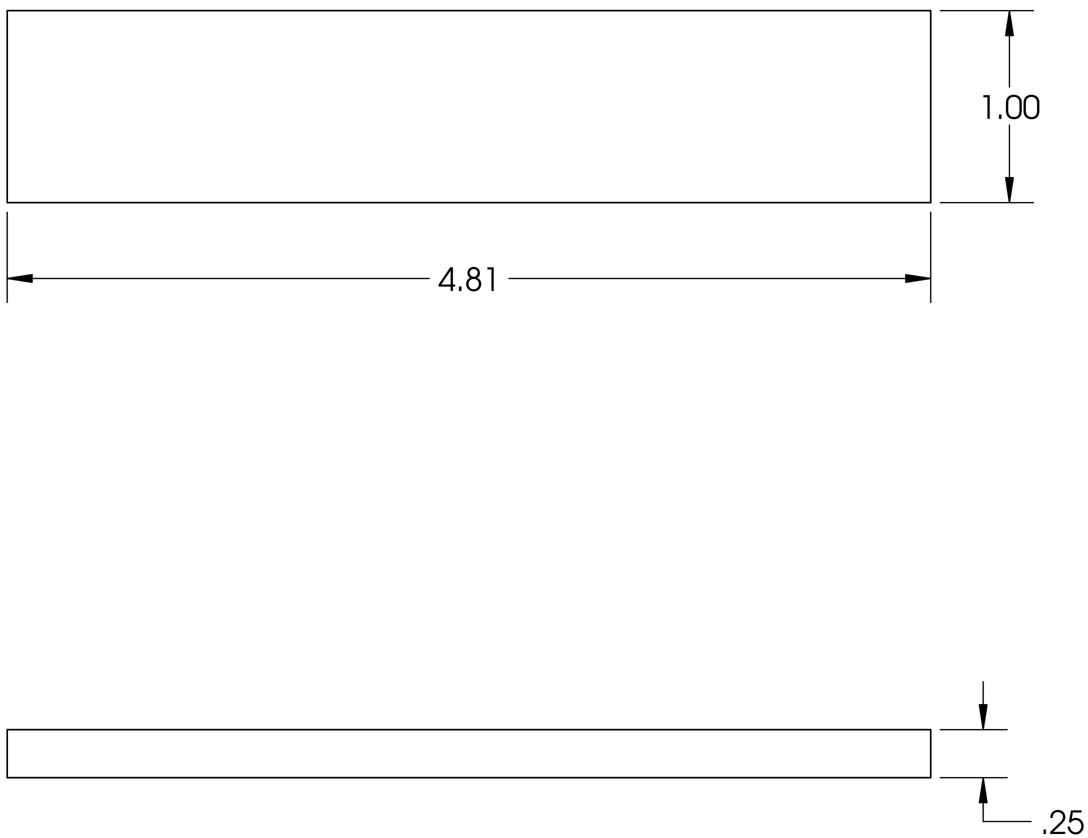
Actual Precision

Mechanical Engineering

DATE: 3-6-12	SCALE 1:2	MATERIAL: Carbon Steel
TOLERANCE: ±0.01	NEXT ASSY: GHT500	DRAWING #: GHT502
UNITS: INCHES	TITLE: Encoder Bottom Flange	SIGNATURE:

4 3 2 1



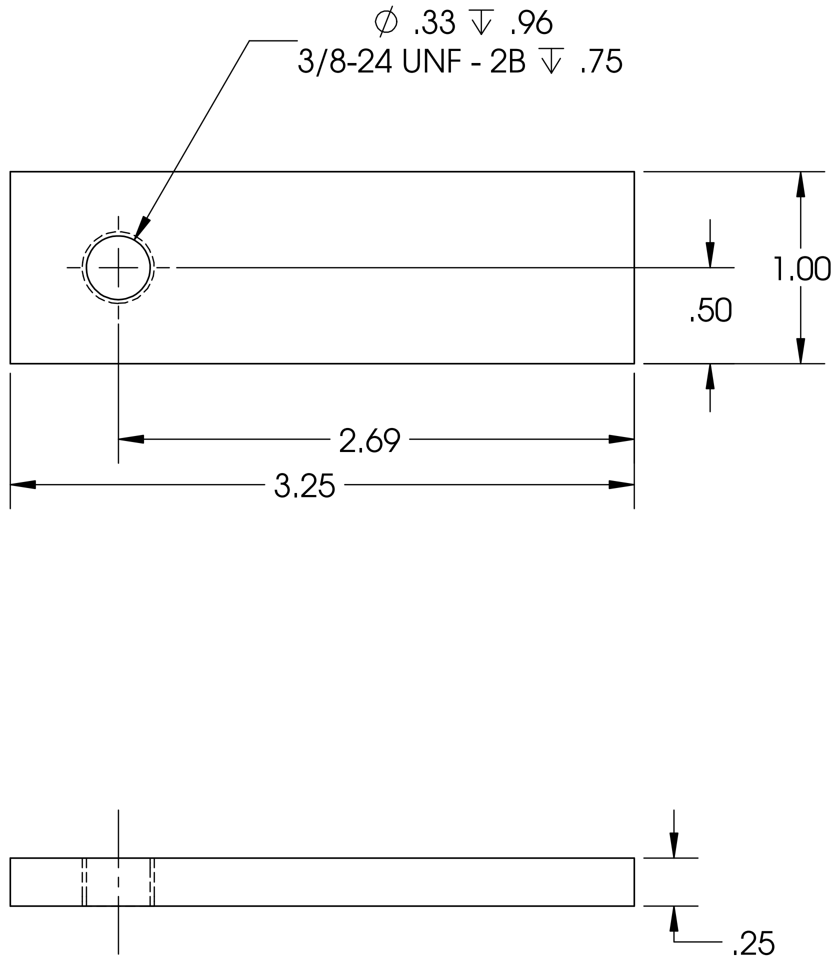


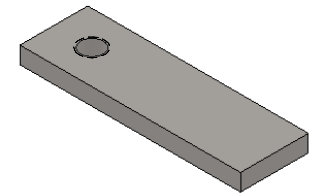
NOTES
UNLESS OTHERWISE SPECIFIED
1. ALL DIMENSIONS IN INCHES
2. TOLERANCES
X.XX = ±.01
X.XXX = ±.005
ANGLES = ±2°
3. BREAK ALL SHARP EDGES .03 MAX
4. 63 FAO

Actual Precision

Mechanical Engineering

DATE: 3-6-12	SCALE 1:1	MATERIAL: Carbon Steel
TOLERANCE: ±0.01	NEXT ASSY: GHT500	DRAWING #: GHT503
UNITS: INCHES	TITLE: Encoder Middle Flange	SIGNATURE:





NOTES
UNLESS OTHERWISE SPECIFIED
1. ALL DIMENSIONS IN INCHES
2. TOLERANCES
X.XX = ±.01
X.XXX = ±.005
ANGLES = ±2°
3. BREAK ALL SHARP EDGES .03 MAX
4. 63 √ FAO

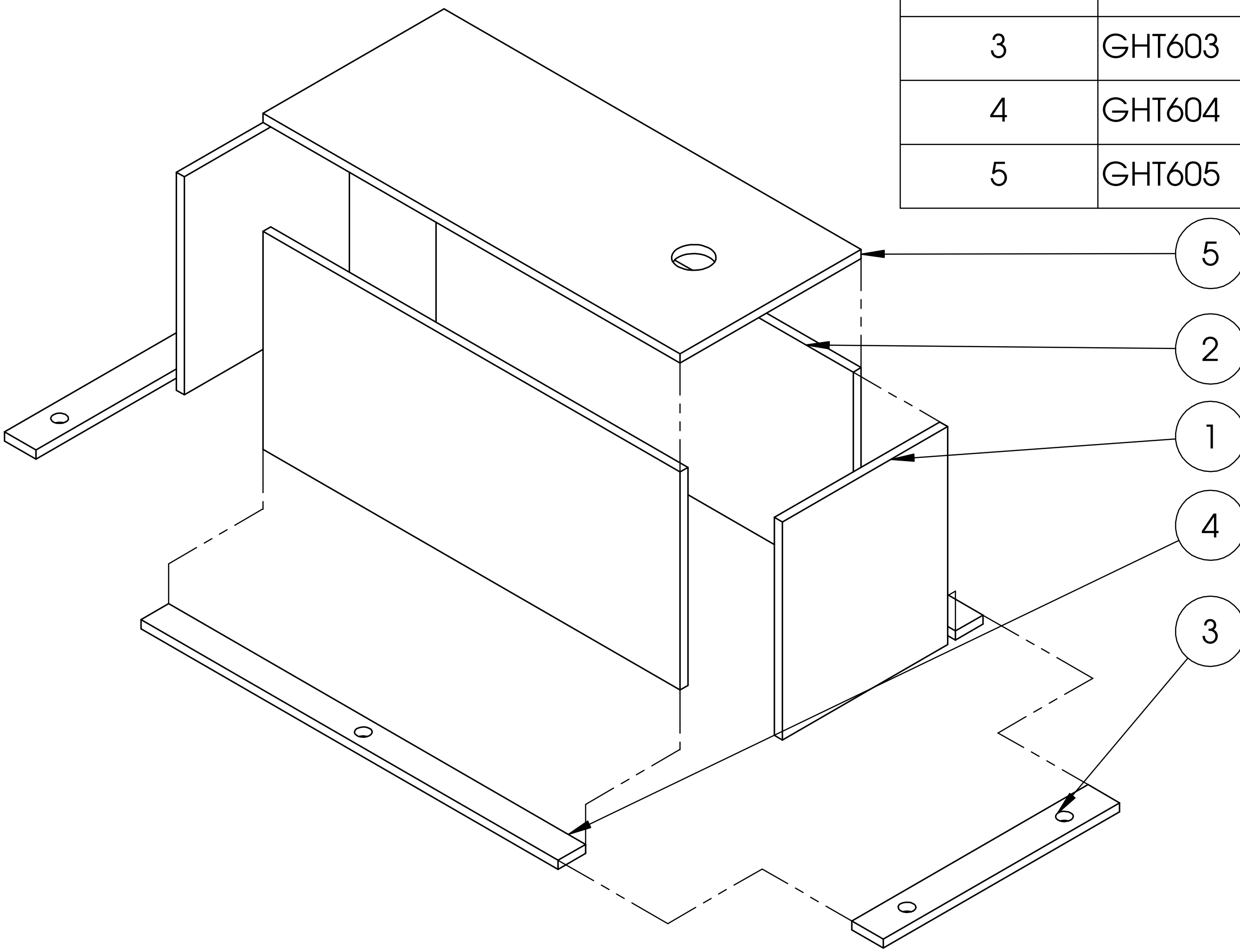
Actual Precision

Mechanical Engineering

DATE: 3-6-12	SCALE 1:1	MATERIAL: Carbon Steel
TOLERANCE: ±0.01	NEXT ASSY: GHT500	DRAWING #: GHT504
UNITS: INCHES	TITLE: Encoder Top Flange	SIGNATURE:

4 3 2 1

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	GHT601	Housing End Plate	2
2	GHT602	Housing Side Plate	2
3	GHT603	Housing End Flange	2
4	GHT604	Housing Side Flange	2
5	GHT605	Housing Bottom Plate	1

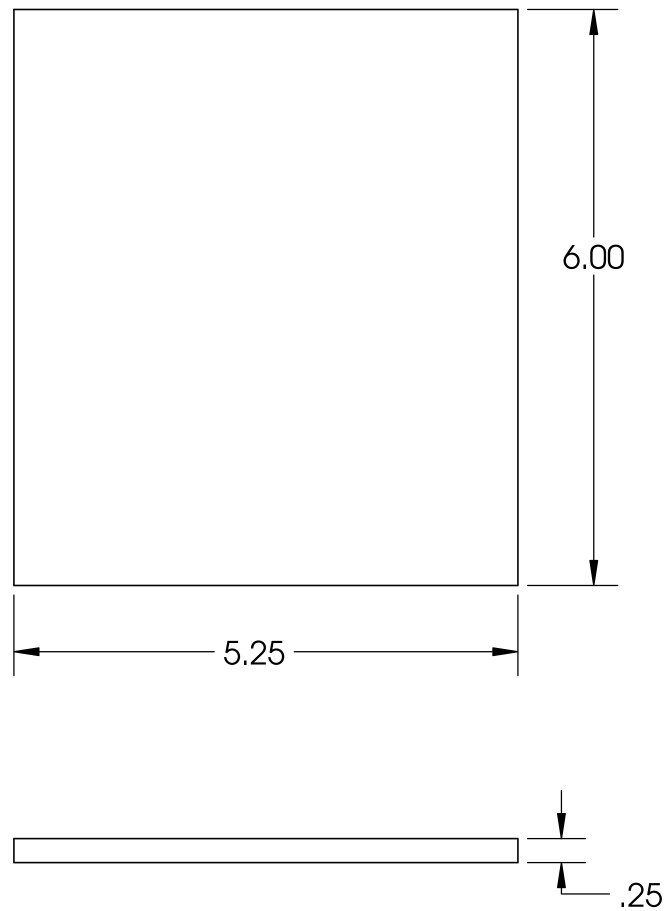


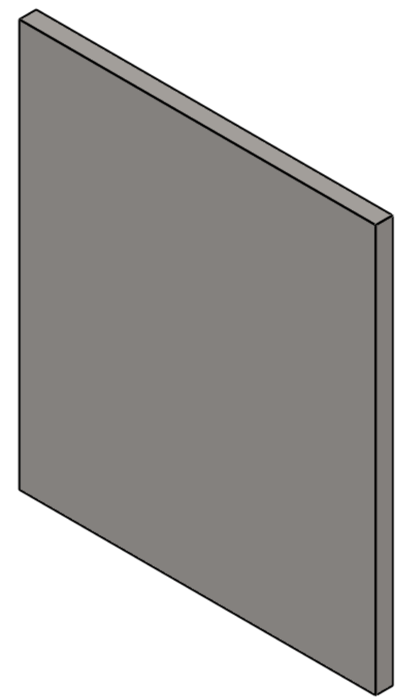
Actual Precision

Mechanical Engineering

DATE: 3/7/12	SCALE: 1 : 4	TITLE: Housing Assembly
		DWG #: GHT600
UNITS: INCHES	NEXT ASSY: GHT000	SIGNATURE:

4 3 2 1





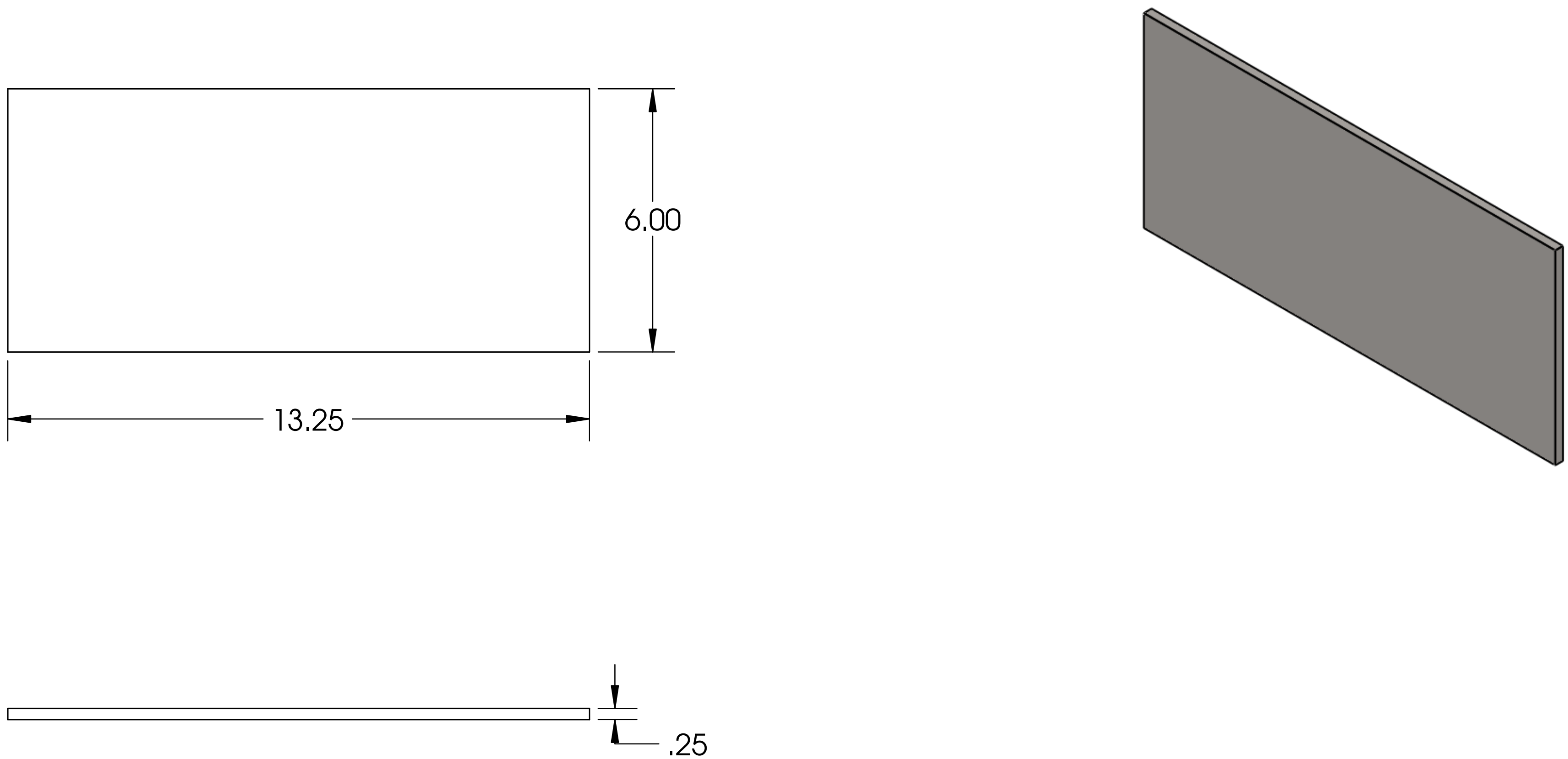
NOTES
UNLESS OTHERWISE SPECIFIED
1. ALL DIMENSIONS IN INCHES
2. TOLERANCES
X.XX = ±.01
X.XXX = ±.005
ANGLES = ±2°
3. BREAK ALL SHARP EDGES .03 MAX
4. 63√ FAO

