EML4552 - Spring 2009

Senior Design

Group 12

Bevel Gear Testbed

Final Analysis

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Abstract

Harris Corporation has requested the design and development of a fully adjustable bevel gear testbed capable of testing the life-cycle of a wide range of bevel gear sizes, styles and materials while meeting several alignment and loading characteristics. The foundation for this project is the result of a previous industrial application at Harris Corp. where the bevel gears involved failed well before the expected life cycle. The possible problems of this failure include but are not limited to misalignment of the shafts, anodic coating failure on the gears, and overloading of the gears. While these complications from previous experience are a concern, Harris Corp. has also asked that the speed of the input gear be controlled up to 100 rpm, the resistive load on the output gear be controlled up to 50 inch-pounds, the mounting distance of the gears be controlled within ± 0.001 inch, the gears be able to rotate up to at least ± ½ degree both clockwise and counter-clockwise accurate to ± 1/20 of a degree, and that the testbed be capable of testing gears from 1/3 inch to 5 inches in diameter. In addition to the product specifications, the entire system must not exceed $1500.00.

The testbed fabrication has been designed and built primarily to accommodate these considerations and finished $64.10 under budget. The finished testbed currently meets 4 of the original 6 product specifications, including the variable speed, variable shaft angle, shaft angle accuracy, and gear size range. The mounting distance accuracy is over specification by less than ± 0.00005 inch due to the propagation of error in the alignment table and calibration device; both components are within specification individually, but when used in conjunction, the accuracy cannot be guaranteed within specification. Additionally the variable torque requirement has yet to be met due to pending electrical impediments with the motor controller. Once this complication is resolved however, the torque requirement is expected to be within specification.

The primary concerns in the coming week are achieving the variable torque specification by correctly operating the motor controller and collecting and analyzing adequate data to determine the effectiveness of the testbed. These concerns and the results of further testing will be addressed in an addendum submitted on the date of open house.

The Harris Corporation

Harris Corporation is an international communications and information technology company, founded on December 23, 1895 as the Harris Automatic Press Company in Niles, OH. Through the years HAPC began to acquire various communications technologies companies until the 1970s when the name was changed to the Harris Corporation and the corporate headquarters moved to Melbourne, FL. Harris serves government and commercial markets in more than 150 countries, with annual revenues of $5.3 billion and the employment of 16,500 employees, including about 7,000 Engineers and Scientists. The primary contact for this project is Brent Stancil, a Mechanical Engineer working for Harris Corporation. Previously, Harris Corp. tested bevel gears for a similar project and failed to achieve the expected results. The possible problems included: misalignment of the shafts, anodic coating failure on the gears, and overloading of the gears.

Bevel Gears

The most common and convenient way of changing the direction of shaft rotation is by the use of bevel gears. Typically the shafts are aligned perpendicularly, but the gears can be designed to work at any desired shaft angle. The teeth of the gears themselves can be straight, spiral, or hypoid depending on the application for the gears and ease of design.

The pitch surface and pitch angle are very important concepts in bevel gear applications. The pitch surface of the gear is an imaginary toothless surface that results from averaging the peaks and valleys of the individual teeth. Usually the pitch surface of a gear resembles the shape of a cut-off cone. The pitch angle of a gear is the angle between the face of the pitch surface and the axis.

Bevel gear applications can be found on locomotives, marine applications, automobiles (differentials), printing presses, and power plants. One can also find bevel gear applications in any common hand drill.

Bevel Gear Design Calculations

In the design of any bevel gear application, the forces exerted on the entire gear train must be accounted for. Depending on the range of torque being transmitted into the gears as well as the size and gear ratio of the application, the gears may experience a tangential, radial, and axial force. The magnitudes of these forces are the primary influence in designing the shaft strengths required and the bearings that must be used. The design calculations used to select the proper shaft material, shaft diameter and bearings for this senior design project can be found in Appendix I.

Senior Design Project

The purpose of this senior design project is to design and build a fully adjustable bevel gear test bed for the Harris Corporation. Harris Corp. sponsored a similar senior design project at a different university, but the bevel gears being tested failed sooner than their expected life cycles. There are some possibilities as to why the previous system failed to achieve the expected results. Some of these possibilities include overloading the gears, misalignment in the gears, and/or a breakdown of the anodic coating on the surface of the teeth causing the teeth to become abrasive. It is unlikely that any of these problems were accounted for in the initial design calculations of the previous group, and it is part of the scope of the current project to prevent these possible problems.

While trying to prevent the problems of the previous group is a concern, the initial product specifications (Table 1) given by the sponsor of this project were quite demanding. Due to budget restrictions and high degree of difficulty, the product specifications were revised (Table 2). The primary difference between the initial and final product specifications is the reduction of the motor torque and speed. Due to a budget restriction of $1500.00, a set of motors to run at this specification is far too much for the budget of this project.

Table 1: Initial Product Specifications

Specifications	U.S. Units
Variable Speed	0 rpm – 1000 rpm
Variable Torque	0 in·lb - 100 in·lb
Mounting Distance Accuracy	± 0.001 in.
Variable Shaft Angle Range	± 0.5 degree
Shaft Angle Increments	± 0.001 degree
Gear Size Range	1/3 in. – 5 in.

Table 2: Final Product Specifications

Specifications	U.S. Units
Variable Speed	0 rpm – 100 rpm
Variable Torque	0 in·lb - 50 in·lb
Mounting Distance Accuracy	± 0.001 in.
Variable Shaft Angle Range	± 0.5 degree
Shaft Angle Increments	± 0.05 degree
Gear Size Range	1/3 in. – 5 in.

Concept Generation

The initial concept generation included six general design ideas. The focus was on how to adjust the walls supporting the gear shafts. The first concept is a drawer slider concept which operates on the same concept as a household drawer. The wall translates in one direction within a specific geometric design. The second concept incorporates the rotation of the walls. The wall rotates along an angular slotted path. The third concept is a combination of the first two concepts, having the wall itself translate and rotate. The fourth concept is similar to the drawer slider except instead of the wall being flat and the slider being similar to a drawer slider, the wall will be curved to make the angular movement of the shafts easier. In addition, the slider will be round to reduce the contact area between the slider and surrounding material, thus reducing friction. The fifth concept is a rack and pinion style design with two varying connection methods. The first has a driving pinion and second a driven pinion. The last conceptual design is a worm gear design where a worm gear is in mesh with a spur gear and will produce translational motion of the walls.

Drawer-Slider

The drawer slider design is a simple translational motion design that is an imitation of the design of a drawer slider. The slider moves along a horizontal track. It moves with a relatively smooth motion but will be hard to control to within 0.001 inch. In addition, the design does not consider the desired rotational motion. Figure 1 below is an illustration of the drawer slider design.

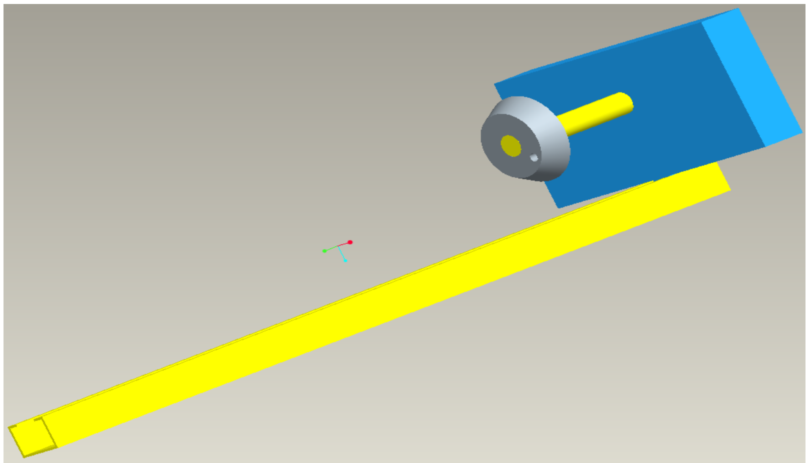

Figure 1: Drawer-Slider Design

Rotator

The rotator design is a simple rotational motion design. The rotator moves along a circular track in the horizontal plane. It moves with a relatively smooth motion but fails to incorporate the desired translational motion. Figure 2 is an illustration of the rotator design.

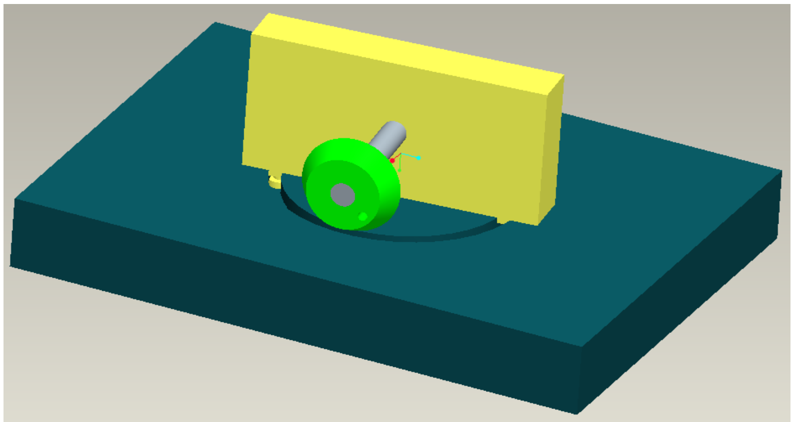

Figure 2: Rotator Design

Rotator-Slider

The rotator-slider design is a combination of the rotator and slider design. The translational portion of the motion is accomplished by a horizontal slot in the lower base plate. The rotational portion of the motion is achieved through a circular slot in the upper base plate. Figure 3 is an illustration of the rotator-slider design.

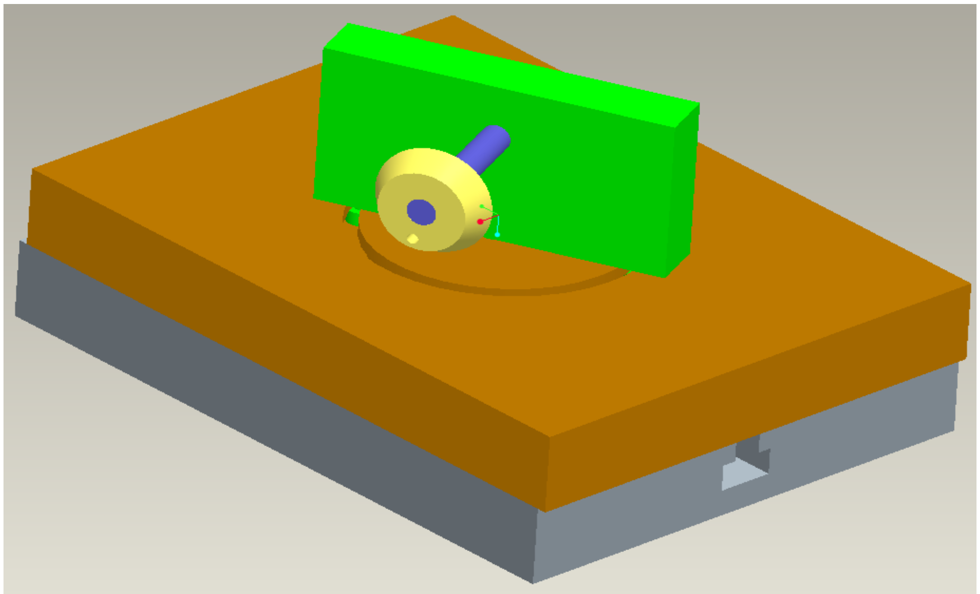

Figure 3: Rotator-Slider Design

Curved Wall

The curved wall design is closely based on the design concept of the drawer slider wall idea. This design concept will use the same type of slider concept to adjust the mounting distance of the bevel gears being tested. However instead of a square cross section being used for the slider piece, a circular slider piece will be used. This is because a rounded slider piece should be less difficult to machine to within a 0.001 inch tolerance. The drawer slider concept will achieve the rotational motion by some method of rotating the wall while still being able to adjust mounting distance. The problem is that it adds extra parts to the overall system, which will increase the degree of difficulty for machining and assembly of the test bed.

By adding curvature to the wall, the need for extra parts that allow the wall to rotate will be eliminated. Based on a calculated radius of curvature and a rectangular shaft slot, the gear shaft can be set to a pivot point to achieve the slight shaft angles requested by the client. Having a curved wall with an exact radius of curvature will increase the difficulty level in machining the testbed. Figure 4 shows how the curved wall design would look.

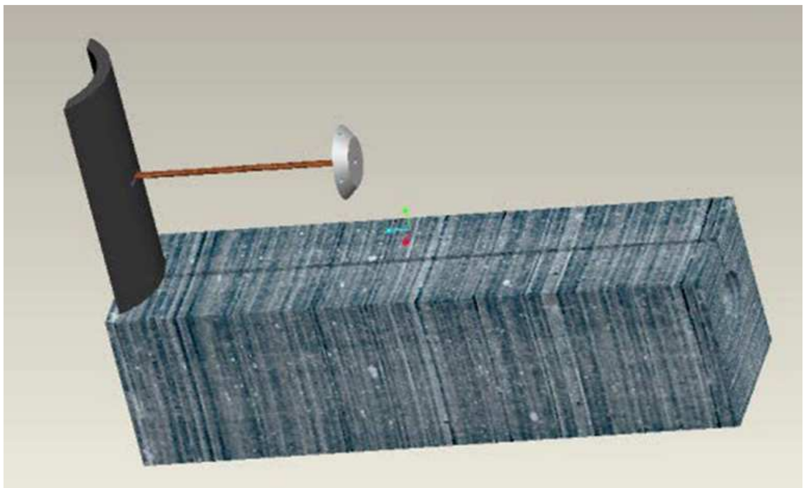

Figure 4: Curved Wall Design

<u>Rack-and-Pinion</u>

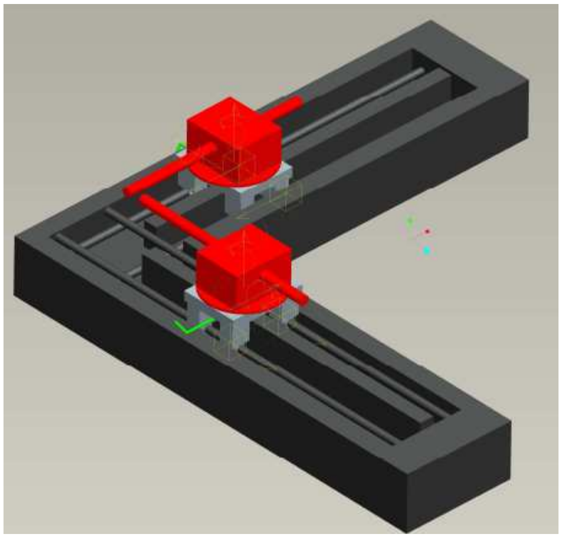

Figure 5: Rack-and-Pinion - Concept A

Concept A

The first idea is to mount the gear-shaft housing on a rack-and-pinion style gear assembly. In this concept, a pinion is mounted underneath a gear-shaft housing in mesh with a static rack, mounted to the base of the gearbox. To move the gear-shaft housing there will be a crank connected directly to the pinion which allows for translational movement with an excellent degree of control along the stationary pinion based on the translational rotation. The precise translational movement associated with a particular rotation of the crank can be determined empirically, or can be achieved through the use of a digital sensor to measure the displacement of the gear-shaft housing. The benefit of this system is that it would be relatively simple to assemble as there is only one moving part in the gear train to translate the housing. However this is also the biggest disadvantage of the system, since having a pinion being both the driving and driven gear poses reliability concerns and may produce a lot of backlash and unnecessary wear on the teeth of the gears.

