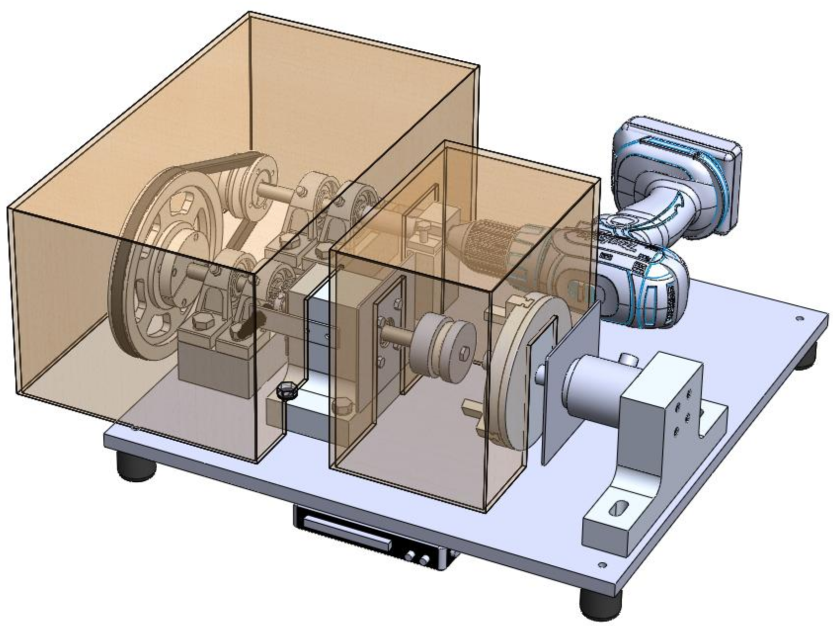
Friction Test Machine

Sponsors: Eric Polcuch & Abbas Charafeddine
Parker Hannifin, Aerospace Group

June 13, 2014



Final Design Report

Kyle Raymon Schless

Lynda Beatriz Tesillo

Mustapha Nadri

Statement of Disclaimer

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Executive Summary

Parker Hannifin is a world leader in the design and production of electromechanical actuators for aerospace use. These systems are designed to have as little drag and friction as possible in order to improve aerodynamic functionality. One source of drag and friction can originate from the bearings and mechanical seals used.

In response to an effort to reduce these losses, our team has been asked by Parker Hannifin to design and fabricate a test apparatus to measure the viscous drag associated with bearings and seals as a function of shaft speed, temperature, and size.

We have generated several potential design solutions over the course of the last nine months, as will be discussed later in this report. However, the final design was chosen and thoroughly analyzed in order to assure our sponsor that the test apparatus will function properly. This design has been manufactured and implemented to achieve the key functionality required.

The final design utilizes a drill motor to drive a dual stage drive train in order to achieve speeds up to 15,000 RPM. The output shaft of the drive train will rotate various bearing or seal testing attachments, which accommodate the numerous inner diameter sizes to be tested. An external jaw chuck will hold the outer race of the test bearing or seal still and rotate in accordance to the amount of drag torque the test specimen experiences. A reaction torque sensor will measure that rotation and output the drag torque to the data acquisition system, which will also be receiving temperature and output shaft speed measurements. The data will be logged using LabVIEW.

This test machine will allow Parker Hannifin to accurately measure the drag associated with the bearings and seals integrated into their systems and, as a result, provide them with the opportunity to make their products perform better.

Chapter 1 – Introduction

Sponsor Background and Needs

Parker Hannifin's Aerospace Group engineers various electromechanical systems for numerous aerospace applications. Although their products already achieve incredible feats, Parker Hannifin continues to strive to improve their systems, including their electromechanical actuators. These systems must produce little to no drag or friction in order to aerodynamically improve the overall system. Therefore, Parker Hannifin would like to investigate the drag associated with the bearings and mechanical seals in the system caused by temperature, speed, and size differences. The drag in question is the viscous drag that occurs between the inner and outer races of a bearing or seal when one of the races is rotating and the other is held still.

Formal Problem Definition

Parker Hannifin has asked our team to design and build a test fixture that will measure the viscous drag generated by bearings and seals as a function of temperature, shaft speed, and size. As a result, Parker Hannifin will be able to pinpoint and reduce the frictional losses in their actuator design.

Specification Development

Our team developed a set of engineering requirements derived from the customer requirements of Parker Hannifin. These engineering requirements were developed using the Quality Function Deployment (QFD) method, a design technique that has been proven to assist in the development of engineering specifications. The main output of using this method is the "House of Quality". This is a diagram that displays the correlation between each customer and engineering requirement and benchmarks how well other existing solutions to the customer problem fulfill the customer requirements. The "House of Quality" created by our team for this project, as well as an explanation about its features, can be found in Appendix A.

Our team was able to learn many things by using the "House of Quality". Looking at the correlations that were made, it can be seen that the customer requirements of being able to test different sizes of bearings and seals were the most important, since these two requirements had the greatest amount of correlations. Also, the requirement of being safe to

use was very important and frequently correlated to other requirements. Looking at the weighting of the customer requirements, it can be seen that being safe to use was the most important requirement to fulfill according to our team. The next highest weightings were the ability of the fixture to test different torque values, speed values and bearing sizes. However, it can also be seen that the lowest weighted requirements for our team were the requirements that the fixture only need one operator and that it can test samples using different greases. This simply means that, compared to the other requirements, these were the requirements that were slightly less likely to be focused on. This does not by any means indicate that these requirements will not be fulfilled. We also learned that the hand-testing method currently in use (described in Chapter 2) is not adequate, as it does not fulfill all of the customer requirements. Therefore, there is a need for a better solution.

Our team visited the Parker Hannifin facilities in Irvine, CA on October 11th, 2013. We were given insight into the purpose of this test machine, as well as the application of the bearings and seals being tested. This visit was crucial in determining the requirements and specifications for the project. The interface between the lab's thermal control system, the Thermotron, and this friction test machine was observed, as well as the lab environment that the fixture will be operating in.

The bearing and seal sizes to be tested were requested from Parker Hannifin, as the sizes of the samples would play a large part in the design of the machine. It was determined that the machine will need to test all sizes of bearings that are compatible with the shaft requirements provided. Therefore, the machine will need to accommodate a continuous range of bearing sizes. However, there are a discrete number of sizes of seals that will need to be tested. The list of these discrete seal sizes provided by Parker Hannifin can be found in Appendix A in Table 7. All of the technical drawings for these seals were provided to our team, allowing us to design with specific sizes and tolerances in mind. Therefore, our team will need to design the machine to test a large continuous array of bearing sizes and discrete seal sizes.

The customer and engineering requirements generated through our visit to Parker Hannifin's facilities, as well as from communication with our sponsor, are listed in Appendix A and are also explained below.

The first customer requirement was that the test fixture needed to be safe to use. This requirement is absolutely the most important. In order to determine the engineering specifications required to accomplish this, we needed to consider all the components of the potential system and how they could cause harm. The fixture will definitely involve moving and rotating machinery, so all these components will need to be shielded. Since there will also be

electrical components to the system, they will need to be properly grounded. These electrical components will also need to be compatible with a typical wall outlet, so excessive voltage and current values will be avoided. There will also be a thermal chamber around the test sample in order to control the testing temperature, so this will act as one of the guards for the rotating test sample. However, the temperature range is from -55°C to 70°C, which means that the thermal chamber will need to be well insulated so as not to cause harm to the operator when it is touched. Also, the customer requested that there be a single control station for a single operator. In order to accomplish this, we will need to make sure that the single operator will be able to start, stop, and operate the fixture by themselves with ease.

The customer also required that the fixture be able to test different sizes of bearings and seals. There are several dimensions of the fixture that will be dependent on the size specifications of the bearings and seals, so it is very important that the design considers all potential sizes. Based on the bearings and seals typically used by Parker Hannifin, the shaft size will need to accommodate a range of inner diameters from ¼” to 2 ½” for the bearings and up to about 3.5” for the seals. Additionally, the fixture will need to accommodate the average widths and outer diameters for bearings and seals typically used in Parker Hannifin’s electromechanical actuators.

It was also required that the fixture test bearings and seals with different kinds of greases. In order to be compliant with this requirement, we will ensure that all materials used in the building of the test fixture will be compatible with the various greases to be used.

The test fixture must also test the bearings and seals at different speeds. According to customer specifications, the large bearings will be tested at low speeds, while the small bearings will be tested at high speeds. So, the speed will need to reach up to 15,000 RPM.

The bearings and seals will also need to be tested at different temperatures. In order to accurately represent the range of temperatures that the bearings and seals will experience in actual use, the range will be from -55°C to 70°C. In order to ensure that the temperature will remain constant during testing, an insulated thermal chamber will be placed around the test sample that will attach to Parker Hannifin’s Thermotron system.

In order for the bearings and seals to be tested in realistic conditions, the shaft to which they are mounted needs to have the same surface conditions as the shafts in the actuators. Therefore, the shaft in the test fixture will need to be ground, smoothed, and polished to standard specifications.

Since the test runs will be quite short in order to capture quick behaviors (such as breakout torque), it is required that the fixture have a Data Acquisition (DAQ) system with a good sampling rate. We determined that a 100 Hertz sampling rate would provide complete data sets. However, since each of the test runs will span a short amount of time (i.e. just a few seconds), it is acceptable to utilize a much higher sampling rate, if available. Noise should not be a major issue if the test runs are short. We need to ensure that all of the behaviors of the samples are recorded.

Another customer requirement is that the fixture allow for variable acceleration of the shaft. This is so the testing can be performed in ramp-up and ramp-down speed conditions. Also, because of the need to test breakout torque behavior, it is desired to have the ability to start the speed at a lower value and, once the breakout torque has occurred, accelerate the motor to the testing speed as quickly as possible so the testing can be performed before the test sample heats up. Therefore, in order to accomplish this, the motor selected will need to have variable speed capabilities.

The test fixture will also need to be compact enough to fit on a desk top. Parker Hannifin has determined that the fixture should be approximately 18"x18"x18" in size, so this is will be the engineering specification for this customer requirement. The compact size is very important for the customer because it will allow Parker Hannifin engineers to maneuver the fixture easily around the testing labs on a rolling cart or desktop.

Considering that this fixture will be used for testing to reduce drag in Parker Hannifin's future actuator designs, it is very important that this fixture be able to perform many tests and have a long lifespan. It would not be worthwhile for this fixture to be designed to only perform a small amount of testing. So, this fixture will be designed to last for 1000 test runs for a duration of 5 minutes each. Since each test run will not typically last 5 minutes, but instead will last for less than a minute, designing the fixture with this lifespan will help ensure that it will benefit Parker Hannifin for at least this amount of time. Of course, as soon as the average duration of testing is determined, more accurate fatigue and lifespan calculations will be performed.

For the purpose of easy, universal use by Parker Hannifin and our team, it is imperative that the fixture be compatible with a standard wall outlet for power supply. Wall outlets provide approximately 115 Volts and 20 Amps. However, if needed, a 220 Volt wall power supply will also fulfill Parker Hannifin's requests. In order to ensure the compatibility with a wall outlet, we will need to minimize the amount of electrical components and machinery as much as possible. Parker Hannifin has also requested that the fixture prevent all electrical interference between measurement signals. In order to fulfill this requirement, shielded wiring will be used for all

measurement signal wiring. Electrical interference causes error in measurement signals, so, to maintain high accuracy in the measured data, it will be prevented.

The temperature control system will be provided by Parker Hannifin by means of the Thermotron system on site. However, the fixture will need to effectively interface with this system. A full thermal chamber will be built around the test sample, in which the Thermotron will control the temperature of the test sample environment.

Because many of the measured behaviors that are to be tested will occur very quickly, it is very important that the fixture be able to take accurate measurements of all test variables. The accuracy of the temperature and speed has been set to allow an error of ± 2.0%, while the torque has been allowed an error of ± 5.0%. These low percent errors will ensure the measurements are accurate, as well as aid in creating accurate representations of the trends and correlations that may occur.