Actual Precision

Mechanical Engineering

DATE: 3-6-12	SCALE 1:2	MATERIAL: Carbon Steel
TOLERANCE: ±0.01	NEXT ASSY: GHT600	DRAWING #: GHT601
UNITS: INCHES	TITLE: Housing End Plate	SIGNATURE:

4 3 2 1

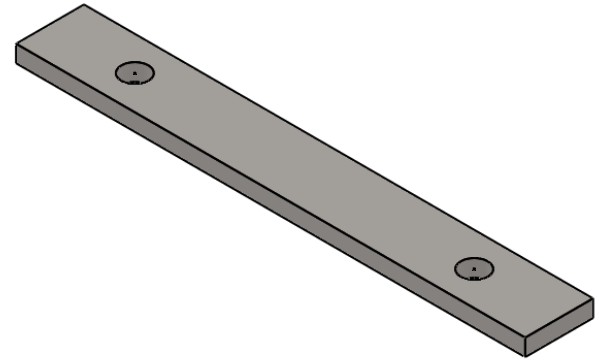


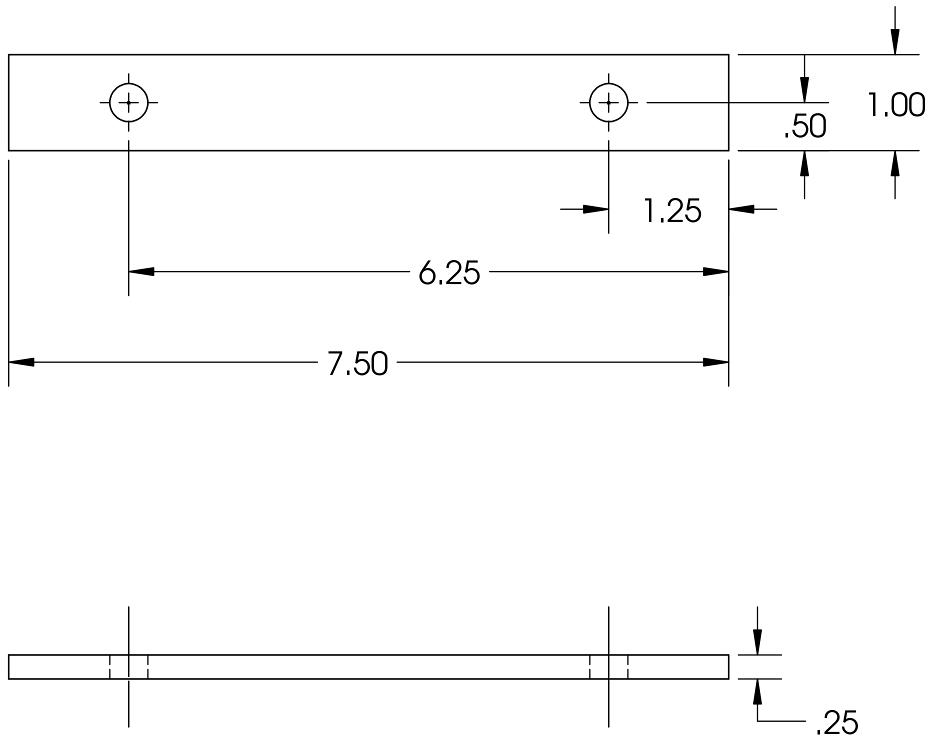
Actual Precision

Mechanical Engineering

DATE: 3-6-12	SCALE 1:4	MATERIAL: Carbon Steel
TOLERANCE: ±0.01	NEXT ASSY: GHT600	DRAWING #: GHT602
UNITS: INCHES	TITLE: Housing Side Plate	SIGNATURE:

4 3 2 1





NOTES

UNLESS OTHERWISE SPECIFIED

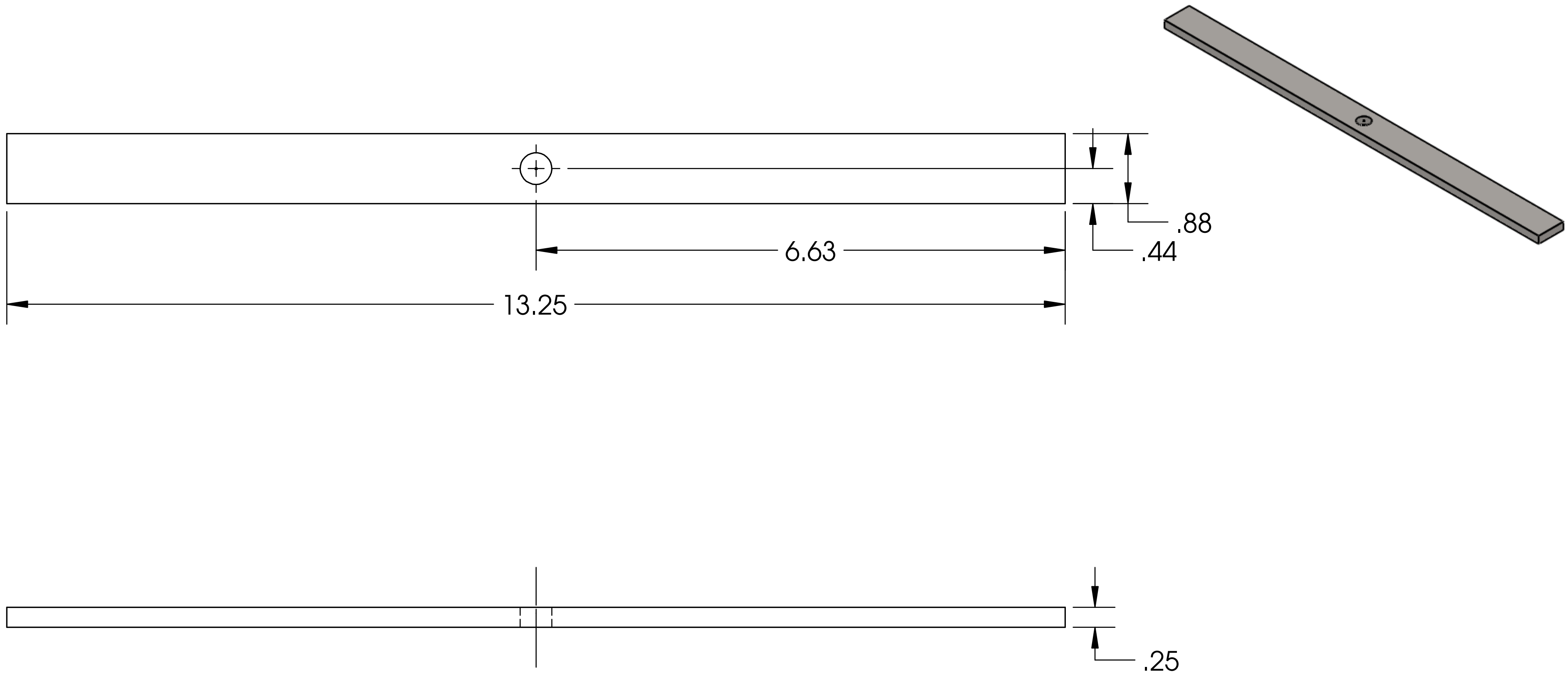
1. ALL DIMENSIONS IN INCHES
2. TOLERANCES
   X.XX = ±.01
   X.XXX = ±.005
   ANGLES = ±2°
3. BREAK ALL SHARP EDGES .03 MAX
4. $\sqrt[63]{}$ FAO

Actual Precision

Mechanical Engineering

DATE: 3-6-12	SCALE 1:2	MATERIAL: Carbon Steel
TOLERANCE: ±0.01	NEXT ASSY: GHT600	DRAWING #: GHT603
UNITS: INCHES	TITLE: Housing End Flange	SIGNATURE:

4 3 2 1



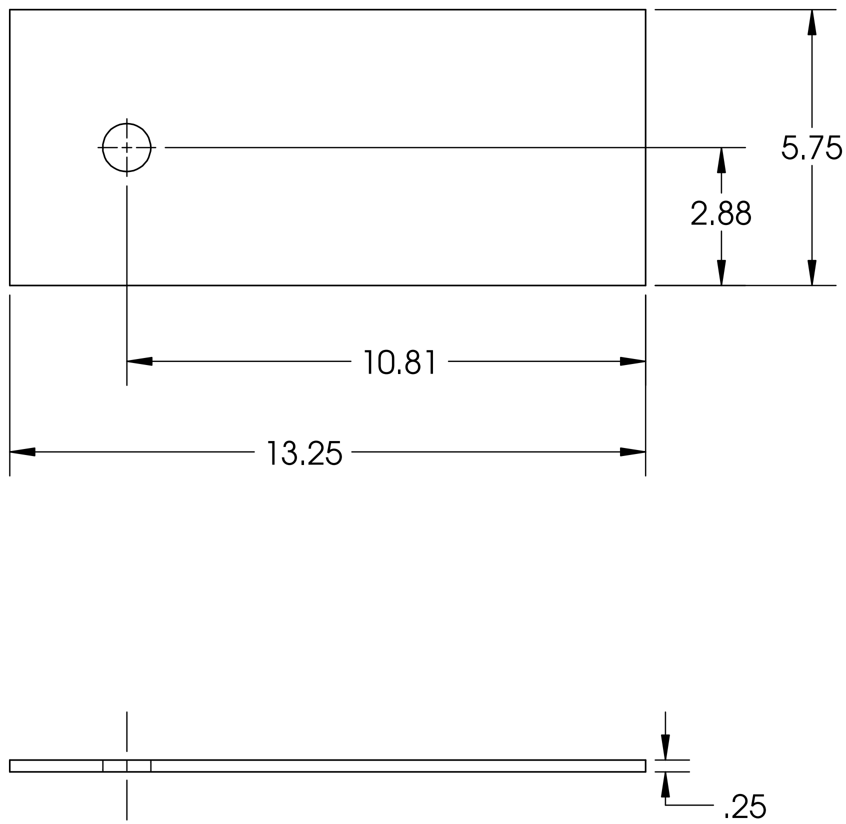
NOTES
UNLESS OTHERWISE SPECIFIED
1. ALL DIMENSIONS IN INCHES
2. TOLERANCES
X.XX = ±.01
X.XXX = ±.005
ANGLES = ±2°
3. BREAK ALL SHARP EDGES .03 MAX
4. 63 FAO

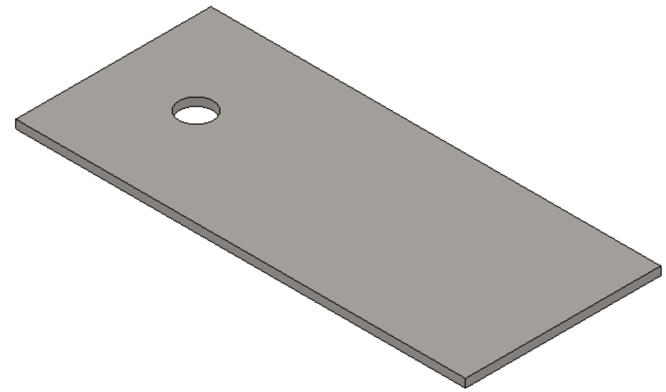
Actual Precision

Mechanical Engineering

DATE: 3-6-12	SCALE 1:2	MATERIAL: Carbon Steel
TOLERANCE: ±0.01	NEXT ASSY: GHT600	DRAWING #: GHT604
UNITS: INCHES	TITLE: Housing Side Flange	SIGNATURE:

4 3 2 1





NOTES
UNLESS OTHERWISE SPECIFIED
1. ALL DIMENSIONS IN INCHES
2. TOLERANCES
X.XX = ±.01
X.XXX = ±.005
ANGLES = ±2°
3. BREAK ALL SHARP EDGES .03 MAX
4. 63 FAO

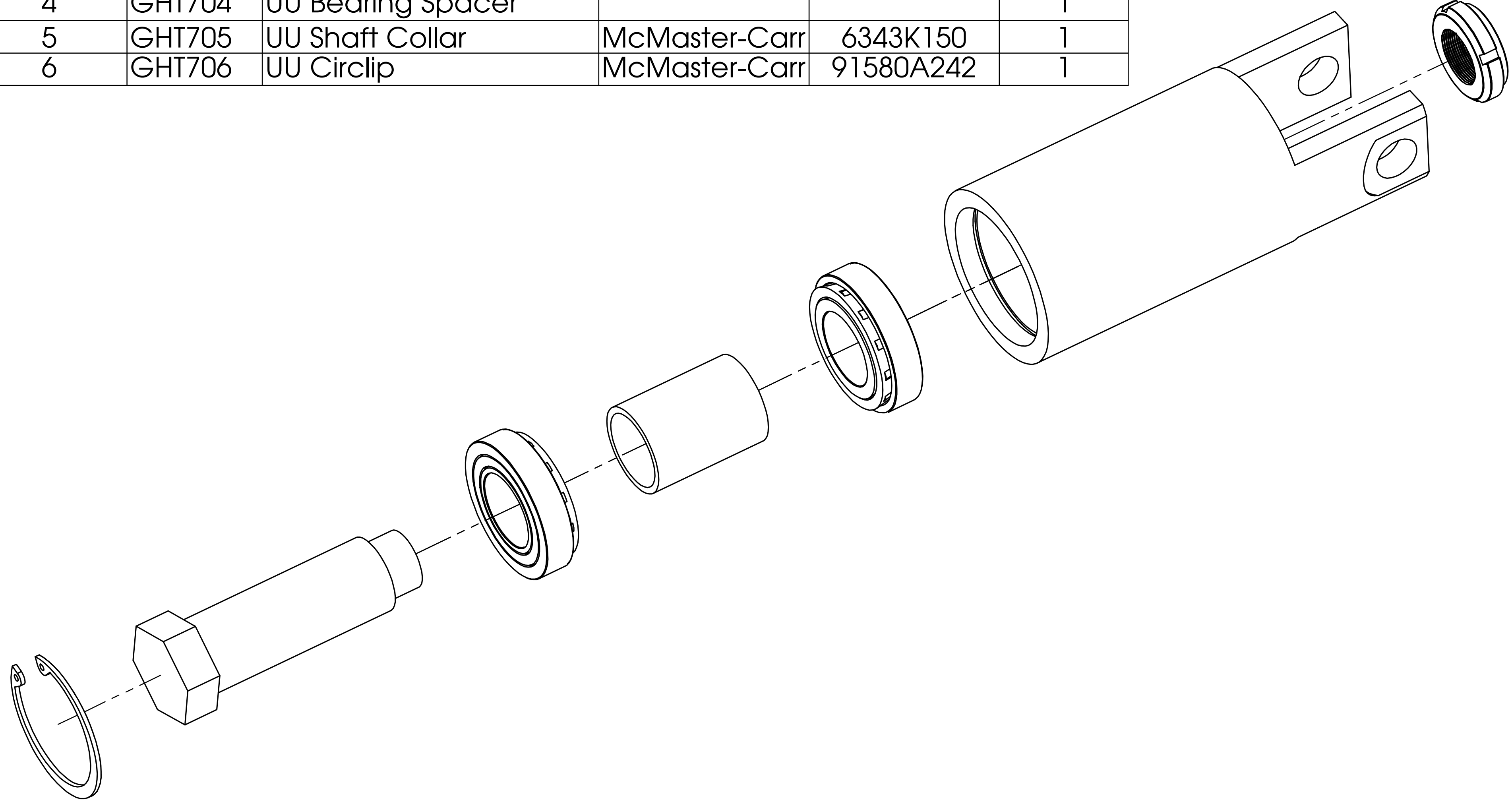
Actual Precision

Mechanical Engineering

	DATE: 3-6-12	SCALE 1:4	MATERIAL: Carbon Steel
	TOLERANCE: ±0.01	NEXT ASSY: GHT600	DRAWING #: GHT605
	UNITS: INCHES	TITLE: Housing Bottom Plate	SIGNATURE:

4 3 2 1

ITEM NO.	PART NUMBER	DESCRIPTION	Vendor	VendorNo	QTY.
1	GHT701	UU Yoke			1
2	GHT702	UU Shaft			1
3	GHT703	UU Thrust Bearing	McMaster-Carr	5709K210	2
4	GHT704	UU Bearing Spacer			1
5	GHT705	UU Shaft Collar	McMaster-Carr	6343K150	1
6	GHT706	UU Circlip	McMaster-Carr	91580A242	1