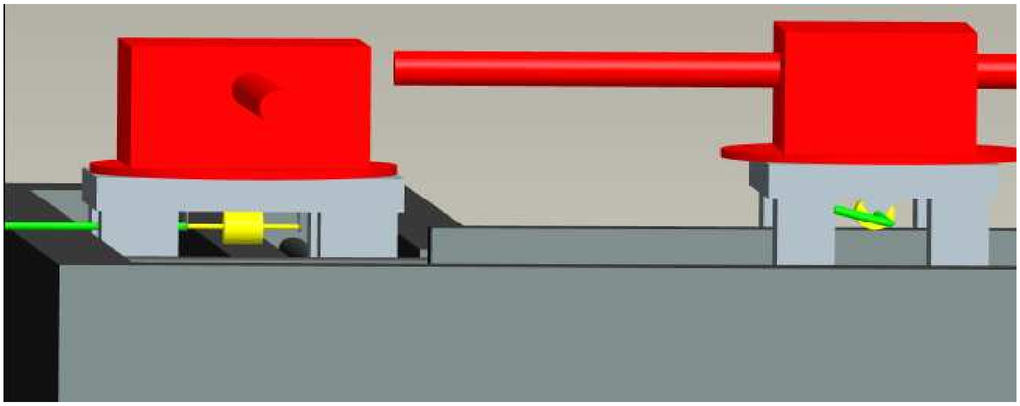

Figure 6: Close up of Rack-and-Pinion - Concept A

Concept B

The second of the two ideas is a more traditional rack-and-pinion where the pinion is mounted onto the primary baseplate of the gearbox and the rack in this case will be attached underneath the gear-shaft housing. A crank will remain attached to the pinion to control the translational motion of the housing, although instead of a driving pinion rotating and translating on a stationary rack, this concept has a rack being driven by a pinion to achieve translational motion. The advantage of this concept is that it is a more traditional use of a rack-and-pinion and would thus not pose as large of a reliability concern. The drawback of this design is that the rack needs to be much longer than the housing wall to achieve the full range of motion necessary for the mounting and interchanging of different gear sets within the test bed, thus making the testbed much more cumbersome.

<u>*Worm Gear*</u>

The alternative to the rack-and-pinion ideas is to replace the rack-and-pinion with a worm gear and mated to a pinion. The worm gear is mounted within the base plate of the gearbox with a crank to control the translational motion of the pinion, which would be mounted underneath the gear-shaft housing, similarly to *Concept A* above. The difference from this concept and *Concept A* is that the worm gear is the driving gear and the pinion need only be driven. Since the worm gear is always in direct contact with its mating pinion, there is absolutely no backlash in the gear set and therefore, extremely high precision can be achieved with repeatability. The displacement associated with each revolution of the crank can be determined empirically or via a digital sensor, both of which would be simple enough to execute and as such, the digital sensor will most likely be preferred, but most likely be extra strain on the budget.

The biggest advantage of the worm gear assembly over the rack-and-pinion assembly is the repeatability of motion without compromising accuracy. Since there is no backlash in the worm gear, interchanging and re-aligning different gears and/or shafts can be done with a high degree of precision. The biggest disadvantage of the worm gear is that it will likely cost more than the rack-and-pinion, since it requires its own bearings and alignment, whereas the rack-and-pinion needs only one set of bearings for the pinion. Despite these differences, both gear sets can provide highly accurate methods of producing translational motion for the gear-shaft housing. However, the worm gear will be more expensive.

Commercially Ordered Alignment Tables

Since the design and assembly of an alignment system is not only expensive, but involves a high degree of precision, the best decision is to instead buy a commercially available alignment table. Since the mounting distance accuracy is the driving factor, a highly precise translational table is needed. To provide this type of precision alignment, a good option is a micrometer linear motion stage. Figure 7 below shows a UMR8.25 Micrometer linear stage available from Newport for $278.46 which will meets the mounting distance accuracy parameter of the system. However this product can only support a 17N load axially (see product specifications in Appendix II under UMR Micrometer Stage) while the calculated gear loading for the gears being tested is over 18N (see design calculations in Appendix I). This is only for the largest gear size being tested in this project which is much smaller than the maximum gear size this testbed is capable/required to accommodate.

Figure 7: UMR Series Precision Double-Row Ball Bearing Linear Stage

To account for the gear loading and still allow a mounting distance accuracy of ± 0.001 inch, a Cross-Slide Rotary Table (see Figure 11) is the primary alignment component in this testbed. More on the Cross-Slide Rotary Table will be discussed in the Current Design section.

Gear-to-Shaft Connections

Since every aspect in the design is affected by the mounting distance precision (±0.001 in.), precision in the gear-to-shaft connections is just as important as in mounting and alignment. To ensure that each bevel gear being tested is connected to its accompanying shaft precisely and is able to be run within the gear-shaft housing, several connection methods have been investigated. The different types of connections are as follows:

Concept 1 – Variable shaft diameters with an adjustable chuck

The first concept design for gear connection is to allow for variable shaft diameters as per variable gear sizes. This is achieved by mounting an adjustable chuck onto a fixed diameter shaft. From this a variable diameter shaft could be interchanged from the chuck depending on the size of the gear. This allows for the gearbox to integrate any sized gear-shaft pair into the testbed as long as it fits within the space provided, and the shaft fits inside the chuck. This method requires that the chuck be fixed to the motor and inside a large accompanying bearing to account for the added weight of the chuck, to ensure reliability. This is a very simple method of interchanging different sized shafts. The biggest drawback for this idea is the precision of the chuck, since it is unlikely that it would be precise to 0.001 inch, thus ruining the alignment of the gears. Upon further investigation, it is decided that a high precision chuck would be quite difficult to find and afford, while the manufacturing of a custom chuck would be much too complex and time-consuming for the group to take on.

Concept 2 – Constant shaft diameter with variable shaft adapters

The second concept is to use a constant diameter large stock shaft and supplementary bearings. Variable diameter adapters are machined to mount onto the input and output shafts to fit any size bevel gear being tested, as shown in Figure 8 below. This method requires that a new adapter be machined for each size gear-shaft to be tested, which costs time and money; however it is less costly in terms of both time and money than designing and manufacturing of an adjustable chuck. In addition, if done properly, this method should be much more accurate than an adjustable chuck.

Figure 8: Gear to Shaft Adapters

The idea is to drill a tapped hole into the adapter with an accompanying flat section on the shaft that connects by way of a set screw. This approach will be simple to machine and can be machined to the required accuracy.

The most likely approach is to go with Concept 2 since it will be simplest to assemble, and is also the most reliable without too much unnecessary cost. It will be very rigid, and the mounting distance is capable of being controlled to the necessary tolerance. It also is not very expensive in comparison to purchasing or machining a chuck, and allows for variable gear-shaft sizes.

Current Design

The current design is much more intricate than the initial concept designs. This is expected however. A detailed explanation of the primary componentry required to meet the design parameters is explained below with a labeled image of the finished testbed at the end of the section.

The system has a 24 volt DC input motor (Figure 9) to supply the pinion a variable speed, along with a 24 volt DC resistive motor to supply the gear with a variable torque. The torque acts as a resistive load on the gear system to simulate active loading on the gears. The variable torque is supplied to the DC motor via a potentiometer connected to a motor controller. By varying the resistance on the motor controller, the current can be regulated to control the torque output of the resistive motor, since motor torque is based on current supplied. The controller used for this project is a Brush Type PWM Servo Amplifier, supplied by Advanced Motion Controls (Figure 10).

Figure 9: Pittman 24 Volt DC Motor

Figure 10: AMC – Brush Type PWM Servo Amplifier

According the product specifications in Table 2, the mounting distance on the gears must be accurate to ± 0.001 inch, and the shaft angle variation requires a range of at least ± 0.5 degrees accurate to ± 0.05 degrees. Since this tolerance is far too tight for the machine shop at the College of Engineering, a Cross-Slide Rotary Table (CSRT) from McMaster-Carr (Figure 11) is the primary component in the design. The Cross-Slide Rotary Table provides motion in the transverse and lateral directions, as well as rotation about the vertical, while still providing the required precision for alignment. Both the transverse and lateral motion of the CSRT are capable of 4 inches of travel accurate to 0.001 inch, and the rotation is accurate to 0.05 degrees. In addition, the CSRT is built to handle higher loading from the gears (approximately 20 lbf radially), as opposed to a micrometer stage which can technically meet the design parameters, but not withstand the subsequent loading. According to McMaster-Carr, "You can bolt these tables to your drill press, milling machine, or grinder for milling, routing, shaping, grinding, and cutting slots and keyways." (McMaster-Carr Catalog pg. 2531) Unfortunately, despite the relative robustness of the CSRT, the table does have a slight amount of play in the rotational component, which is clearly undesirable. Fortunately the table can be locked down via two black ¼-20 socket head screws, eliminating a majority of the wobble.

Figure 11: Cross-Slide Rotary Table (CSRT)

-Material	*Cast Iron*
-Surface Finish	*Ground*
-Height	*5 ½"*
-Base Size	*6 ½" x 7 ½"*
-Table Diameter	*8"*
-Keyway Size	*$^{5}/_{8}$"*
-Gear Ratio	*40:1*
-Mounting Holes Diameter	*½"*
-Mounting Holes Quantity	*4*
-Table Travel (X x Y)	*4" x 4"*
-Translational Graduations	*0.001"*

-Rotational Range	*360º*
-Rotational Graduations	*0.05"*

While the CSRT itself is accurate to the required specification (± 0.001 inch) the accuracy of the milling machine and lathe used to fabricate the remaining components is approximately ± 0.005 inch (does not meet the specifications). This can be corrected by calibrating the reference point of the bevel gears by the use of an edge-finder. The design uses a modified electronic edge-finder supplied by McMaster-Carr (Figure 12) to calibrate a local reference for the alignment of the gears being tested. Since the edge-finder is mounted horizontally on the shaft, a conventional edge-finder will not work in this circumstance. In addition an electronic edge-finder also provides an extremely high tolerance (± 0.0003 inch) while remaining extremely user-friendly and requires no rotation of the shaft. The modification to the edge-finder is simply removing the internal battery casing for an external battery casing to allow the edge-finder to fit on the shaft better, and is also inset in an adapter to slide over top of the tooling that connects the gears to the shafts, as seen in Figure 12 below. For more information on how the edge-finder works, reference Steps 18-21 of the "Assembly and Calibration Manual".

Figure 12: Electronic Edge-Finder

The majority of the fabrication for this project was done at the College of Engineering Machine Shop as well as at the NHMFL (Magnet Lab). The various other components, such as the gears and bearings, are located in Appendix II under Additional Components.

To monitor the variable speed of the input shaft/pinion, a digital optical tachometer which can monitor the rpm without contact is used. According to the manufacturer, this tachometer (see Figure 13 below) can measure a range of 2 rpm - 99,999 rpm and is accurate to ± (0.05% reading + 1 digit) with a resolution of 0.1 rpm. (Calright Instruments) The speed in rpm from the tachometer is used to determine the length of time necessary to run a gear set for a certain number of cycles, as well as used as a reference to observe the torque supplied, as seen in Appendix II.

Figure 13: Laser Tachometer from Calright Instruments

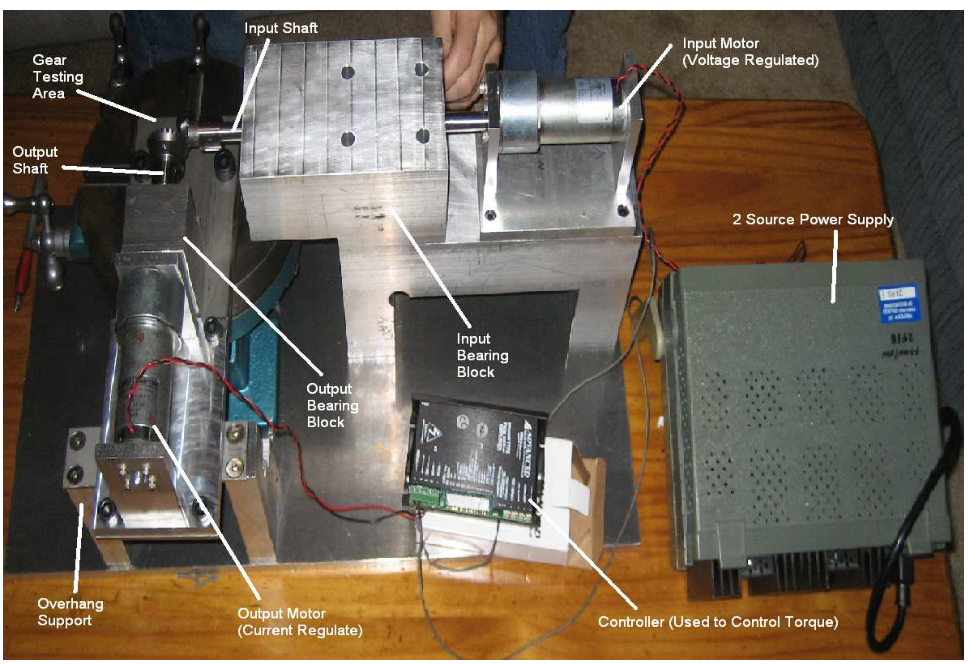

Figure 14: Final Design Assembly and Componentry

Cost Analysis

Upon completion of the initial design, the project exceeded the budget by more than $500.00. The majority of this cost was based on purchasing gears, motors, and micrometer translational stages. To correct this problem, Harris Corp. was able to supply three gear sets and several motors for free. In addition this, the micrometer translation tables were removed from the design, in favor of the Cross-Slide Rotary Table, which is significantly less expensive than a set of micrometer stages. As can be seen in Figure 15 below, the project is currently under budget, including all of the necessary components.

