Bevel Gear Testbed

Fall Report 2008

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Introduction

Harris Corporation is an international communications and information technology company. They serve government and commercial markets in more than 150 countries. Headquartered in Melbourne, Florida, they receive annual revenues of $5.3 billion and provide work for 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.

Brent Stancil and Harris Corp. have asked for the design and development of a bevel gear testbed capable of testing various life-cycle characteristics of bevel gears used to open satellites in space. Previously, Harris Corp. engineers tested bevel gears for a project and were dissatisfied with the results. The possible problems included: misalignment of the shafts, anodic coating failure on the gears, and overloading of the gears. The design should have these problems accounted for and controlled.

Problem Definition

The primary objective of the current testbed is to be compatible with a wide range of gear sizes and materials while maintaining high mounting and aligning precision. Additionally the testbed should be fully functional and designed to allow for various upgrades to be made by Harris Corp. upon completion, such as equipment to measure vibration, heat generation, etc.

Product Specifications

The following parameters and values are the guidelines that must be followed to produce a working testbed for Harris Corporation.

Motor/Power Source

For this bevel gear test bed, a power source and motor will be needed to power the system. Harris has an extra DC motor that they'd be willing to lend for the project, but it has yet to be determined if it will work with our design. Otherwise either a DC or a Servo motor will be used to run the gear train. Examples of both are shown in the figures below.

Figure 1: DC Motor

Figure 2: Servo Motor

Variable Speed and Torque

The power source and driving motor will need an adjustable speed control. The maximum amount of rotational speed needed for this project will be 100rpm or $10.47\frac{rad}{s}$. A controller will be used to display the current speed of the rotating shaft. An example of a digital readout is shown below.

Figure 3: Torque and Speed LED Display

The power source and resistive motor that will provide our driven gear with a resistive torque will need a variable torque controller. The maximum torque needed for the testbed is 50 in*lb (5.6 N*m). A controller with a digital display is preferred by Harris Corp. to show the current torque output.

Gear Sizes and Material

Figure 4: Various Bevel Gear Sizes

This test bed is designed to test a wide range of bevel gears. The gears range from 1/3 in. (8.467 mm) to 5 in. (127 mm). These gears could be made from any of the following materials:

- Stainless Steel
- Hardened Steel
- Bare Aluminum
- Anodized Aluminum

Mounting Distance

To prevent misalignment and reduce wobble in the gears, the mounting distance must be accurate to 0.001 in. (0.0254 mm). As a result the precision of this testbed is much higher than most common testbeds, and as such is the greatest concern in the design.

Variable Shaft Angle

The variable shaft angle will be used to misalign the gears to verify that the actual test results are similar to the data in corresponding property tables. For this test bed, a variable shaft angle of ±0.5 degrees (±0.00873 radians) will be used to test for misalignment. To ensure that the misalignment is controlled, Harris Corp. originally asked for 0.001 degree incremental shaft angle variation, however this parameter would be far too expensive for the means of this project and as such a compromise of 0.1 degrees was reached.

<u>Product Specification Summary Table</u>

Specifications	**U.S. Values**	**SI Values**
Variable Torque	0 in.-lb - 50 in.-lb	0 Nm – 5.6 Nm
Variable Speed	0 rpm - 100 rpm	0 rad/s – 10.47 rad/s
Gear Size Range	1/3 in. to 5 in.	8.467 mm - 127 mm
Mounting Distance Accuracy	+/- 0.001 in.	+/- 0.0254 mm
Variable Shaft Angle Range	+/- 0.5 degrees	+/- 0.00873 rad
Shaft Angle Increments	0.02 degrees	2.909*10^-4 rad

Table 1

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 will rotate along an angular slotted path. The third concept is a combination of the first two concepts, having the actual wall 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 5 below is an illustration of the drawer slider design.