It has been requested that the fixture be operable by only one person. This will help ensure safety, as the entire fixture will be able to be started, operated, and stopped by one person only. If the fixture were designed for two people to operate, it would be necessary to ensure that it could only be started and operated by two people. However, in order to avoid the possibility of injury or harm due to one person attempting to operate the fixture alone, it should only need to be fully operated by one person. In order to fulfill this requirement, the fixture will be designed with a single control station.

In order to prevent noise in the measurements and inaccuracy in the data, Parker Hannifin has requested that the fixture be designed with minimal vibration while operating. In order to achieve this, our team will need to design and build the fixture with a low and feasible alignment tolerance. Parker Hannifin asked our team to reduce the need for couplings along the shaft, so we will need to properly and carefully align all of the components of the fixture when it is being fabricated. This will mainly be handled during the machining process, as well as in the detail design process with tolerance determination.

Lastly, the bearings and seals will need to be tested at different torque values. The range of values that the bearings and seals typically are exposed to is from 1 in-lb to 20 in-lb of torque. The larger bearings and seals will be tested at lower speeds and higher torque values, while the smaller bearings and seals will be tested at higher speeds and lower torque values. However, in order to accommodate the wide range of sizes of test samples, the fixture should be able to handle and measure this wide range of values.

Table 1 below lists each requirement specified, assesses the risk of accomplishing this requirement in the final design, and details the means of compliance. A risk level of "L" indicates that there should be little to no trouble in accomplishing this requirement in the overall design. A risk level of "M" indicates that there may be a small amount of difficulty in implementing the requirement in the design. A risk level of "H" means that there is a high risk of the requirement not being met. Thankfully, our team does not believe any of the requirements are high risk. As for the compliance method, "A" indicates that the design requirement will be met through analysis. "T" indicates that the requirement will be confirmed during testing. "I" indicates that the requirement will be met after visual inspection.

Table 1. Requirement Details with Assessment of Risk and Means of Compliance

Specification #	Parameter Description	Requirement or Target	Tolerance	Risk	Compliance
1	Safety	Very Safe	-	L	T, I
2	Seal Size	Given Sizes	-	M	A, T, I
3	Bearing Size	Standard Sizes	-	M	A, T, I
4	Grease Type	Standard Types	-	L	A, T, I
5	Speed	15,000 RPM	Max	M	A, T
6	Temperature	-55 to 70°C	Min to Max	L	A, T
7	Shaft Finish	Standard Finish	-	L	A, T, I
8	Sampling Rate	100 Hertz	Min	L	T, I
9	Acceleration	Rates TBD	-	M	A, T
10	Size	5832 in^3	Max	M	A, I
11	Lifespan	1000 Cycles	Min	L	A, T, I
12	Power Source	115 V, 20 A	Max	M	A, T
13	Electrical Interference	Shielding Wires	-	M	T, I
14	Thermotron Interface	Specs Provided	-	M	A, I
15	Accuracy	±2% , ±5%	Max	M	A, T, I
16	Number of Operators	1 Person	Max	L	I
17	Vibration	None	Max	M	A, T, I
18	Torque	20 in-lb	Max	M	A, T

Project Management

We managed the project by dividing the design into a list of tasks and assigning each task to the most qualified person for that task. The design duties were split into two categories: team duties and design duties, as shown in Table 2 below. Some of these tasks are representative of the subsystems of the design. Each team member has been focused on designing specific subsystems and integrating them into the system effectively.

Table 2. Team Member Responsibilities for Subsystems, Design, and Business

<table>
<tr><th></th><th>Mustapha Nadri</th><th>Kyle Schless</th><th>Lynda Tesillo</th></tr>
<tr><th>Specific Design Area</th><td>Motor Selection/Implementation</td><td>Fixture and Mechanical Design</td><td>Sensor and Data Acquisition Selection/Programming</td></tr>
<tr><th rowspan="5">Design Duties</th><td>Stress/Strength Analysis</td><td>3D Modeling and Technical Drawings</td><td rowspan="2">Electrical Wiring and Wiring Diagrams</td></tr>
<tr><td>Temperature Analysis</td><td>Manufacturing Considerations</td></tr>
<tr><td>Vibration Analysis</td><td rowspan="2">Prototype Fabrication and Aesthetics</td><td>Data Output/Input</td></tr>
<tr><td>Lifetime Calculations</td><td>Material Selection</td></tr>
<tr><td>Testing Plan</td><td>Refining the Model</td><td>Control System Design</td></tr>
<tr><th rowspan="3">Team Duties</th><td>Maintain Team Budget</td><td>Maintain Information Repository</td><td>Sponsor Communication</td></tr>
<tr><td rowspan="2">Organize Product Purchases Through School</td><td>Meeting Minutes Documentation</td><td>Meeting Organization</td></tr>
<tr><td>Meeting Agenda Generation</td><td>Organize Product Purchases Through Sponsor</td></tr>
</table>

All of the team members have performed background research and gathered information about old methods of measuring friction in bearings and seals. Furthermore, all the team members have gathered detailed information about the project requirements and specifications.

Table 3 below lists the important milestones in the design process. All of the tasks listed have been completed and fulfilled. The project Gantt chart, attached in Appendix D, details the tasks and milestones completed, along with their durations. Also included in the Gantt chart is the total amount of hours worked on each project task.

Table 3. Timetable of Milestones

Date	Milestone
10/24/2013	Project Proposal Submitted
11/21/2013	Pre-Concept Review
12/5/2013	Conceptual Design Report Submitted
12/11/2013	Conceptual Design Review
2/7/2014	Critical Design Report Submitted
2/17/2014	Critical Design Review
3/4/2014	Manufacturing and Test Review
4/1/2014 - 4/10/2014	Assembly Period
4/11/2014 - 4/27/2014	Integration Period
4/28/2014 - 6/6/2014	Testing Period
5/31/2014	Senior Design Expo
6/13/2014	Final Report and Machine Delivery

Chapter 2 – Background

Existing Solutions

Parker Hannifin has conducted a similar friction test on bearings in the past. The test was a result of the need to find sources of frictional losses in the actuator systems and quantify the viscous drag associated with the use of various lubricants. This test was conducted using pre-existing, non-specific test equipment, as well as ad-hoc test procedures. Major takeaways from this previous test will be applied to our project, including expected torque values, conditions that produce the best data (ramp-up speeds), and the need for a higher degree of safety.

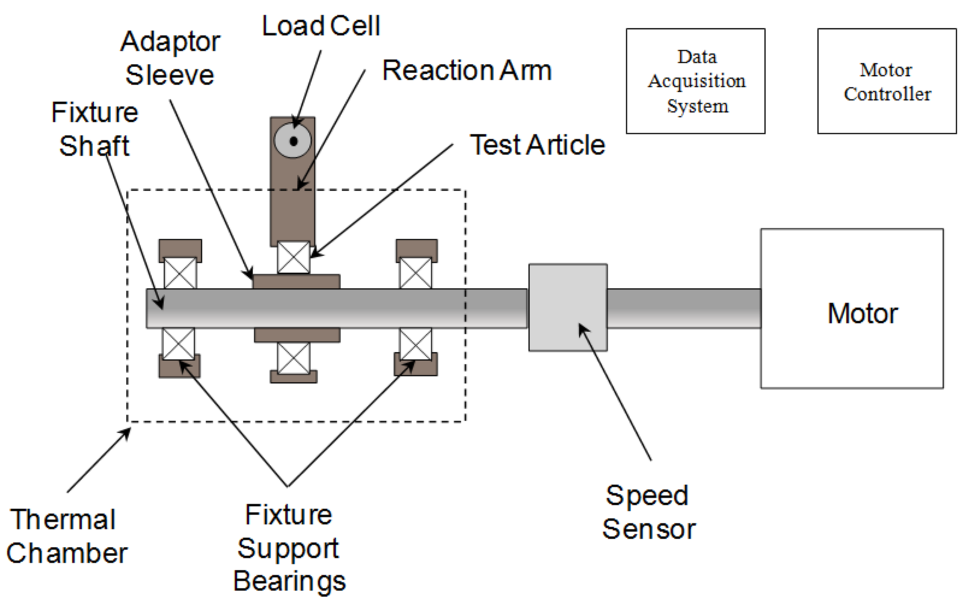


Figure 1. Initial Proposed Solution from Parker Hannifin, Aerospace Group

When Parker Hannifin presented the project to our group, we were provided with a proposed solution to the problem. This proposed solution can be seen above in Figure 1. The proposed fixture consisted of a motor, equipped with a speed sensor and controller that would drive a shaft containing the test article. The fixture would come with adaptor sleeves that would allow various sizes of bearings and seals to be mounted on the test shaft. The friction drag generated by the test sample would be measured using a load cell attached to a reaction arm, which would be in contact with the sample's outer race. The fixture would measure speed, temperature, and torque from the load cell through a data acquisition system. Our team has considered this solution, but we have chosen to propose an alternate solution.

Current State of the Art

Besides the previous testing that Parker Hannifin performed, there have not been many new technologies that measure the drag torque in bearings and seals. A patent search was done and no documents registered within the last ten years were found. Also, we were unable to find any patents that directly achieved the goals of this project.

Specific Technical Data

Because there are not many machines that test the drag torque of bearings and seals, the only technical data benchmark we have is the data obtained by Parker Hannifin in their previous testing. Unfortunately, because these tests were performed inconsistently and were not standardized, the results could be seen as somewhat unreliable. So, we may compare our obtained values to those obtained by Parker Hannifin mainly to verify that similar trends occur and the measured values are in the correct range of values.

Applicable Standards

Our team has investigated various resources for bearing and seal friction testing standards, including the Society of Automotive Engineers (SAE) and the American Society for Testing and Materials (ASTM). We were only able to find the *Standard Test Method for Determination of the Breakaway Friction Characteristics of Rolling Element Bearings* from ASTM. This standard determines the breakaway friction torque of various bearings in operation. However, this does not match the objective of our project very well. Although the measurement of the friction torque in the bearings will, indeed, log the breakaway friction point, we will not be exclusively focusing on when the breakaway occurs or the corresponding friction. We were unable to find any standards for seal drag and friction testing. The measurement of the drag that occurs between the inner and outer races of bearings and seals is not something that has explicitly been measured, based on our research. Therefore, we will not be basing our design on any testing standards. However, we will be adhering to the American Society of Mechanical Engineers (ASME) Code of Ethics. This document details our responsibility to design this test apparatus to the highest degree of safety and uphold ethics throughout its design and production.

Chapter 3 – Design Development

Discussion of Conceptual Design

During the conceptual design stage, the design for the test machine was divided into several subsystems. A motor was necessary to rotate the test samples. In order to fluctuate the speed and torque output from the motor, a drive train was needed in conjunction with a speed controller. A torque sensor system was needed to detect the frictional torque from the bearings and seals. A temperature sensor system was needed to measure and monitor the testing temperature conditions. A speed sensor system was also needed to measure and monitor the output shaft speed to the test sample. A clamp was needed to hold the outer race of the bearing or seal still in order for the friction torque to be detected. Because we needed to accommodate a continuous range of bearing sizes, an appropriate shaft design was made for bearing testing. Because there were a discrete number of seal sizes to be tested and the shaft-seal inner race interface was necessary for accurate friction drag measurements, a different shaft design was made for seal testing.

For idea generation, we generated as many possible solutions as we could that fulfilled the customer and engineering requirements. The best solutions were chosen and were integrated into the conceptual design, as seen below in Figure 2.