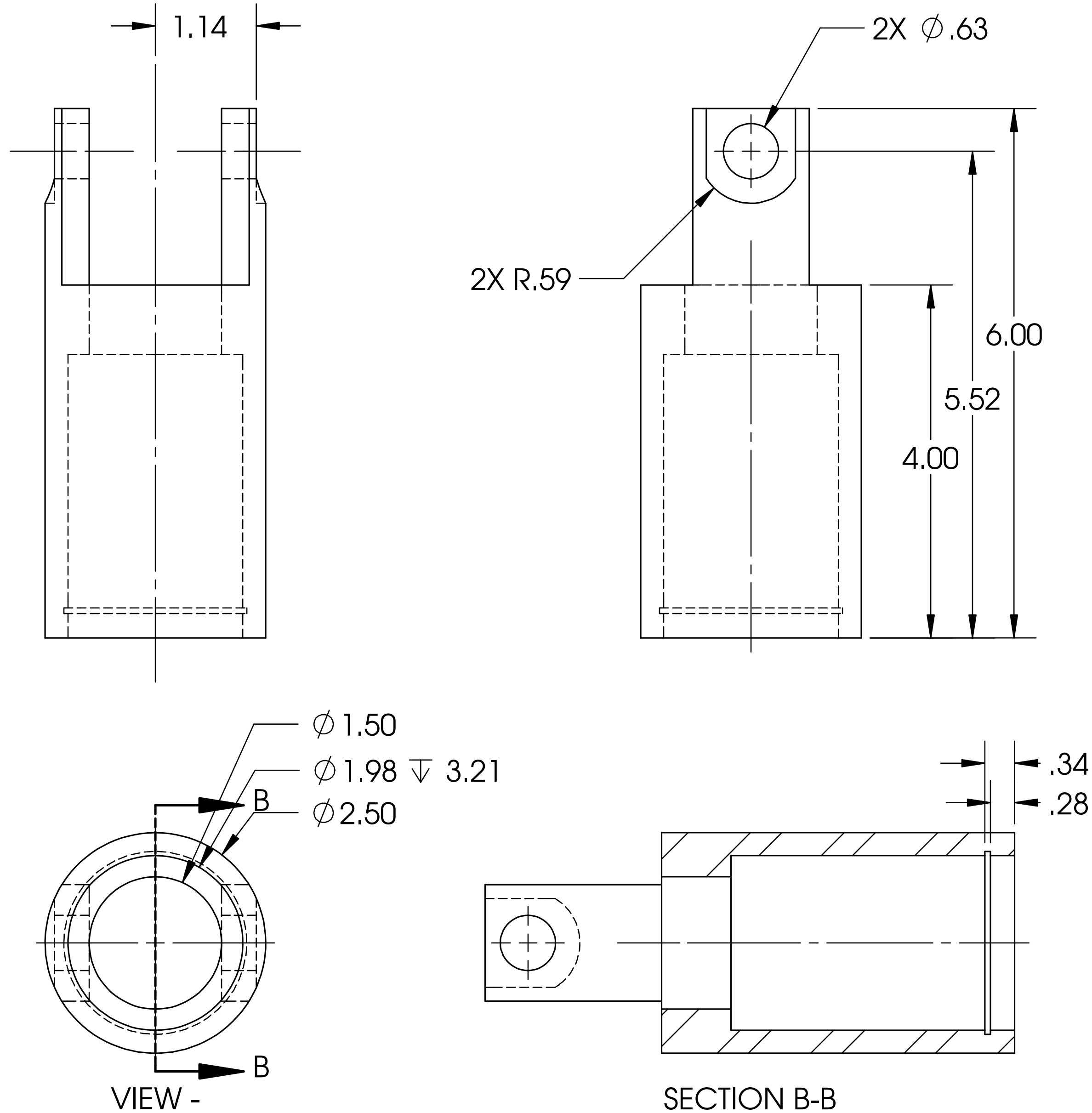


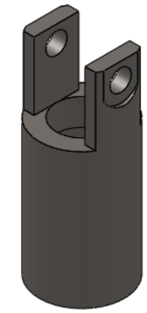
Actual Precision

Mechanical Engineering

DATE: 3-6-12	SCALE 1:2	MATERIAL: Carbon Steel
TOLERANCE: ±0.01	NEXT ASSY: GHT750	DRAWING #: GHT700
UNITS: INCHES	TITLE: UU Yoke with Bearings	SIGNATURE: Brett Hartt

4 3 2 1





NOTES
UNLESS OTHERWISE SPECIFIED
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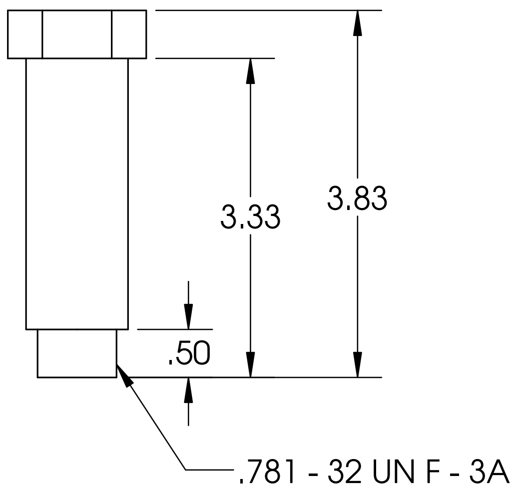
2. TOLERANCES
X.XX = ±.01
X.XXX = ±.005
ANGLES = ±2°
3. BREAK ALL SHARP EDGES .03 MAX
4. 63 √ FAO

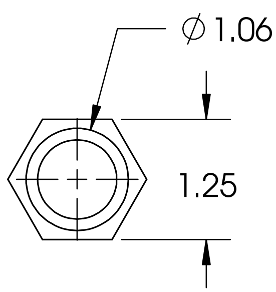
Actual Precision

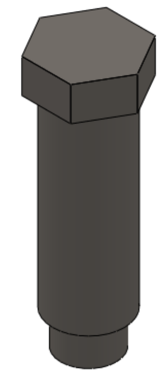
Mechanical Engineering

DATE: 3-6-12	SCALE 1:2	MATERIAL: Carbon Steel
TOLERANCE: ±0.01	NEXT ASSY: GHT700	DRAWING #: GHT701
UNITS: INCHES	TITLE: U-U Yoke	SIGNATURE:

4 3 2 1





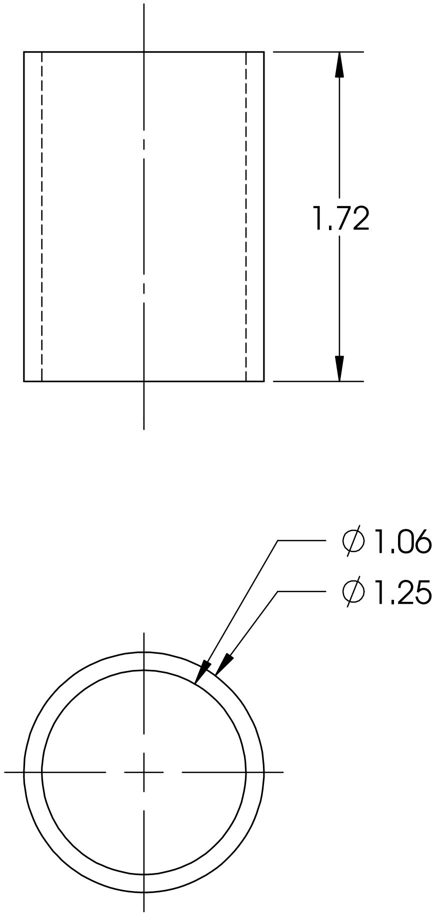


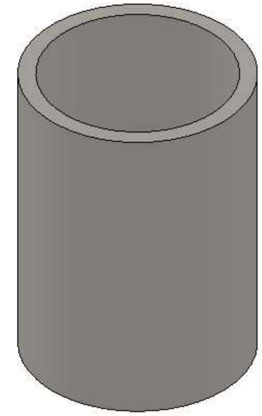
Actual Precision

Mechanical Engineering

DATE: 3-6-12	SCALE 1:2	MATERIAL: Carbon Steel
TOLERANCE: ±0.01	NEXT ASSY: GHT700	DRAWING #: GHT702
UNITS: INCHES	TITLE: Yoke Shaft	SIGNATURE:

4 3 2 1





NOTES
UNLESS OTHERWISE SPECIFIED
1. ALL DIMENSIONS IN INCHES
2. TOLERANCES
X.XX = ±.01
X.XXX = ±.005
ANGLES = ±2°
3. BREAK ALL SHARP EDGES .03 MAX
4. 63√ FAO

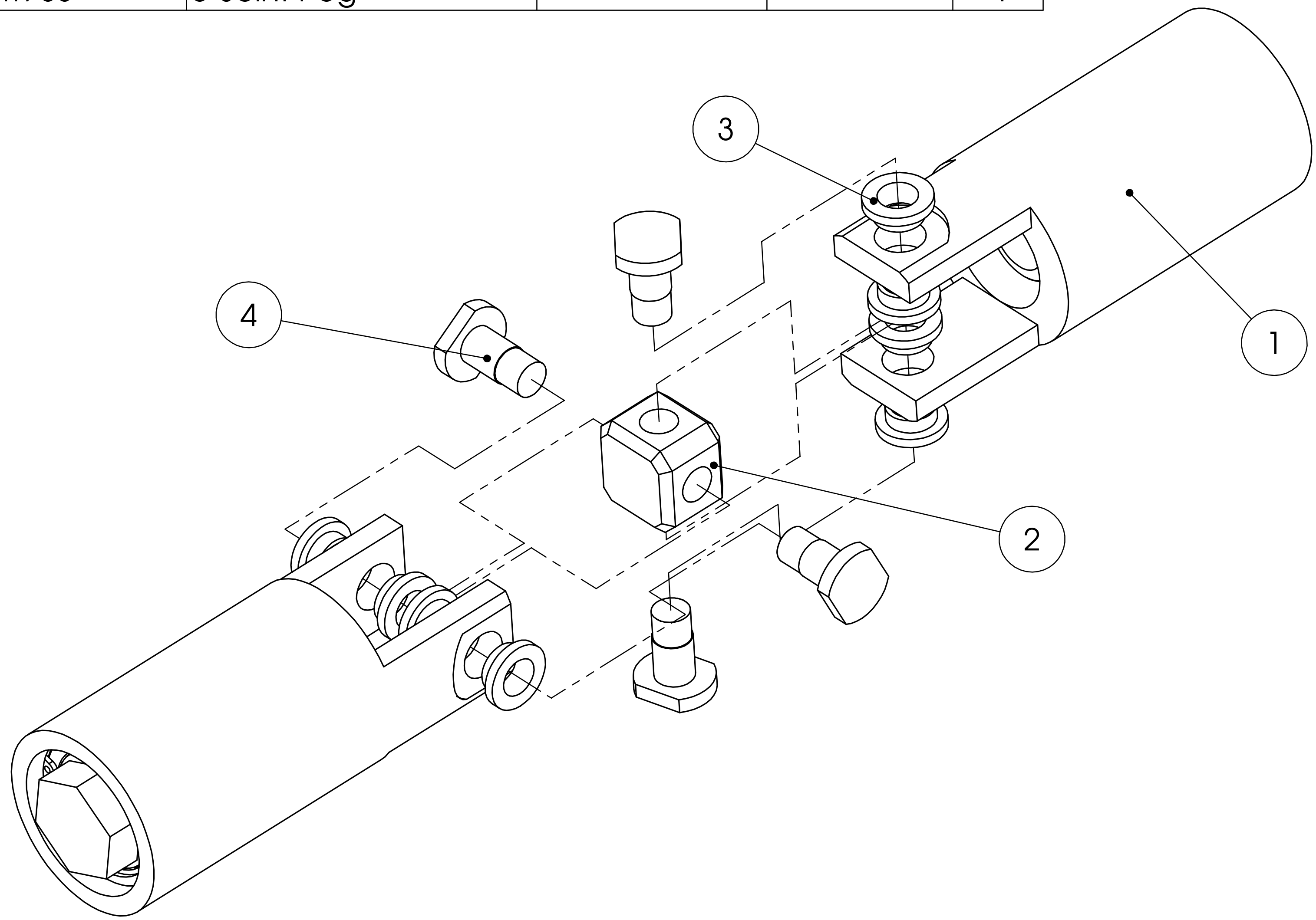
Actual Precision

Mechanical Engineering

DATE: 3-6-12	SCALE 1:1	MATERIAL: Carbon Steel
TOLERANCE: ±0.01	NEXT ASSY: GHT700	DRAWING #: GHT704
UNITS: INCHES	TITLE: Bearing Spacer	SIGNATURE:

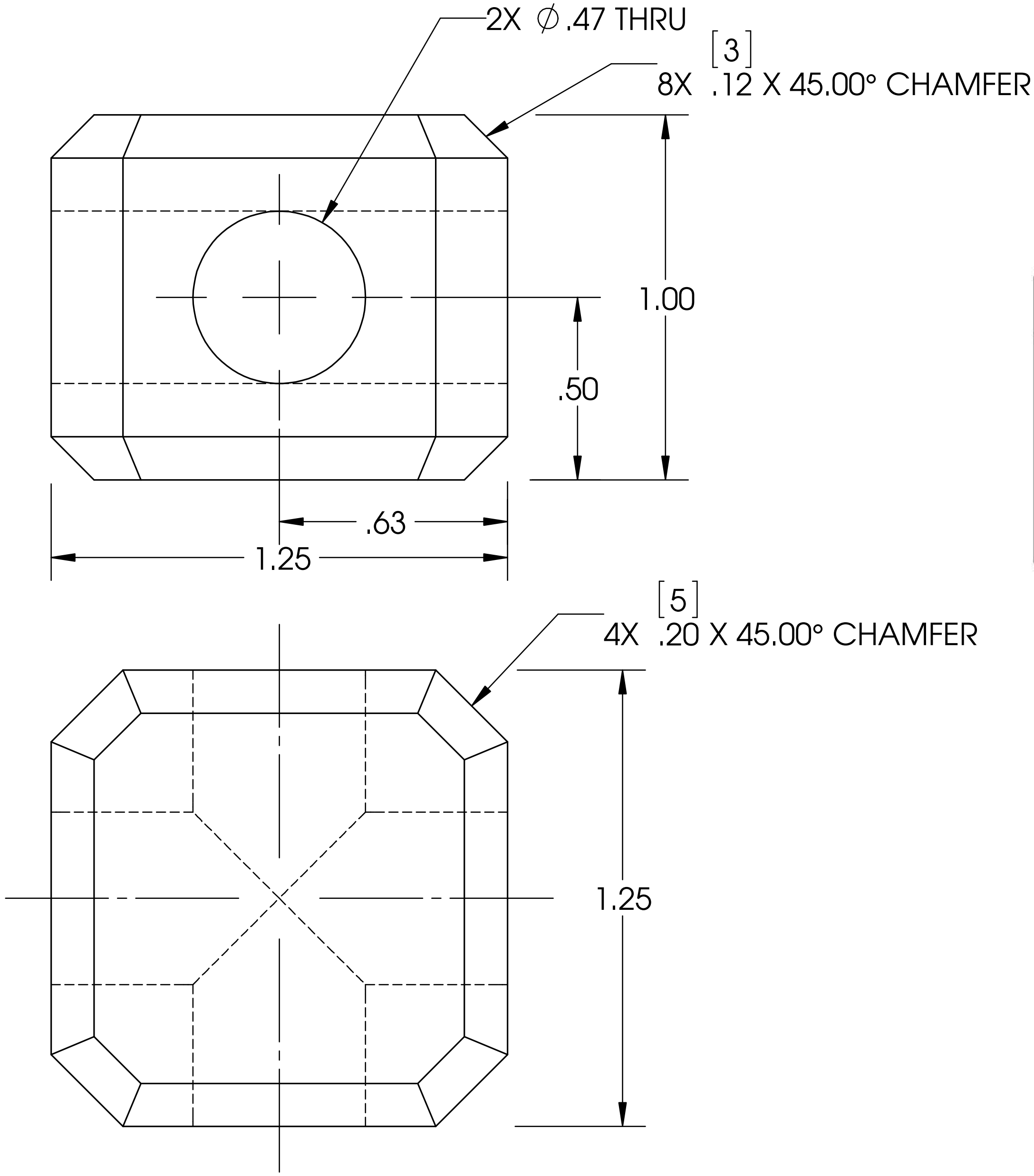
4 3 2 1

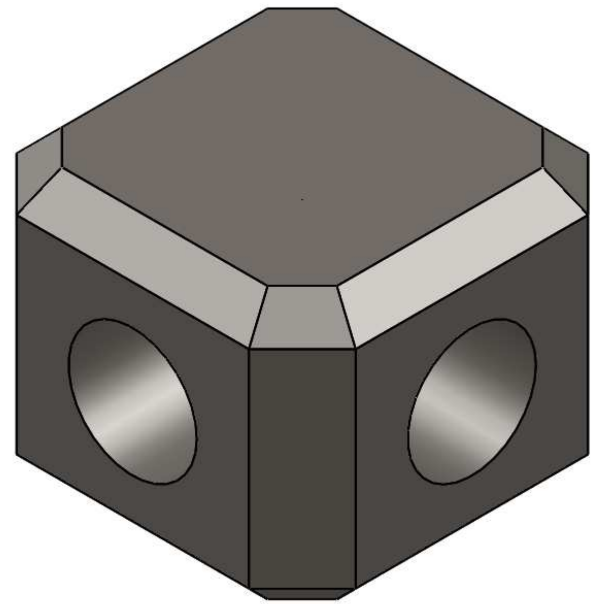
ITEM NO.	PART NUMBER	DESCRIPTION	Vendor	VendorNo	QTY.
1	GHT700	UU- Yoke with Bearings			2
2	GHT751	U-Joint Cross Block			1
3	GHT752	U-Joint Bushing	McMaster-Carr	2938T11	8
4	GHT753	U-Joint Peg			4



Actual Precision	DATE: 3-6-12	SCALE 1:2	MATERIAL: Carbon Steel
Mechanical Engineering	TOLERANCE: ±0.01	NEXT ASSY: GHT000	DRAWING #: GHT750
	UNITS: INCHES	TITLE: Universal Universal Joint	SIGNATURE: Brett Hartt

4 3 2 1





NOTES

UNLESS OTHERWISE SPECIFIED

1. ALL DIMENSIONS IN INCHES
2. (DIMENSIONS) ARE IN MM
3. TOLERANCES
   X.XX = ±.01
   X.XXX = ±.005
   ANGLES = ±2°
4. BREAK ALL SHARP EDGES .03 MAX
5. 63√ FAO