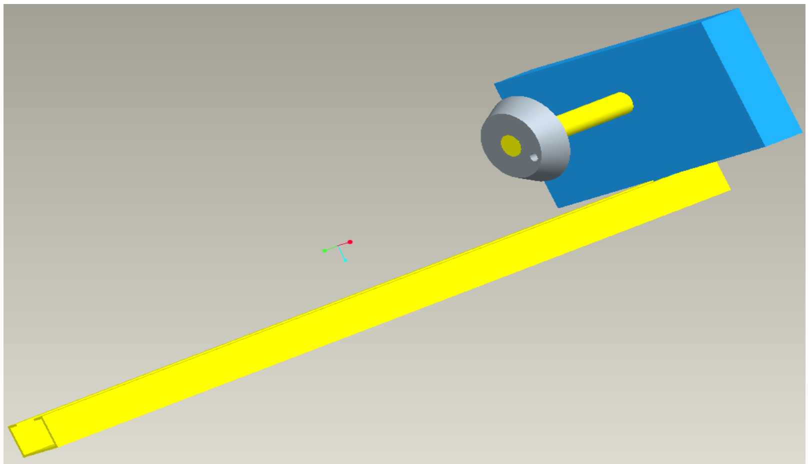

Figure 5: Drawer-Slider Design

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 6 is an illustration of the rotator design.

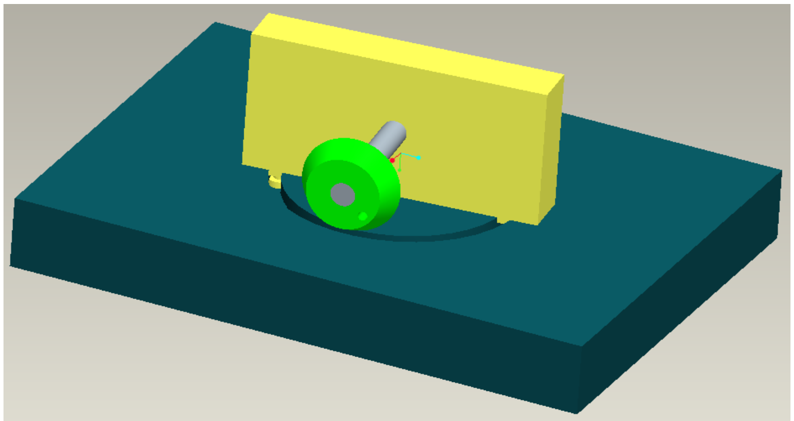

Figure 6: Rotator Design

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 7 is an illustration of the rotator-slider design.

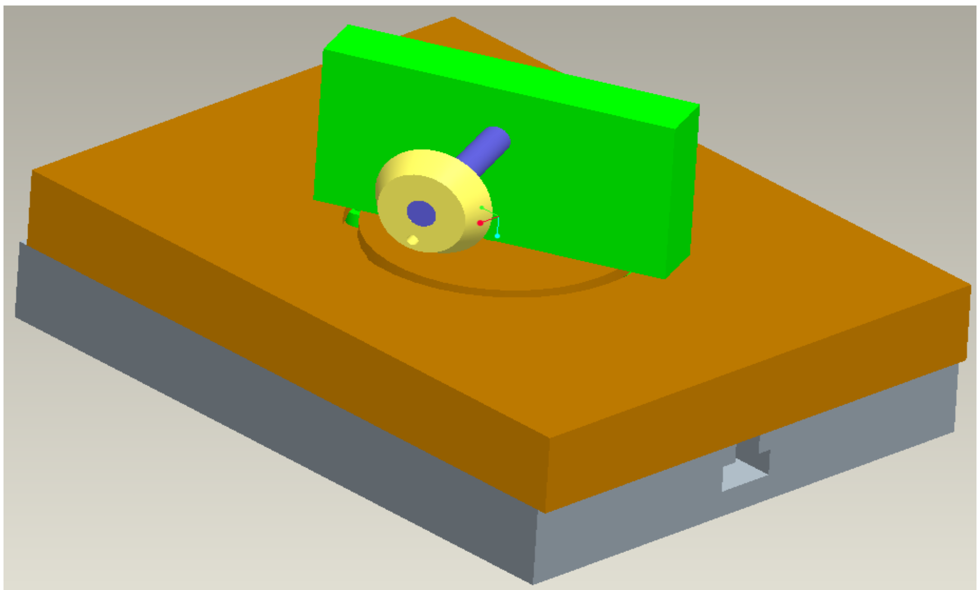

Figure 7: Rotator-Slider Design

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 8 shows how the curved wall design would look.