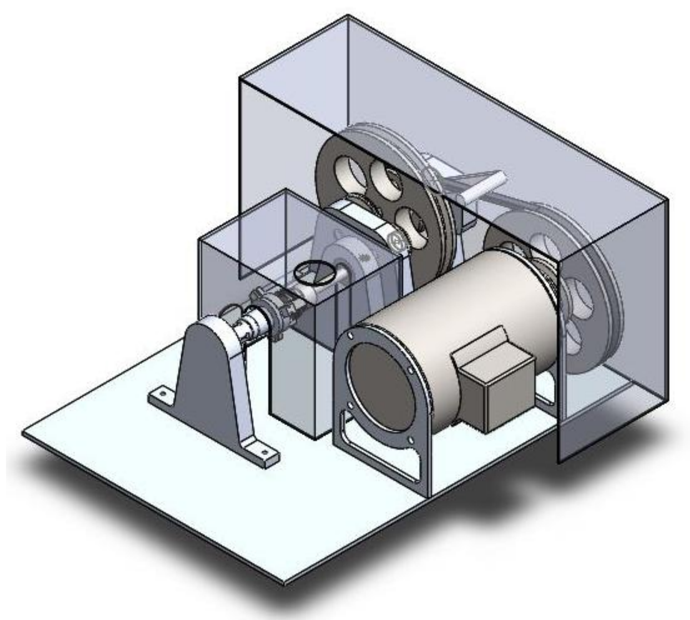


Figure 2. Conceptual Design SolidWorks Model

The layout of the conceptual design was chosen in order to provide a safe, compact, and accessible design. The output from the motor attached directly to a shaft that held two pulleys. The intermediate gearing shaft included two pulleys and a gear, held between two bearing supports, as shown in Figure 3. The tensioning mechanism attached directly to one of the bearing supports to provide simplicity and ease of access. The main support bracket included bearing races for the intermediate gear shaft and the output shaft, as well as a speed sensor bracket. The output shaft would screw into to the test fixtures and locate center with a shoulder. Both the bearing and seal attachments would locate the bearing in the same position axially, so the outer race clamp would not have to slide.

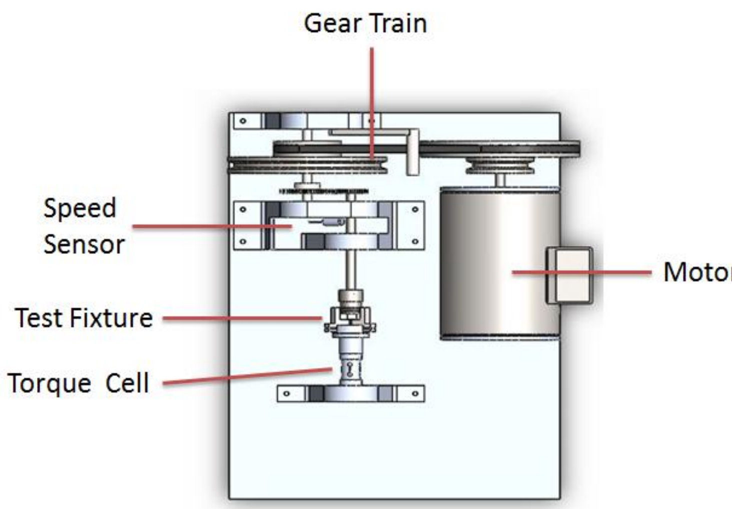


Figure 3. Top View of Conceptual SolidWorks Model with Subsystems Identified

The selected motor was a Baldor ¼ horsepower brushed AC motor, seen in Figure 4, that has a maximum speed of 3450 RPM with a corresponding output torque of about 5 in-lb. This motor would allow us to have reasonable gear ratios, which would satisfy Parker Hannifin’s speed and torque range requirements.

Figure 4. Baldor L3356 ¼ Horsepower Brushed AC Motor

The selected speed controller was the Dart Controls AC Motor Speed Controller, shown in Figure 5 below. Parker Hannifin required us to obtain a rotational velocity as low as 200 RPM for an output torque of 20 in-lb. With the motor and gear ratios selected, this could not be achieved without a speed controller. When the torque is 20 in-lb, the rotational output speed is approximately 720 RPM, as detailed in the motor calculations in Appendix E. Therefore, the speed needed to be reduced, while keeping the same output torque level.

Figure 5. Dart Controls AC Motor Speed Controller

Figure 6 and 7 below show the selected drive train design, which utilized a V-belt pulley drive system and a single gear ratio. V-belts were selected because they are relatively inexpensive, capable of higher speeds than other belts and easily interchangeable. However, because the maximum peripheral speed that the belts can safely experience is 5000 ft/min (6366 RPM) for the minimum pulley diameter of 3” according to *Shigley's Mechanical Engineering Design*, the belts could not be used to perform the final increasing ratio up to the higher rotational speeds. Therefore, a spur gear pair was needed to achieve the higher speeds. When using V-belts, there is a need to include a tensioner, which tensions the belt for proper use. When the tensioner is not in contact with the belt, the belt would have some slack in it to ease the movement of the belt from one pulley set to the other.

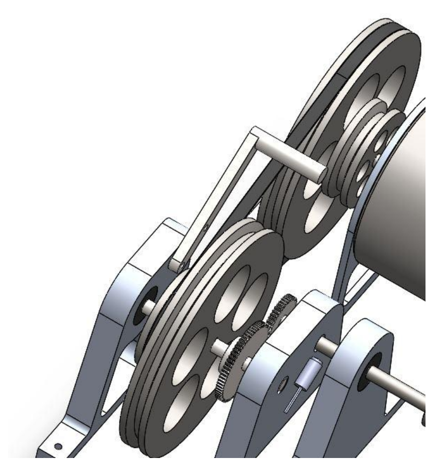


Figure 6. Close Up View of Selected Drive Train Design

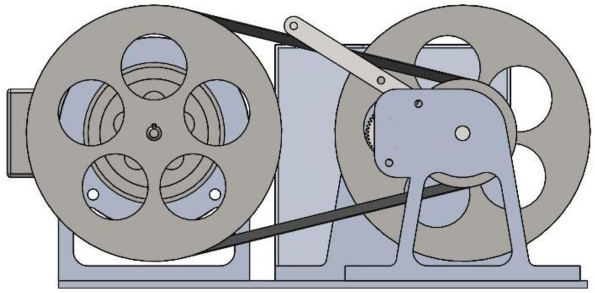


Figure 7. Side View of Selected Drive Train Design

Figures 8 and 9 below show the selected bearing-shaft interface design. There would be a selection of cones to accept a larger range of continuous bearing inner diameter sizes. The cones would hold the inner race of the bearings securely and ensure concentricity with the output shaft. It must be kept in mind, however, that the fabrication of the cones must be done with great care. Any slight asymmetric feature/flaw would cause imbalance in the bearing testing, thus altering the recorded data and alignment of the system.

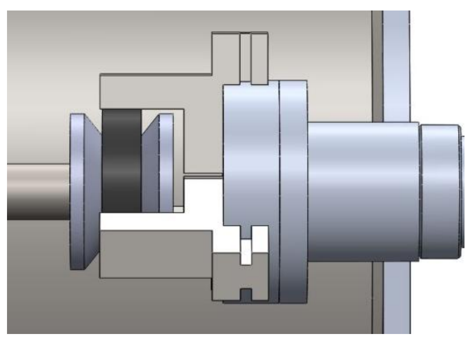


Figure 8. Side View of Selected Bearing-Shaft Interface Design

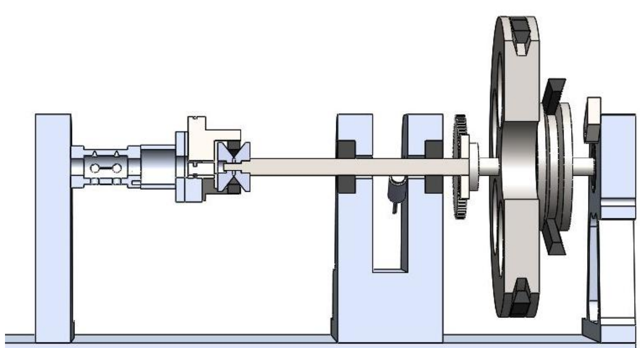


Figure 9. Cross Sectional View of Selected Bearing-Shaft Interface Design

Figures 10 and 11 below show the selected seal-shaft interface design. With the high tolerances required for the shaft sizes, a multiple-stepped shaft would cause many manufacturing difficulties. Also, multiple steps would require the testing location of the seal to change with respect to where the appropriate shaft size is compared to the end of the output shaft. A simple single step shaft design would create consistency for seal test placement, thus allowing the outer race chuck to not have to move linearly. It would also allow for and promote future expansion for this test machine to include a greater selection of shaft and seal sizes.

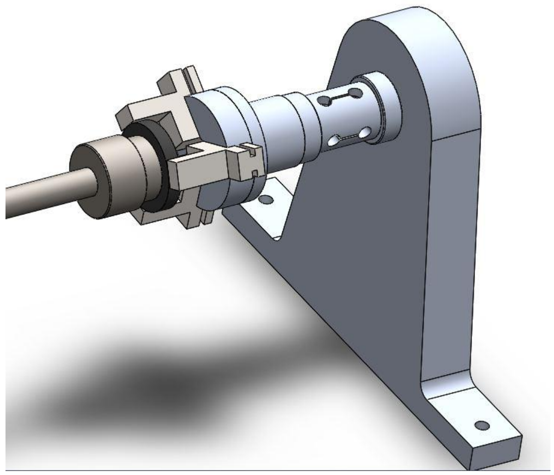


Figure 10. Close Up View of Selected Seal-Shaft Interface Design

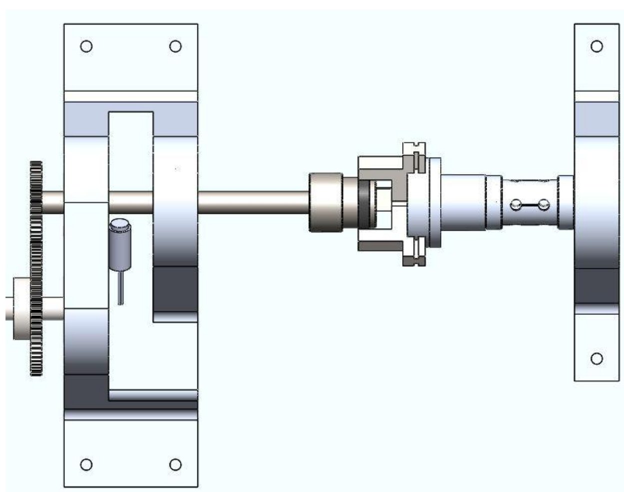


Figure 11. Top View of Selected Seal-Shaft Interface Design

The data acquisition system selected was the NI USB-6000, as seen in Figure 12 below. This DAQ had eight analog voltage inputs and was small in size. It was compatible with LabVIEW software, which is available for our team through the university. Having this access through Cal Poly would be very useful for the testing phase of the design process. The voltage input range was ±10 Volts, which exceeded the voltage outputs expected from sensors. Its sampling rate

was around 10000 samples/second, which far exceeded the required sampling rate of 100 samples/second (or 100 Hertz). Although the initial intent was to have a smaller sampling rate to eliminate the potential for noise, the test runs will be happening quickly, so it was important to record the behaviors of the samples as accurately as possible. Additionally, this DAQ connected to any PC or laptop equipped with LabVIEW via a USB connection. This made setup for the operator very easy and ensured that the machine would be portable and lightweight. The power source came from the PC or laptop, decreasing the amount of power required to operate the machine through a wall outlet.

Figure 12. National Instruments USB-6000 Data Acquisition System

To measure the temperature inside of the thermal chamber, a thermocouple would be placed inside the chamber and wired back to the data acquisition system, located outside of the chamber. With a desired temperature range from -55 to 70 degrees Celsius, a thermocouple that operates in this range would need to be used. A Type T thermocouple would work well for the temperature range expected, as it has a useful range from -250 to 350 degrees Celsius. The measuring side of the thermocouple would be inside the thermal chamber, while the opposite end would be connected to the data acquisition system for the measurements to be read and recorded. The DAQ input voltage range was ±10 Volts, while the thermocouple maximum output was around 20 mV. Therefore, this thermocouple was compatible with the DAQ and required temperature range.