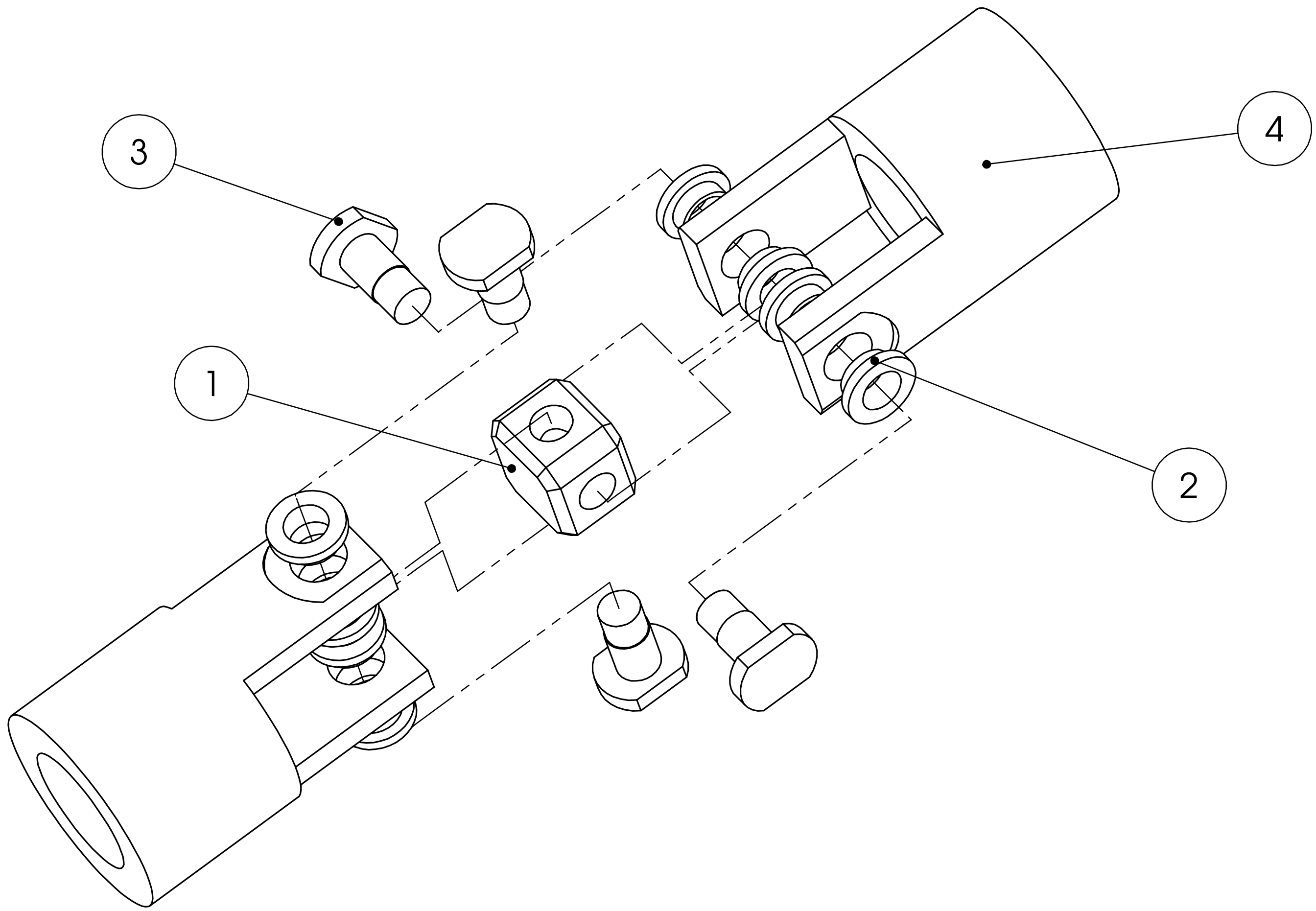
Actual Precision

Mechanical Engineering

DATE: 3-6-12	SCALE 2:1	MATERIAL: Carbon Steel
TOLERANCE: ±0.01	NEXT ASSY: GHT750	DRAWING #: GHT751
UNITS: INCHES	TITLE: Cross Block	SIGNATURE:

4 3 2 1

ITEM NO.	PART NUMBER	DESCRIPTION	Vendor	VendorNo	QTY.
1	GHT751	U-Joint Cross Block			1
2	GHT752	U-Joint Bushing	McMaster-Carr	2938T11	8
3	GHT753	U-Joint Peg			4
4	GHT801	U Yoke			2

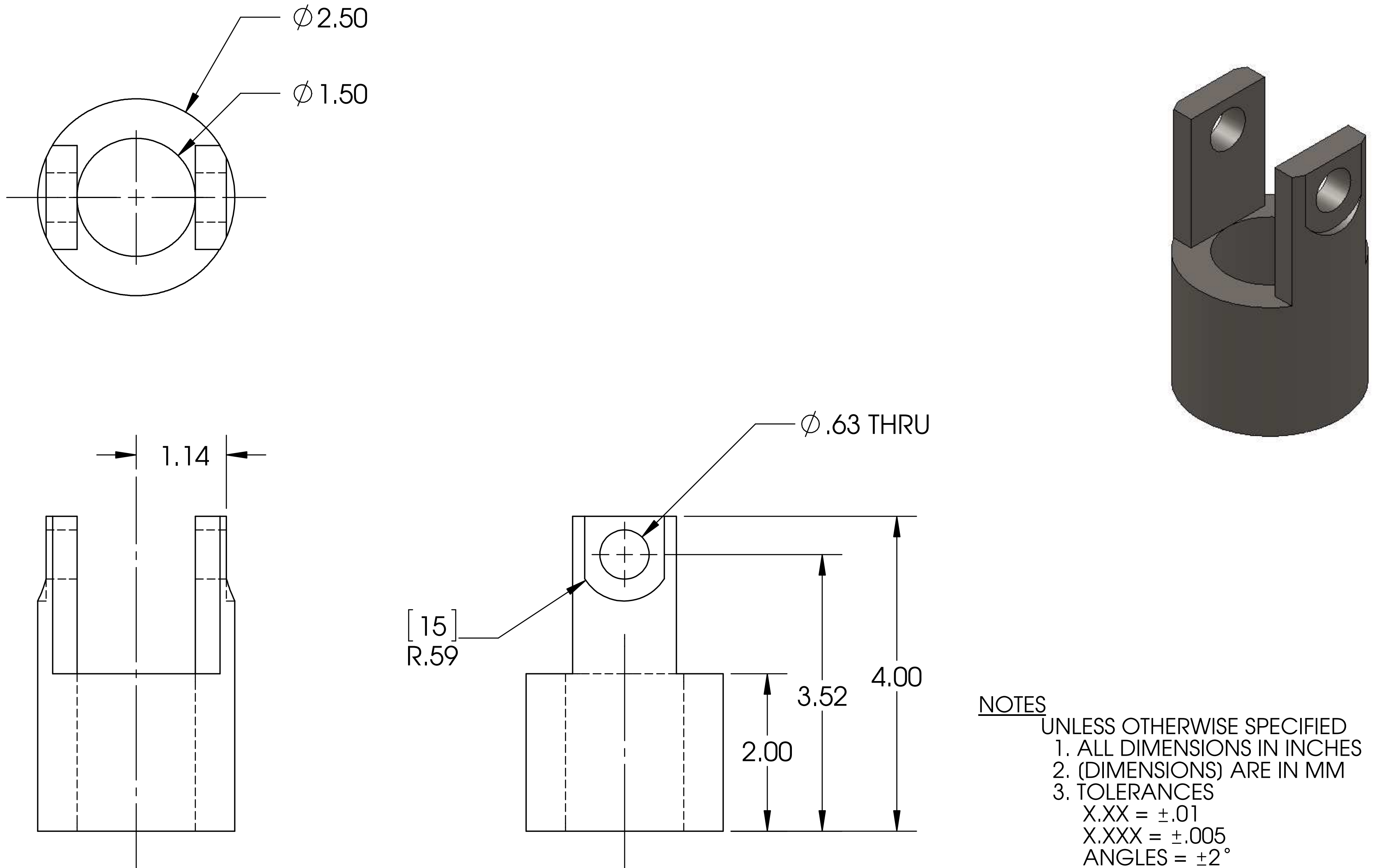


Actual Precision

Mechanical Engineering

DATE: 3-6-12	SCALE 1:2	MATERIAL: Carbon Steel
TOLERANCE: ±0.01	NEXT ASSY: GHT000	DRAWING #: GHT800
UNITS: INCHES	TITLE: Universal Joint	SIGNATURE:

4 3 2 1



NOTES
UNLESS OTHERWISE SPECIFIED
1. ALL DIMENSIONS IN INCHES
2. (DIMENSIONS) ARE IN MM
3. TOLERANCES
X.XX = ±.01
X.XXX = ±.005
ANGLES = ±2°
4. BREAK ALL SHARP EDGES .03 MAX
5. 63 √ FAO

Actual Precision Mechanical Engineering	DATE: 3-6-12	SCALE 1:2	MATERIAL: Carbon Steel
	TOLERANCE: ±0.01	NEXT ASSY: GHT800	DRAWING #: GHT801
	UNITS: INCHES (mm)	TITLE: U Joint	SIGNATURE:

4 3 2 1

Appendix C – List of Vendors, Contact Information, and Pricing

Vendor	Item	Part Number	Price ($)	Phone Number	Website
McMaster-Carr	Ball Screw	5966K8	232.23	(805) 928-4044	Applied.com
	Ball Nut	5966K47	101.18		
	Pillow Mount Block	60755K15	451.04		
	U-Joint Bushing	2938T11	0.86	(330) 342-6100	McMasterCarr.com
	UU-Joint Thrust Bearing	5709K15	12.89		
	UU -Joint Shaft Collar	6343K15	14.64		
	UU-Joint Circlip	91580A242	5.01		
US Digital	Angular Encoder	S5-1000-236-NE-D-B	91.73	(360) 397-9999	usdigital.com
Anaheim Automation	Electric Motor	BDPG-60-110-24V-3000-R168	194	(714)-992-0471	anaheimautomation.com
SDP	Belt Pulley	A 6A14M32D20	51.3	(800) 819-8900 Ext. 491	sdp-si.com
	Timing Belt	A 6Z13MD0700	93.62		
VXB Ball Bearings	Linear Guide way System	KIT8282	69	(800) 928 - 4430	vxb.com
		KIT8512	9.95		

Appendix D – Vendor supplied Component Specifications and Data Sheets

Electric Motor

BDPG-60-110 Series Planetary Gearmotor

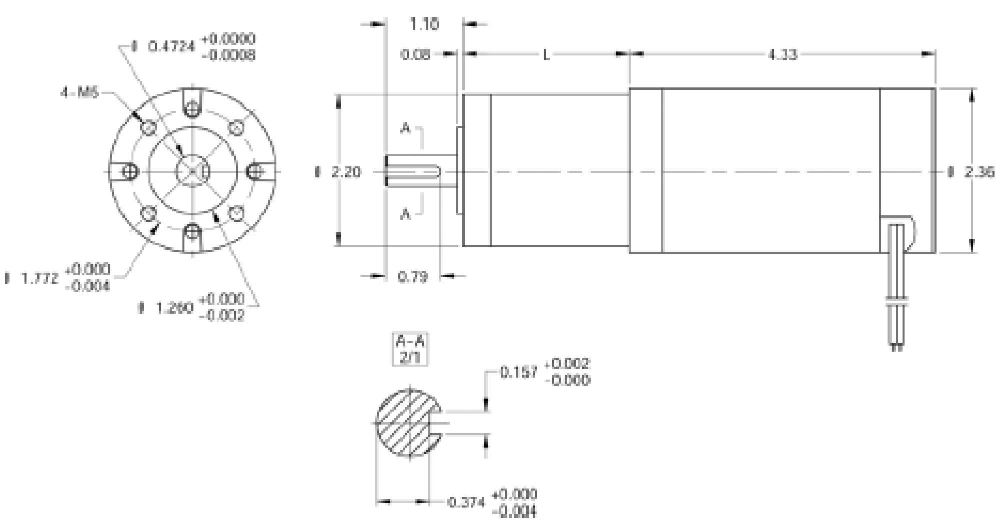
- DC Brush Planetary GearMotor
- Value Line - designed for high quality and a low price
- Perfect for OEM applications
- Up to 4166 oz-in (300 kg-cm) of Continuous Torque
- 60 mm motor body diameter
- Custom versions are available for orders over 100 pieces

BDPG-60-110-24V-2000-Rxx Specifications

Gear Ratio Rxx	3.6	4.25	13	15	18	47	55	65	76	168	198	234	276	326
Length (L) (in)	1.79	1.79	1.79	1.79	1.79	2.22	2.22	2.22	2.22	3.01	3.01	3.01	3.01	3.01
No Load Speed (RPM)	555	470	154	133	111	42	36	30	26	12	10	8.5	7.2	6.1
Rated Speed (RPM)	460	365	119	103	86	33	28	24	20	9.2	7.8	6.6	5.6	4.7
Rated Torque (kg-cm)	4.3	5.0	14	17	20	50	58	73	81	170	200	237	279	300
Peak Torque (kg-cm)	30	30	100	100	100	300	300	300	300	600	600	600	600	600

BDPG-60-110-24V-3000-Rxx Specifications

Gear Ratio Rxx	3.6	4.25	13	15	18	47	55	65	76	168	198	234	276	326
Length (L) (in)	1.79	1.79	1.79	1.79	1.79	2.22	2.22	2.22	2.22	3.01	3.01	3.01	3.01	3.01
No Load Speed (RPM)	833	706	230	200	167	64	54	46	39	18	15	13	11	9.2
Rated Speed (RPM)	694	588	192	167	139	53	45	38	33	15	12	10	9.0	7.7
Rated Torque (kg-cm)	4.1	4.8	14	16	19	48	56	66	77	163	192	227	268	300
Peak Torque (kg-cm)	30	30	100	100	100	300	300	300	300	600	600	600	600	600



910 E. Orangefair Ln. Anaheim, CA 92801 Tel (714) 992-6990 Fax (714) 992-0471
www.anaheimautomation.com

Optical Shaft Encoder

RoHS COMPLIANT

Description

The **S5** series optical shaft encoder is a non-contacting rotary to digital converter. Useful for position feedback or manual interface, the encoder converts real-time shaft angle, speed, and direction into TTL-compatible quadrature outputs with or without index. The encoder utilizes an unbreakable mylar disk, metal shaft and bushing, LED light source, and monolithic electronics. It operates from a single +5VDC supply.

Three shaft torque versions are available. The standard torque version has a sleeve bushing lubricated with a viscous motion control gel to provide torque and feel that is ideal for front panel human interface applications.

The no torque added option has a sleeve bushing and a low viscosity lubricant (that does not intentionally add torque) for low RPM applications where a small amount of torque is acceptable.

The ball bearing version uses miniature precision ball bearings that are suitable for high speed and ultra low torque applications.

A secure connection to the **S5** series encoder is made through a 5-pin (single-ended version) or 10-pin (differential version) finger-latching connector (sold separately). The mating connectors are available from US Digital with several cable options and lengths.

For differential version: the internal differential line driver (26C31) can source and sink 20mA at TTL levels. The recommended receiver is industry standard 26C32. Maximum noise immunity is achieved when the differential receiver is terminated with a 110 Ω resistor in series with a .0047 μF capacitor placed across each differential pair. The capacitor simply conserves power; otherwise power consumption would increase by approximately 20mA per pair, or 60mA for 3 pairs.

Features

- Small size
- Low cost
- Optional differential / line-driver output
- Positive finger-latching connector
- 2-channel quadrature, TTL squarewave outputs
- 3rd channel index option
- Ball bearing option tracks to 10,000 RPM
- -40C to +100C operating temperature
- Single +5VDC supply

Product Specifications

- Mechanical
- Phase Relationship
- Single-ended Electrical
- Differential Electrical
- Pin-outs
- Product Change Notifications

Parameter	Sleeve Bushing	Ball Bearing
Acceleration	250,000 rad/sec²	250,000 rad/sec²
Vibration	5 Hz to 2 kHz, 20 G	5 Hz to 2 kHz, 20 G
Shaft Speed	100 RPM max. continuous	10,000 RPM max. continuous
Shaft Rotation	Continuous and reversible	Continuous and reversible
Shaft Torque	0.5 ± 0.2 in. oz. 0.3 in. oz. max. (**N**-option)	0.05 in. oz. max.
Shaft Loading	2 lbs. max. dynamic 20 lbs. max. static	1 lb. max.
Bearing Life	> 1,000,000 revolutions	$L_{10} = (19.3/F_r)^3$ * Where L_{10} = bearing life in millions of revs, and F_r = radial shaft loading in pounds
Weight Single-ended Differential	1.01 oz. 1.28 oz.	1.15 oz. 1.42 oz.
Shaft Runout	0.0015 T.I.R. max.	0.0015 T.I.R. max.

* only valid with negligible axial shaft loading.

Product Specifications

- Mechanical
- Phase Relationship
- Single-ended Electrical
- Differential Electrical
- Pin-outs
- Product Change Notifications

Single-ended Electrical

B leads A for clockwise shaft rotation, and A leads B for counterclockwise rotation viewed from the shaft side of the encoder (*see the EM1 page*).

Product Specifications

- Mechanical
- Phase Relationship
- Single-ended Electrical
- Differential Electrical
- Pin-outs
- Product Change Notifications

- Specifications apply over entire operating temperature range.
- Typical values are specified at Vcc = 5.0Vdc and 25°C.
- For complete details see the EM1 and HEDS product pages.

	Supply Current	Output voltage low	Output voltage high	
Resolution	Typ / Max	Max	Min	Based on
50,96, 100, 110, 120, 192, 200, 250, 256, 360, 400, 500, 512, 540 CPR, non-index	17 / 40 mA	0.4 volts @ 3.2mA	2.4 volts @ -200uA	Low-res HEDS
1000, 1016, 1024 CPR, non-index	57 / 85 mA	0.5 volts @ 8mA	2.4 volts @ -40uA	High-res HEDS
32 CPR, with index	27 / 30 mA	0.5 volts @ 8mA	2.0 volts @ -8mA	EM1
50,96, 100, 192, 200, 250, 256, 360, 400, 500, 512 CPR, with index	57 / 85 mA	0.5 volts @ 8mA	2.4 volts @ -40uA	High-res HEDS
720, 900, 1000, 1024, 1250 CPR, with index	55 / 57 mA	0.5 volts @ 8mA	2.0 volts @ -8mA	EM1

Product Specifications

- Mechanical
- Phase Relationship
- Single-ended Electrical
- Differential Electrical
- Pin-outs
- Product Change Notifications

Parameter	Min.	Typ.	Max.	Units	Notes
Supply	4.5	5.0	5.5	Volts	
Current Consumption - Index: 32 CPR	-	28	53	mA	No load
Current Consumption - Index: 720, 900, 1000, 1250 CPR	-	56	59	mA	No load
Current Consumption - Index: All Other Resolutions	-	58	88	mA	No load
Current Consumption - Non-index: <2000 CPR	-	18	43	mA	No load
Current Consumption - Non-index: >=2000	-	58	88	mA	No load
Output Voltage - Sourcing to +5	2.4	3.4	-	Volts	@ -20mA
Output Voltage - Sinking to Ground	-	0.2	0.4	Volts	@ 20mA

For complete details see the EM1 / HEDS page.