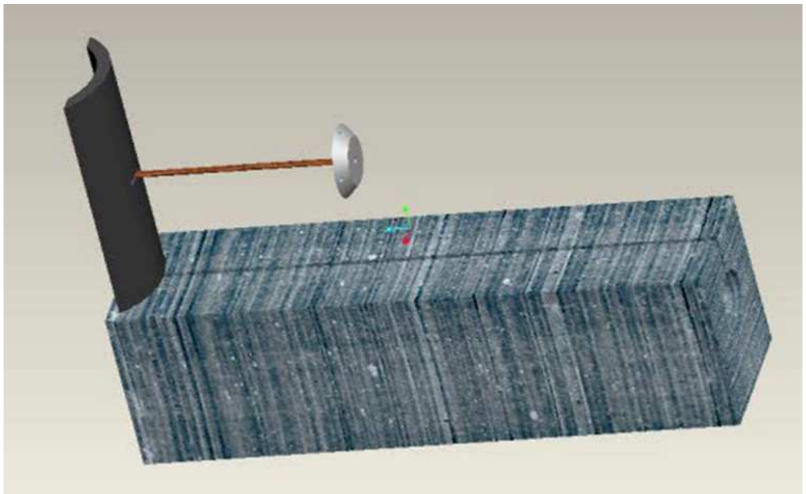

Figure 8: Curved Wall Design

Rack-and-Pinion

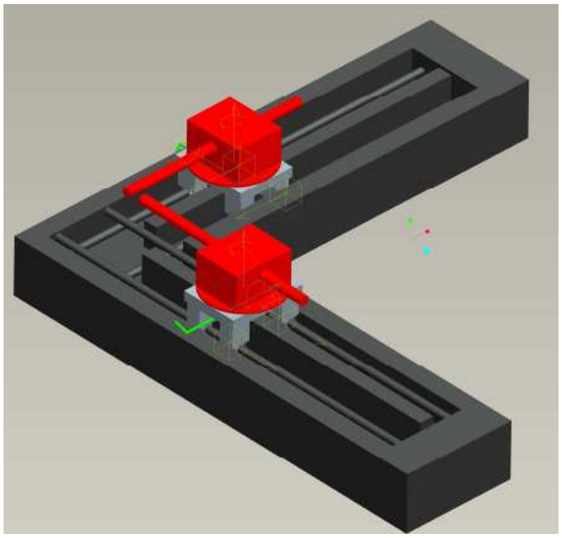

Figure 9: Rack-and-Pinion - Concept A

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.