The selected torque sensor was the Futek TFF325 Flange to Flange Reaction Torque Sensor, seen in Figure 13 below. The maximum measurable torque was 50 in-lb, which exceeded the maximum expected bearing and seal friction torque of 20 in-lb. Also, the output was measured in Volts and was able to be read directly from the DAQ. However, the output was rated from a range of voltages of 1 to 18 Volts DC. Because the maximum voltage that can be read by the DAQ is only 10 Volts, this could have presented an issue. Thankfully, the maximum output would be associated with the maximum torque, which would never be reached. The maximum friction torque expected from the bearings and seals was less than half of the maximum output of the sensor, thus implying that the maximum voltage output expected would be less than 9

Volts DC. The torque sensor would be located outside of the thermal chamber, so it would remain within the compensated temperature range, ensuring accurate measurements for testing. Because the outer diameter of the bearings and seals would be held by a clamp and not rotating, a reaction torque cell was required as opposed to a rotary torque cell. The other end of the clamp would be attached to the torque sensor, allowing for the friction torque experienced by the test sample and clamp to be measured by the sensor directly. Since the sensor was a flange-to-flange sensor, the non-measuring end would need to be held stationary to act as the datum. This end would be secured to an upright plate bracket.

Figure 13. Futek TFF325 Flange to Flange Reaction Torque Sensor

In order to measure the speed of the output shaft, the Allegro Microsystems A1324 Hall Effect Speed Sensor, shown in Figure 14 below, was chosen. This speed sensor had an analog output, meaning the output would be a voltage. This would be compatible with the DAQ selected. A Hall Effect sensor generates a signal when a magnet passes by the sensor, creating an electromagnetic field. The magnet would be placed on the output shaft and would trigger the sensor with every rotation, creating a voltage output that was proportional to the rotational speed of the shaft.

Figure 14. Allegro Microsystems A1324 Hall Effect Speed Sensor

The outer race of both the seals and bearings needed to be held stationary in order to measure the reaction torque. The clamping mechanism must work for both types of test specimens and be adjustable from 0.35” to 3.75” (as per the specified seal and bearing dimensions). Torque values of up to 20 in-lb would be reacted at the jaws, so they must provide a strong grip. The conceptual design featured a cantilevered torque cell, so a lightweight clamping mechanism was required in order to reduce the induced moment on the instrumentation. The Grizzly

H8033 3" – 3 Jaw wood lathe chuck, shown in Figure 15 below, met all of the requirements and could be procured with no lead time and at a low cost. The jaws would be reversible in order to accommodate the variable sizes. Chuck lathes are already balanced, so there would be no issue with concentricity. A 3-jaw model would provide sufficient clamping force.

Figure 15. Grizzly H8033 3'' – 3 Jaw Wood Chuck

The thermal chamber should be able to create a suitable environment that replicates the real environments in a smaller scale. The thermal chamber would be made of insulated plywood, which has a small thermal conductivity. However, the thermal chamber would have slots where the output shaft would pass through. The exposed area would be insulated with additional removable plywood or insulation, minimizing the heat losses. Also, the thermal chamber would be placed where it would not interfere with the output shaft or rotating equipment.

Supporting Preliminary Analysis

With the given rotational speed range from 0-15,000 RPM that Parker Hannifin required, we needed to find a motor that could be stepped up or down to the maximum and minimum speeds easily. We found the Baldor ¼ HP motor with a rated speed of 3,450 RPM. Using basic gear and torque ratio laws, this motor would output a torque value of 1.05 in-lb at 15,000 RPM and 22.84 in-lb at 630 RPM. This was calculated using the gear ratios of 1:4.31 and 5:1, respectively. We would need to step the motor speed down from the minimum output speed of 630 RPM to 200 RPM using a speed controller. The rotational speed and torque values were higher than the required values, which would ensure that we could overcome the friction losses from the support bearings and shafts. The detailed calculation for the conceptual design motor verification can be found in Appendix E.

Discussion of Critical Design

The critical design improves upon and fixes problems found in the conceptual design.

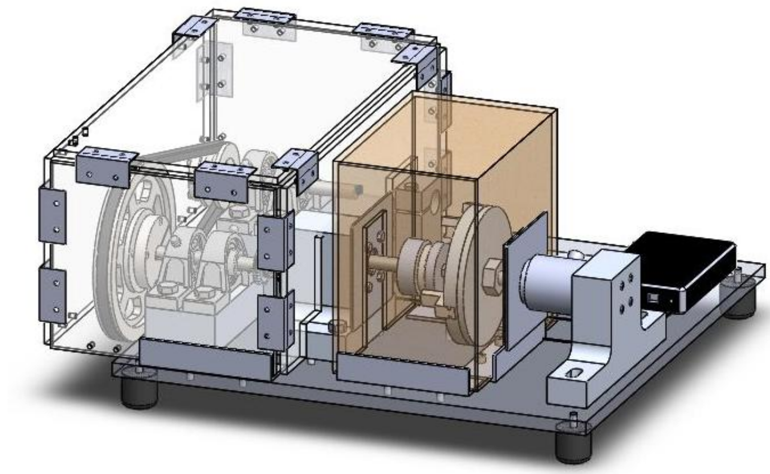


Figure 16. Full Assembly of Critical Design

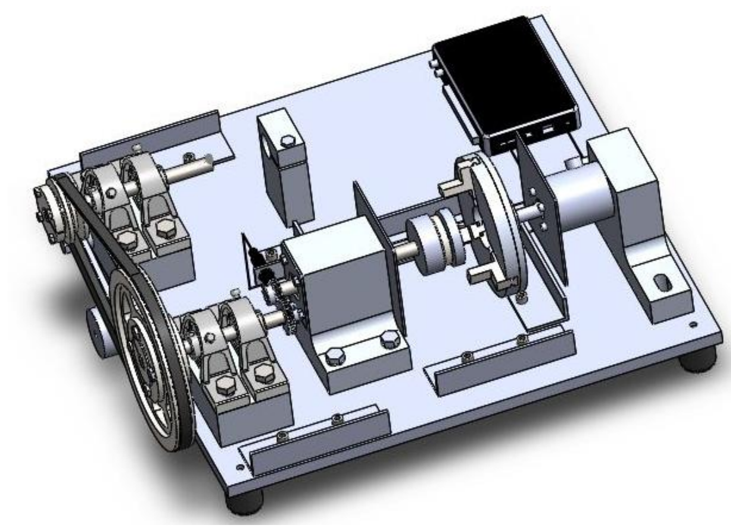


Figure 17. Final Detailed Critical Design Layout

Our critical design, as seen above in Figures 16 and 17, is a system that is able to measure drag torque and shaft speed under different temperatures of a wide range of bearings and seals. It essentially performs a simulation on the bearings and seals to undergo the real life stresses that they are submitted to. The power source is a brushed DC motor (not shown above) in the form of a corded drill assembly that is able to generate the power, torques, and speeds required for our test. The power is transmitted through a drive train containing a V-belt pulley system and

spur gears. The drive train is shielded for safety purposes, due to the presence of high speed rotating equipment. A Thermotron will pump air through flanges located in the walls of a thermal chamber that is insulating the test bearing or seal. To obtain torque, speed and temperature data, we will be using a torque and speed sensor and thermocouple, whose output signals will be fed into a DAQ. The attachments used to test bearings will be sets of two cones and the attachments used to test seals will be various sizes of stepped shafts. An external jaw chuck will be attached to the torque sensor and will grab the outer diameter of the seals and bearings in order to measure the drag torques they experience.

The rotational speed range required by Parker Hannifin was 0-15000 RPM, meaning that we would need to find a motor that could be stepped up or down to this range. Also, this motor should have a speed controller. The originally selected motor was a Baldor ¼ HP, 3,450 RPM motor that cost almost $400. The problem with industrial motors is that they are expensive and were designed to run at a constant speed. Most of these motors used start capacitors, making selection of a speed controller difficult because the speed controllers that we could find on the market were not able to function with a start capacitor motor. The appropriate speed controller cost nearly $50. We thought we could improve the motor, so we decided to keep looking for other options to minimize the price and maximize the performance. The best motor we could find was a Ridgid Drywall Drill for $20. Drill motors have a tendency to heat up relatively quickly. A two hour test would overheat the windings in the drill's brushed motor and potentially create a fire hazard. However, our test will only run for about 30 seconds, which ensures the drill is appropriate for our design. Knowing that other components of our system, such as the high-sensitivity torque cell, will cost us a relatively large amount of money, this motor was financially the best choice. The drill will be engaged with either a large or small pulley, depending on whether the user desires to increase or decrease the speed. The user will be able to vary the speed via a speed controller included in the drill. This speed controller is able to lock into various speed values to allow the user to obtain constant speeds. To minimize any overshoot in speed, we have determined that the motor speed should be slowly adjusted. We need to reach steady state speeds without exceeding the limitations of the system to avoid any unexpected failure. We chose a corded model because we would not need to worry about charging the motor. The drill will be coupled to the input shaft of the drive train with the use of a coupling housing, which will also secure the drill down for stability during testing.

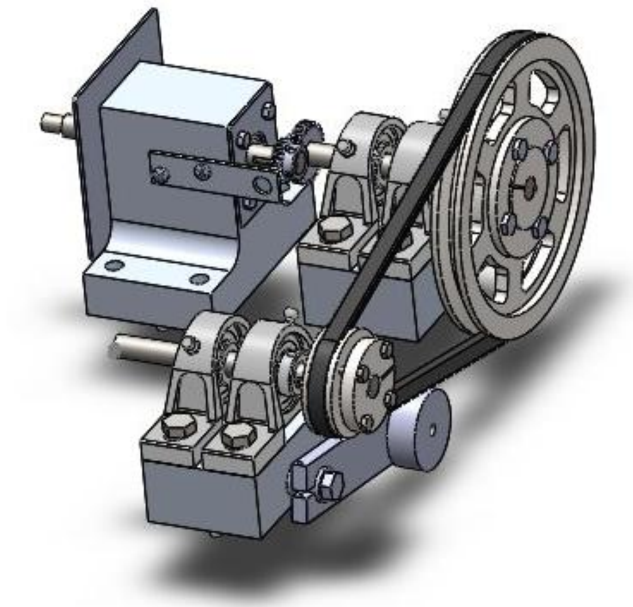


Figure 18. Drive Train Sub-Assembly

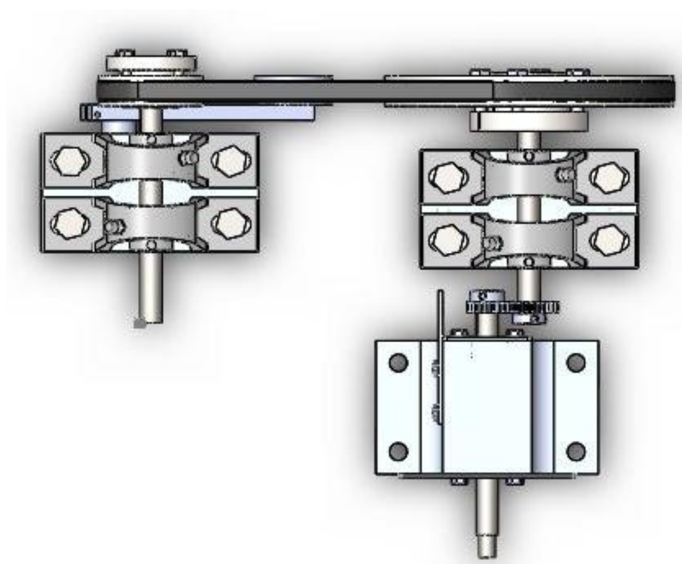


Figure 19. Top View of Finalized Drive Train Design

Parker Hannifin required us to achieve a torque of 20 in-lb at low speeds. Therefore, we needed to design a high performance drive train that would accomplish the task. High speed rotations cause unwanted vibration that may disturb the accuracy of our system. The chances of a system failure increases as well. Vibration would disturb the load cell measurements, as well as the

speed sensor measurements. Since Parker Hannifin required us to obtain precise data measurements, this is not desired. So, we decided to use a reversible V-belt pulley system design as the first stage reduction (2.35:6.5 ratio) or increase (6.5:2.35 ratio) because they are cheap, easy to maneuver, and reliable. We can switch the orientation of the belt, which gives us reduction and increase capabilities. For the second stage reduction, we decided to use a spur gear system (1.5:1 ratio) because of space constraints and the high speeds that they could withstand. Moreover, we were unable to find any pulleys that could withstand 15,000 RPM. The selection process for the first stage gear ratio led into finding the critical speeds at which these pulleys would fail. Thus, we tried to find the smallest pulley possible and then adjust the other pulley's size in accordance to the desired ratio. This is due to the fact that the small pulleys can run at very high speeds compared to larger ones. The spur gears have a maximum speed limitation of 20,000 RPM. However, our maximum required angular speed output is 15,000 RPM, meaning that using the V-belts and gears in this orientation, as seen above in Figures 18 and 19, will achieve the necessary performance values. A belt tensioner and idler pulley will be used to tension the belt during testing and loosen it for switching the desired ratio and maintenance.