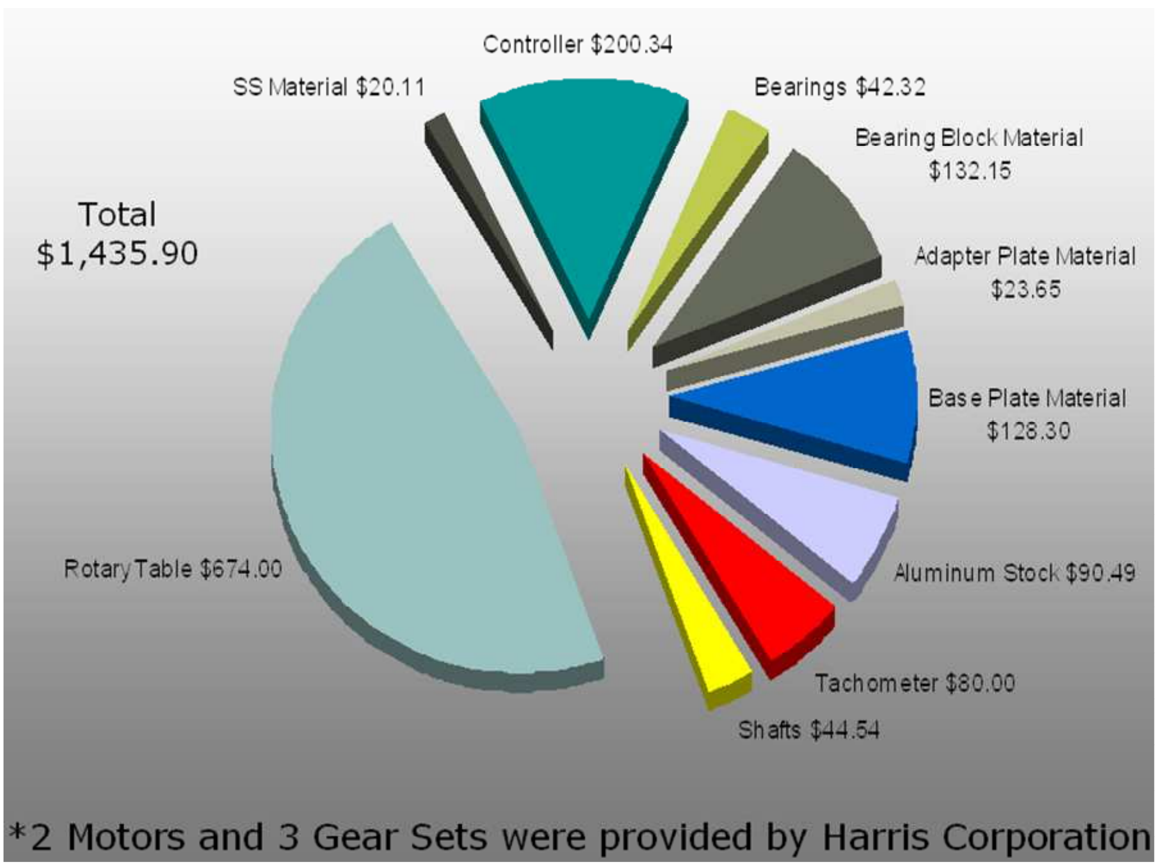

Figure 15: Cost Analysis Summary

Testing and Analysis

Due to the time constraints in fabricating and assembling the testbed, little to no parametric testing has been done at present. The majority of the testing done thus far is essentially troubleshooting. Ensuring that the assembly components work/fit as expected and that the gears can be run in mesh is the primary concern of testing presently, although no data has been collected thus far. Additionally the shaft axes have been properly aligned to exactly ninety degrees using an edge-feeler gauge accurate to ± 0.0005 inch (well within the angular precision of ± 0.005 inch). As such an addendum will be submitted at a later date with more conclusive data comparing the testbed results to the product specifications.

Determining the resistive torque load on the gears, the motor controller must supply the resistive motor with a controlled variable current. To determine the total resistive load on the gears, a combination of resistance loads are combined. Even when the resistive motor is not resisting the rotation of the gears, there is still a resistive load associated with turning a shaft against a motor. This resistive load is determined by balancing a beam of known radius with a known mass attached to the end of it, to determine the torque required to turn the shaft inside the motor. This is the unloaded resistance of the system. When the resistive motor is running however, there is significantly more resistive loading on the gears, which is difficult to quantify empirically. This can be done however by using a Torque vs. Current and RPM plot supplied by the motor manufacturer which is located in Appendix II. Combining these two loads yields the total resistive torque applied to the gears.

Propositions for Improvement

Precision Alignment Resolution

A major concern in the fabrication of this testbed is the propagation of error in the individual components. Each machined component is accurate to ~ ± 0.005 inch. This deviation compounds with every piece attached to another, resulting in a tolerance well outside the range deemed acceptable for this design. To combat this problem, an edge-finder is used to minimize the error propagation in the transverse and lateral (x and y) directions, however any deviation from alignment in the vertical is controlled only by the accuracy of the machining process. There are a number of things capable of improving the alignment resolution, and are described as follows.

One of the simplest and cheapest improvements to the alignment precision would be to plane every surface of every component used to construct the testbed. Many of the raw materials received were assumed to be flat, based on the look and feel of the cut. Instead of taking this assumption as relative truth, it would have been beneficial to properly face each surface of the components from the original material stock. A prime example of this is the baseplate, which is made of solid aluminum, ½ inch thick. The baseplate appears to have planar surfaces, but after considerable study it can be seen that it is in fact somewhat warped. Regardless of this and several other flaws in the materials, all of the components still fit together and appear to operate smoothly. It is because of this reason that the problem posed by this is hard to quantify, but it is likely that this concern can eventually become problematic after long duration tests at high loading.

Another possible improvement in the alignment resolution is to try to replace several parts connected together with one solid piece of stock. This also is not too expensive of a design option, though still more expensive, but would have been much more difficult to do based on the geometry of the testbed. The input assembly especially would have been hard to fabricate as such a large piece of material out of one solid piece. The difficulty in this is rather large, in addition to increased cost, since the majority of the current bed is made from recycled material stock. This method, although more accurate, is more expensive in terms of adding more wasted material to the budget, and less practical due to increased degree of difficulty in machining.

A third improvement to the fabrication precision is to incorporate alignment pins into the assembly process. Since alignment pins align components to an extremely high precision, the uncertainty in the assembly precision is greatly reduced. This is because without the use of alignment pins, the components are aligned only by the surfaces and screws holding them together. Conversely with alignment pins the components are still held together by the screws, but the screws/threads are not used for alignment, only rigidity. While it is usually hard to visibly notice any misalignment from screw-based alignment, there is a pre-determined acceptable amount or error in screws/threads, which can cause mild deviations from the design alignment. Alignment pins have a much tighter tolerance than any screw threads and as such are a much more precise method of aligning components.

The single greatest improvement to the precision alignment resolution is to build the entire testbed on more accurate machinery. If enough of the budget was allotted to outsourcing the machine work to a shop with more precise tolerances, the components in the system would have higher precision. The best case scenario would be for each of the components to be fabricated on an automated CNC milling machine, since these machines usually have the highest precision, because they eliminate user error. Essentially the best, yet most expensive, way to improve the alignment precision is to have a more experienced machinist with better equipment fabricate the design.

Independent X – Y Translation Slides

A major flaw in the current testbed is the inability to rotate the gear alignment about the pitch cone center (PCC). This is a result of the method of manually positioning the gears for alignment, by use of the Cross-Slide Rotary Table. The CSRT is the primary component in the design of this testbed because it can handle the maximum gear loading with relative ease, while still meeting the alignment criteria and fitting well into the budget. The problem with the table is that only the output assembly can move, therefore unless the PCC is perfectly aligned with the center of the rotary table, any rotation of the gears will be non-uniform. Basically since the purpose of the rotary component in the design is to allow for controlled misalignment in testing, and the PCC might not always line up over center, then it is very hard to control that misalignment. To improve this, the best approach is to provide translational motion independently for the input and output assemblies, with one of the two having a rotational component. This requires two linear stages each with only one direction of travel and one rotary table, able to connect in some manner to one of the linear stages. In addition, each of these motion tables must be able to withstand the maximum gear forces and have a method of locking down once aligned, unless there is significant resistance to back-driving forces. This is a feasible improvement, but based on the budget of the project it is too expensive to provide such a system of motion

Increased RPM and Resistive Torque:

While the speed and torque requirements on this project had to be greatly reduced on account of the budget limitations, realistically Harris requires specifications on the order of the initial requirements (speed x 10 and torque x 2) to obtain testing results worth obtaining. This is a simple fix however, requiring little more than the funding to purchase a more powerful motor and perhaps redesigned motor mounts to fit the new motor.

Increased Range of Gear Sizes:

Due to the total translational displacement of the CSRT, the range of gear sizes capable of being tested is limited. By implementing a larger translation table the gear size range for testing increases. This would require a larger budget and more time to integrate the new table into the current design. The range of gear sizes able to be tested currently just barely meets the product specifications on the larger end (~5.0 inch).

Possible Future Plans

There are a few additions that can be added to the test bed to make it more useful in testing for a wide range of common gearing problems. Due to a very tight budget, the main objective for this project is to design, fabricate and assemble a fully adjustable bevel gear testbed to operate to tight mounting specifications and variable speed and loading requirements. Anything beyond this scope is an extended goal, and as such is more of a future consideration than a concern of this project. Additionally the budget of the project does not adequately accommodate the implementation of the following.

Vibration Testing:

By implementing accelerometers at certain key areas of the test bed, critical vibrations (chatter) can be quantified. This could prove helpful in determining the magnitude of the disturbance forces that may add unnecessary loading on the gears. Additionally if the magnitudes of vibration in the testbed are quite large, there will be significant chatter in the gears, reducing the life span of the gears being tested. The use of accelerometers in this project was limited primarily by budget, in addition to unwarranted benefit vs. the added complexity. While implementing the accelerometer is not particularly complex (simply rigidly connect it to any region of the system worth studying). The complexity is mostly associated with analyzing the data it produces. An example of one such accelerometer is the Premium Grade, Low Profile Accelerometer from Omega for $315.00, as shown in Figure 16. (*Omega.com*)

Figure 16: Accelerometer from Omega.com

Heat Generation:

The heat generated by two gears in mesh operating for a long range of time can eventually induce significant loading, based on a wide range of factors. Misalignment or excessive chatter in the gears and breakdown of surface coating, leading to abrasion can play a significant role in the heat generation in a gear train. Gear materials with a large coefficient of expansion and/or a large heat capacity are especially vulnerable to excess loading by heat generation. The gears in this project are all stainless steel and running at relatively low speed, so heat generation for the purposes of this system is negligible. However for a company like Harris Corp. which requires high speed, lightweight gears, the heat generated by the gears is a significant design consideration. The simplest way to measure the heat generated by the gears is to shine an infrared laser on the teeth in mesh, calibrated over the range of temperatures that the gears are expected to run at. The infrared laser will allow the user to observe a temperature increase, which can be directly related to the heat generated between the gears. A commercial infrared laser capable of this is widely available at many hardware stores, such as Home Depot for $49.97. (*HomeDepot.com*) Although these systems are cheap and easy to implement, they are used mostly for quick reference temperatures and not to be taken as absolute. To get a more accurate reading, thermal sensors such as thermocouples can be attached to the gears, although this would require the design of a set of rotating wires, which complicates the design.

Figure 17: Infrared Laser Thermometer

Rotational Alignment Calibration:

While the calibration of the translational alignment is controlled by the incorporation of the edge finder, there is currently no precision method to control the alignment of the rotational component of the testbed. The only method used currently to control the rotational precision is the use of a feeler gauge to verify that the input and output shafts are perpendicular, but this method will not work for gears that require atypical mounting angles. The Cross-Slide Rotary Table can adjust the vertical rotation to ± 0.05 degree, however since the pitch cone center is not always over the center of the CSRT, the rotation of the CSRT may not be equal to the rotation at the PCC.

One method to allow for non-ninety degree mounting angles is to machine a block of material that has close fit holes drilled at the specified mounting angle. Using this block, the shafts can be aligned to relatively close precision at any specified mounting angle, assuming the block is machined with close fit holes. Although this is a relatively simple approach, it is still slightly difficult to accurately machine a block of material at a given non-ninety degree angle. The disadvantage is the need for a new block of material for every mounting angle desired for testing.

The best method to align the rotational component of the assembly is to use a professional alignment system. Modern professional systems use laser alignment for precision that typically cannot be replicated with mechanical components. These systems are well outside of the budget of this project, but can vastly improve the precision well above the required specifications in both manufacturing and alignment.

Conclusion

While fabrication of the testbed is finished and most of the design parameters are met, the conclusive results are limited. This is largely due to the lack of physical data and some unresolved issues. Calibrating the reference origin for the input and output shafts using the edge-finder and controlling and quantifying the resistive motor torque are the main unresolved issues at present. Data acquisition is primarily dependent on solving the motor controller problem.

Using the edge-finder to calibrate the x-y translational reference locations (the origin) is unresolved simply because the edge-finder bulb failed due to a short in the ground wire terminal after it was modified to fit the testbed. Since the bulb cannot be replaced, a replacement edge-finder is currently in transit. Once the new edge-finder is obtained, this will be a non-issue as long as more care is taken to electrically insulate the terminals.

To control the resistive torque according to the parameter specifications, the motor controller uses a potentiometer which is essentially a variable resistor, by varying the current supplied to the resistive motor. Unfortunately, the most recent test of the controller was unsuccessful in providing a variable current by means of a varying resistance. Essentially the potentiometer resistance was adjusted with no change in current supplied to the resistive motor. To solve this problem, the controller vender, Advanced Motion Controls, will be contacted for further technical support. Additionally the quantification of the torque load supplied to the gears by the resistive motor is provided in the method described in the Testing and Analysis section of this report. This however is dependent on solving the issue in operating the motor controller.

The data collection is scheduled to take place this weekend and the days leading up to open house. As stated in the Testing and Analysis section, an addendum will be provided at a later date, providing empirical data comparing various runs of the gears, including loaded and unloaded gear trials (pending motor controller functionality), and life cycle testing. The gears have been run for an indefinite length of time (~3 hours) and have run quite smooth, without considering any of the specific parameters other than verifying the rpm with the optical tachometer. This is solely for the knowledge that the system works as well as troubleshooting, rather than trying to provide any substantial data.

As previously stated a majority of the design specifications (Table 2) are accounted for in the fabricated testbed. The variable speed (0-100 rpm), variable shaft angle range (± 0.5 degree), shaft angle accuracy (± 0.05 degree) and gear size range (1/3 in. – 5 in.) are all attainable in the current system. Only the variable torque (0 in·lb - 50 in·lb) and mounting distance accuracy (± 0.001 in.) have yet to be met. The torque specification has yet to be met on account of the problems with the motor controller, discussed previously. There is a strong belief however that once the motor controller is wired properly and its operation fully understood, that this requirement will be met.

The deviation in the mounting distance accuracy is primarily the result of error propagation in the componentry. The linear translation tables on the CSRT are accurate to ± 0.001 inch in each direction. This is just the accuracy of the table itself and does not account for the accuracy of the edge-finder which is used in conjunction with the CSRT to reference the local origins of the translational axes. If the edge-finder accuracy is included (± 0.0003 in.) and using the root of the squares of these accuracies is considered, the total mounting distance accuracy is ± 0.001044 inch. While this is a very tight tolerance for the purpose of testing gears in the speed and torque range of this project, it is still outside the specifications from the sponsor. This is due in large part to budget driven design. Since the Cross-Slide Rotary Table by itself just barely exceeds the mounting distance specification of the project, this table cannot be used to meet this parameter. However, this table is approved by the sponsor (Brent Stancil) since it is the only table found that can approximate the accuracy requirements while still handle the gear loads and remain within the budget for the project.