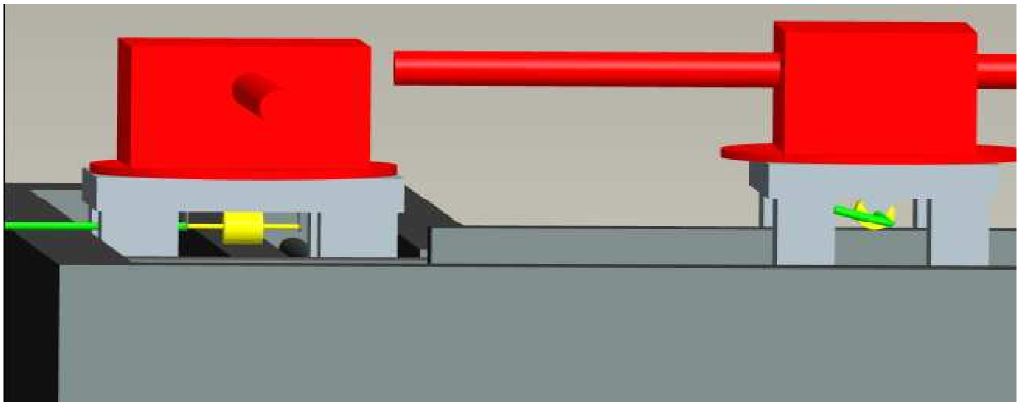

Figure 10: Close up of Rack-and-Pinion - Concept A

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.

Worm Gear

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 would be 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.

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 are just as important as that of 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 will be 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 would allow 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 was found 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 will be machined to mount onto the constant diameter shaft to fit any size bevel gear being tested. This method will require that a new adapter be machined for each size gear-shaft to be tested, which costs time and money, however it will cost much less of both than the designing and manufacturing of an adjustable chuck. In addition, if done properly, this method should be much more accurate than an adjustable chuck, but it is not as simple, since multiple extra pieces need to be designed, manufactured, and then assembled as needed. Within this concept there are different methods to attach the adapter to the shaft:

a) The first of the ideas under consideration is to drill a clearance hole into the adapter with an accompanying tapped hole on the shaft that will be connected via a threaded bolt, to act as a set screw. This approach will be simple to machine and can be machined to the required accuracy. However it is possible that the set screw might not make the adapter completely rigid to the main shaft, and the extra un-centered weight of a set screw can add wobble to the shaft and cause added wear on the gears.

b) To account for the possible problems associated with the above concept, the adapter and the shaft can connect by having one side threaded on its outer diameter and the other side threaded on its inner diameter. The threads will be designated the appropriate hand (right-hand or left-hand threads) as to allow the two pieces to self-tighten from the rotation of the shaft by the motor and the interactions between gears. This will be an extremely simple connection method, and would also mesh the two components quite rigidly. The only drawback is that it will be difficult to control the length of the adapters on the shaft to ± 0.001 in., although it is possible. Also since there is no set screw, there is no worry of extra weight on the shaft.

The most likely approach is to go with Concept 2b 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.

Concept Selection

When deciding which design to use, we first wanted to set up a Quality Function Deployment chart to relate our customer requirements to our engineering requirements. Our customer requirements include the following: machinability, durability, looks good, accuracy, reliability, ease of assembly, and our power source. Our engineering requirements include the following: rotational velocity, resistive torque, elastic modulus, anodic coating thickness, weight, deflection, yield strength, shaft mounting distance, and cost. These relationships are demonstrated in Table 2 on the following page.

		Engineering Requirements								
		Rotational Velocity	Resistive Torque	Elastic Modulus	Anodic Coating Thickness	Weight	Deflection	Yield Strength	Mounting Shaft Distance	Cost
Customer Requirements	Machinability			X	X	X		X		X
	Durability						X	X		X
	Looks Good			X	X					X
	Accuracy				X	X	X	X	X	X
	Reliability	X	X			X	X	X	X	X
	Assembled	X	X	X	X					
	Power Source	X	X							X
	Units									
		rpm	in.-lb.	MPa	μm	lb.	mm	MPa	in.	$
		1000	100							**≤1500**
		Engineering Targets								

Table 2

Next, we compared the different concepts by using 6 different criterions to judge which concept will best suit the project. The overall concept selection is based upon the total rating of each concept using our discretion as to scaling each criterion to a specific importance percentage and rating each criterion from each concept from 4 to 8. We arranged our concept grading in the concept screening matrix in Table 3 below. As seen in the table, the worm gear design is the best choice of the 7 original conceptual designs.