We made sure to find bearings that would be suitable for our design. So, we calculated the dynamic bearing rating C_{10}, which can be found in Appendix E, and selected bearings accordingly. We determined which bearing will be under the largest load and calculated the dynamic rating, then used it to select the other bearings. The selection of bearings with good lubrication is necessary to minimize adding any friction losses to our overall system. The bearings that we selected are very light and are able to withstand the high speed conditions to be experienced by the drive train. The entire test fixture will be mounted onto a base plate which will enable the design to be portable. This is why minimizing the weight is a necessity.

The fixture also should be able to contain the hot or cold air generated by the Thermotron. So, we designed a thermal chamber, seen in Figure 20 below, which will contain the air output from the Thermotron. Using plywood for the thermal chamber design was preferred because it is cheap and abundant. Insulation will be added to the inside surface of the thermal chamber by Parker Hannifin. The thermal chamber has two 4'' flanges to allow the Thermotron output air to interface with the test sample. We attempted to minimize the heat losses out of the thermal chamber by reducing the amount of holes and gaps. We have shielded the various components, including the torque sensor, in close proximity to the thermal chamber because the humidity in the output air can potentially cause rust, decreasing our system lifespan. Furthermore, the electrical components need to be protected from the Thermotron air because the humidity would cause damage and failure. We also do not want these potentially extreme temperature conditions to harm the user.

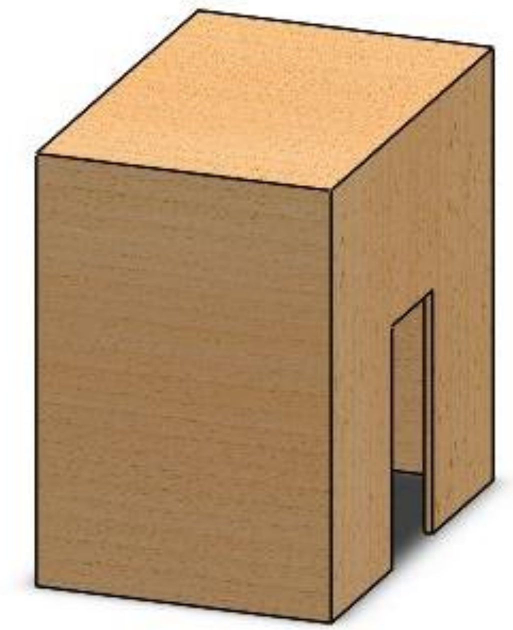


Figure 20. Thermal Chamber

In order to ensure that the tested seals would remain concentric with the output shaft during testing, we designed output stepped shaft attachments that would interface with the inner race of the seals, causing the seal to rotate. We made these attachments lightweight to avoid imbalance within the drive train, which would cause potential error in acquired measurements. To accommodate the range of seals sizes provided by Parker Hannifin, several attachments of different sizes were created. The external jaw chuck jaws will be able to grab the outer diameter of the seal to measure the drag torque experienced by the seal as a result of various testing conditions.

We also designed a few sets of cones that will enable testing for a continuous range of bearing sizes. These cones will be bolted onto the output shaft and tightened against the inner race of the bearing, creating two circles of contact, as can be seen in Figure 21 below. The cones were made to be thin and lightweight to avoid imbalance within the drive train, which would cause potential error in acquired measurements. Again, the external jaw chuck jaws will grab the outer diameter of the bearing to measure the drag torque experienced as a result of various testing conditions.

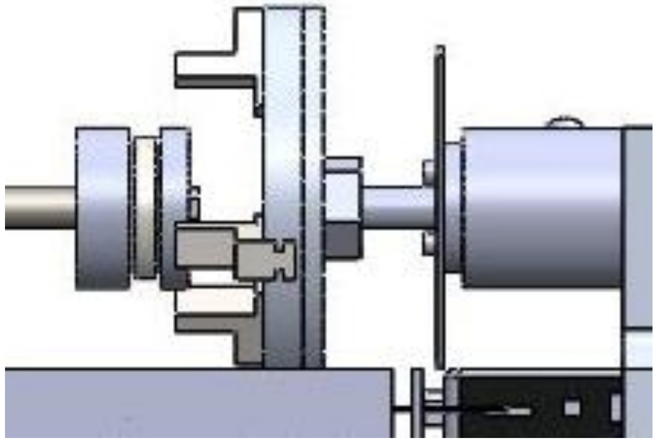


Figure 21. Bearing Cone Attachment Configuration

The torque sensor selected during the conceptual design stage was unable to be implemented into the critical design. This is because it was an OEM model, meaning that it was not meant for final testing use. The strain gauges within the sensor and lead wires were fully exposed, which posed a problem since the air from the thermal chamber could significantly damage and rust the sensor. So, we selected the Futek TFF400 Reaction Torque sensor, which is the finalized version of the OEM model selected before. The torque sensor will be attached via a flange to a housed bearing from one side and to the external jaw chuck from the other side. The external jaw chuck will be placed onto a slotted shaft connected to the end to the torque sensor, which will allow the chuck to adjust axially to the position of the test sample. We also made sure to shield the torque sensor from any air coming out from the Thermotron by placing a plate between the thermal chamber face and the sensor, as seen below in Figure 22.

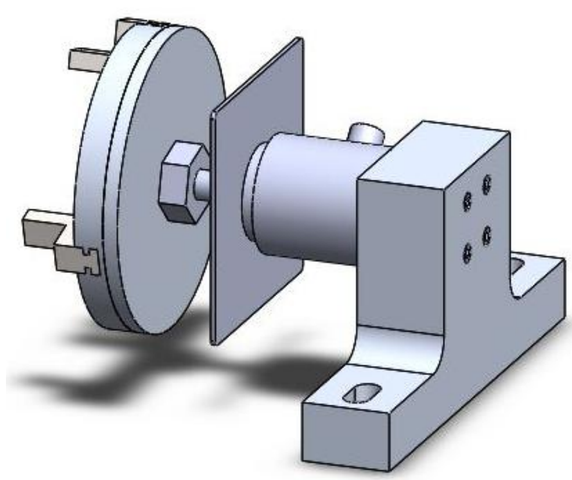


Figure 22. View of Torque Sensor Placement

The DAQ originally selected during the conceptual design phase was unable to be implemented into the critical design. This is due the fact that the sensors were changed, which required different output power and input requirements from the DAQ. The torque sensor was changed to a final design model, as previously detailed, but the output and power requirements remained the same. The Type-T thermocouple output signal could not be read by the previous DAQ without additional circuitry and wiring. Also, the previously selected Hall Effect speed sensor was changed to a Hall Effect gear tooth sensor, which will detect the rotation of the final 20 tooth gear and create a pulsing output signal. However, this output signal could not be properly interpreted with the previous DAQ because it utilized a software-timed counter. This means that the timing of the signals would be dependent on the software operations instead of interpreted in real time. In order to accommodate these changes, the National Instruments myRIO DAQ has been chosen. Although this system is slightly more expensive than the previous DAQ, it is able to handle the thermocouple and gear tooth sensor signals appropriately. The DAQ has an FPGA processor that enables it to be programmed to accurately interpret real time frequencies. It also has an available module that can provide internal cold junction compensation and wiring so the thermocouple leads can be directly input into the terminals.

Supporting Analysis

The first and main calculation performed was to verify the motor that would be able to achieve the torque and rotational speed values needed. After selecting the Ridgid drill motor, we were able to determine that it would easily exceed the necessary values because of its maximum output speed of 4,000 RPM and 1 HP rating. Using the finalized drive train ratios of 6.5:2.35 and 3:2 to obtain the highest speed output, the motor would output about 17,000 RPM and 3.8 in-lb of torque. Using the finalized drive train ratios of 2.35:6.5 and 3:2 to obtain the highest torque output, the motor would output about 30 in-lb of torque, which a speed that can be reduced to very low values using the speed controller in the drill. A spreadsheet was created to calculate the expected speed and torque values at various parts of the drive train, including at the pulleys to ensure that the pulley maximum speeds were not exceeded. This spreadsheet table can be found as Table 8 in Appendix E, along with the corresponding theoretical and basic hand calculations for the Ridgid drill capabilities.

Next, we needed to find the appropriate drive train hardware that would be able to withstand these high angular velocities. Appropriate high speed rated bearings were selected based on the calculated dynamic C_{10} value corresponding to the selected shaft sizes. This calculation can be found in Appendix E. Additionally, we needed housed bearings so they could easily be mounted onto the base plate. For the output shaft, two bearings were used to minimize

deflections and vibration and increase stability. In order to house these two bearing together, we designed a housing that would accommodate both together.

Another area of concern was the cantilevered output shaft's deflection, which would cause undesired vibration within the system. In order to help mitigate this problem, we selected a short, stout output shaft, and then conducted deflection analysis in accordance with *Shigley's Mechanical Engineering Design*. We needed to be within an allowable range of slopes caused by the loads. To perform the analysis, we only considered linear slopes, while, in fact, the shaft could experience deflection slopes in two dimensions. The deflection in the other dimension would be caused by misalignment in the shaft layout, which should be reduced and avoided during the fabrication and assembly stages. The linear model created allowed us to approximate the deflection values for various loading conditions. The numerical results of the deflection calculations can be found in Table 9 in Appendix E, along with the corresponding theoretical and basic hand calculations.

The critical speeds of the output shaft were also analyzed using the Rayleigh method, which determines the critical speed of the first mode of vibration. The results for the bearing and seal testing setups can be seen below in Figures 23 and 24 and in Table 9 in Appendix E along with the corresponding and theoretical and basic hand calculations. We predicted that the critical speeds would decrease as the sizes and weights of the test samples increased. This is because the critical speed is inversely proportional to deflection. After plotting the critical speeds, we observed this trend, which confirmed that the spreadsheet calculations performed were reasonable. However, testing will be crucial in determining the actual behavior of the assembled system.