Despite the various parametric constraints in the design and development of this project, a majority of the design parameters have been met and a fully functional testbed is ready for submission. In addition, an addendum will be submitted before Open House to provide more information on the experimental results of the testbed and how it meets life cycle expectations. This is a very challenging problem and a great deal has been learned in designing, developing and troubleshooting this project, in addition to providing an excellent opportunity to practice semi-formal, industry-type presentations and progress reports.

Acknowledgements

Dr. Hollis

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- Design guidance

Brent Stancil

- Providing the motors and gear sets
- Clarifying critical specifications
- Teleconference meetings for further design guidance

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- Machining Guidance

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- Pro-E assistance
- Supply of miscellaneous hardware
- Machining Assistance

Angela Scharnetski / Rene Ymzon

- Providing a student discount on the controller
- Controller application and implementation guidanc

References

1. Harris history and background: http://www.harris.com/company-history.html
2. General gear information: http://science.howstuffwork.com/gear4.htm
3. Rotary Table Information: http://www.mcmaster.com/#rotary-tables/=1bx20r
4. Accelerometer Information: http://www.omega.com/ppt/pptsc.asp?ref=ACC797&Nav=prek02
5. Gear Ordering Information: http://www.bostongear.com
6. Gear Ordering Information: http://www.wmberg.com
7. Tachometer Ordering Information: http://www.calright.com/pd_1055.aspx
8. Infrared Laser Thermometer: http://www.homedepot.com/webapp/wcs/stores/servlet/ProductDisplay?storeId=10051&langId=-1&catalogId=10053&productId=100651817&N=10000003+90401+500061

Appendix I

Design Calculations:

Gear Calculations

$T_{pmax} := 10in \cdot lbf$ $\quad rpm := \frac{2 \cdot \pi}{min}$ $\quad$ *http://www.wmberg.com/catalog/product.aspx*

$\omega_p := 25rpm$ $\quad d_p := 1.25in$ $\quad r_p := \frac{d_p}{2}$ $\quad r_g := r_p$

Bore : 1/2
Material : Aluminum Anodized (Pinion)/Aluminum Anodized (Gear)
Type : MITER & BEVEL GEARS
Style : PIN HUB
Pitch Diameter : 1.250 (PINION) / 1.250 (GEAR)
Hub Diameter : 1
Set Screw Location : 1/4
Ratio : 1 to 1
Module/Pitch : 16
Hub Length : 7/16
Pressure Angle : 20
AGMA Quality : 10
Teeth : 20/20

Mounting Distance (Gear) : 1.25
Mounting Distance (Pinion) : 1.25
Face Width : 5/16
PDF File name : B05B224
Catalog Page Number : B224

$K_a := 1$ (Application factor, assumed to be "1" for generally smooth-running devices)

$K_s := 1$ Size factor, assumed to be "1" unless the designer wants to raise the value, usually to accomodate for very large teeth.

$V_t := r_p \cdot \omega_p \cdot \frac{s}{m}$ $\quad V_t = 0.042$ $\quad Q_v := 10$

M16P-1A
BEVEL GEAR SET

$B := \frac{(12 - Q_v)^{\frac{2}{3}}}{4}$ (for Qv between 6 and 11) $\quad A_1 := 50 + 56 \cdot (1 - B)$

$K_v := \left(\frac{A_1}{A_1 + \sqrt{200 \cdot V_t}}\right)^B$ $\quad K_v = 0.987$

$K_m := 1.6$

$F_1 := \frac{5}{16} in$ $\quad N_p := 20$ $\quad N_g := 20$ $\quad \phi := 20deg$

$\nu_p := .345$ $\quad \nu_g := .345$ $\quad x_p := 0$ $\quad E_p := 70GPa$ $\quad E_g := 70GPa$

$\alpha_p := 45deg$ $\quad \alpha_g := 45deg$ $\quad p_d := \frac{16}{in}$ $\quad m_1 := \frac{1}{p_d}$

$$C_p := \sqrt{\frac{1}{\pi \cdot \left[\left(\frac{1 - \nu_p^2}{E_p}\right) + \left(\frac{1 - \nu_g^2}{E_g}\right)\right]}} \qquad C_p = 1.125 \times 10^5 \frac{kg^{0.5}}{m^{0.5} \cdot s}$$

$$\rho_p := \sqrt{\left(r_p + \frac{1 + x_p}{p_d}\right)^2 - \left(r_p \cdot \cos(\phi)\right)^2} - \frac{\pi}{p_d} \cdot \cos(\phi) \qquad \rho_p = 4.391 \times 10^{-3} \, m$$

$C_1 := 1.25in$ $\quad \rho_g := C_1 \cdot \sin(\phi) - \rho_p$

$$I_1 := \frac{\cos(\phi)}{\left(\frac{1}{\rho_p} + \frac{1}{\rho_g}\right) \cdot d_p} \qquad I_1 = 0.077 \qquad J_1 := 0.2$$

$C_a := K_a$ $\quad C_m := K_m$ $\quad C_v := K_v$ $\quad C_s := K_s$

$$L_1 := \frac{r_p}{\sin(\alpha_p)} \qquad d_g := d_p$$

$$m_G := \frac{N_g}{N_p} \qquad d_m := \frac{d_p + d_g}{2}$$

$$\omega_g := \frac{\omega_p \cdot N_p}{N_g}$$

(Transmitted force for both are the same)

$$W_t := 2 \cdot \frac{T_{pmax}}{d_m}$$

$$W_{ap} := W_t \cdot \tan(\phi) \cdot \sin(\alpha_p) \qquad W_{ag} := W_t \cdot \tan(\phi) \cdot \sin(\alpha_g) \qquad W_{ap} = 18.317\ \mathrm{N}$$

$$W_{rp} := W_t \cdot \tan(\phi) \cdot \cos(\alpha_p) \qquad W_{rg} := W_t \cdot \tan(\phi) \cdot \cos(\alpha_g) \qquad W_{rp} = 18.317\ \mathrm{N}$$

$$W_1 := \frac{W_t}{\cos(\phi)}$$

$$K_x := 1$$

$$\sigma_{bp} := \frac{2T_{pmax} \cdot K_a \cdot K_m \cdot K_s}{d_p \cdot F_1 \cdot m_1 \cdot J_1 \cdot K_v \cdot K_x} \qquad \sigma_{bp} = 4.58 \times 10^7\ \mathrm{Pa}$$

$$\sigma_{bg} := \frac{2T_{pmax} \cdot K_a \cdot K_m \cdot K_s}{d_p \cdot F_1 \cdot m_1 \cdot J_1 \cdot K_v \cdot K_x} \qquad \sigma_{bg} = 4.58 \times 10^7\ \mathrm{Pa}$$

$C_b := .634$ $\qquad C_f := 1$ $\qquad C_{xc} := 1$ (Assuming uncrowned teeth)

$$N_1 := 1 \cdot 10^5$$

$K_L := 1.6831 \cdot N_1^{-.0323}$ $\qquad C_L := 1$ $\qquad K_T := 1$ $\qquad K_R := 1.5$ $\qquad C_H := 1$

$C_{md} := 3.6$ (because cantilever uncrowned)

$K_L = 1.16$ $\qquad S_{fbprime} := 80\mathrm{MPa}$ $\qquad S_{fcprime} := 450\mathrm{MPa}$

$C_T := K_T$ $\qquad C_R := K_R$ $\qquad S_{fb} := \dfrac{K_L \cdot S_{fbprime}}{K_T \cdot K_R}$

$$T_D := \frac{F_1 \cdot I_1 \cdot C_v \cdot S_{fcprime}{}^2 \cdot d_p{}^2 \cdot (0.774 \cdot C_H)^2}{2 \cdot C_s \cdot C_{md} \cdot C_f \cdot C_a \cdot C_{xc} \cdot C_p{}^2 \cdot C_b{}^2 \cdot C_T{}^2 \cdot C_R{}^2} \qquad S_{fb} = 6.189 \times 10^7\ \mathrm{Pa}$$

$$T_D = 0.9\ \mathrm{J}$$

$T_{pmax} = 1.13\,J$ $\quad z := 1$ $\quad S_{fc} := \dfrac{C_L \cdot C_H \cdot S_{fcprime}}{C_T \cdot C_R}$

$\sigma_c := C_p \cdot C_b \cdot \sqrt{\dfrac{2 \cdot T_{pmax} \cdot C_a \cdot C_m \cdot C_s \cdot C_f \cdot C_{xc}}{F_1 \cdot I_1 \cdot d_p^{\,2} \cdot C_v}}$ $\quad S_{fc} = 3 \times 10^8\,Pa$

$N_{bp} := \dfrac{S_{fb}}{\sigma_{bp}}$ $\quad N_{bp} = 1.351$ $\quad \sigma_c = 1.734 \times 10^8\,Pa$

$N_{bg} := \dfrac{S_{fb}}{\sigma_{bg}}$ $\quad N_{bg} = 1.351$

$N_c := \left(\dfrac{S_{fc}}{\sigma_c}\right)^2$ $\quad N_c = 2.993$

Modulus of rigidity = 26.8GPa
Yield strength = 414 MPa (tensile)
414 MPa (compressive)
172 MPa (shear)

Ultimate strength = 469 MPa (tensile)
= 469 MPa (compressive)
= 290 MPa (shear)

Bearing Calculations

$L_{bb} := 1 \cdot 10^3$ (Life cycle of 100,000 cycles) $\quad a := 3$ (ball bearings)

$W_{ap} = 18.317\,N$ $\quad W_{rp} = 18.317\,N$ $\quad W_{ag} = 18.317\,N$ $\quad W_{rg} = 18.317\,N$

$F_r := W_{rp}$ $\quad F_r = 18.317\,N$ $\quad F_a := W_{ap}$ $\quad F_a = 18.317\,N$

$F_{bb} := \sqrt{F_a^{\,2} + F_r^{\,2}}$ $\quad F_{bb} = 25.904\,N$ $\quad C_{bb} := F_{bb} \cdot L_{bb}^{\frac{1}{a}}$ $\quad C_{bb} = 0.259\,kN$

Shaft Deflection

$F_\delta := \sqrt{\left(W_{rp}^{\,2}\right) + \left(W_t^{\,2}\right)}$ $\quad D_{shaft} := 0.5in$ $\quad r_{shaft} := \dfrac{D_{shaft}}{2}$

$L := C_1$ $\quad C_{constant} := 3$ $\quad I := \dfrac{\pi \cdot r_{shaft}^{\,4}}{4}$

$E_{steel} := 200GPa$ $\quad E_{aluminum} := 69GPa$

$$Deflection_{steel\_shaft} := \frac{F_{\delta} \cdot L^3}{C_{constant} \cdot E_{steel} \cdot I}$$

(The constant is 3 because the load on the shaft is an end load)

$$Deflection_{aluminum\_shaft} := \frac{F_{\delta} \cdot L^3}{C_{constant} \cdot E_{aluminum} \cdot I}$$

$$Deflection_{steel\_shaft} = 1.209 \times 10^{-4} in$$

$$Deflection_{aluminum\_shaft} = 3.503 \times 10^{-4} in$$

Error Propagation:

Mounting Distance Error Propagation:

Accuracy of CSRT in each direction:

$$e_{table} := 0.001in$$

Accuracy of Edge-Finder

$$e_{e\_find} := 0.0003in$$

Total Accuracy of System:

$$e_{total} := \sqrt{e_{table}^2 + e_{e\_find}^2}$$

$$\boxed{e_{total} = 1.044 \times 10^{-3} in}$$

Appendix II

Additional Components:

Bearings

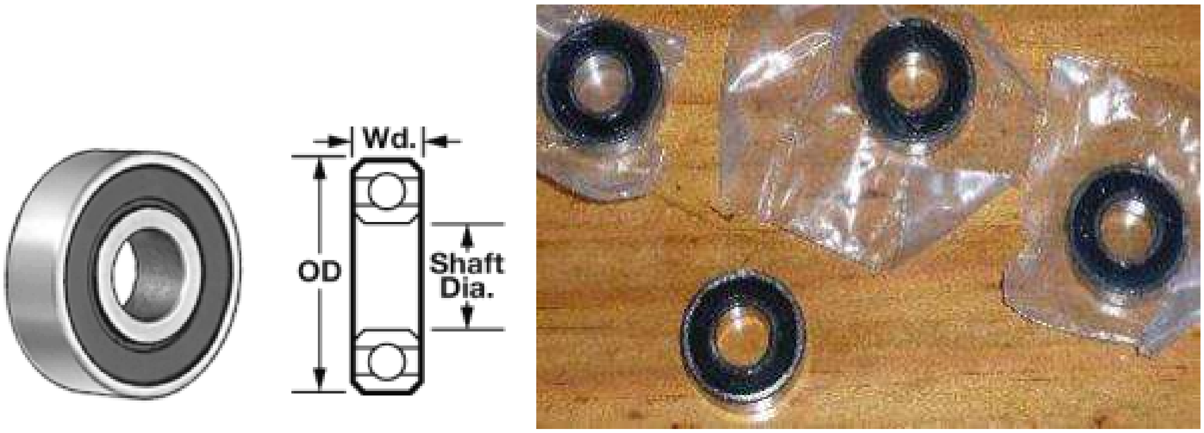


Figure 18: Bearings

-Type	*Ball Bearings*
-Ball Bearing Style	*Double Sealed*
-Ball Bearing Type	*Perma-Lube*
-System of Measurement	*Inch*
-For Shaft Diameter	*1/2"*
-Outside Diameter	*1-1/8"*
-Width	*5/16"*
-ABEC Precision Bearing Rating	*ABEC-1*
-ABEC-1 Precision Rating	*Regular*
-Bearing Trade Number	*R8*
-Dynamic Radial Load Capacity, lbs.	*1,148*
-Dynamic Radial Load Capacity Range, lbs.	*1,001 to 1,500 lbs.*
-Maximum rpm	*24,300*
-Maximum rpm Range	*15,001 to 30,000*
-Temperature Range	*+10° to +200° F*
-Bearing Material	*Steel*

-Seal Material *Buna-N*

-Specifications Met *Not Rated*

<u>*Gears*</u>

Set 1

M64N-1S

BEVEL GEAR SET

Bore : 3/16
Material : Stainless Steel (Pinion)/Stainless Steel (Gear)
Type : MITER & BEVEL GEARS
Style : PIN HUB
Pitch Diameter : .500 (PINION) / .500 (GEAR)
Hub Diameter : 3/8
Set Screw Location : 7/64
Ratio : 1 to 1
Module/Pitch : 64
Hub Length : 7/32
Pressure Angle : 20
AGMA Quality : 10
Teeth : 32/32
Mounting Distance (Gear) : 0.562
Mounting Distance (Pinion) : 0.562
Face Width : 5/32
PDF File name : B05B228
Catalog Page Number : B228

Return to Product Locator

Figure 19: Bevel Gear Set 1

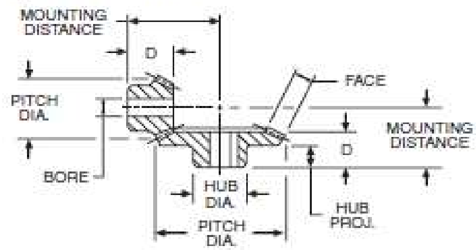
Set 2

SPIRAL BEVEL GEARS

30 THROUGH 8 DIAMETRAL PITCH
STEEL—UNHARDENED AND HARDENED

20° PRESSURE ANGLE
35° SPIRAL ANGLE

All Hardened steel gears have teeth only hardened and are equipped with standard keyways and setscrews, except as noted. All pinions are left hand.