Criteria	Weight	Concepts													
		Drawer Slider		Rotator		Slider-Rotator		Curved Wall		Rack and Pinion (Pinion Driven)		Rack and Pinion (Rack Driven)		Worm Gear	
		Rating	Rating x Weight	Rating	Rating x Weight	Rating	Rating x Weight	Rating	Rating x Weight	Rating	Rating x Weight	Rating	Rating x Weight	Rating	Rating x Weight
Machinability	10%	4	0.40	3	0.30	3	0.30	1	0.10	4	0.40	4	0.40	4	0.40
Looks Good	5%	3	0.15	3	0.15	2	0.10	4	0.20	4	0.20	4	0.20	5	0.25
Cost	10%	3	0.30	2	0.20	1	0.10	2	0.20	2	0.20	2	0.20	2	0.20
Reliability	25%	2	0.50	2	0.50	2	0.50	2	0.50	4	1.00	3	0.75	4	1.00
Ease of Assembly	10%	4	0.40	4	0.40	3	0.30	3	0.30	2	0.20	2	0.20	3	0.30
Accuracy	40%	2	0.80	2	0.80	2	0.80	2	0.80	4	1.60	4	1.60	5	2.00
Total Score		2.55		2.35		2.1		2.1		3.6		3.35		4.15	

Table 3

Current Design

Our current design employs some of our previous designs and concepts, but also includes many new concepts, adaptations, and innovations as well.

INPUT PRO-E PICTURE IN HERE………..

To begin, there is a stationary motor and bearing block. These will be used to support the input motor, input driving shaft, and pinion. The shaft coming out of the motor will be attached to a 0.50 inch shaft using a set screw or keyway. For gears with a bore size of 0.50 inch, this shaft will be used for gear attachment. For gears with a smaller bore size, adaptors will be used in which one side of the adaptor will slip over top the 0.50 inch shaft and the other side of the adaptor will be machined down to the size of the bore of the gear in question. The adaptor will be attached to the shaft using a set screw or keyway. The adaptor will be attached the gear using a set screw or keyway as well.

The output cantilever is a little more complex. The adjustable bearing block table will be mounted to an adaptor plate. This adaptor plate is then attached to the base plate. The adjustable bearing block table translates in the x and y direction and can also rotate. The accuracy in the translational directions is 0.001 inch. The accuracy in the rotational portion of the table is 0.05 degrees. I FINISHED HERE!!!!!!!! This can be used to make sure that the gears are aligned perfectly or to misalign the gears as needed for testing. The rotation table is accurate to 1 arc minute, which equals a precision near 0.02 degrees. On top of the rotation table will be a lock-down plate which will take the majority of the axial loading from the gear set to ensure that the translation and rotation tables do not exceed their maximum loading. The bearing block for the output gear will be mounted to this lock-down plate and will contain two bearings for the output shaft, one on each side. The lock-down plate is illustrated in the figure below.

HOW THAT MOUNTING PLATE LOOKS AND WORKS HERE

The resistive motor and output bearing block will work similar to the input motor and bearing block, except that the bearing block for the output shaft is much smaller since it sits atop the translation and rotation tables and it will not be directly attached to the primary baseplate. Additionally the output motor must be capable of moving along with the tables and bearing block. Therefore the support for the motor will be on a sort of wheeled cart and be capable of to being secured to the base plate. The shaft coming out of the motor will be attached to a 0.50 inch shaft using a set screw or keyway. For gears with a bore size of 0.50 inch, this shaft is to be used for gear attachment. For gears with a smaller bore size, adaptors are to be used in which one side of the adaptor will slip over top the 0.50 inch shaft and the other side of the adaptor will be machined down to the size of the bore of the gear in question. The adaptor will be attached to the shaft using a set screw or keyway. The adaptor will be attached the gear using a set screw or keyway as well.