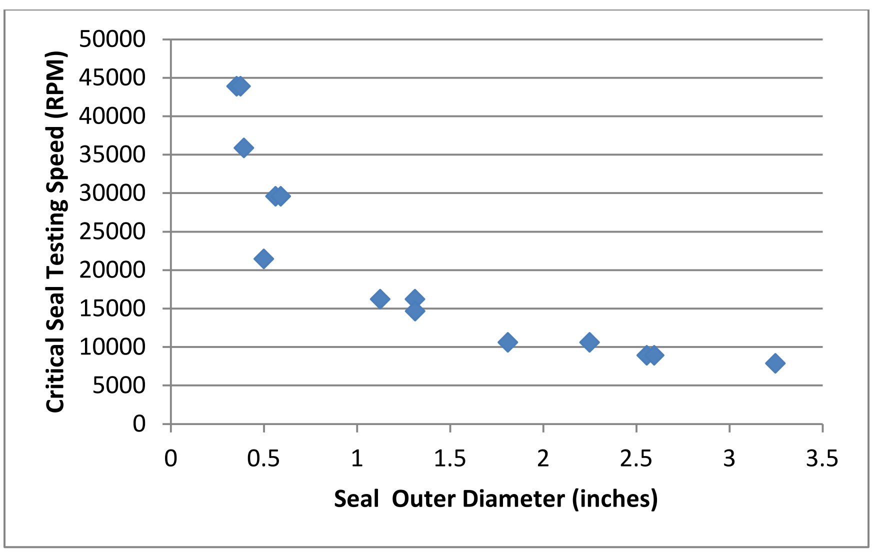


Figure 23. Calculated Critical Speeds for Desired Seal Outer Diameters

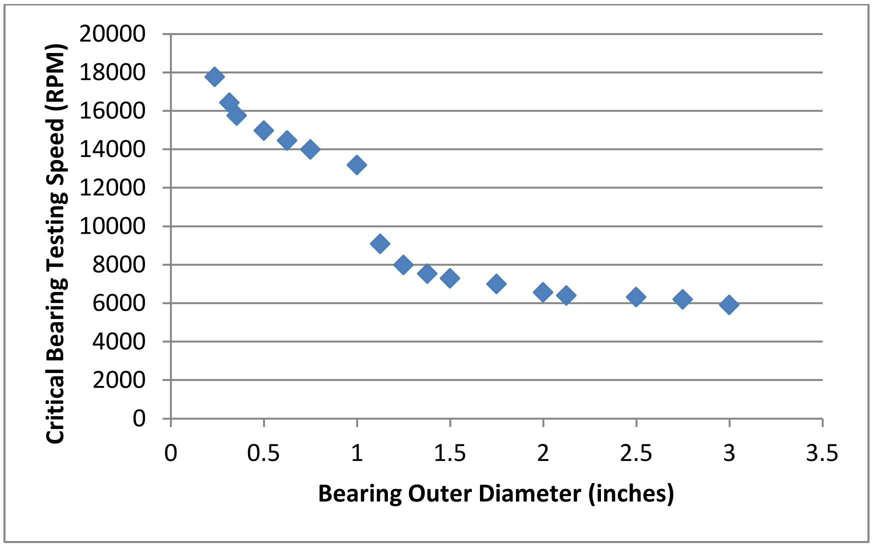


Figure 24. Calculated Critical Speeds for Desired Range of Bearing Outer Diameters

Concept Selection

All the initially generated solutions were benchmarked in terms of how well they met the customer and engineering specifications using Pugh Matrices. The Pugh Matrices created by our team for this project during the conceptual design phase, as well as an explanation about its use and features, can be found in Appendix A.

There were four ideas considered to implement the drive train design. The first option involved using a multiple shift gearbox. This would be a gearbox with functionality similar to a manual transmission. The self-contained, pre-assembled and purchased gearbox would contain a variety of gear ratios that would allow for the increase and reduction of speed and torque. A lever would shift the gears and engage new ratios very easily and quickly. The second option was a manually reversible gearbox. This would be a gearbox with a ratio of 5:1, for example, that could be manually rotated by the operator to create a ratio of 1:5. The gearbox would be attached on both ends with easily removable couplers. The third option was to use only a variable speed motor. This configuration would use the direct output of the motor to rotate and test the sample. The last option was to use belt drives. There would be a two-stage pulley system that would allow for a quick change between an increase and decrease in speed by simply changing the belt configuration. In order to select a solution, a Pugh Matrix was created for the drive train subsystem. The matrix can be found as Table 4 in Appendix A.
The results of the Pugh Matrix indicated that a multiple shift gearbox would be a good solution. However, upon further inspection, this option was bulky, expensive, and difficult to maintain

and repair. All of these negative attributes made the multiple shift gearbox a poor solution. The manually reversible gearbox may have reduced the system losses, but it was also slightly expensive and was not quick to assemble or change the orientation. This was because the gearbox would need to be removed from the couplers, rotated, and reattached to the couplers. This would also possibly cause alignment issues. Using only the variable speed motor allowed for the full range of speeds but only allowed for the nominal torque output level of the motor, thus not achieving the required torque range. The motor could also overheat or stall at the extreme operating conditions. The belt drives met all of the criteria. They were easy to repair and assemble and fairly inexpensive. There are losses introduced by using the belts but these losses were assumed to be minimal. In addition, the motor output torques would exceed the expected friction torques, accounting for any additional losses the motor must overcome. Therefore, the belt drives were found to be the best solution.

There were a total of two ideas generated to measure the friction torque of the bearings and seals. The first idea utilized the proposed solution from Parker Hannifin of a load cell with a reaction arm. This configuration would consist of a reaction arm attached to the outer race of the bearing or seal that will engage a load cell when a friction torque is detected. The force detected by the load cell would be proportional to the friction torque experienced by the bearing or seal. The other idea was to use a reaction torque sensor. A reaction torque sensor directly measures the torque experienced by an object. This configuration would consist of a reaction torque sensor attached to the outer race clamp. This clamp would experience torque because it would be moving with the outer race. Therefore, the torque sensor would measure the torque experienced by the outer race that is transmitted through the clamp. In order to select a solution, a Pugh Matrix was created for the Torque Sensor subsystem. The matrix can be found as Table 5 in Appendix A.

From the Pugh Matrix, it could be seen that many of the concepts seemed to be great options. However, upon further inspection, it could be seen that three of the concepts were worse than the datum when it came to measurement accuracy. Because of this, these concepts were not good solutions, as accuracy in the measurements was a very important customer requirement. Using the reaction torque cell outside the Thermotron and attached to the inside race of the sample was not a solution that was compatible with the bearing and seal configurations. Therefore, this option was ruled out. Using the reaction torque cell outside of the Thermotron and measuring from outside race of the sample was seemingly the best solution. Compared to the datum, this option met or exceeded the criteria standard set by the datum. Therefore, using a reaction torque cell outside of the Thermotron measuring the outer race of the sample was the selected solution.

There were six concepts generated for the bearing-shaft and seal-shaft interface designs. The first idea for both the bearing and seal designs was to use interchangeable sleeves. This would consist of using numerous sleeves that are slid onto and secure to the output shaft, essentially increasing the shaft diameter to the necessary size. The next proposed concept for the bearing and shaft designs consisted of using different sizes shafts. This configuration would necessitate a shaft for every possible inner diameter size, making each shaft compatible with only one bearing/seal inner diameter. One concept for the bearing-shaft design was to use an expandable chuck to grab and rotate the inner race. The chuck would be able to adjust to fit the continuous range of inner diameters sizes for the bearings. Another concept for the bearing-shaft design is to use a dual cone design. The bearings would have a circular line of contact between the inner race and the cone, holding and rotating the bearing with cones. The cones are able to accommodate a continuous range of inner diameters due to their inherent continuously increasing outer diameter. For the seal-shaft interface design, one concept utilizes stepped shafts to fit a few discrete sizes of on the same attachment shaft. The seals would be mounted onto the appropriate sized step. The final design concept for the seal-shaft design consisted of an array of threaded shaft attachments for each needed inner diameter size. This concept was also incorporated into others because a threaded attachment would significantly reduce the amount of time needed for setup. In order to find a solution, a Pugh Matrix, found as Table 6 in Appendix A, was created for the Bearing/Seal-Shaft Interface subsystem.

From the Pugh Matrix, it could be seen that all of the concepts, except for the dual cone concept, were more expensive than the datum. Also, only two concepts showed to be stable, which was a large requirement since balance and alignment in our system was crucial. Therefore, it can be seen that the best option for the bearings was an adjustable cone design. The best design for the seal interface was a variety of stepped shafts, as this option was practical and easy to maintain and repair.

Chapter 4 – Description of Final Design

Overall Description and Layout

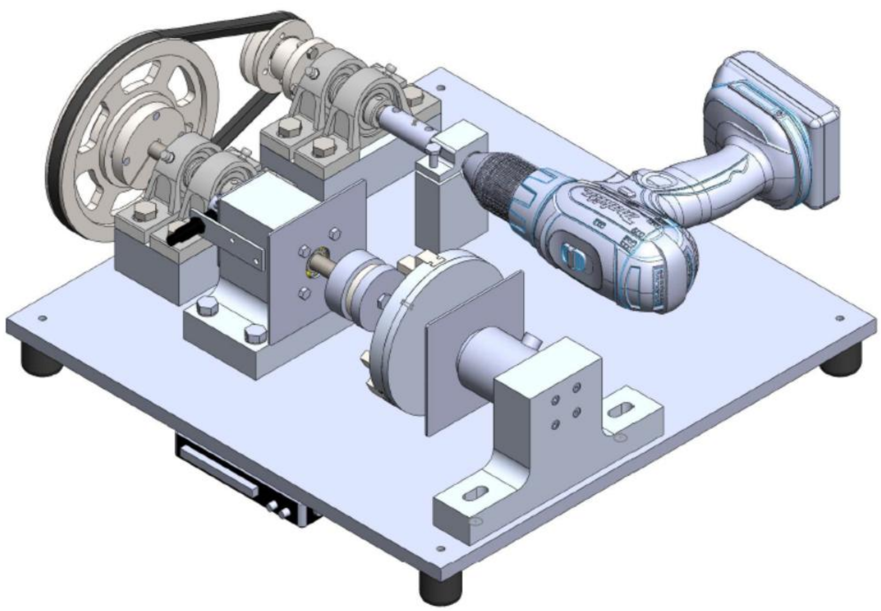


Figure 25. Final Design

The final design, seen above in Figure 25, is driven by a ¼ HP drill motor with an integrated variable resistance speed controller. The motor shaft is attached to the input shaft using a helical coupler. Two pairs of support bearings support the input and intermediate shafts radially. A pulley system, consisting of a V-belt and two bushings and pulleys, is attached to the end of both the input and intermediate shafts. The second stage of the drive train system consists of a pair of spur gears with a 1.5:1 ratio. A hall effect sensor is placed in close proximity to the 20 tooth gear on the output shaft in order to measure the output shaft speed. The output shaft is supported by two bearings that are press fit into an aluminum housing. The cantilevered end of the output shaft has a threaded hole, into which the bearing or seal attachments are secured for testing. The jaws of a 3-jaw chuck are secured onto the test bearing or seal outer race and holds it still while the inner race is rotating with the attachment. The chuck is secured to the face of a reaction torque sensor, which will measure the rotational displacement of the chuck that occurs due to the viscous drag torque in the test sample. This rotational displacement input to the torque sensor is sent through a strain gage amplifier to amplify and condition the signal, then output to the data acquisition system. The DAQ interfaces with a LabVIEW program that interprets the analog signal from the torque sensor

and the digital signal from the hall effect sensor and converts them into drag torque and output speed readings, respectively. A thermal chamber is placed over the test sample and interfaces with Parker Hannifin's Thermotron via two 4" flanges. A wooden cover is placed over the drive train to protect the operator from the high speed rotating machinery. Since the weight of the 3-jaw chuck is cantilevered off of the torque sensor, its weight must be vertically supported to avoid exceeding the bending moment rating of the sensor. So, a steel wire secured to a wooden support fixture through an eyebolt is tied to a hose clamp that is fastened around the chuck. The entire system is mounted on a 18"x18" aluminum plate using bolts and washers. All of the detail drawings for the parts we manufactured can be found in Appendix B. The specification sheets for the ordered components can be found in Appendix F.

Detailed Design Description

Figure 26. Ridgid Drywall Screwdriver

A Ridgid drywall corded drill, seen in Figure 26 above, was used to drive the system. This drill was chosen for its low cost, high speed operation, and ease of use. Fastening the drill to the base plate is an aluminum clamping support which grips the neck of the drill. The drill is coupled to the input shaft using a flexible helical shaft coupler, which adjusts for any misalignment in the two shafts. The drill operates using an internal clutch mechanism, so in order to bypass it for direct-drive operation, the head of the drill is preloaded into the coupler.