Product Specifications

- Mechanical
- Phase Relationship
- Single-ended Electrical
- Differential Electrical
- Pin-outs
- Product Change Notifications

5-pin Single-ended:

Pin	Description
1	Ground
2	Index
3	A channel
4	+5VDC power
5	B channel

10-pin Differential Standard:

Pin	Description
1	Ground
2	Ground
3	Index-
4	Index+
5	A- channel
6	A+ channel
7	+5VDC power
8	+5VDC power
9	B- channel
10	B+ channel

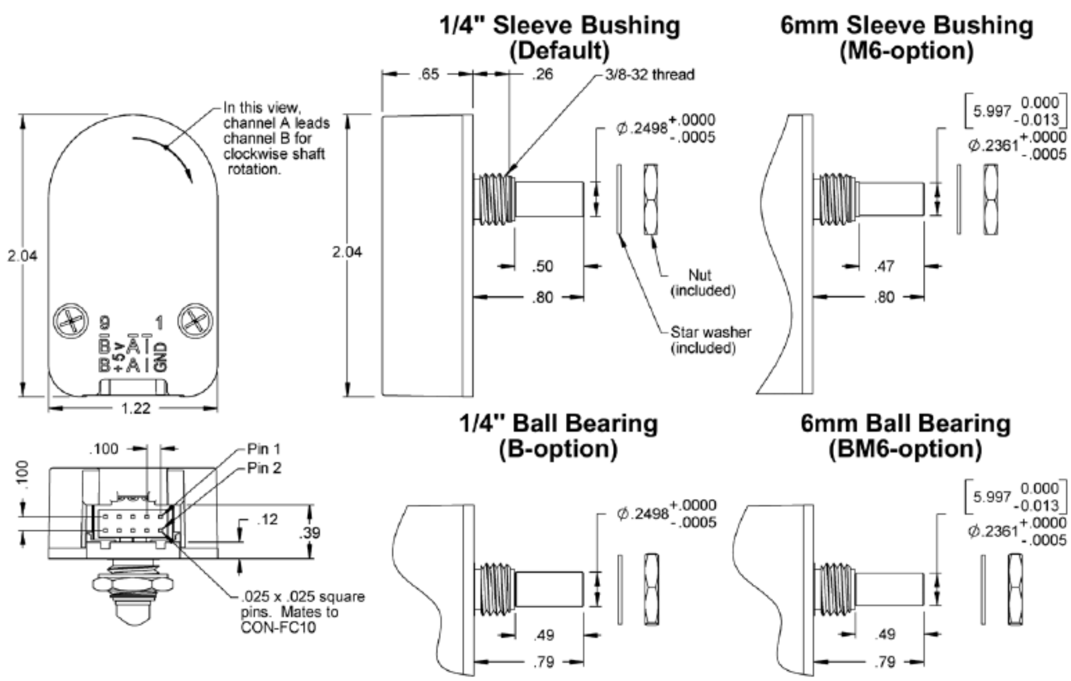
Attention: For wire descriptions see the 5-pin Finger-latching or 10-pin Finger-latching Cables / Connectors page.

Product Specifications

- Mechanical
- Phase Relationship
- Single-ended Electrical
- Differential Electrical
- Pin-outs
- Product Change Notifications

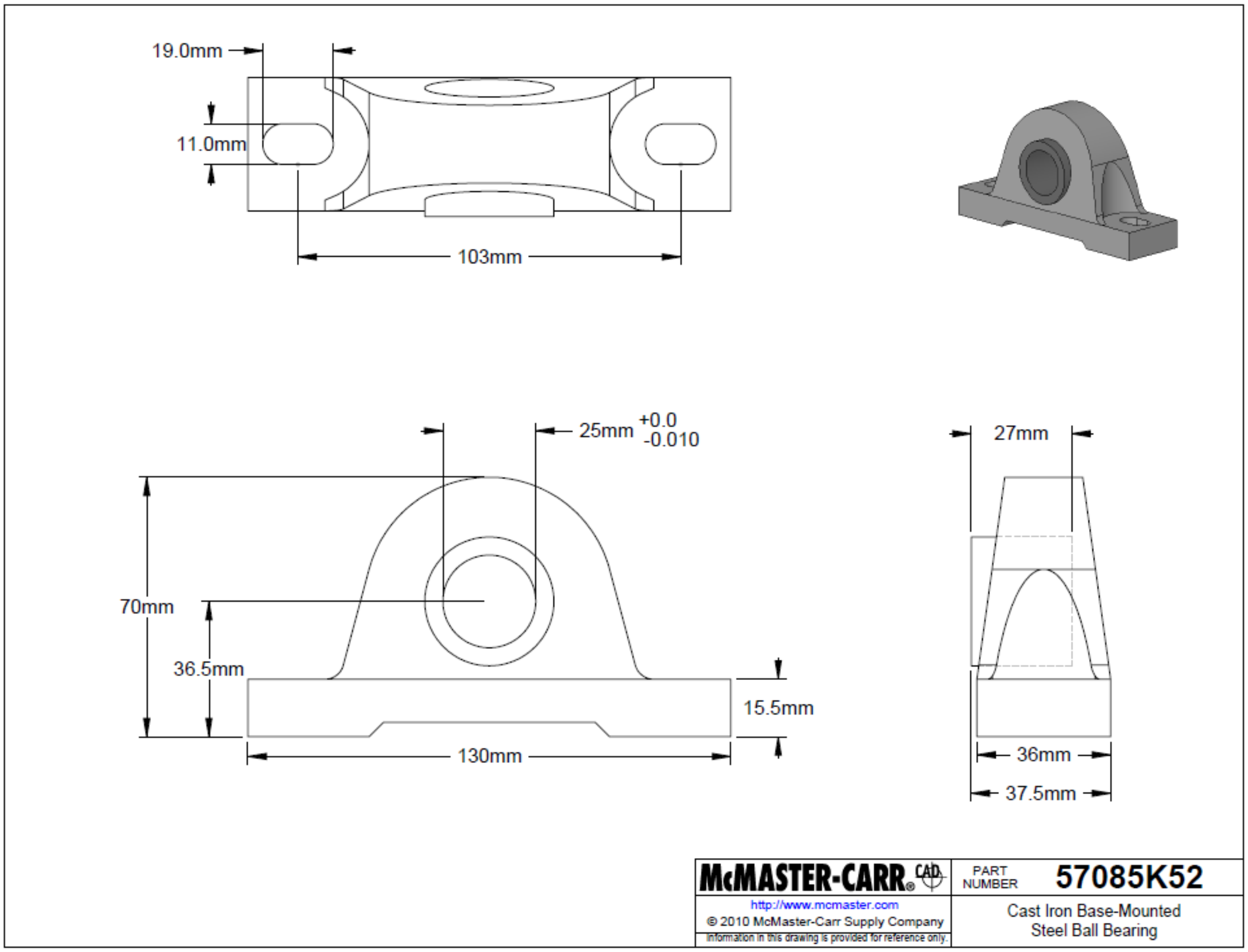
Title	Date	Description	Download
E5 Insert Overmold - PCN 1008	8/23/2011	In an effort to enhance the robustness of our E5 encoder; the four threaded inserts pressed into the base are being replaced with similar threaded nuts that will be insert-molded into the encoder base. This change in process will retain the insert with much greater strength.	Download
E5 Laser Marking - PCN 1009	8/23/2011	The primary purpose for this change is to create a more durable and longer lasting solution compared to the previous stick on label solution. The E5 encoder covers will now have the US Digital logo, part number, lot code, and pin-outs laser marked onto the top surface.	Download
E5 Mold Update - PCN 1007	8/23/2011	The plastic E5 base and covers have been redesigned for improved moldability and aesthetics. Design changes are primarily alteration of surface drafts, additional or increased corner radii and additional coring out of thick regions. This update was carefully done to preserve the size and shape of the encoder. The new parts are dimensionally equivalent and will fit within the envelope of the previous parts. Only the E-option covers and the G-option bases have features with dimensional changes.	Download

S5 Optical Shaft Encoder Drawing



US DIGITAL
1400 NE 136th Avenue
Vancouver, Washington 98684, USA
info@usdigital.com
usdigital.com
Local: 360.260.2468
Toll-free: 800.736.0194

Base Mount Bearing

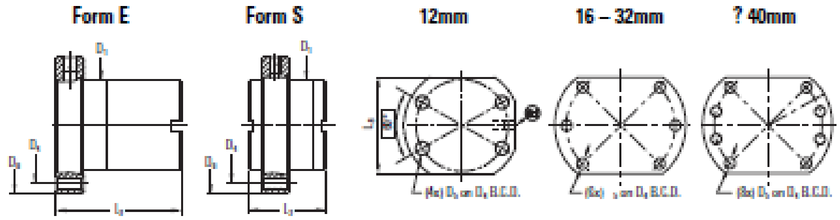


Thomson Ball Screw

Metric Ball Screws Product Overview

Flanged Ball Nuts

Fineline: FK, FH
NEFF: KGF-D
Return: Internal
Style: Flanged
Mounting: Flanged
Backlash: Z1, Z2 and Z3
Thread Direction: Right Hand



Note: KGF-D 2525 and 4040 models have round flanges.

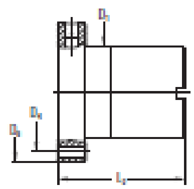
Nominal Diameter	Lead	Dynamic Load Capacity (C_{am})	Length L_2	Width/ Diameter D_1	Ball Nut Form	Flange Diameter D_6	Bolt Hole Circle D_4	Hole Diameter D_5	Ball Nut	Nut P/N	Screw P/N	Catalog Page
(mm)	(mm)	(kN)	(mm)	(mm)		(mm)	(mm)	(mm)				
12	5	5.6	46.5	24.0	Form S	40.0	32.0	4.6	FK	**7832773**	**7832772**	118
16	5	9.5	48.5	28.0	Form S	48.0	38.0	5.5	FK	**7832777**	**7832776**	118
16	5	9.3	42.0	28.0	Form E	48.0	38.0	5.5	KGF-D	**KGF-D-1605-RH-EE**	**KGS-1605-050-RH**	122
16	10	15.4	55.0	28.0	Form E	48.0	38.0	5.5	KGF-D	**KGF-D-1610-RH-EE**	**KGS-1610-050-RH**	122
20	5	11.5	48.5	36.0	Form S	58.0	47.0	6.6	FK	**7832780**	**7832779**	118
20	5	10.5	42.0	36.0	Form E	58.0	47.0	6.6	KGF-D	**KGF-D-2005-RH-EE**	**KGS-2005-050-RH**	122
20	20	10.8	36.0	36.0	Form S	58.0	47.0	6.6	FH	**7832784**	**7832783**	118
25	5	13.1	49.0	40.0	Form S	62.0	51.0	6.6	FK	**7832787**	**7832786**	118
25	5	12.3	42.0	40.0	Form E	62.0	51.0	6.6	KGF-D	**KGF-D-2505-RH-EE**	**KGS-2505-050-RH**	122
25	10	22.9	51.0	40.0	Form S	62.0	51.0	6.6	FH	**7832791**	**7832790**	118
25	10	13.2	55.0	40.0	Form E	62.0	51.0	6.6	KGF-D	**KGF-D-2510-RH-EE**	**KGS-2510-050-RH**	122
25	20	13.0	35.0	40.0	Form S	62.0	51.0	6.6	KGF-D	**KGF-D-2520-RH-EE**	**KGS-2520-050-RH**	122
25	25	13.1	39.0	40.0	Form S	62.0	51.0	6.6	FH	**7832794**	**7832793**	118
25	25	16.7	35.0	40.0	Form S	62.0	51.0	6.6	KGF-D	**KGF-D-2525-RH-EE**	**KGS-2525-050-RH**	122
25	50	15.4	58.0	40.0	Form S	62.0	51.0	6.6	KGF-D	**KGF-D-2550-RH-EE**	**KGS-2550-050-RH**	122
32	5	19.3	57.0	50.0	Form S	80.0	65.0	9.0	FK	**7832796**	**7832795**	118
32	5	21.5	55.0	50.0	Form E	80.0	65.0	9.0	KGF-D	**KGF-D-3205-RH-EE**	**KGS-3205-050-RH**	122
32	10	26.4	73.0	50.0	Form S	80.0	65.0	9.0	FK	**7832799**	**7832798**	118
32	10	33.4	69.0	53.0	Form E	80.0	65.0	9.0	KGF-D	**KGF-D-3210-RH-EE**	**KGS-3210-050-RH**	122
32	20	47.2	83.0	56.0	Form S	86.0	71.0	9.0	FH	**7832803**	**7832802**	118
32	20	29.7	80.0	53.0	Form E	80.0	65.0	9.0	KGF-D	**KGF-D-3220-RH-EE**	**KGS-3220-050-RH**	122
32	32	19.7	42.0	56.0	Form S	86.0	71.0	9.0	FH	**7833300**	**7833301**	118
32	32	18.0	42.0	50.0	Form S	80.0	65.0	9.0	KGF-D	**KGF-D-3232-RH-EE**	**KGS-3232-050-RH**	122
40	5	26.3	66.0	63.0	Form S	93.0	78.0	9.0	FK	**7832805**	**7832804**	118
40	5	23.8	57.0	63.0	Form E	93.0	78.0	9.0	KGF-D	**KGF-D-4005-RH-EE**	**KGS-4005-050-RH**	122
40	10	64.9	88.5	63.0	Form S	93.0	78.0	9.0	FK	**7832809**	**7832808**	118
40	10	38.0	71.0	63.0	Form E	93.0	78.0	9.0	KGF-D	**KGF-D-4010-RH-EE**	**KGS-4010-050-RH**	122
40	20	52.2	83.0	63.0	Form S	93.0	78.0	9.0	FH	**7832812**	**7832811**	118
40	20	33.3	80.0	63.0	Form E	93.0	78.0	9.0	KGF-D	**KGF-D-4020-RH-EE**	**KGS-4020-050-RH**	122
40	40	59.7	104.0	70.0	Form S	100.0	85.0	9.0	FH	**7832815**	**7832814**	118
40	40	35.0	85.0	63.0	Form S	93.0	78.0	9.0	KGF-D	**KGF-D-4040-RH-EE**	**KGS-4040-050-RH**	122
50	10	66.4	92.0	75.0	Form S	110.0	93.0	11.0	FK	**7832818**	**7832817**	118
50	10	68.7	95.0	75.0	Form E	110.0	93.0	11.0	KGF-D	**KGF-D-5010-RH-EE**	**KGS-5010-050-RH**	122
50	20	78.8	85.0	75.0	Form S	110.0	93.0	11.0	FH	**7832821**	**7832820**	118
50	20	60.0	95.0	85.0	Form E	125.0	103.0	11.0	KGF-D	**KGF-D-5020-RH-EE**	**KGS-5020-050-RH**	122
63	10	93.8	103.5	90.0	Form S	125.0	108.0	11.0	FK	**7832823**	**7832822**	118
63	20	103.1	86.0	95.0	Form S	135.0	115.0	13.5	FK	**7832826**	**7832825**	118
80	10	121.9	121.0	105.0	Form S	145.0	125.0	13.5	FK	**7832828**	**7832827**	118
80	20	176.4	160.5	125.0	Form S	165.0	145.0	13.5	FK	**7832831**	**7832830**	118

Metric Ball Screws Product Overview

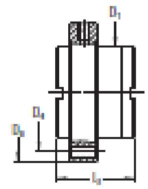
Flanged Ball Nuts

Fineline FS
NEFF KGF-N
Return: Internal
Style: Flanged
Mounting: Flanged
Backlash: Z1, Z2 and Z3
Thread Direction: Right Hand