Base plate and Bearing Blocks

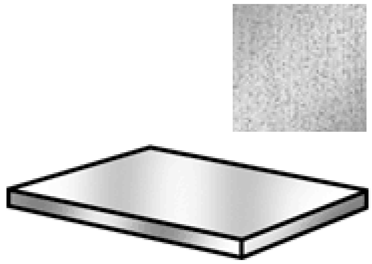

Figure 15 - Baseplate

The base plate was chosen based on the total size of our design. Also, the material was chosen to help keep our design rigid. The base plate will require some machining to make it fit our design. The specifications of the base plate are as follows:

-Material	*Aluminum (Alloy 5083)*
-Finish/Coating	*Unpolished (Mill)*
-Shape	*Sheets, Bars, Strips, and Cubes*
-Sheets, Bars, Strips, and Cubes Type	*Plain*
-Edge Type	*Square*
-Tolerance	*Standard*
-Thickness	*1/2"*
-Thickness Tolerance	*±023"*
-Length	*24"*
-Length Tolerance	*±1/4"*
-Width	*12"*
-Width Tolerance	*±1/4"*
-Material Certification	*Without Material Certification*
-Temper	*H116 (1/8 hard)*
-Hardness	*Not Rated*
-Yield Strength	*31,000 psi*
-Flatness Tolerance	*Not Rated*
-Temperature Range	*Not Rated*

-Specifications Met	*ASTM*
-ASTM Specification	*ASTM B209 or ASTM B928*

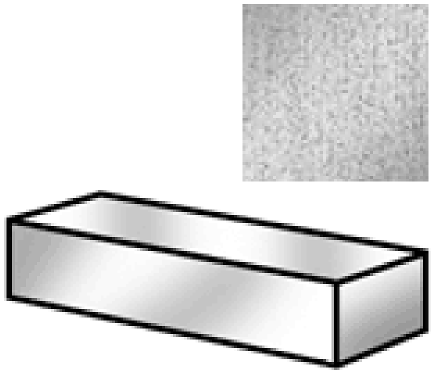

Figure 16 – Bearing Block

The bearing blocks were chosen because it will fit the design of our bevel gear test bed the best. These bearings blocks also have tolerances that are higher than the forces that they will be subjected too. This is based on calculations we did. The specifications of the bearings blocks are as follows:

-Material	*Alloy 6061*
-Finish/Coating	*Unpolished (Mill)*
-Shape	*Sheets, Bars, Strips, and Cubes*
-Sheets, Bars, Strips, and Cubes Type	*Plain*
-Edge Type	*Square*
-Tolerance	*Standard*
-Thickness	*2"*
-Thickness Tolerance	*±024"*
-Length	*6"*
-Length Tolerance	*±1"*
-Width	*4"*
-Width Tolerance	*±034"*
-Material Certification	*Without Material Certification*
-Temper	*T6511*

-Hardness	*60-95 Brinell*
-Yield Strength	*35,000 psi*
-Flatness Tolerance	*Not Rated*
-Temperature Range	*-320° to +300° F*
-Specifications Met	*ASTM*
-ASTM Specification	*ASTM B221*

<u>*Bearings*</u>

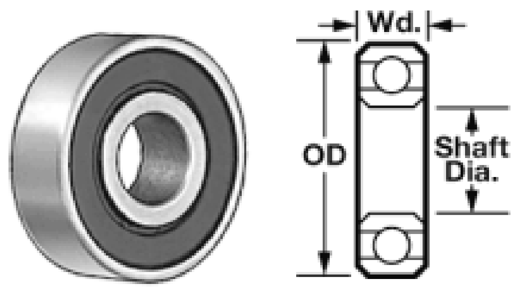

Figure 17 - Bearings

These bearings were chosen because they fill the shaft diameter we are going to use. These bearings also have tolerances that are far better that the forces that they will be subjected too. This is based on the bearing calculations values that we calculated. The specifications of the bearings are as follows:

-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*

Shaft and Gear Sets

The shaft sets were chosen because it will fit the design of our bevel gear test bed the best. These shaft sets also have tolerances that are higher than the forces that they will be subjected too. This is based on calculations we did. The specifications of the shaft sets are as follows:

-Material	*440 STAINLESS STEEL*
-Type	*SHAFT*
-Style	*GROUND & CASE HARDENED*
-Diameter	*0.375*
-Overall Length	*12*
-Overall Length Tolerance	*+/- 1/32*
-Nominal Fractional Diameter	*1/2*
-Diameter Tolerance	*+.0000/.0005*
-Surface Finish	*12-24*
-Straightness	*.001-.002/ft*
-Max Length Available	*11 FT*
-PDF File name	*B05I021*
-Catalog Page Number	*I21*

Figure 18 – 1:1 Bevel Gear Set

The gear sets were chosen by the Harris Corporation and they will be provided for us. These gear sets also have tolerances that are higher than the forces that they will be subjected too. This is based on calculations we did. The specifications of the gear sets are as follows:

Set 1

-Bore	*3/16*
-Material	*Stainless Steel Pinion & 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	*B22*

Set 2

<u>*Gear*</u>

-Description	*19 DP, 20 PA, 2:1 R, 26 Teeth*
-KWY/Setscrew/Other	*NO KWY OR SETSCREW*
-Pitch Diameter (IN)	*1.370*
-Bore Diameter (IN)	*0.500*
-Outside Diameter (IN)	*1.400*
-Mounting Distance (IN)	*1.000*
-Mating Gear	*SS192-P / 2:1 RATIO*

<u>*Pinion*</u>

-Description	*19 DP, 20 PA, 2:1 R, 13 Teeth*
-KWY/Setscrew/Other	*NO KWY OR SETSCREW*
-Pitch Diameter (IN)	*0.680*
-Bore Diameter (IN)	*0.313*
-Outside Diameter (IN)	*0.800*
-Mounting Distance (IN)	*1.062*
-Mating Gear	*SS192-G / 2:1 RATIO*

Set 3

<u>*Gear*</u>

-Description	*16 DP, 20 PA, 24 TEETH, STEEL*
-KWY/Setscrew/Other	*NO KWY OR SETSCREW*
-Pitch Diameter (IN)	*1.500*
-Bore Diameter (IN)	*0.500*
-Outside Diameter (IN)	*1.540*
-Mounting Distance (IN)	*1.000*
-Mating Gear	*L148Y-P / 2:1 RATIO*

<u>*Pinion*</u>

-Description	*16 DP, 20 PA, 12 TEETH, STEEL*

-KWY/Setscrew/Other	*NO KWY OR SETSCREW*
-Pitch Diameter (IN)	*0.750*
-Bore Diameter (IN)	*0.375*
-Outside Diameter (IN)	*0.900*
-Mounting Distance (IN)	*1.125*
-Mating Gear	*L148Y-G / 2:1 RATIO*

Motor and Controller

The motors will be provided for us by the Harris Corporation. The motor has requirements of outputting 50 in.-lb. of torque while spinning at 100 rpm. The motor is one of the most expensive pieces of this project and having it provided will allow us to use the money for other parts. The controller that the Harris Corporation used to control these motors was given to us as a recommendation. This controller is manufactured by Advanced Motion Controls and is compatible with the motors that we will be given.

Summary Table of Parts and Cost

Part Name	**Part Number**	**Vendor**	**Quantity**	**Price Per Item (less S&H)**	**Total Cost (less S&H)**
Adaptor Plate	8975K115	McMaster-Carr	1	$23.33	$23.33
Adjustable Table	5207A4(7)	McMaster-Carr	1	$674.00	$674.00
Base Plate	4058T52	McMaster-Carr	1	$128.30	$128.30
Bearings	2342K86	McMaster-Carr	4	$9.80	$39.20
Bearing Blocks	8975K565	McMaster-Carr	2	$40.00	$80.00
Controller	25A8	A-M-C	2	$295.00	$590.00
Shaft Sets	LMS-46-12	WMBerg	1	$25.00	$25.00
Motors	--------	Harris	2	Free	Free
Gear Sets	--------	Harris	3	Free	Free
Total					**$1559.83**

Table 4 – Parts and Costs Summary