REFERENCE PAGES

Alterations — 151
Horsepower Ratings — 71
Lubrication — 151
Materials — 152
Selection Procedure — 67

STANDARD TOLERANCES

DIMENSION		TOLERANCE
BORE	All	±.0005

ALL DIMENSIONS IN INCHES
ORDER BY CATALOG NUMBER OR ITEM CODE

Ratio	No. of Teeth	Pitch Dia.	Face	Bore	MD *	D	Hub Dia.	Hub Proj.	Catalog Number	Item Code	Catalog Number	Item Code
30 DIAMETRAL PITCH									STEEL UNHARDENED		STEEL HARDENED	
19 DIAMETRAL PITCH												
2:1	26 13	1.37 .68	.25	.500 .3125	1.000 1.062	.730 .623	1.12 .62	.42 .33	SS192-G SS192-P	11934 11936	— —	— —

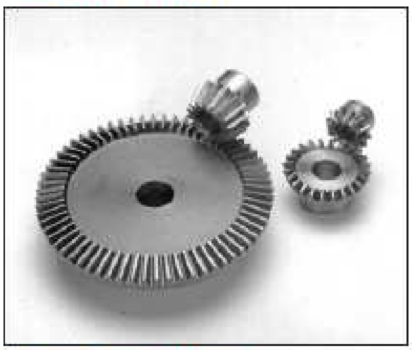
Figure 20: Bevel Gear Set 2

Set 3

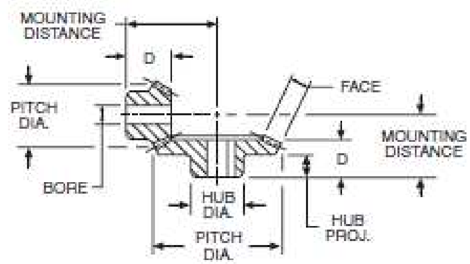
BEVEL GEARS

16 THROUGH 12 DIAMETRAL PITCH
STEEL—UNHARDENED AND HARDENED
AND CAST IRON

20° PRESSURE ANGLE



All gears have "Coniflex"® tooth form.
All Hardened steel gears have teeth only hardened and are equipped with standard keyways and setscrews.



REFERENCE PAGES

Alterations — 151
Horsepower Ratings — 70
Lubrication — 151
Materials — 152
Selection Procedure — 67

STANDARD TOLERANCES

DIMENSION		TOLERANCE
BORE	All	±.0005

ALL DIMENSIONS IN INCHES
ORDER BY CATALOG NUMBER OR ITEM CODE

Ratio	No. of Teeth	Pitch Dia.	Face	Bore	MD *	D	Hub Dia.	Hub Proj.	Catalog Number	Item Code	Catalog Number	Item Code	Catalog Number	Item Code
16 DIAMETRAL PITCH									STEEL UNHARDENED		STEEL HARDENED		CAST IRON GEARS STEEL PINIONS	
2:1	24	1.500	.20	.500	1.000	.625	1.00	.44	L148Y-G	12238	HL148Y-G	11858	—	—
	12	.750	.19	.375	1.125	.575	.66	.34	L148Y-P	12240	HL148Y-P	11860	—	—

Figure 21: Bevel Gear Set 3

Rotary Table

32803 & 34102

5207A4

CROSS SLIDE ROTARY TABLES

DESCRIPTION

Palmgren Cross Slide Rotary Tables (32803 & 34102) accurately position work on the X and Y axis, as well as 360° top rotation. All movements are infinitely adjustable by turning one of the handles. Rotary movement is calibrated in degrees around the full circumference of the top. Rotary feed dials are graduated in 3-minute increments. Tables can be locked against rotation. Cross travel screws are precision-cut Acme thread. All handles clear machine table work surface. These rugged, precision-made tables of industrial quality will allow the operator to do milling, routing, shaping, indexing, cutting slots and keyways plus many other jobs.

DIMENSIONS

Refer to Figure 1.

Stock No.		32803	34102
Overall Length	A	11 27/32″	12 25/32″
Overall Width	B	11 11/16″	12 5/8″
Table Centerbore	C	1 5/8″ x 3/4″ Deep	1 5/8″ x 3/4″ Deep
Base Width	D	6 11/32″	6 11/32″
Mounting Centers	E	5 3/8″	5 3/8″
Top Diameter	F	8″	10″
Height	H	5 5/8″	5 9/16″
Base Keyway	I	5/8″	5/8″
Top Keyway	J	5/8″	5/8″
Handle Length	K	3″	3″
Base Length	L	7 5/8″	7 5/8″
Mounting Centers	M	6 5/8″	6 5/8″
Longitudinal Travel		4″	4″
Cross Travel		4″	4″
Dial Division-Linear		0.001″	0.001″
Linear Travel per Turn		0.100″	0.100″
Dial Division-Rotary		3 Minutes	3 Minutes
Rotation per Turn		9°	9°
Gear Ratio		40:1	40:1

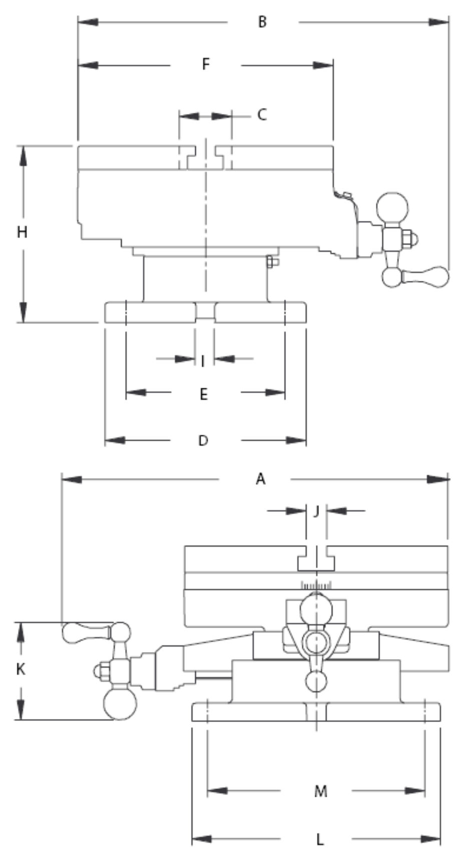


Figure 1 - Dimensions

Figure 22: CSRT Schematic

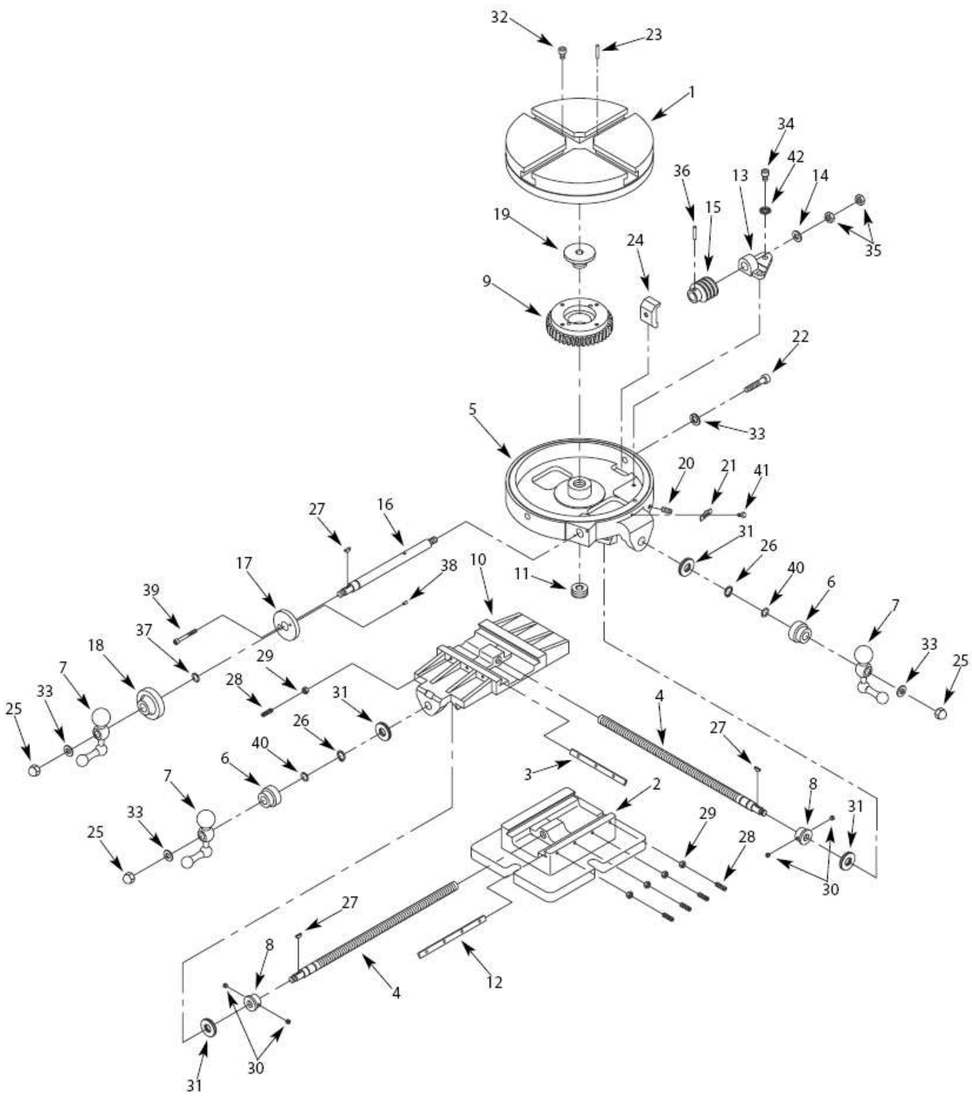


Figure 2 - Replacement Parts Illustration for Cross Slide Rotary Tables

Figure 23: CSRT Exploded View

REPLACEMENT PARTS LIST FOR CROSS SLIDE ROTARY TABLES

Ref. No.	Description	Part Number for: 32803	34102	Qty.
1	Top	12817.00	12810.00	1
2	Base	12921.09	12921.09	1
3	Gib Plate	12834.00	12834.00	1
4	Feed Screw	12838.00	12838.00	2
5	Top Slide	12803.09	12801.09	1
6	Dial	12826.00	12826.00	2
7	Handle	12934.00	12934.00	3
8	Threaded Collar	12830.00	12830.00	2
9	Worm Gear	12819.00	12819.00	1
10	Cross Slide	12922.09	12922.09	1
11	Set Screw	12880.00	12880.00	1
12	Gib Plate	12835.00	12835.00	1
13	Worm Shaft Bracket	12809.00	12809.00	1
14	Spacer	12848.00	12848.00	1
15	Worm	12824.00	12824.00	1
16	Worm Shaft	12833.00	12821.00	1
17	Collar	12829.00	12822.00	1
18	Dial	12828.00	12828.00	1
19	Retainer Screw	12832.00	12832.00	1
20	6-1.0 x 16mm Set Screw	*	*	1
21	Pointer	12878.00	12886.00	1
22	8-1.25 x 35mm Socket Head Bolt	*	*	2
23	5 x 25mm Dowel Pin	12852.00	12852.00	2
24	Locking Clamp	12812.00	12812.00	2
25	8-1.25mm Acorn Nut	*	*	3
26	3AMI-12 Retaining Ring	12860.00	12860.00	2
27	2.5 x 10mm Woodruff Key	12879.00	12879.00	3
28	5-0.8 x 20mm Set Screw	*	*	7
29	5-0.8mm Hex Nut	*	*	7
30	5-0.8 x 5mm Set Screw	*	*	4
31	1226 Thrust Bearing Assembly	12864.10	12864.10	4
32	6-1.0 x 20mm Socket Head Bolt	*	*	4
33	Spacer	12883.00	12883.00	5
34	6-1.0 x 16mm Socket Head Bolt	*	*	2
35	8-1.25mm Hex Nut	*	*	2
36	4 x 20mm Dowel Pin	12863.00	12863.00	1
37	O-Ring	12898.00	12898.00	1
38	3 x 6mm Dowel Pin	12867.00	12867.00	1
39	4-0.7 x 12mm Socket Head Bolt	*		1
39	4-0.7 x 35mm Socket Head Bolt		12896.00	1
40	O-Ring	12885.00	12885.00	2
41	4-0.7 x 10mm Pan Head Screw	*	*	1
42	6mm Serrated Washer	*	*	2

* Standard hardware item available locally.

3

Figure 24: CSRT Part Numbers

Motors

GM9236S026

Lo-Cog® DC Gearmotor

Assembly Data	Symbol	Units	Value	
Reference Voltage	E	V	24	
No-Load Speed	S_{NL}	rpm (rad/s)	71	(7.4)
Continuous Torque (Max.)[1]	T_C	oz-in (N-m)	480	(3.4E+00)
Peak Torque (Stall)[2]	T_{PK}	oz-in (N-m)	2585	(1.8E+01)
Weight	W_M	oz (g)	20.7	(586)
Motor Data				
Torque Constant	K_T	oz-in/A (N-m/A)	6.49	(4.58E-02)
Back-EMF Constant	K_E	V/krpm (V/rad/s)	4.80	(4.58E-02)
Resistance	R_T	Ω	2.49	
Inductance	L	mH	2.63	
No-Load Current	I_{NL}	A	0.16	
Peak Current (Stall)[2]	I_P	A	9.64	
Motor Constant	K_M	oz-in/√W (N-m/√W)	4.11	(2.90E-02)
Friction Torque	T_F	oz-in (N-m)	0.80	(5.6E-03)
Rotor Inertia	J_M	oz-in-s² (kg-m²)	1.0E-03	(7.1E-06)
Electrical Time Constant	τ_E	ms	1.06	
Mechanical Time Constant	τ_M	ms	8.5	
Viscous Damping	D	oz-in/krpm (N-m-s)	0.053	(3.5E-06)
Damping Constant	K_D	oz-in/krpm (N-m-s)	12.5	(8.5E-04)
Maximum Winding Temperature	θ_{MAX}	°F (°C)	311	(155)
Thermal Impedance	R_{TH}	°F/watt (°C/watt)	56.3	(13.5)
Thermal Time Constant	τ_{TH}	min	13.5	
Gearbox Data				
Reduction Ratio			65.5	
Efficiency[3]			0.80	
Maximum Allowable Torque		oz-in (N-m)	500	(3.53)
Encoder Data				

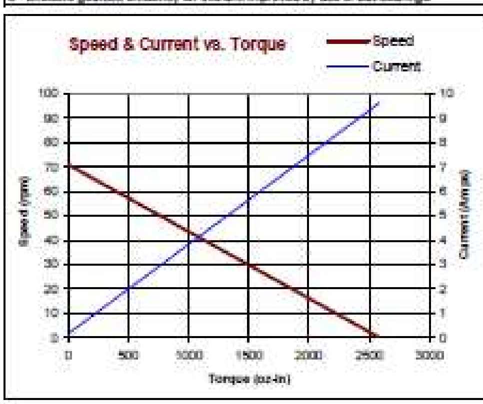
1 - Specified at max. winding temperature at 25°C ambient without heat sink. 2 - Theoretical values supplied for reference only.
3 - Effective gearbox efficiency for this unit improved by use of ball bearings.