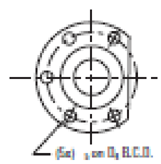
Form E



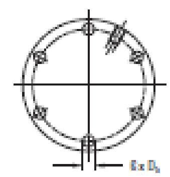
Form S



FS



KGF-N

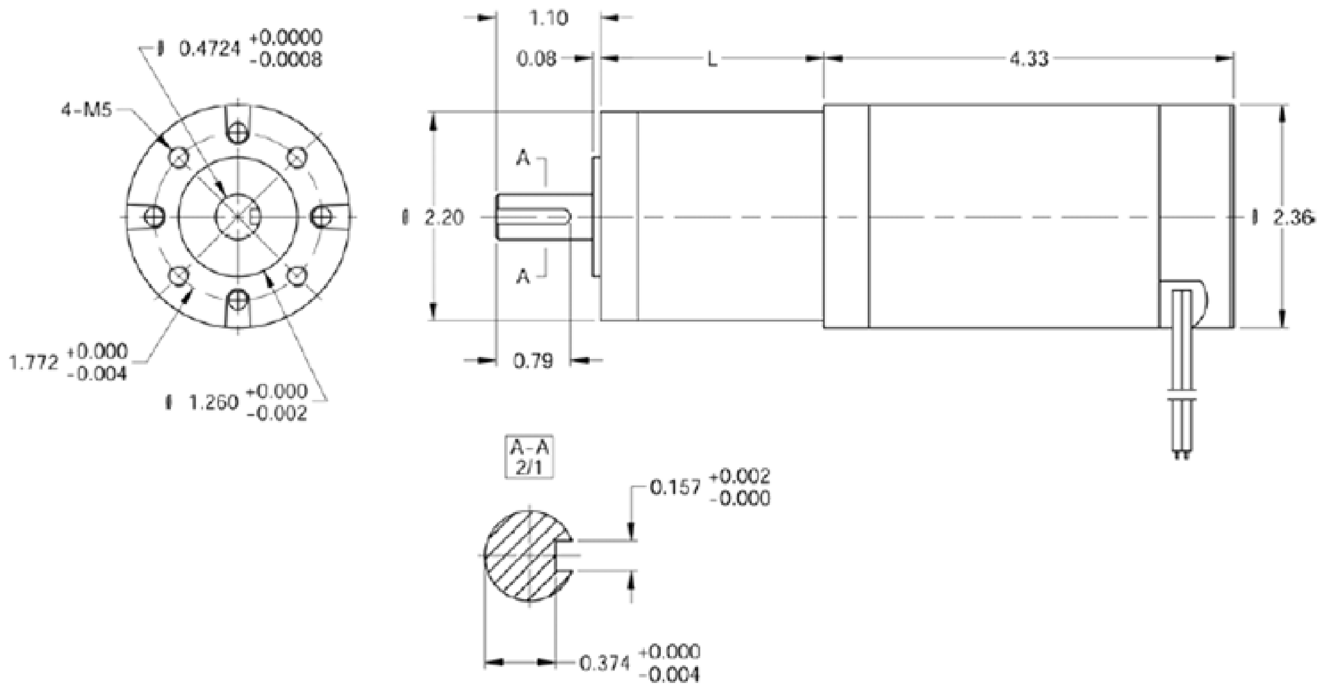


Nominal Diameter	Lead	Dynamic Load Capacity (C_{am})	Length L_2	Width/ Diameter D_1	Ball Nut Form	Flange Diameter D_6	Bolt Hole Circle D_4	Hole Diameter D_5	Ball Nut	Nut P/N	Screw P/N	Catalog Page
(mm)	(mm)	(kN)	(mm)	(mm)		(mm)	(mm)	(mm)				
16	5	9.3	44.0	28.0	Form E	48.0	38.0	5.5	KGF-N	**KGF-N-1605-RH-EE**	**KGS-1605-050-RH**	124
20	5	11.1	48.0	33.0	Form E	58.0	45.0	6.6	FS	**7832782**	**7832779**	116
20	5	10.5	44.0	32.0	Form E	55.0	45.0	7.0	KGF-N	**KGF-N-2005-RH-EE**	**KGS-2005-050-RH**	124
20	20	11.5	56.5	40.0	Form E	63.0	50.0	6.6	FS	**7832785**	**7832783**	116
20	20	11.6	30.0	35.0	Form S	62.0	50.0	7.0	KGF-N	**KGF-N-2020-RH-EE**	**KGS-2020-050-RH**	124
20	50	13.0	56.0	35.0	Form S	62.0	50.0	7.0	KGF-N	**KGF-N-2050-RH-EE**	**KGS-2050-050-RH**	124
25	5	13.1	53.0	38.0	Form E	63.0	50.0	6.6	FS	**7832789**	**7832786**	116
25	5	12.3	46.0	38.0	Form E	62.0	50.0	7.0	KGF-N	**KGF-N-2505-RH-EE**	**KGS-2505-050-RH**	124
32	5	21.5	59.0	45.0	Form E	70.0	58.0	7.0	KGF-N	**KGF-N-3205-RH-EE**	**KGS-3205-050-RH**	124
32	10	26.4	72.0	50.0	Form E	80.0	65.0	9.0	FS	**7832801**	**7832798**	116
32	10	33.4	73.0	53.0	Form E	80.0	68.0	7.0	KGF-N	**KGF-N-3210-RH-EE**	**KGS-3210-050-RH**	124
32	40	14.9	45.0	53.0	Form S	80.0	68.0	7.0	KGF-N	**KGF-N-3240-RH-EE**	**KGS-3240-050-RH**	124
40	5	21.7	53.0	56.0	Form E	80.0	68.0	6.6	FS	**7832807**	**7832804**	116
40	5	23.8	59.0	53.0	Form E	80.0	68.0	7.0	KGF-N	**KGF-N-4005-RH-EE**	**KGS-4005-050-RH**	124
40	10	38.0	73.0	63.0	Form E	95.0	78.0	9.0	KGF-N	**KGF-N-4010-RH-EE**	**KGS-4010-050-RH**	124
40	20	39.7	62.0	75.0	Form E	110.0	93.0	11.0	FS	**7832813**	**7832811**	116
40	40	29.8	106.0	72.0	Form E	110.0	93.0	11.0	FS	**7832816**	**7832814**	116
50	10	68.7	97.0	72.0	Form E	110.0	90.0	11.0	KGF-N	**KGF-N-5010-RH-EE**	**KGS-5010-050-RH**	124
63	10	76.0	99.0	85.0	Form E	125.0	105.0	11.0	KGF-N	**KGF-N-6310-RH-EE**	**KGS-6310-050-RH**	124

Motor - BDPG-60-110-24V-3000-R168

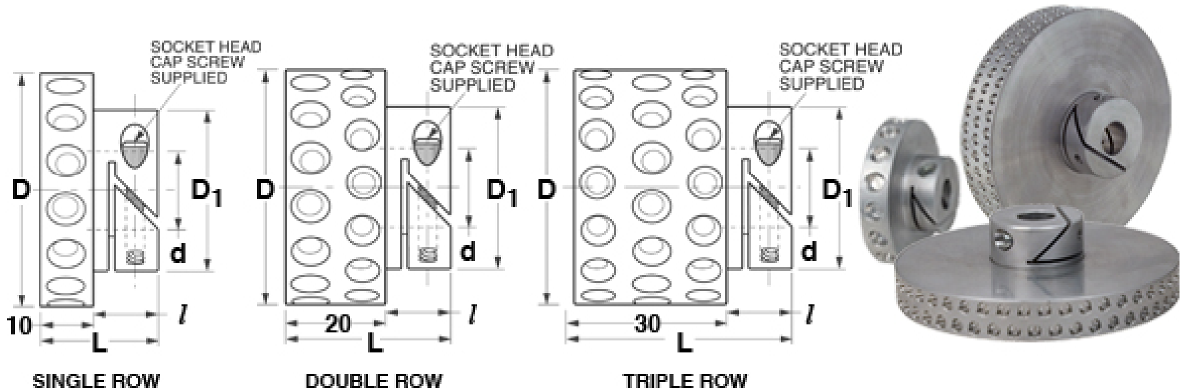
BDPG-60-110-24V-3000-Rxx Specifications

Gear Ratio **Rxx**	3.6	4.25	13	15	18	47	55	65	76	168	198	234	276	326
Length (L) (in)	1.79	1.79	1.79	1.79	1.79	2.22	2.22	2.22	2.22	3.01	3.01	3.01	3.01	3.01
No Load Speed (RPM)	833	706	230	200	167	64	54	46	39	18	15	13	11	9.2
Rated Speed (RPM)	694	588	192	167	139	53	45	38	33	15	12	10	9.0	7.7
Rated Torque (kg-cm)	4.1	4.8	14	16	19	48	56	66	77	163	192	227	268	300
Peak Torque (kg-cm)	30	30	100	100	100	300	300	300	300	600	600	600	600	600



SDP - Timing Belt Pulley

Description
10mm Pitch, Double Row, 32 Cavities per Row Conidrive Pulley With 20 mm Bore



Product Details	
Part Number	A 6A14M32D20
Unit	Metric
Pitch	10 mm
No. Of Cavities/row	32
Material	Aluminum Alloy
Belt Type	Double Row
Bore Size	20 mm
Hub Configuration	FAIRLOC® HUB
Outside Dia.	100.42 mm
Overall Width	42 mm
Hub Dia.	49.5 mm
Hub Projection	22 mm

Price Information	
Quantity	**Price**
1 to 24	$51.30
25 to 49	$44.15
50 and up	$41.18
Availability	In Stock
Sell Unit	Each
Quantity	

ADD TO CART

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CAD Models / Catalog Pages	
Specs from printed catalog	PDF
PTC PartsLink	3D CAD Models

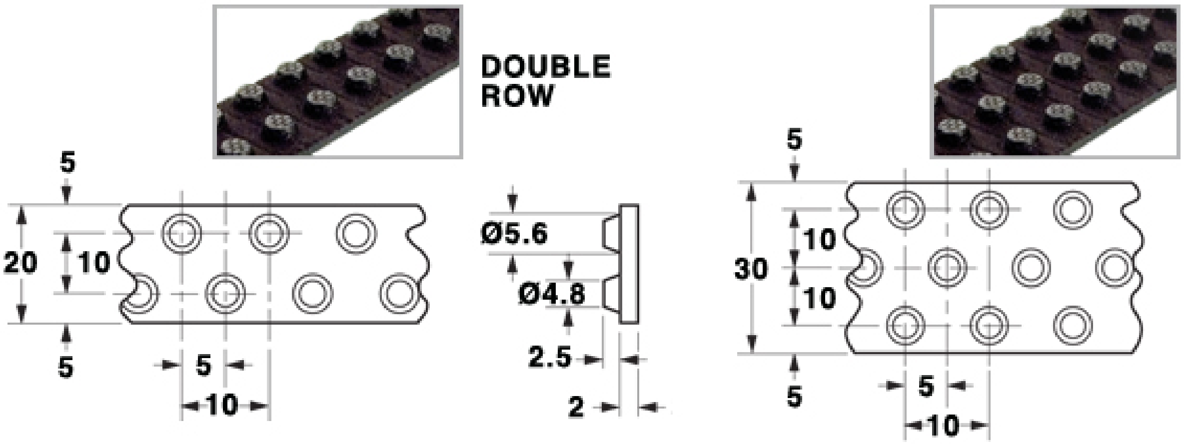
For Breaks Not Shown & Out Of Stock Items

Please call or email our sales department for delivery schedule and higher price breaks.

Phone: (800) 819-8900X491 Fax: (516) 326-8827 Email: support@sdp-si.com

SDP – Timing Belt

Description
10mm Pitch, 20mm wide, 700mm Long,Double Row Self Guiding Conidrive belt



Product Details	
Part Number	A 6Z13MD0700
Unit	Metric
Belt Type	Double Row
Pitch	10 mm
Belt Width	20 mm
Material	Polyurethane
Tension Member	Steel Tensile Cords
Pitch Length	700 mm

Price Information	
Quantity	**Price**
1 to 9	$93.62
10 to 24	$91.75
25 and up	$89.60
Availability	Out of Stock
Sell Unit	Each
Quantity	
	ADD TO CART
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CAD Models / Catalog Pages	
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10 mm PITCH
SELF-GUIDING
CONTINUOUS ROLLING ACTION
LOW NOISE & VIBRATION

MATERIAL:
Polyurethane Reinforced with Steel Tensile Cords

OPERATING TEMPERATURE:
-30°C to +80°C

FEATURES:
Self-guiding system: no side flanges needed on pulleys
Nondirectional: same meshing performance in both directions of belt travel
Polygon-free: smooth rolling around pulleys thanks to contact with flat belt area
Noise-minimized & low-vibration: continuous rolling, smooth meshing of conical projections into recesses
Homogenous distribution of forces in the belt: no force components acting laterally thanks to symmetrical cone geometry and balanced tension member arrangement (S/Z winding)

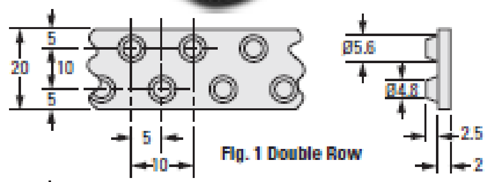


Fig. 1 Double Row

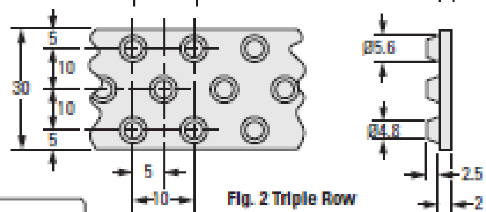


Fig. 2 Triple Row

METRIC COMPONENT

Catalog Number		Length mm
Fig. 1 Double Row	Fig. 2 Triple Row	
A 6Z13MD0500	A 6Z13MT0500	500
A 6Z13MD0600	A 6Z13MT0600	600
A 6Z13MD0700	A 6Z13MT0700	700
A 6Z13MD0800	A 6Z13MT0800	800
A 6Z13MD0900	A 6Z13MT0900	900
A 6Z13MD1000	A 6Z13MT1000	1000

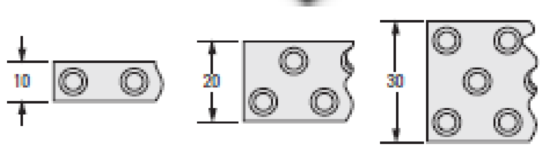
TIMING BELT STOCK - 10 mm Pitch

MATERIAL:
Polyurethane Reinforced with Steel Tensile Cords

OPERATING TEMPERATURE:
-30°C to +80°C

APPLICATIONS:
Metering, positioning, conveying and oscillating drives where belt lengths required are longer than standard endless belts.

Priced per Meter

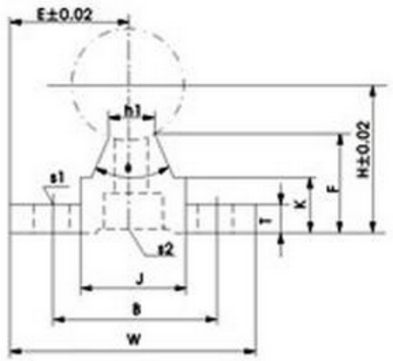


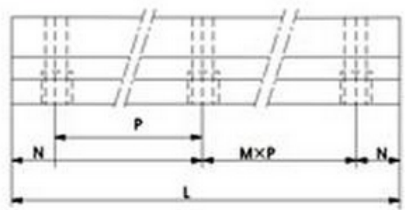
Single Row Double Row Triple Row

METRIC COMPONENT

Catalog Number	No. of Rows	Allowable Static Tensile Load N	Max. Available Length m
A 6Z13MCS	Single	650	50
A 6Z13MCD	Double	1300	50
A 6Z13MCT	Triple	1950	50

Linear Guide way System





Shaft (mm)	Dimensions(mm)									Mounting Dimensions(mm)				
	H	E	W	L	F	T	K	J	h1	B	N	M*P	S1	S2
12	20.46	17	34	**1,117.6**	15	4.5	9.8	15	6	25	50	1x100	4.5	M4

U-Joint

Bushing

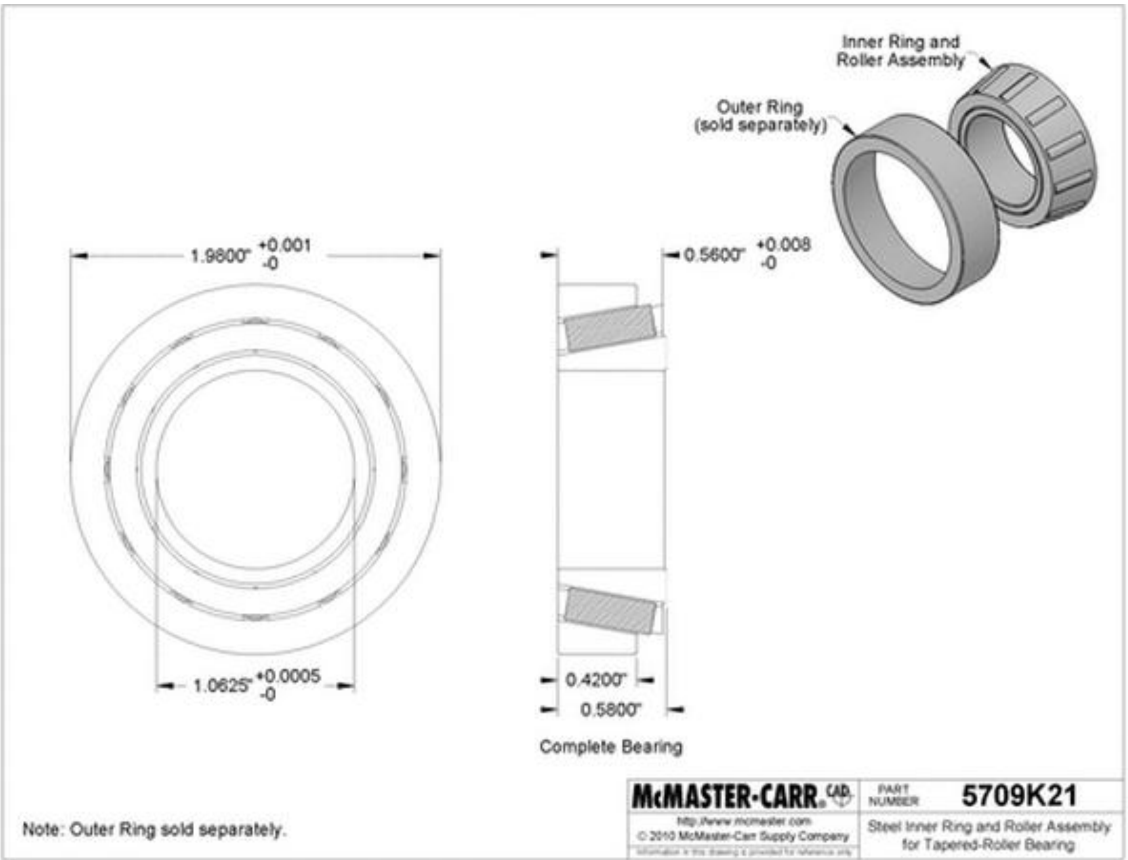
SAE 863—2938T

For Shaft Dia.	OD	Tolerance for Shaft Dia.	OD Tolerance
3/16"	5/16"	–0.001" to +0.002"	–0.001" to +0.0015"
1/4"	3/8"	+0.001" to +0.002"	+0.001" to +0.002"
5/16"	7/16"	+0.000" to +0.001"	+0.0015" to +0.0025"
5/16"	9/16"	–0.001" to +0.0025"	–0.001" to +0.003"
5/16"	5/8"	–0.001" to +0.005"	–0.001" to +0.010"
3/8"	1/2"	+0.001" to +0.002"	+0.002" to +0.003"
3/8"	5/8"	+0.001" to +0.002"	+0.002" to +0.003"
7/16"	9/16"	–0.001" to +0.0015"	–0.001" to +0.0025"
7/16"	5/8"	–0.001" to +0.0015"	–0.001" to +0.003"
1/2"	5/8"	+0.001" to +0.002"	+0.001" to +0.002"
1/2"	3/4"	+0.002" to +0.003"	+0.002" to +0.003"
5/8"	3/4"	+0.000" to +0.001"	+0.002" to +0.003"
5/8"	7/8"	–0.001" to +0.002"	–0.001" to +0.004"
3/4"	7/8"	+0.000" to +0.001"	+0.001" to +0.002"
3/4"	1"	+0.001" to +0.002"	+0.003" to +0.004"
7/8"	1 1/8"	–0.001" to +0.002"	–0.0015" to +0.003"
1"	1 1/4"	+0.000" to +0.001"	+0.001" to +0.002"
1 1/4"	1 1/2"	–0.0015" to +0.0035"	–0.002" to +0.004"
1 3/8"	1 5/8"	–0.0015" to +0.002"	–0.002" to +0.003"
1 1/2"	1 3/4"	–0.002" to +0.004"	–0.002" to +0.005"
1 3/4"	2 1/4"	–0.002" to +0.003"	–0.002" to +0.004"

Length	Tolerance
1/4"	±0.010"
3/8"	±0.010"
1/2"	±0.010"
5/8"	±0.010"
3/4"	±0.010"
1"	±0.010"
1 1/4"	±0.010"
1 1/2"	±0.010"
2 1/2"	±0.015"