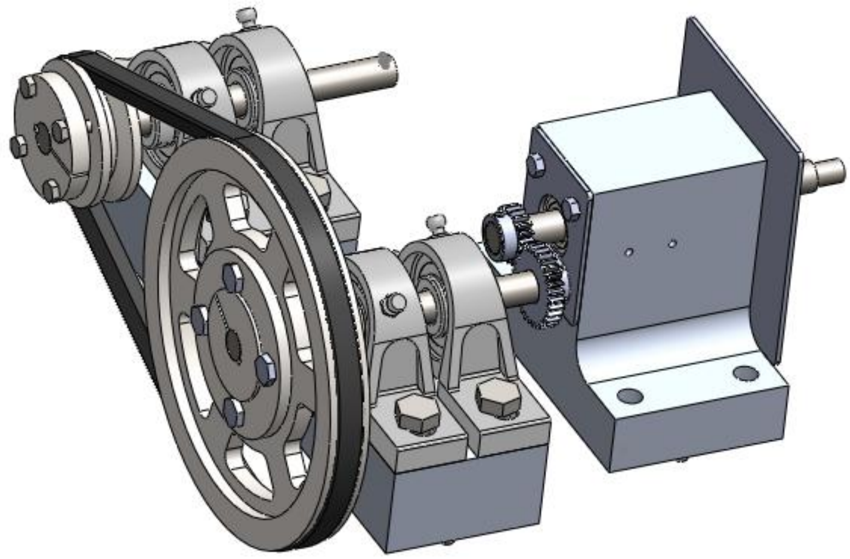


Figure 27. Drive Train Final Design

Reversible V-belt industrial pulleys were used for the drive train, seen above in Figure 27, to allow for two different speed configurations. By switching the locations of the larger and smaller pulley, we are able to achieve our speed requirements. The pulleys are attached to the shafts using a wedged bushing-sheave system. The belt is tensioned by expanding the distance between the two pairs of support bearings using the slotted pillow blocks. This eliminated the need for a tensioner and idler pulley, as was included in the critical design. The final drive consists of two steel spur gears with a 1.5:1 ratio. This entire setup allows us to meet our 15,000 RPM maximum speed, while still maintaining the necessary torque at the low speeds.

The input, intermediate, and output shafts were faced to length and turned down to the appropriate diameters. We utilized interference fits between the shafts and the support bearings in order to mitigate vibrations or slipping. We chose to use short steel shafts to minimize the deflections, especially on the cantilevered portion of the output shaft. The input and intermediate shafts are each supported by a pair of support bearings, while the stepped output shaft is supported by two housed high-speed ball bearings.

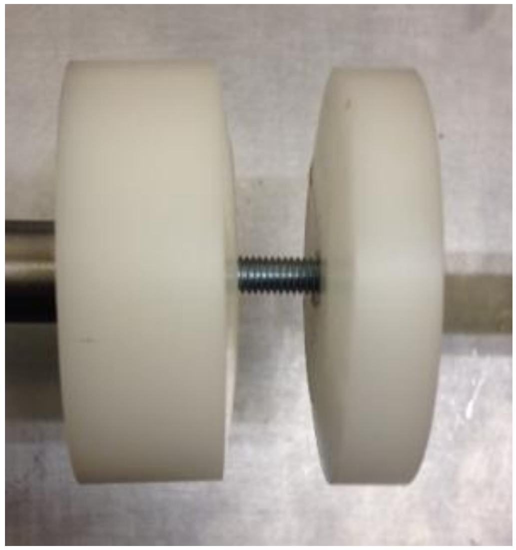


Figure 28. One Set of Delrin Bearing Attachments Made

The bearing testing attachments consist of three sets of Delrin cones, which securely grip the inner race. One of the pairs of cones can be seen above in Figure 28. These cones are screwed and secured onto the output shaft with a machine screw. The detail drawings for these attachments can be found in Appendix B.

Figure 29. Seal Attachment

Since there are discrete sizes of seals that will be tested, a number of tools needed to be designed. One of the seal attachments made can be found above Figure 29. In order to reduce the amount of manufacturing required, similarly sized tools were combined so that each seal attachment could test two different seal sizes. The detail drawings for these attachments can be found in Appendix B.

A thermal chamber was made similarly to the critical design. The wooden thermal chamber covers the test sample during testing and insulates the cold or hot air input by the Thermotron. There is one slot on each side in order to allow the output shaft and the chuck adapter to rotate freely. Two 4" holes were placed in the chamber to allow for the interface between the Thermotron and the test sample environment.

Figure 30. Grizzly 5" 3-Jaw Chuck

A Grizzly 5" 3-Jaw chuck, seen above in Figure 30, was used to grip the outer races of the test samples. The 3" jaw chuck from the critical design needed to be upgraded to a 5" chuck because of the need to grip the outer races of bearings as large as 3.5", which could not be accomplished with the 3" chuck. A chuck was chosen so that concentricity is maintained with the seals, as the seals are not self-centering. The chuck selected was actually heavier than anticipated, since the weight of the chuck was not specified in any of the product specifications. Because of this a chuck support was needed to hold up the weight of the chuck to avoid exceeding the bending moment limits of the torque sensor. This chuck support, seen below in Figure 31, is a wooden fixture with two vertical legs and a horizontal cross beam with an eyebolt located in the middle of it. The eyebolt holds a steel wire that goes down through the thermal chamber and attaches to a hose clamp secured around the chuck during testing.

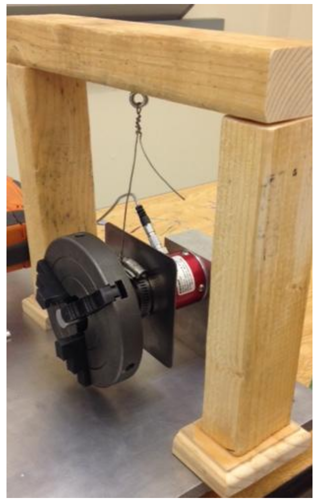


Figure 31. Chuck Support Fixture

A wooden safety cover was also designed to be placed around the drive train during testing. There are two slots to allow the input and intermediate shafts to pass through. This cover acts as a shield for the operator in case any components fly off and prevents the operator from being injured by the high speed rotating components.

Figure 32. National Instruments myRIO-1900 Data Acquisition System

The data acquisition system chosen was the National Instruments myRIO-1900, seen above in Figure 32. This student edition DAQ has several analog and digital inputs and outputs so we are able to input the torque, speed, and temperature signals easily while also powering the hall effect sensor with 5 Volts. This DAQ also has FPGA (Field Programmable Gate Array) capabilities, which essentially allows for precision timing in hardware instead of software timing. Since software timing can sometimes be unreliable and slow, it was imperative to use a system with hardware timing functionality. This DAQ interfaces with LabVIEW, which is the program used by Parker Hannifin.

Figure 33. Hamlin Gear Tooth Hall Effect Sensor

To measure the speed of the output shaft, the Hamlin gear tooth hall effect sensor, seen above in Figure 33, was used. The sensor measured the presence of a gear tooth, so it was able to output a digital pulse every time one of the 20 gear teeth passed it. From this, the frequency of the pulse could be used to calculate the speed of the output shaft.

Figure 34. Futek TFF400 Reaction Torque Sensor

The torque sensor chosen was the Futek TFF400 Reaction Torque Sensor, as seen in Figure 34 above. This torque sensor can measure up to 160 in-oz. In order to measure the smaller torques more accurately, we chose a model that measures a smaller range. However, since this does not accomplish measuring up to 20 in-lb as required, we made the torque sensor easily removable and interchangeable with a torque sensor that Parker Hannifin has that will measure the higher range of torques.

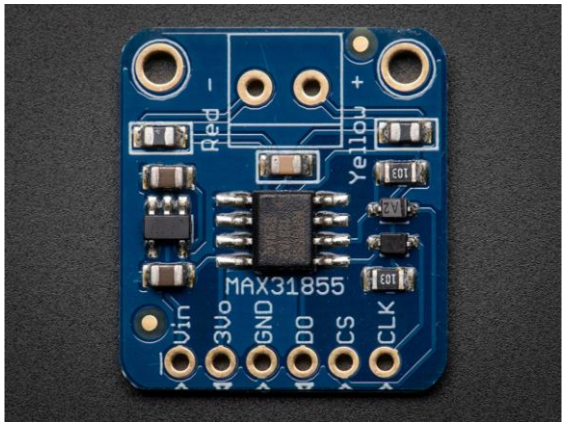


Figure 35. Adafruit MAX31855 Thermocouple Module Breakout Board

In order to measure the ambient, steady state temperature inside the thermal chamber, we needed external wiring to input the thermocouple signal into the DAQ. Because the DAQ did not have an internal module for thermocouple measurements as originally thought, an external module was used. Due to the limitations on the type and cost of modules available, a module for use with a Type K thermocouple was selected. Type K will be able to measure the expected range of temperatures, but with a larger resolution than the Type T originally selected. The output of this module is a direct readout of the temperature in degrees Celsius.

The programming for the DAQ was done through LabVIEW. In order to measure the hall effect sensor output accurately at high speeds, the DAQ's FPGA was used. The FPGA program can be seen below in Figure 36.

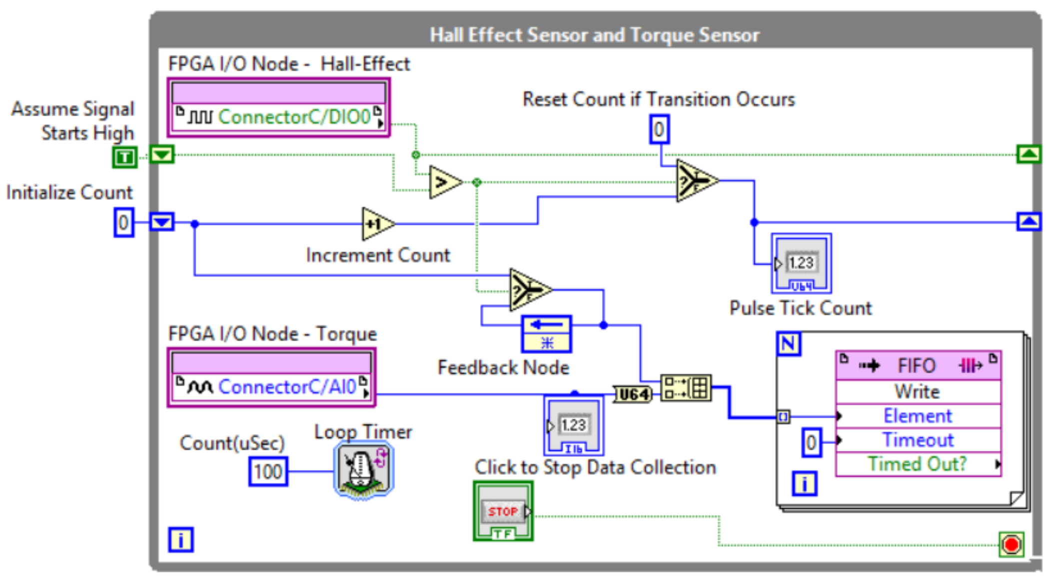


Figure 36. LabVIEW FPGA Block Diagram

This program reads the input signals from the torque sensor and hall effect sensor. The hall effect signal comes in as a Boolean digital signal, so the frequency of this signal is determined by finding when the value of the signal goes from low to high, thus detecting rising edges. The amount of pulses between each of these transitions is the period of the signal. This signal and the analog torque sensor signal are sent through the FIFO write command, which writes all of the data to a module that transfers the data to the DAQ's software modules. So, a program needed to be created using the normal DAQ interface to interpret this data appropriately. This program can be seen below in Figures 37 and 38.