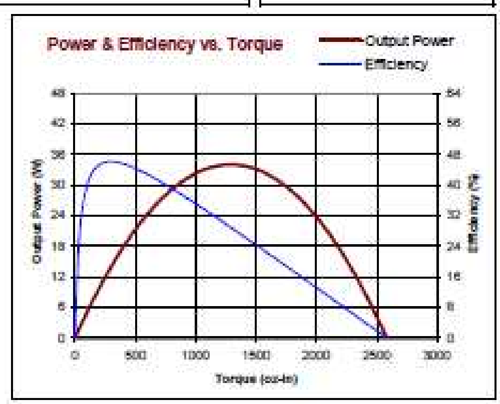
Included Features

2-Pole Stator
Ceramic Magnets
Heavy-Guage Steel Housing
7-Slot Armature
Silicon Steel Laminations
Stainless Steel Shaft
Copper-Graphite Brushes
Diamond Turned Commutator
Motor Ball Bearings
Output Ball Bearing
Wide Face Gears

Customization Options

Alternate Winding
Sleeve or Ball Bearings
Modified Output Shaft
Custom Cable Assembly
Special Brushes
EMI/RFI Suppression
Alternate Gear Material
Special Lubricant
Optional Encoder
Fail-Safe Brake





All values are nominal. Specifications subject to change without notice. Graphs are shown for reference only.

PITTMAN, 343 Godshall Drive, Harleysville, PA 19438, Phone: 877-PITTMAN, Fax: 215-256-1338, E-mail: info@pittmannet.com, Web Site: www.pittmannet.com

Figure 25: Motor Specifications

UNCONTROLLED

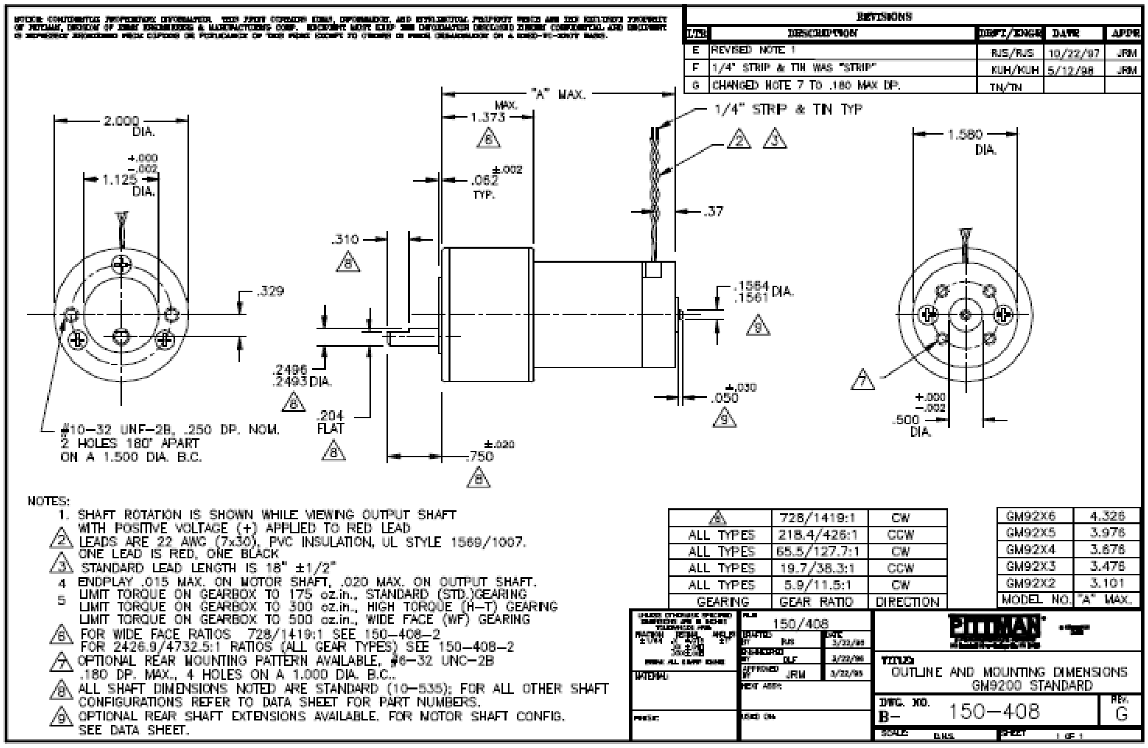


Figure 26: Motor Schematic

Products \ Tachometers / Stroboscopes

Send to a Friend Print This Page

Reed AT-6 Photo Non-Contact Tachometer

Condition: Brand New

The Reed AT-6 Photo Tachometer provides fast and accurate rpm measurements of rotating objects without contact.

Features:

- Laser targeting
- Provides Total measurements
- Built-in memory recalls maximum, minimum values as well as last value stored
- Highly visible 5-digit LCD with backlight
- Includes reflective tape, soft carrying case and batteries

Specifications:

- Measuring Range: 2 to 99,999 rpm
- Accuracy: ±(0.05% rdg. + 1 dgt.)
- Resolution: 0.1 rpm: 2 to 999.9 rpm; 1 rpm: >1000 rpm
- Sampling Time: 0.5 sec over 120 rpm
- Detection Distance: 50 to 500mm
- Power Supply: Single 9V battery
- Dimensions: 160 x 58 x 39mm
- Weight: 151g

Stock Number: R-AT-6
Shipping Weight: 2 lb

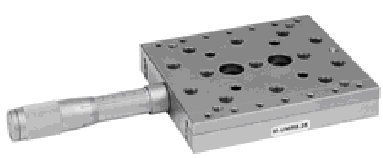
Reed AT-6 Photo Non-Contact Tachometer Add To Cart

(Each)

Qty:	1
Price:	$89.00
Sale:	$80.00

Figure 27: Tachometer Specifications

UMR Series Precision Double-Row Ball Bearing Linear Stages



UMR8.25 (Order actuators separately)

VIEW LARGER IMAGE

- Low-profile design
- Steel construction for high stability and rigidity
- Double-row ball bearings for higher load capacity (except UMR12)
- Angular deviation better than 100–200 µrad
- Threaded micrometer mounting

Product Description | Product Detail | Specifications | Drawings | Catalog PDF | 3-D Model

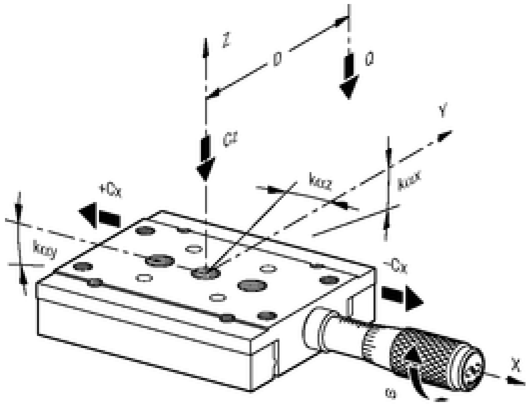
Specifications

Specifications

Model (Metric)	Travel [in. (mm)]	Angular Deviation [in. (mm)]	Load Capacity [lb (N)]	Actuator Sensitivity 1 µm	Actuator Sensitivity 0.1 µm	Fixed Positioning Kit	Carriage Lock	Base Plate
UMR8 Series								
UMR8.25	0.98 (25)	<200	200 (900)	BM17.25	DM17-25	‡	CL12-25	M-PBN8

Figure 28: Micrometer Specifications I

Load Characteristics



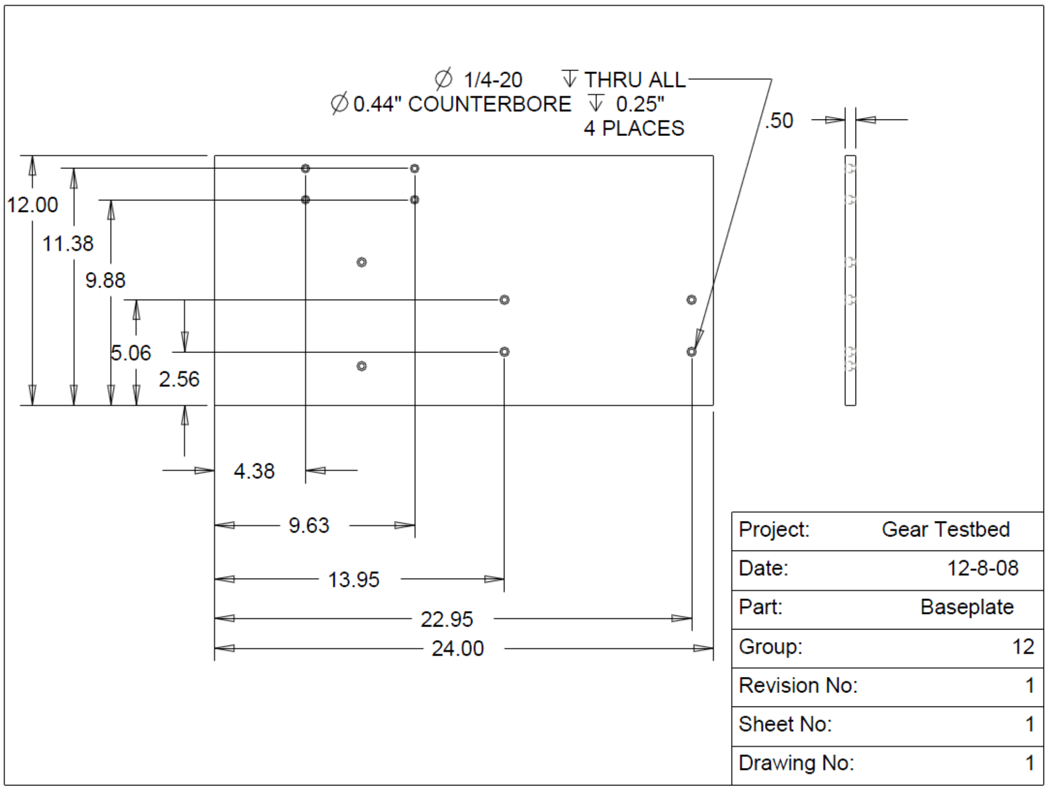
Model (Metric)	Cz (N)	+Cx (N)	-Cx (BM) (N)	-Cx (DM) (N)	a (mm)	ω (Nm)	Kαx (mrad/Nm)	Kαy (mrad/Nm)	Kαz (mrad/Nm)
UMR8.25	900	17	100	50	40	0.07	0.05	0.05	0.05

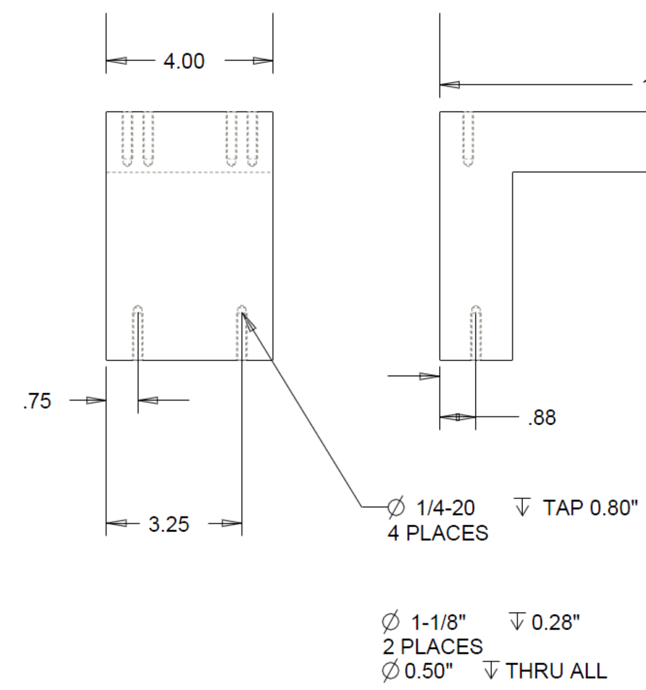
Q	Off-center load, Q≤Cz/(1 + D/a)
Cz	Centered normal load capacity
D	Cantilever distance in mm
a	Bearing constant
-Cx	Axial load capacity in the direction toward the actuator
+Cx	Axial load capacity in the direction away from the actuator
ω	Drive torque for +Cx = 10 N
kαx	Angular stiffness (roll)
kαy	Longitudinal stiffness (pitch)
kαz	Transverse stiffness (yaw)