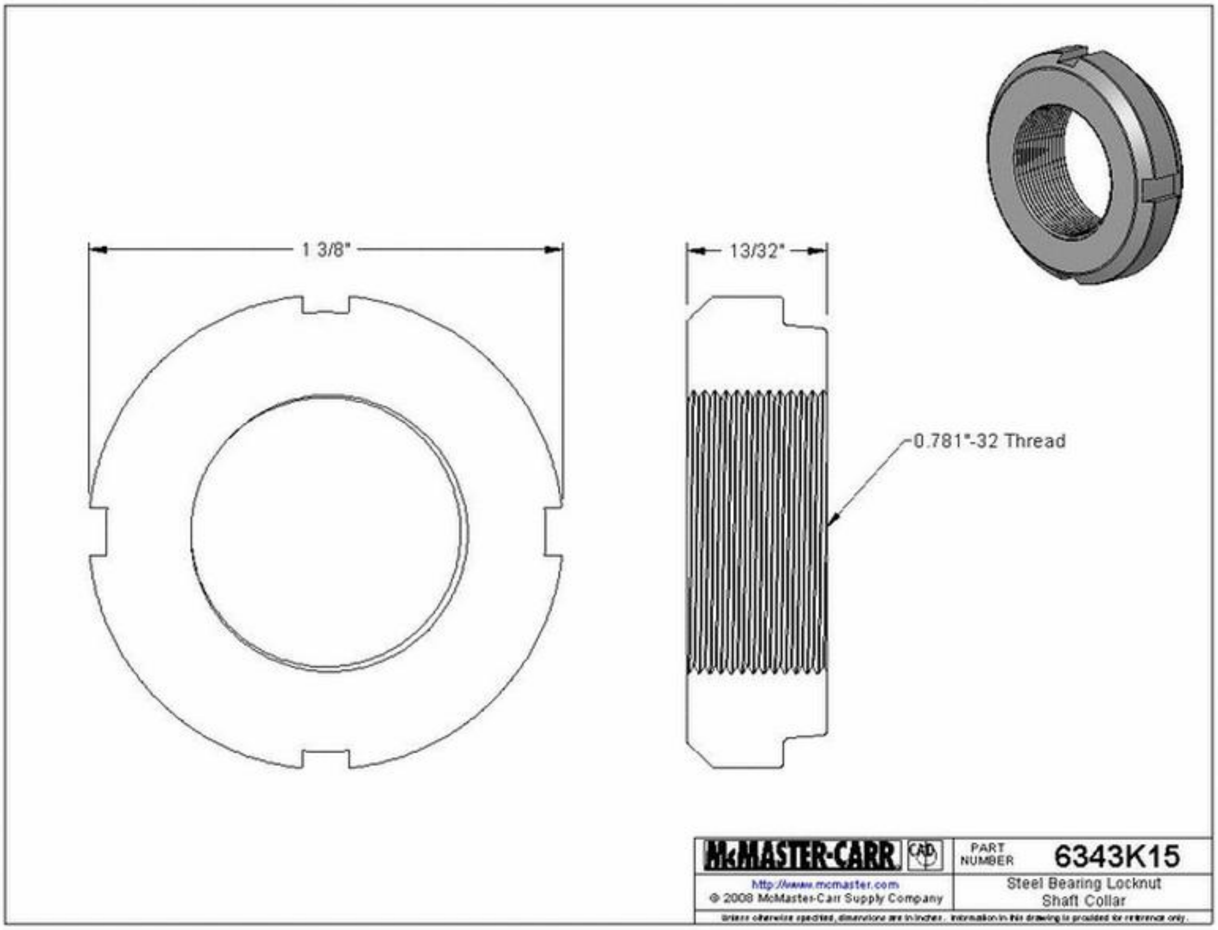
U-U Joint

Thrust Bearing

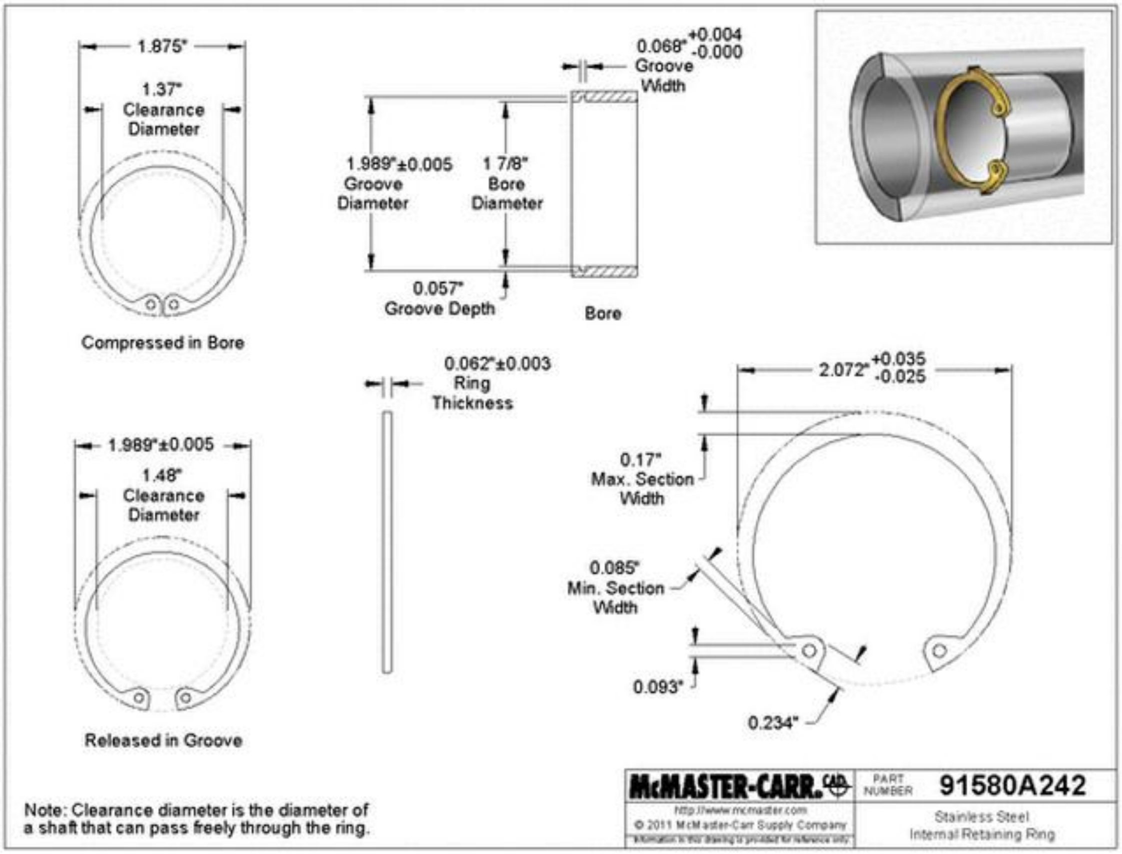


The information in this 3-D model is provided for reference only. Details

Shaft Collar



Circlip

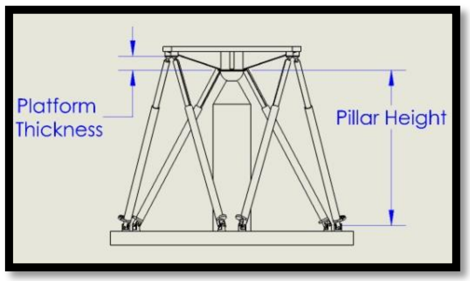


1.875"
1.37" Clearance Diameter
Compressed in Bore
1.989"±0.005
1.48" Clearance Diameter
Released in Groove
0.068" +0.004 -0.000 Groove Width
1.989"±0.005 Groove Diameter
1 7/8" Bore Diameter
0.057" Groove Depth
Bore
0.062"±0.003 Ring Thickness
2.072" +0.035 -0.025
0.17" Max. Section Width
0.085" Min. Section Width
0.093"
0.234"
McMASTER-CARR.
PART NUMBER 91580A242
http://www.mcmaster.com
© 2011 McMaster-Carr Supply Company
Stainless Steel Internal Retaining Ring
Note: Clearance diameter is the diameter of a shaft that can pass freely through the ring.

Appendix E – Detailed Supporting Analysis

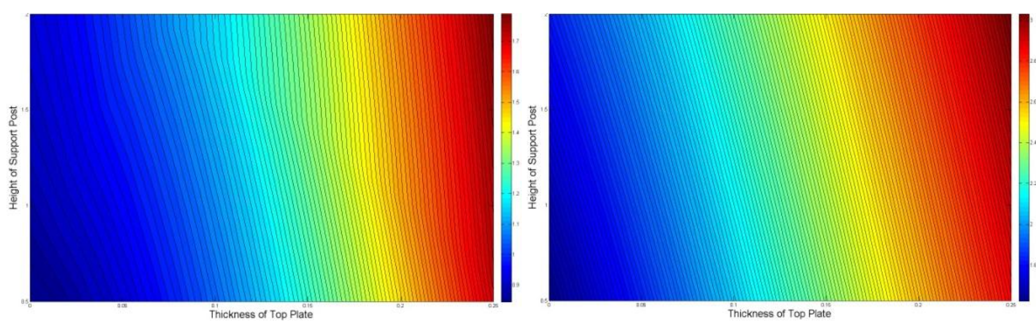
Geometry

Because the width of the platform and base were already determined (1.5 meters and 2.5 meters, respectively), the remaining dimensions to choose were the height of the support pillar and the thickness of the plate. Here, the plate thickness has nothing to do with the actual thickness of the platform, but is meant to describe the vertical distance from the connection point of the platform to the support pillar to the connection point of the actuator, when the hexapod is pointing straight up.



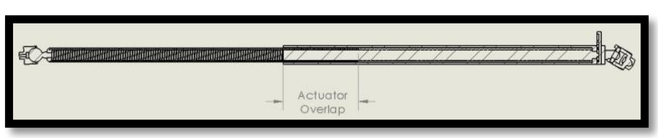
Definition of Geometrical Terms

These two parameters were varied between reasonable values (zero to 0.25 meters for the platform thickness and 0.5 to 2 meters for the pillar), and the results plotted in a contour plot. In this way, we can easily see how altering these parameters affect the hexapod size, actuator size, actuator stability.



Minimum Actuator Length (left) and Maximum Actuator length (right)

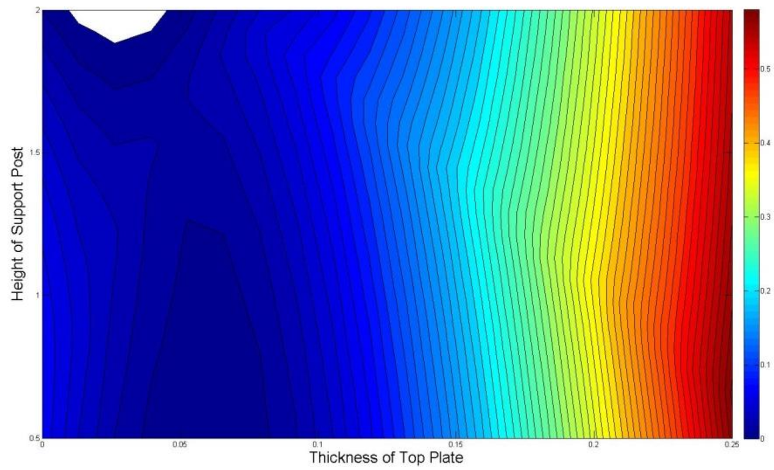
In the above figures, it is shown that there is almost a linear relationship between the maximum length and the pillar height and platform thickness. The same is true for the minimum actuator length, although there is some distortion for thin platform thicknesses. The figure below shows the actuator overlap, which is the amount of the rod that can remain inside the sheave of the fully extended actuator. This parameter was maximized, as it is significantly influences the stability of the actuator.



Definition of Actuator Overlap

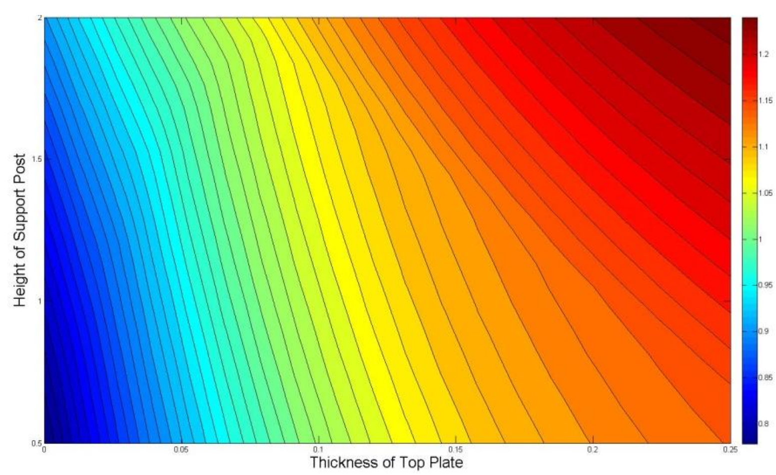
The figure below shows the amount of actuator overlap, and is dependent on the thickness of the top plate and the height of the support pillar. The amount of overlap is important because of imperfections in the manufacturing and the weakness of the material. If the rod and the screw were ideal rigid bodies, there would

need to be almost no overlap. However, because of manufacturing tolerances and material elasticity, it is impossible to get a “perfect” prismatic joint between the rod and screw. By increasing the overlapping area between the two, we can get closer to the ideal state by minimizing bending and distortion due to manufacturing tolerances.



Actuator Overlap

For similar reasons, it is important that the actuator have as small of a stroke as possible. By minimizing the stroke, we can hopefully minimize the compound effect of manufacturing defects, and increase the accuracy of the actuator. With a smaller stroke, we are able to use a screw with a lower pitch angle. This will make the actuator extend and retract slower, but it will greatly increase the precision of the motion. The stroke for varying pillar heights and platform thicknesses is shown in the figure below.



Actuator Stroke Spectrum

Geometry Code

```
%% Hexapod Stroke
% Brian Gilchrist - 10/30/2011
% This Program calculates the minimum stroke (maximum and minimum lengths)
% of the needed actuator, given: positioning of actuators (Beta and Phi),
% hexapod's top plate radius (r), hexapod's bottom plate radius (R), and minimum parallel
% distance between plates (h1).

clear all;
clc;
close all;

Theta = 30; %Angle between centerline of top plate of hexapod and horizontal, in Degrees
b = 100; %Length of arrays: h1 and R
Phi = 55; %Angle between 120 deg and Actuator attachment to the bottom plate (in deg).
Beta = 5; %Angle between 120 deg and Actuator attachment to top plate (in deg).
r = .75; %Hexapod's Top Plate radius (in meters)
h1 = linspace(0,3,b)'; %Minimum parallel distance between two plates (in meters)
R = linspace(.75,2,b)'; %Hexapod's Bottom Plate radius (in meters)
n = 1;
% h1 = 1.25;
% R = 1.25;

for n = 1:b; %Index for h1
    for l = 1:b; %Index for R
    L1(n,l) = sqrt((h1(n,1)^2)+(((R(l,1)*sind(Phi))-(r*sind(Beta)))^2)+((abs((r*cosd(Beta))-(R(l,1)*cosd(Phi))))^2)); %Minimum Length of Actuator (in meters)

    h2(n,l) = h1(n,1) + (r*(cosd(60-Beta)+cosd(Beta)))*sind(90-Theta); %Height of Actuator in extended position, in meters
    BF2(n,l) = (R(l,1)*sind(Phi)) - (r*sind(Beta)); %Base width distance between ends of actuator from Front View, in meters
    Bs2(n,l) = (R(l,1)*cosd(Phi)) + (r*cosd(60-Beta)) - (r*(cosd(60-Beta)+cosd(Beta))*cosd(90-Theta)); %Base width distance between ends of actuators from Side View, in meters

    L2(n,l) = sqrt((h2(n,l)^2)+(BF2(n,l)^2)+(Bs2(n,l)^2)); %Extended length of Actuator, in meters

    S(n,l) = L2(n,l) - L1(n,l); %Actuator Stroke, in meters

   end
end
```

```
 figure(1);
    o=0:0.01:15;
    s=0:.01:2;
    [C,h] = contourf(R,h1,S,s);
    xlabel('Bottom Plate Radius [m]');
    ylabel('Shortest Parallel Plate Distance [m]');
    title('Necessary Stroke [m]')
    colorbar;

figure(2);
    o=0:0.05:15;
    [c,H] = contourf(R,h1,L1,o);
    xlabel('Bottom Plate Radius [m]');
    ylabel('Shortest Parallel Plate Distance [m]');
    title('Minimum Actuator Length [m]')
    colorbar;

 figure(3);
    o=0:0.05:15;
    [s,p] = contourf(R,h1,L2,o);
    xlabel('Bottom Plate Radius [m]');
    ylabel('Shortest Parallel Plate Distance [m]');
    title('Maximum Actuator Length [m]')
    colorbar;
```

Motion Equations

The method of approach and method of approach were taken almost directly from Pedraammehr, S., Mahboubkhah, M. and Pakzad, S. "*An Improved Solution to the Inverse Dynamics of the General Stewart Platform.*" Some of the equations are reproduced below.

For the *i*th actuator, the length vector (the vector representing the length and direction of the actuator)is given by $\mathbf{L}_i$, where

$$\mathbf{L}_i = \mathbf{a}_i - \mathbf{b}_i$$

where $\mathbf{a}_i$ and $\mathbf{b}_i$ are the position vectors of the endpoints of the actuators in the global reference frame. $\mathbf{b}_i$ is static, and does not require any calculations. $\mathbf{a}_i$ is given by

$$\mathbf{a}_i = \mathbf{X} + \mathbf{R} \cdot {}^{\mathbf{P}}\mathbf{a}_i$$

where $\mathbf{X}$ is the position vector of the geometric center of the platform (for our purposes, it is the position of the connection point between the platform and the support pillar). ${}^{\mathbf{P}}\mathbf{a}_i$ is the position vector of the top actuator endpoint in the coordinate system of the top platform. $\mathbf{R}$ is the transformation matrix from the platform reference frame to the global reference frame.

The unit vector for each actuator, $\mathbf{n}_i$, can be found by dividing the length vector, $\mathbf{L}_i$, by the magnitude of the length vector, l_i.

$$\mathbf{n}_i = \mathbf{L}_i / l_i$$

where l_i is given by

$$l_i = \|\mathbf{L}_i\|$$

These equations can then be differentiated with respect to time, to acquire the equations for the velocity and acceleration of the actuators and of the top platform. This is given in great detail in Pedraammehr, S., et al.

Stiffness Calculations

Because the actuators are connected by universal joints at the bottom and by our universal-universal joints at the top, they can be modeled as truss elements, or bars. Because of this, their transverse stiffness is not particularly relevant, except when it comes to buckling in compression. For the sake of this particular discussion it is assumed that the geometry of the actuators act similarly in tension and compression, and buckling is not considered.

Actuators are made up of several different components, and while it may appear as though the sheave plays an important role in the stiffness of the actuator, it in fact only plays a role in the transverse stiffness, and not the axial stiffness considered here. For this reason, we only need to look at the screw and the rod, each of which has its own axial stiffness, given by

$$k = \frac{AE}{l}$$

- k is the stiffness of that particular part
- A is the effective area of that part
- E is the elastic modulus for the material which that component is made up of
- l is the length of the component

Because the screw and the sheave overlap, there is a section where the effective area is increased, and this section must be accounted for separately. Because the three sections (rod, screw, overlap) are all in series, their longitudinal stiffnesses add inversely. That is,

$$k_{actuator} = \left(\frac{1}{k_{rod}} + \frac{1}{k_{screw}} + \frac{1}{k_{overlap}}\right)^{-1}$$

Depending on the position, each actuator will have a different stiffness because the length of the rod, screw, and overlap will be different. For most of the analysis, the hexapod was in the neutral position (pointing straight up to the sky, with all the actuators the same length).

Transverse deflection (motion from side to side) of the platform is the motion that is the most detrimental to the telescope mount. Because of this, FEM analysis was carried out on the hexapod system to observe the side-to-side displacement of the top platform. Because the top platform is not our area of concern, it can be left as a rigid body, with all of the connection points constrained to move the same amount from side to side. Because of the symmetry of the hexapod, and depending on the direction of the applied force, we can greatly reduce the complexity of the analysis by only examining half of the hexapod, and applying symmetrical constraints to the boundaries.

By adding a rigid beam for support of the center of the hexapod, the analysis became much more complicated. Nonetheless, it was found through similar techniques that the stiffness increases by at least ten percent in the worst case. An increase in stiffness by up to 30% has been observed for different angles of the platform; however this has not been fully analyzed and validated just yet.

Screw Diameter Calculations

More Concise Calculations for New, Smaller Design

Engineering Constants

$P = 1500$ [N]

$E_{steel} = 2.07 \times 10^{11}$ [PA]

$L_o = 1$ [m]

$d_{cm,screw} = 2.54$ [cm]

$d_{cm,o,square,rod} = 5.08$ [cm]

$d_{cm,o,round,rod} = 5.08$ [cm]

$K_{screw} = 0.5$

$K_{rod} = 0.5$

$L_{rod} = 1$ [m]

Engineering Equations - Screw

$$\sigma_{screw} = \frac{P}{A_{screw}}$$

$$\sigma_{screw} = E_{steel} \cdot \varepsilon$$

$$A_{screw} = \pi \cdot \frac{d_{screw}^2}{4}$$

$$\varepsilon = \frac{L_{stretch}}{L_o}$$

Engineering Equations - Square Rod

$$\sigma_{square,rod} = \frac{P}{A_{square,rod}}$$

$$\sigma_{square,rod} = E_{steel} \cdot \varepsilon$$

$$A_{square,rod} = d_{o,square,rod}^2 - d_{i,square,rod}^2$$

Engineering Equations - Round Rod

$$\sigma_{round,rod} = \frac{P}{A_{round,rod}}$$

$$\sigma_{round,rod} = E_{steel} \cdot \varepsilon$$

$$A_{round,rod} = \frac{\pi}{4} \cdot d_{o,round,rod}^2 - \left[\frac{\pi}{4} \cdot d_{i,round,rod}\right]^2$$

Euler Buckling - Both Ends Pinned

$$I_{screw} = \frac{\pi}{64} \cdot d_{screw}^{4}$$

$$F_{buckling,screw} = \pi^2 \cdot E_{steel} \cdot \frac{I_{screw}}{\left[L_o \cdot K_{screw} \right]^2}$$

$$I_{square,rod} = \frac{d_{o,square,rod}^{4} - d_{i,square,rod}^{4}}{12}$$

$$F_{buckling,square,rod} = \pi^2 \cdot E_{steel} \cdot \frac{I_{square,rod}}{\left[L_{rod} \cdot K_{rod} \right]^2}$$

$$I_{round,rod} = \frac{\pi}{64} \cdot \left[d_{o,round,rod}^{4} - d_{i,round,rod}^{4} \right]$$

$$F_{buckling,round,rod} = \pi^2 \cdot E_{steel} \cdot \frac{I_{round,rod}}{\left[L_{rod} \cdot K_{rod} \right]^2}$$

$$SF_{screw} = \frac{F_{buckling,total,screw}}{P}$$

$$SF_{square} = \frac{F_{buckling,total,square}}{P}$$

$$SF_{round} = \frac{F_{buckling,total,round}}{P}$$

$$F_{buckling,total,screw} = \pi^2 \cdot E_{steel} \cdot \frac{I_{screw}}{\left[\left(L_{rod} + L_o \right) \cdot K_{screw} \right]^2}$$

$$F_{buckling,total,square} = \pi^2 \cdot E_{steel} \cdot \frac{I_{square,rod}}{\left[\left(L_{rod} + L_o \right) \cdot K_{rod} \right]^2}$$

$$F_{buckling,total,round} = \pi^2 \cdot E_{steel} \cdot \frac{I_{round,rod}}{\left[\left(L_{rod} + L_o \right) \cdot K_{rod} \right]^2}$$

Conversions

$$d_{cm,screw} = d_{screw} \cdot 100 \quad [cm/m]$$

$$d_{cm,o,square,rod} = d_{o,square,rod} \cdot 100 \quad [cm/m]$$

$$d_{cm,o,round,rod} = d_{o,round,rod} \cdot 100 \quad [cm/m]$$

$$t_{cm,square,rod} = t_{square,rod} \cdot 100 \quad [cm/m]$$

$$t_{cm,square,rod} = t_{in,square,rod} \cdot 2.54 \quad [cm/in]$$

$$t_{cm,round,rod} = t_{round,rod} \cdot 100 \quad [cm/m]$$

$$t_{cm,round,rod} = t_{in,round,rod} \cdot 2.54 \quad [cm/in]$$

$d_{i,square,rod} = d_{o,square,rod} - 2 \cdot t_{square,rod}$

$d_{i,round,rod} = d_{o,round,rod} - 2 \cdot t_{round,rod}$

$A_{screw,in} = A_{screw} \cdot 1550$ [in^2/m^2]

SOLUTION

Unit Settings: SI C kPa kJ mass deg

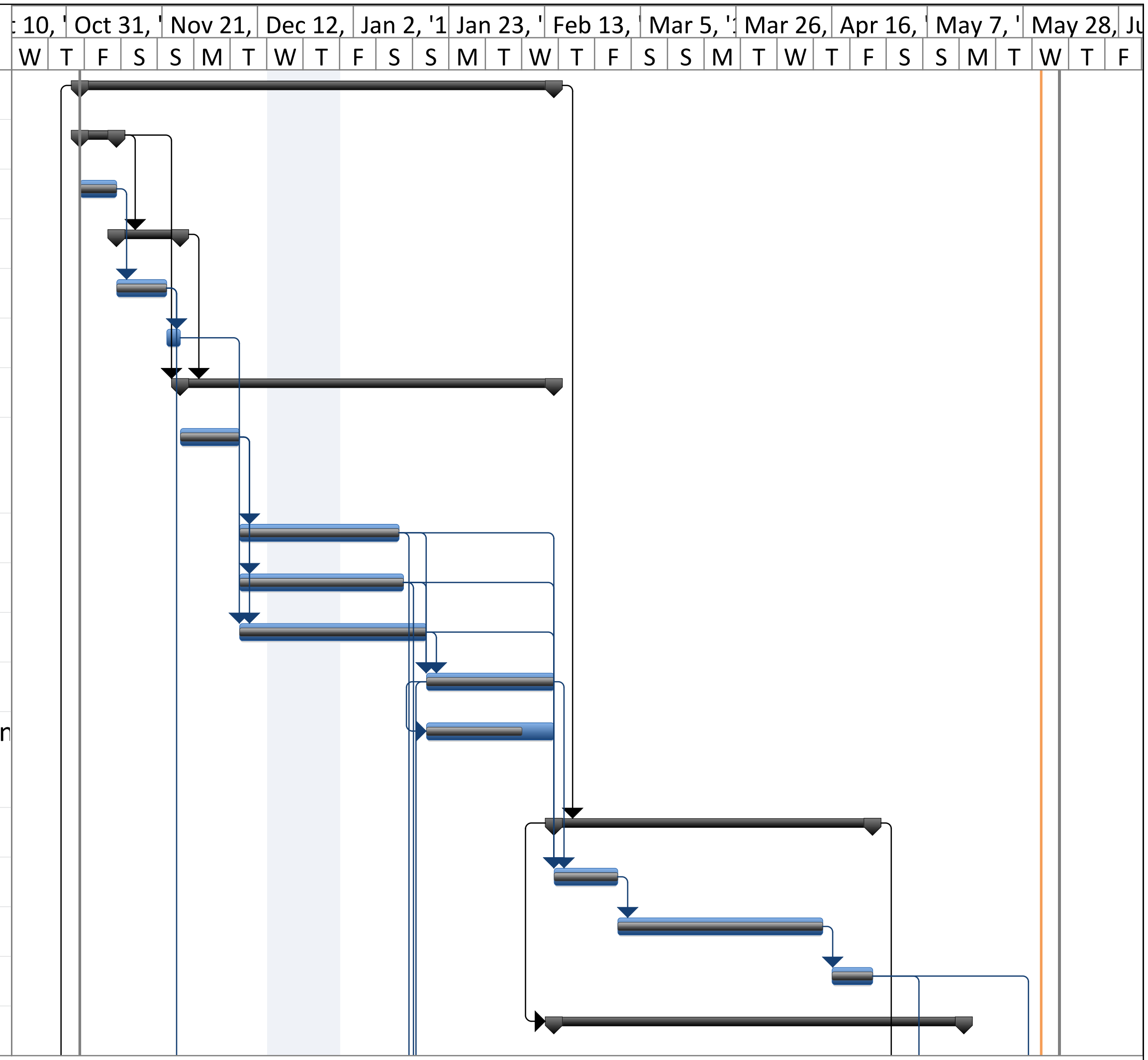
$A_{round,rod}$ = 0.0005067 [m^2]
$A_{screw,in}$ = 0.7854 [in^2]
$d_{cm,o,round,rod}$ = 5.08 [cm]
$d_{cm,screw}$ = 2.54 [cm]
$d_{i,square,rod}$ = 0.04554 [m]
$d_{o,square,rod}$ = 0.0508 [m]
ε = 0.0000143 [-]
$F_{buckling,round,rod}$ = 235384 [N]
$F_{buckling,square,rod}$ = 1.606E+06 [N]
$F_{buckling,total,screw}$ = 41742 [N]
$I_{round,rod}$ = 2.880E-08 [m^4]
$I_{square,rod}$ = 1.965E-07 [m^4]
K_{screw} = 0.5 [-]
L_{rod} = 1 [m]
P = 1500 [N]
SF_{screw} = 27.83
$\sigma_{round,rod}$ = 2.960E+06 [N/m^2]
$\sigma_{square,rod}$ = 2.960E+06 [N/m^2]
$t_{cm,square,rod}$ = 0.263 [cm]
$t_{in,square,rod}$ = 0.1035 [in]
$t_{square,rod}$ = 0.00263 [m]

A_{screw} = 0.0005067 [m^2]
$A_{square,rod}$ = 0.0005067 [m^2]
$d_{cm,o,square,rod}$ = 5.08 [cm]
$d_{i,round,rod}$ = 0.04964 [m]
$d_{o,round,rod}$ = 0.0508 [m]
d_{screw} = 0.0254 [m]
E_{steel} = 2.070E+11 [PA]
$F_{buckling,screw}$ = 166969 [N]
$F_{buckling,total,round}$ = 58846 [N]
$F_{buckling,total,square}$ = 401537 [N]
I_{screw} = 2.043E-08 [m^4]
K_{rod} = 0.5 [-]
L_o = 1 [m]
$L_{stretch}$ = 0.0000143 [m]
SF_{round} = 39.23
SF_{square} = 267.7
σ_{screw} = 2.960E+06 [N/m^2]
$t_{cm,round,rod}$ = 0.0579 [cm]
$t_{in,round,rod}$ = 0.02279 [in]
$t_{round,rod}$ = 0.000579 [m]

No unit problems were detected.

Appendix F – Gantt Chart

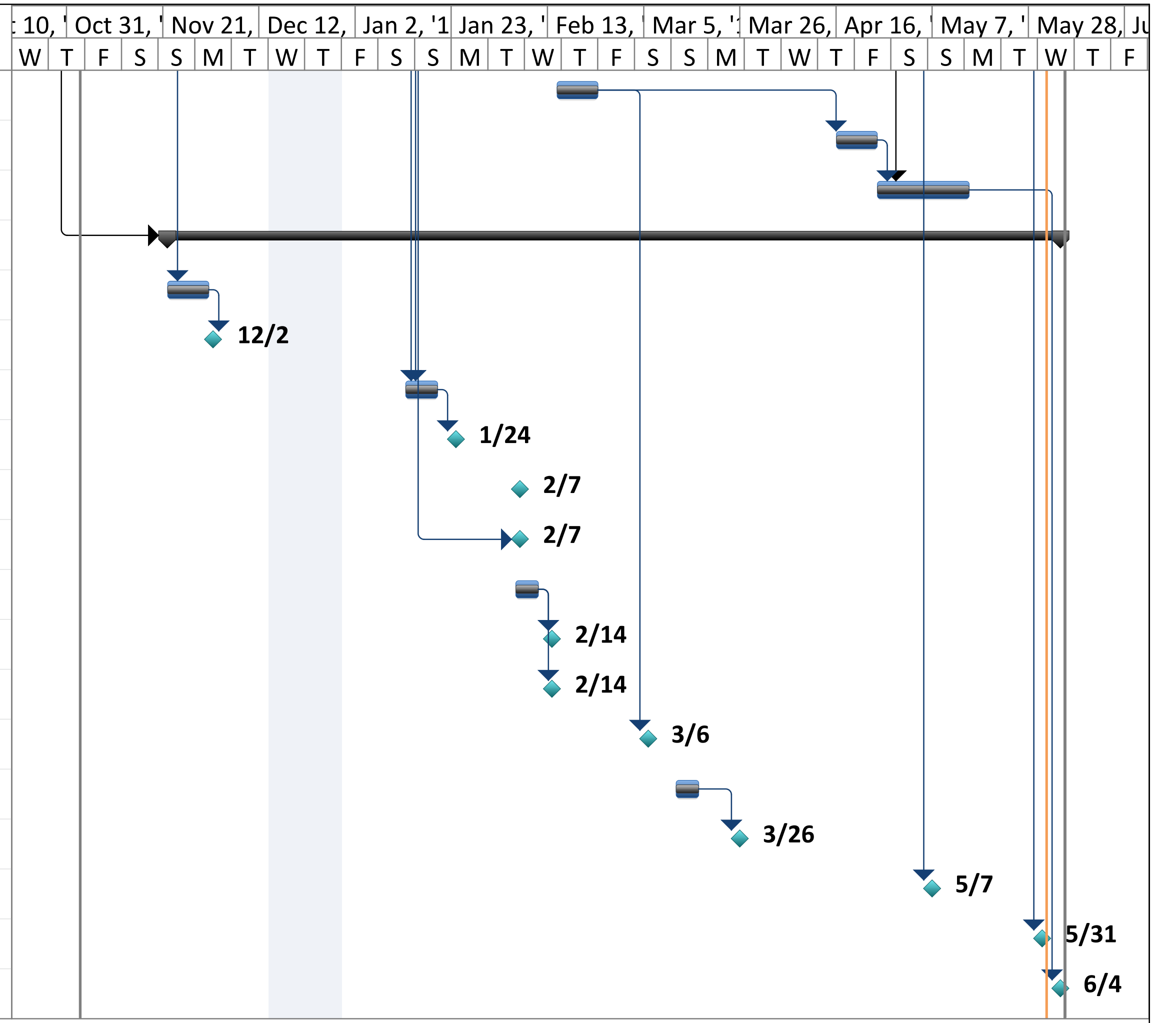
ID	Indicators	Task Mode	Task Name
1			**Design**
2	✓		**Calculate Required Geometries**
3	✓		Define Design Geometry
4			**Force and Stiffness Calculations**
5	✓		Define Required Stiffness
6			Materials Selection
7			**Drive Mechanism**
8	✓		Calculation of Power and Speed Reduction
9	✓		Select Motor
10	✓		Select Gears and Shafts
11	✓		Select Power Screw
12	✓		Design Actuation Cylinder
13			Design of Tracking System & Position Feedback
14	✓		**Build**
15	✓		Find Part and Material Vendors
16	✓		Order and Build all Components
17	✓		Assemble Actuator
18	✓		**Test**



Project: Project Timeline
Date: Fri 6/1/12

Task		External Milestone		Manual Summary Rollup	
Split		Inactive Task		Manual Summary	
Milestone		Inactive Milestone		Start-only	
Summary		Inactive Summary		Finish-only	
Project Summary		Manual Task		Deadline	
External Tasks		Duration-only		Progress	

ID	ℹ	Task Mode	Task Name
19	✓		Define Tests Required
20	✓		Assemble Test Equipment
21	✓		Perform all Required Tests
22			**Reporting and Presenting**
23	✓		Prepare Conceptual Design Report
24			Conceptual Design Report Due
25	✓		Prepare Student Presentation
26			Give Student Presentation
27			Ethics Memo Topic Due
28			Design Report Due
29	✓		Prepare Ethics Presentation
30			Individual Ethics Memo Due
31			Team Ethics Presentations
32			Manufacturing and Test Review
33	✓		Prepare Update for Sponsor
34			Project Update Memo to Sponsor
35			Hardware Demo
36			Senior Design Expo
37			Final Report Due



Project: Project Timeline
Date: Fri 6/1/12

Task	External Milestone	Manual Summary Rollup
Split	Inactive Task	Manual Summary
Milestone	Inactive Milestone	Start-only
Summary	Inactive Summary	Finish-only
Project Summary	Manual Task	Deadline
External Tasks	Duration-only	Progress

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