Figure 29: Micrometer Specifications II

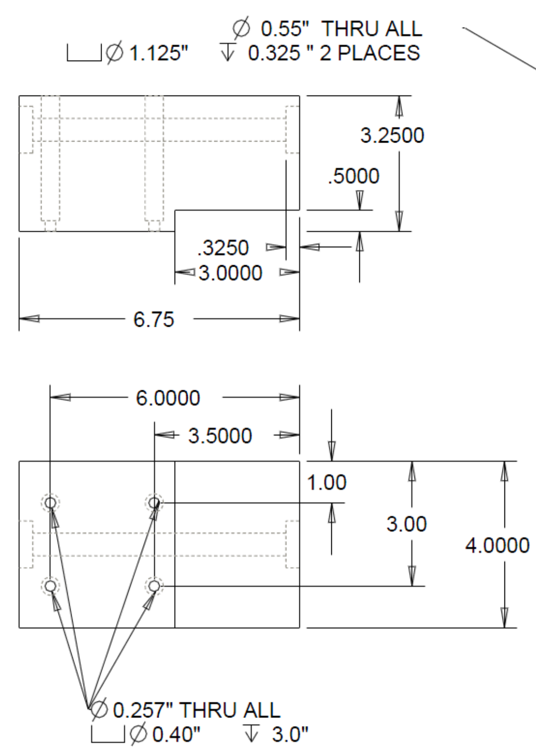
Appendix III

CAD Drawings

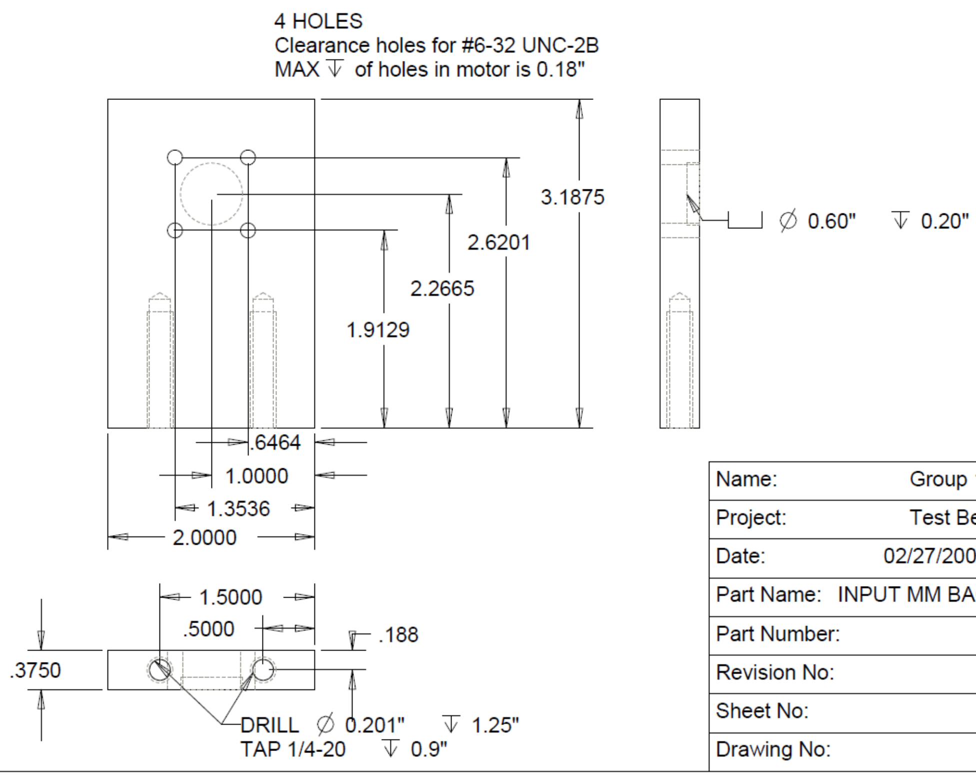




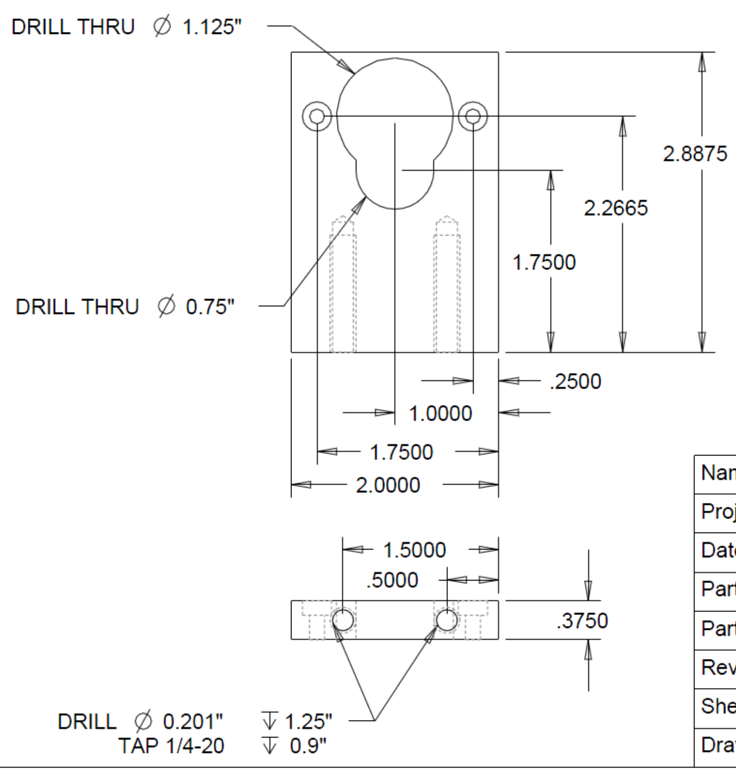
Project:	Gear Testbed
Date:	12-8-08
Part:	Bearing Block In
Group:	12
Revision No:	1
Sheet No:	1
Drawing No:	2



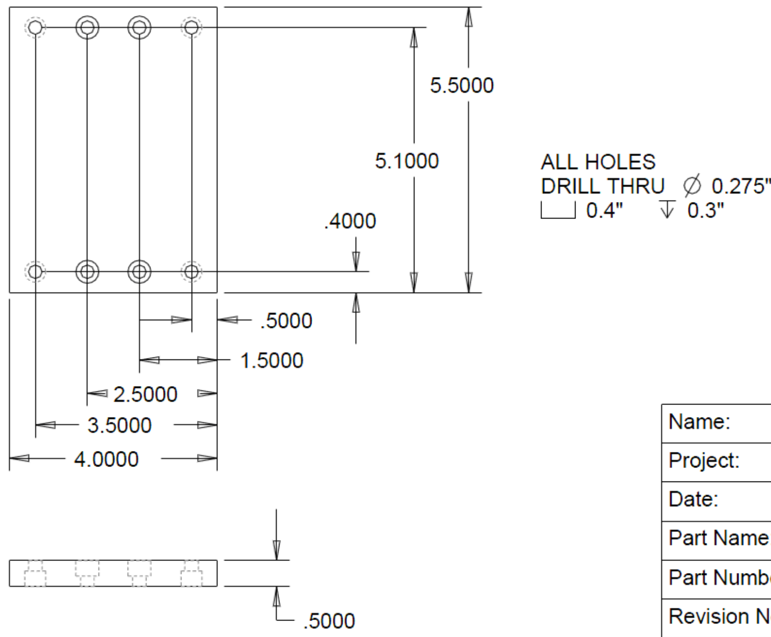
Name:	Group 12
Project:	Test Bed
Date:	02/20/2009
Part Name:	Middle BB
Part Number:	3
Revision No:	1
Sheet No:	1
Drawing No:	3



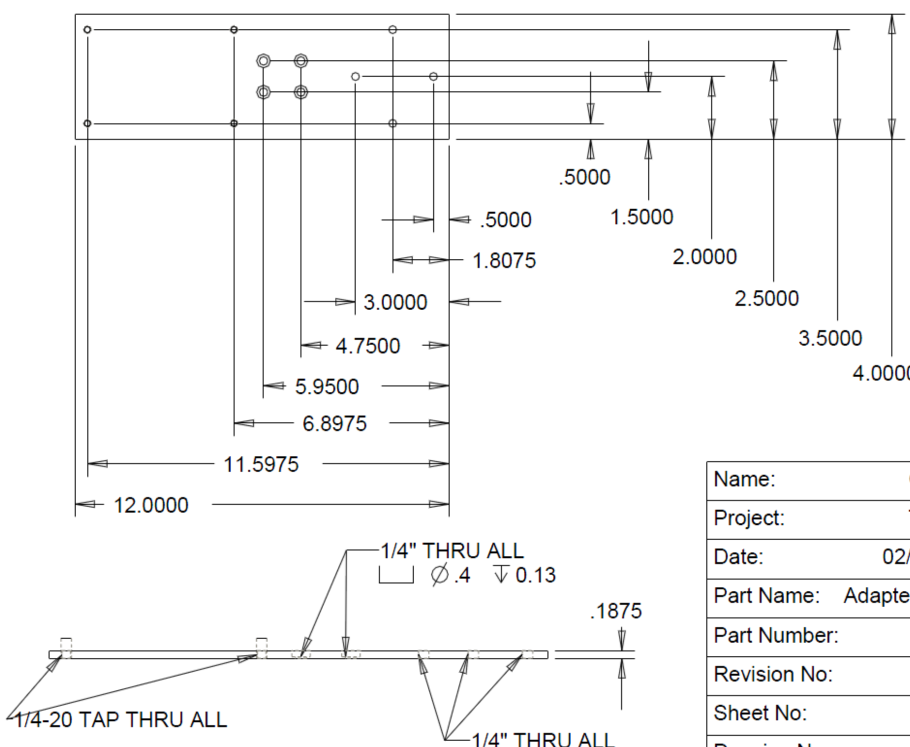
Name:	Group 12
Project:	Test Bed
Date:	02/27/2009
Part Name:	INPUT MM BACK
Part Number:	4
Revision No:	1
Sheet No:	1
Drawing No:	4



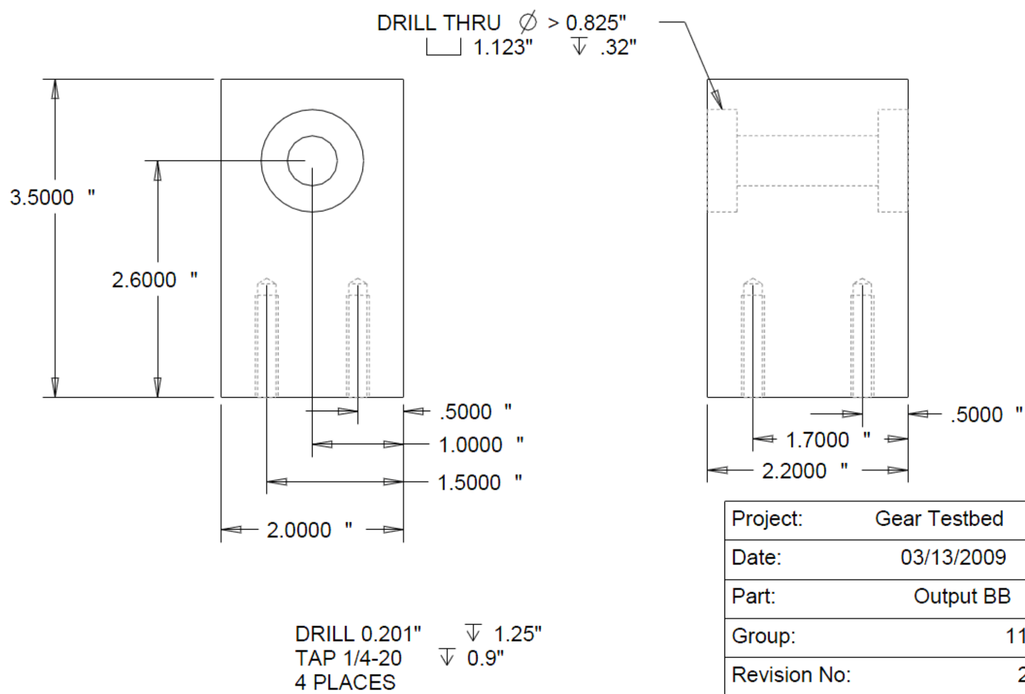
Name:	Group 12
Project:	Test Bed
Date:	02/27/2009
Part Name:	INPUT MM FR
Part Number:	5
Revision No:	1
Sheet No:	1
Drawing No:	5



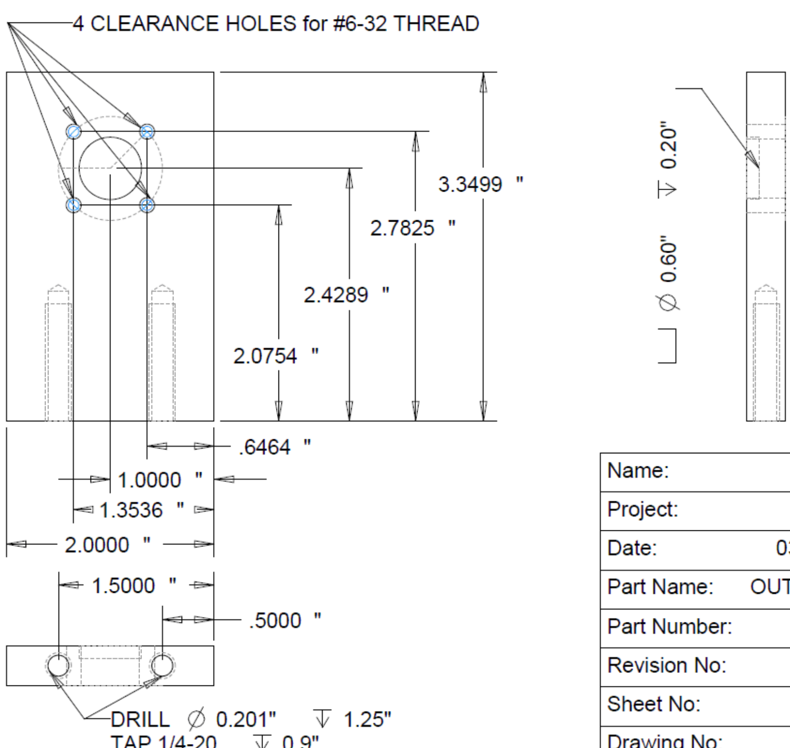
Name:	Group 12
Project:	Test Bed
Date:	02/27/2009
Part Name:	Motor Footing
Part Number:	7/15
Revision No:	1
Sheet No:	1
Drawing No:	6



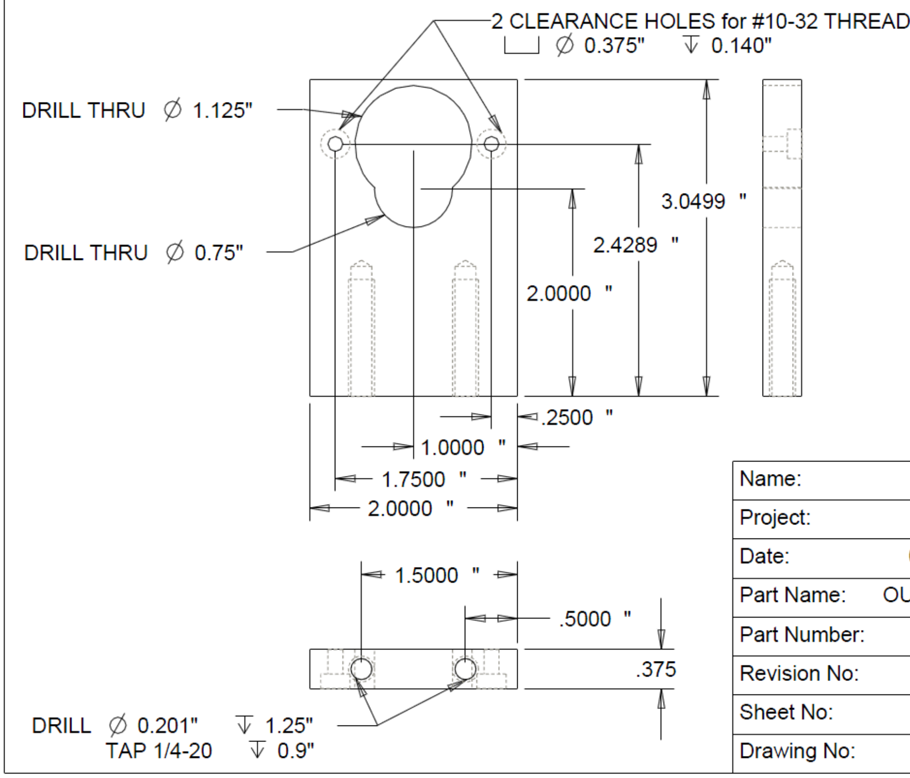
Name:	Group 12
Project:	Test Bed
Date:	02/20/2009
Part Name:	Adapter Plate
Part Number:	10
Revision No:	1
Sheet No:	1
Drawing No:	8



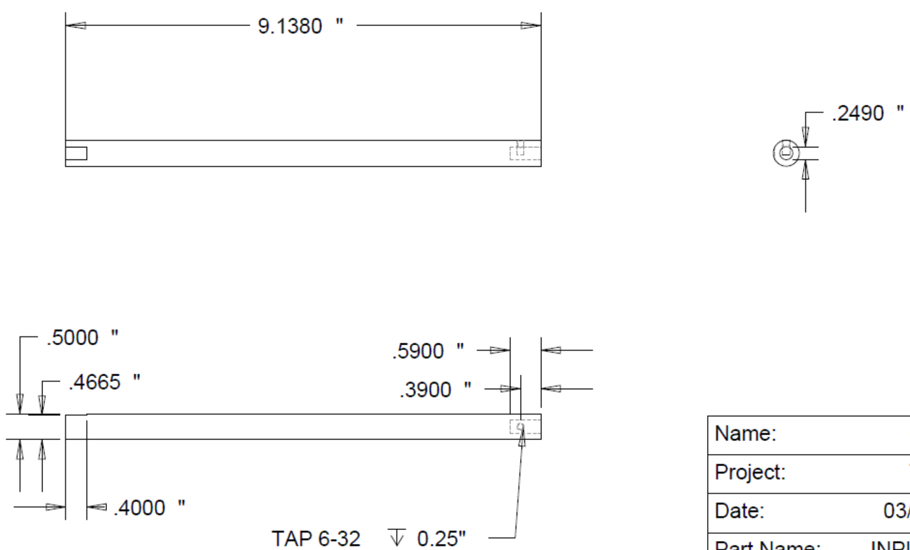
Project:	Gear Testbed
Date:	03/13/2009
Part:	Output BB
Group:	11
Revision No:	2
Sheet No:	1
Drawing No:	9



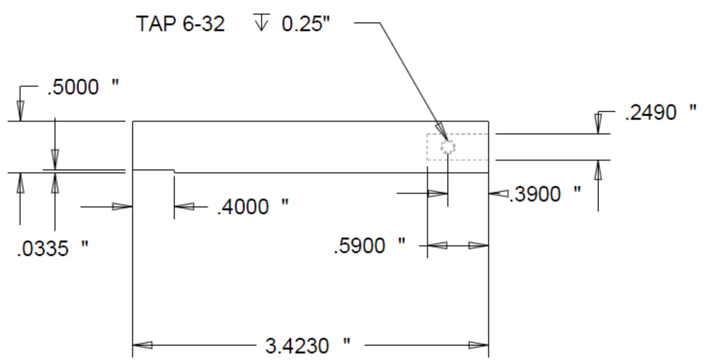
Name:	Group 12
Project:	Test Bed
Date:	03/13/2009
Part Name:	OUTPUT MM BK
Part Number:	12
Revision No:	1
Sheet No:	1
Drawing No:	10



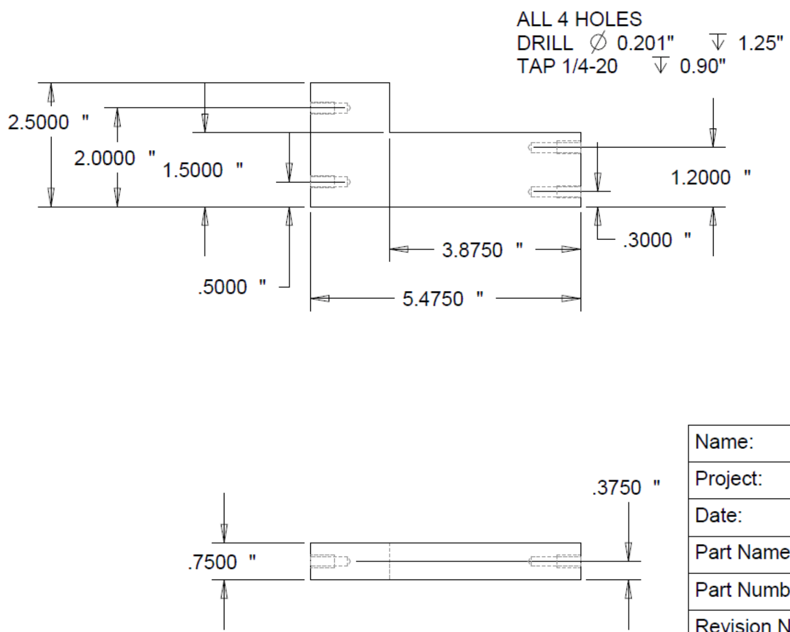
Name:	Group 12
Project:	Test Bed
Date:	03/13/2009
Part Name:	OUTPUT MM FR
Part Number:	13
Revision No:	1
Sheet No:	1
Drawing No:	11



Name:	Group 12
Project:	Test Bed
Date:	03/30/2009
Part Name:	INPUT SHAFT
Part Number:	16
Revision No:	1
Sheet No:	1
Drawing No:	12

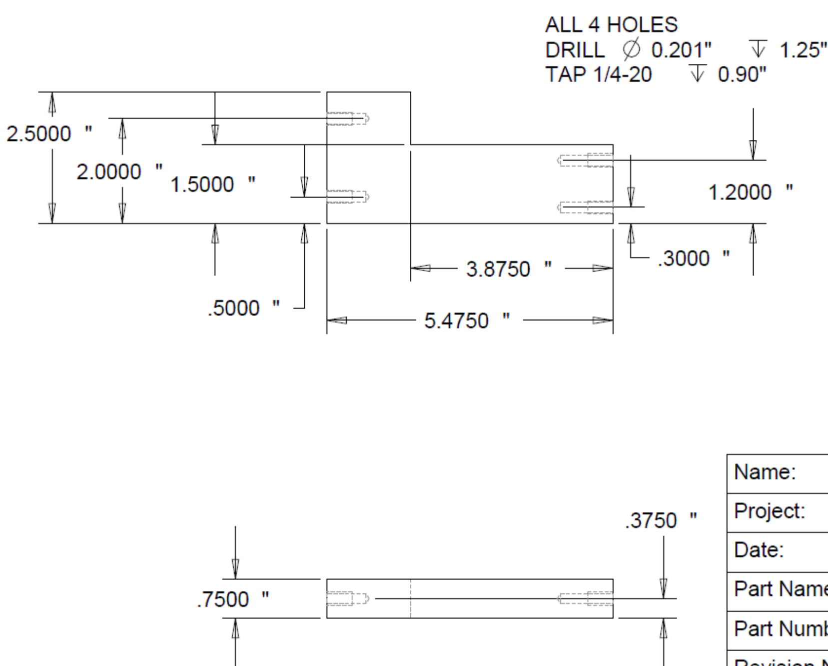


Name:	Group 12
Project:	Test Bed
Date:	03/06/2009
Part Name:	OUTPUT SHAFT
Part Number:	17
Revision No:	1
Sheet No:	1
Drawing No:	13

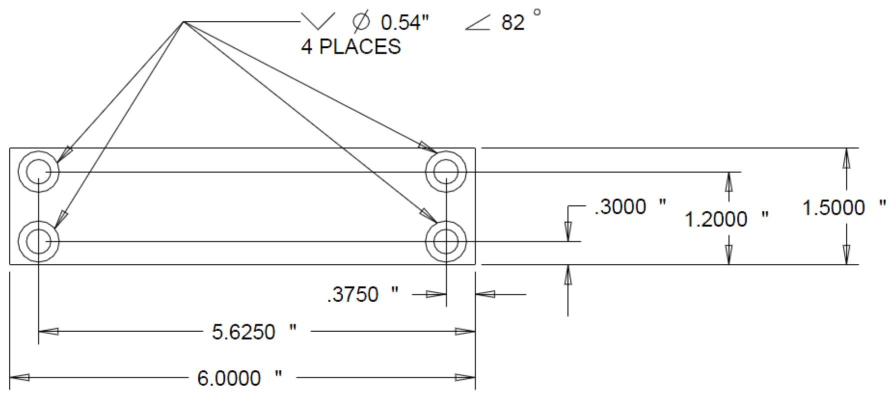


NEED TWO OF THESE PIECES!!!

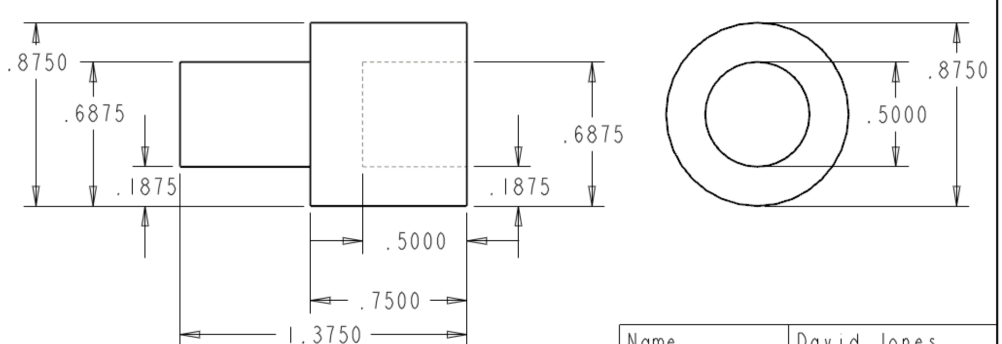
Name:	Group 12
Project:	Test Bed
Date:	03/13/2009
Part Name:	Aluminum H
Part Number:	18
Revision No:	1
Sheet No:	1
Drawing No:	14



Name:	Group 12
Project:	Test Bed
Date:	03/13/2009
Part Name:	Aluminum H
Part Number:	18
Revision No:	1
Sheet No:	1
Drawing No:	14

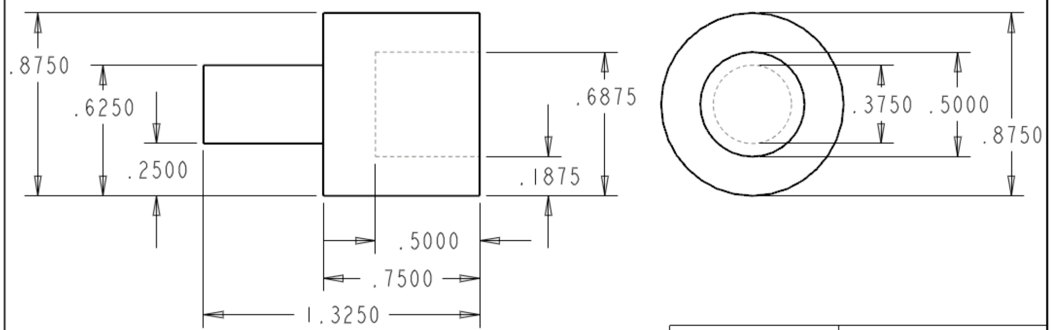


Name:	Group 12
Project:	Test Bed
Date:	03/13/2009
Part Name:	SS OVERHANG
Part Number:	19
Revision No:	1
Sheet No:	1
Drawing No:	15



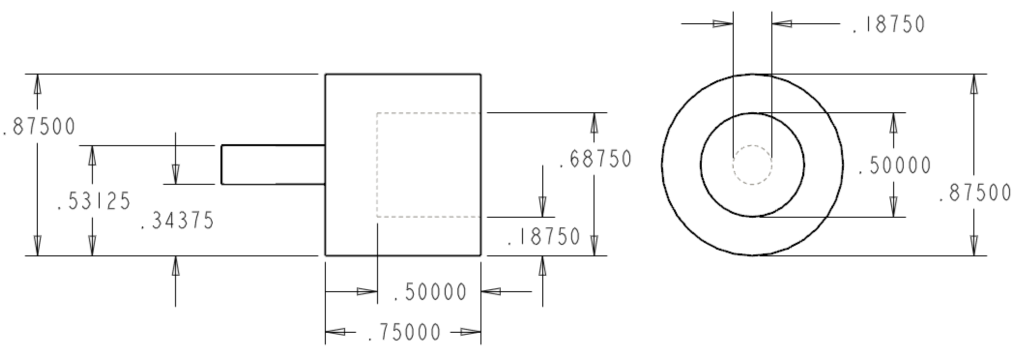
Name	David Jones
Project Name	Test Bed
Part Name	L148Y-G Adapter
Drawing #	1
Sheet #	1
Revision #	1
Last Revision Date	02/16/09
Material	Unhardened Steel

SCALE 2.000



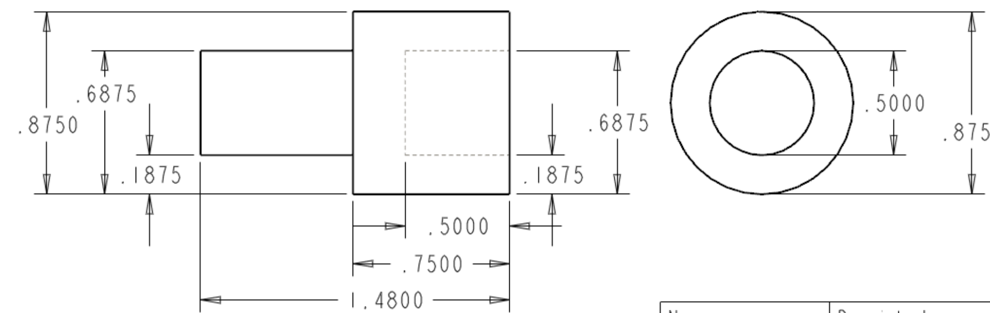
Name	David Jones
Project Name	Test Bed
Part Name	L148Y-P Adapter
Drawing #	1
Sheet #	1
Revision #	1
Last Revision Date	02/16/09
Material	Unhardened Steel

SCALE 2.000



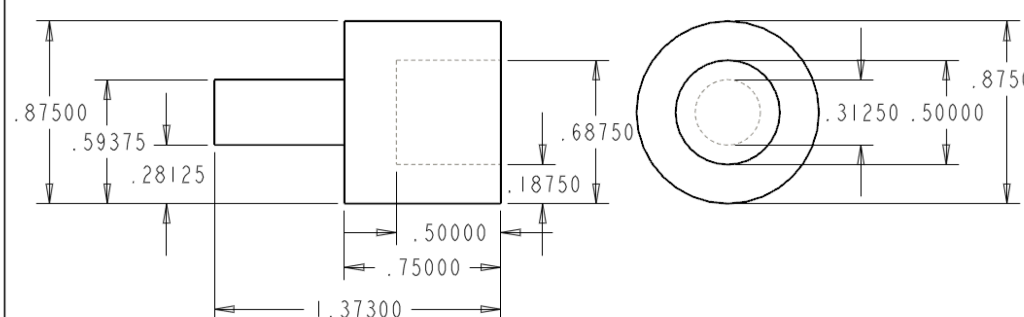
Name	David Jones
Project Name	Test Bed
Part Name	M64N-1S Adapter
Drawing #	1
Sheet #	1
Revision #	1
Last Revision Date	02/16/09
Material	Stainless Steel

SCALE 2.000



Name	David Jones
Project Name	Test Bed
Part Name	SS192-G Adapter
Drawing #	1
Sheet #	1
Revision #	1
Last Revision Date	02/16/09
Material	Unhardened Steel

SCALE 2.000



Name	David Jones
Project Name	Test Bed
Part Name	SS192-P Adapter
Drawing #	1
Sheet #	1
Revision #	1
Last Revision Date	02/16/09
Material	Unhardened Steel

SCALE 2.000