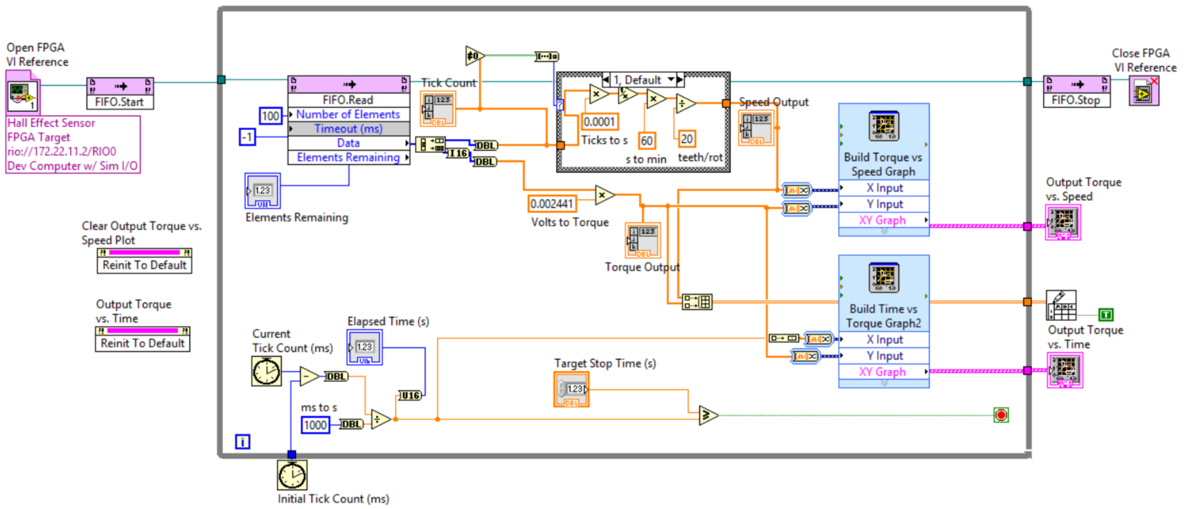


Figure 37. LabVIEW Overall Block Diagram for Case 1

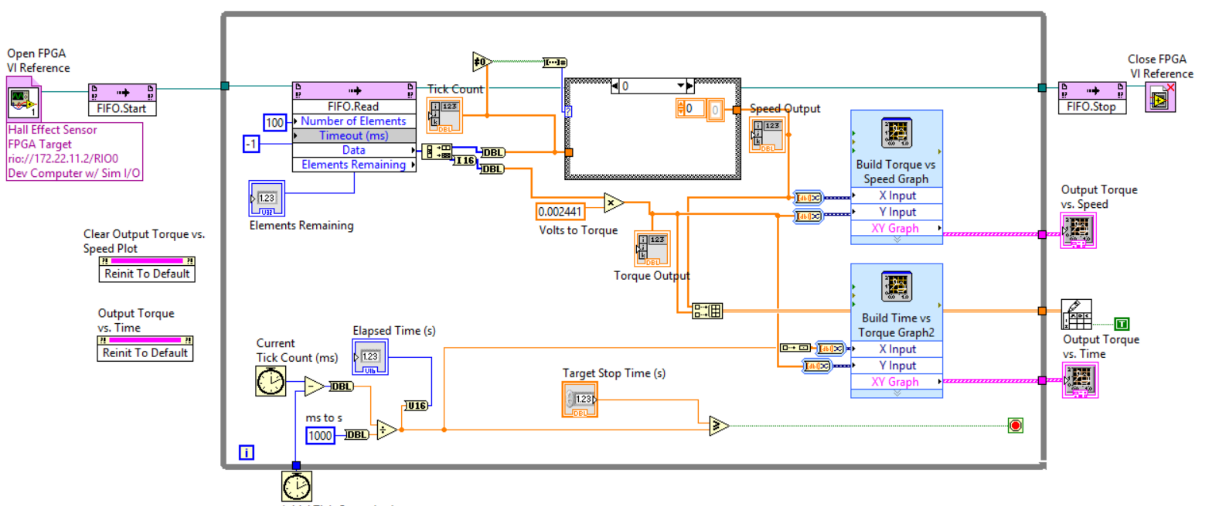


Figure 38. LabVIEW Overall Block Diagram for Case 0

This program uses the FIFO Read function to read the data sent from the FPGA file. It takes this data and calculates speed and torque appropriately. The torque signal is first converted from ADC to Voltage and then calibrated with a gain to obtain the torque value. The hall effect period signal is converted to RPM by calculating frequency and multiplying by the number teeth on the gear being read. However, in order to prevent the program from initializing the speed at infinity due to the period of the initial data point being zero, two cases were created. For case 0, seen in Figure 38, if the period measured is zero, which only occurs upon initialization, the speed output value is set to zero. For case 1, seen in Figure 37, if the period measured is not zero, the rotational output speed is calculated appropriately. These calculated signals are output to a plot and are plotted once the test is complete. The user interface to control these programs can be seen below in Figure 39. On this front panel, the operator can choose the duration of the test and view the current elapsed time, speed output, torque output and period of the gear in ticks. Once the test has finished, the output torque versus time and output torque versus output speed plots are created.

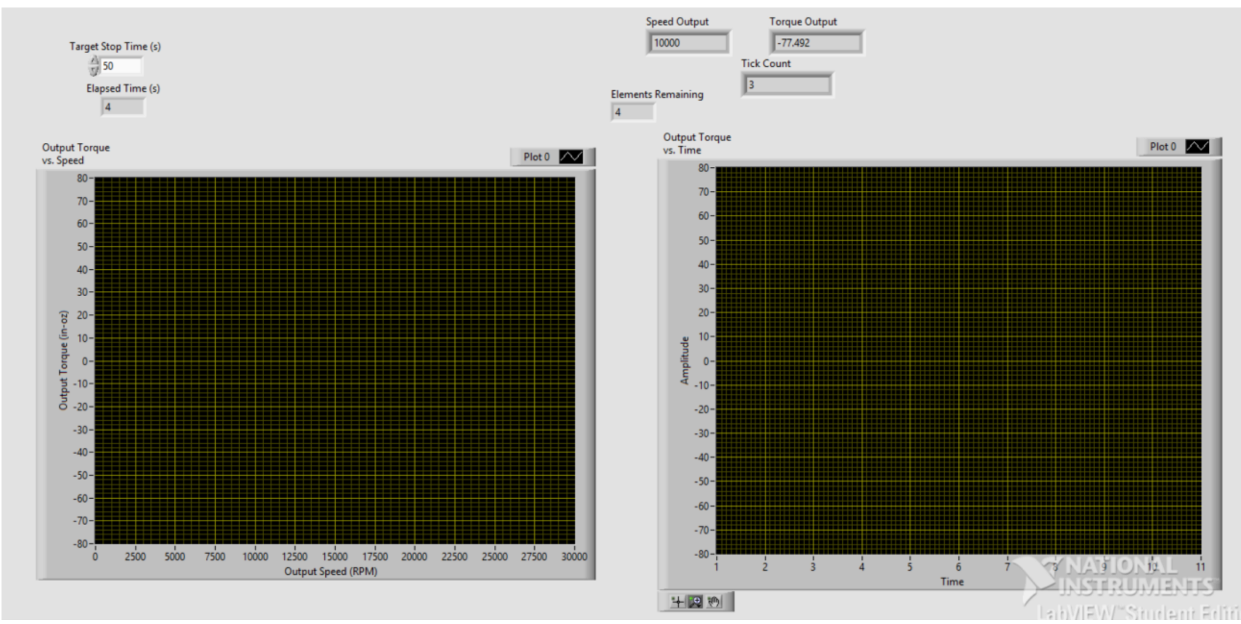


Figure 39. LabVIEW Overall Front Panel

Analysis Results

We were able ramp our system up to full speed. In this process, we noticed that the system would vibrate as one specific speed. Beyond this speed, identified as the critical speed, the vibrations attenuated, indicating that the natural frequency of the system had been passed. Using LabVIEW, we were also able to obtain accurate readings of the rotational output shaft

speed from the final gear tooth using the hall effect sensor. We were also able to display accurate real time readings for the drag torque measurements. However, the graphical data did not match the real time data for some reason unknown to us. We did expect that the drag torque would increase to a maximum value then decrease slightly as the output shaft speed increased. We believe this makes sense, as the static friction in the races would be larger than the kinetic friction, so the breakaway torque would be the maximum value obtained.

Cost Breakdown

We have exceeded the provided budget of $2,000, finishing with a final total cost of $2,760.65. However, we have implemented high-quality components that perform the required tasks outstandingly. All of the expenses for this project came from material purchasing. Because we performed all the machining ourselves, we did not have additional manufacturing costs. We saved money by using the significantly less expensive drill motor with the speed controller included. The torque sensor comprised almost 40% of the overall cost. Parker Hannifin required us to obtain precise torque measurements, which is why we selected the Futek torque sensor. All the selected components are designed and produced by reputable companies, which was important to our team because a company that produces high-quality designs, such as Parker Hannifin, should use testing tools of the same or higher quality. The detailed cost breakdown can be found in Appendix C.

Material Selection

The base plate, drill coupling block, and various cover plates were chosen to be made of aluminum to minimize the overall weight of our system, while keeping it durable and strong. However, the drive train shafts and seal attachments were chosen to be made of steel due to the fact that steel is heavier than aluminum, which would minimize the vibration associated with imbalance in the system. Also, steel will experience smaller deflections than aluminum, which again will decrease the change of imbalances and vibration, especially on the output shaft. Fasteners, such as bolts and screws, were made of steel. The bearing attachments were made out of a lightweight material called Delrin, which will help prevent large deflections or vibration in the output shaft. The seal attachments were made out of steel and hollowed out in order to keep them lightweight, as well.

Wiring Diagram

A wiring diagram has been created in order to detail the wiring performed. This diagram can be found in Appendix B. Wall outlet power is needed for the drill motor, strain gage amplifier, and DAQ. The temperature, torque, and output shaft speed sensors can be powered through the DAQ's output voltage capabilities. The torque and temperature sensor outputs are analog and the speed sensor output is a digital pulsing square wave. All of these signals are read and interpreted by the DAQ. However, some small additions were made to make the circuitry function properly.

The torque sensor output is obtained through a 4 Pin Lemo Connector. This cable's leads will be fed into a strain gage signal amplifier, which amplifies and conditions the output signal from the torque sensor.

The gear tooth sensor, since it is generating a time sensitive pulsing signal, must have its output signal read by the FPGA inputs on the DAQ. These inputs enable the sensor to be counted by an internal hardware processor, as opposed to a software-timed counter, thus reading the sensor accurately. In order to select a DAQ that could handle the output frequency from this sensor, we calculated the frequency of the gear tooth rotation to be measured. This calculation can be found in Appendix E.

All of the components operate at relatively low voltages and currents that all of the components and DAQ can handle, so exceeding power, current, and voltage limitations will not be an issue, except in the extreme use of the gear tooth speed sensor. With this sensor being powered at 25 Volts, the maximum possible output signal will produce a power of 12 mW. The maximum power rating for the FPGA inputs is 10 mW, so in the extreme power conditions, the output signal will damage the DAQ. However, because the sensor cannot be powered by anything higher than a 15 Volt power source from the DAQ, this power rating will not be exceeded. Additionally, all the source impedances from the sensors are within the acceptable limits of the DAQ. The basic electrical power calculations can be found in Appendix E.

Maintenance & Repair Considerations

Most of the components in our design require little to no maintenance or repair, ideally, as the system was designed to withstand a large number of performed tests. All of the components are screwed and bolted onto the base plate, which allows for easy removal and placement. Additionally, all of the components can be ordered easily from various distributors or obtained

from any hardware store. Special care should be taken with the torque sensor and DAQ, as these are the most expensive components in the system. Replacing these components would be relatively easy, but would increase the cost spent on the system.

Safety Considerations

Since we utilized rotating machinery, we needed to protect the user from any potential hazard or harm. Therefore, we shielded the moving parts of the drive train with a wooden box. The test sample will be located inside the thermal chamber. The placement of the thermal chamber and drive train shield can be seen below in Figure 40. Furthermore, all electrical wiring will be shielded in order to protect the user from any electric shocks.

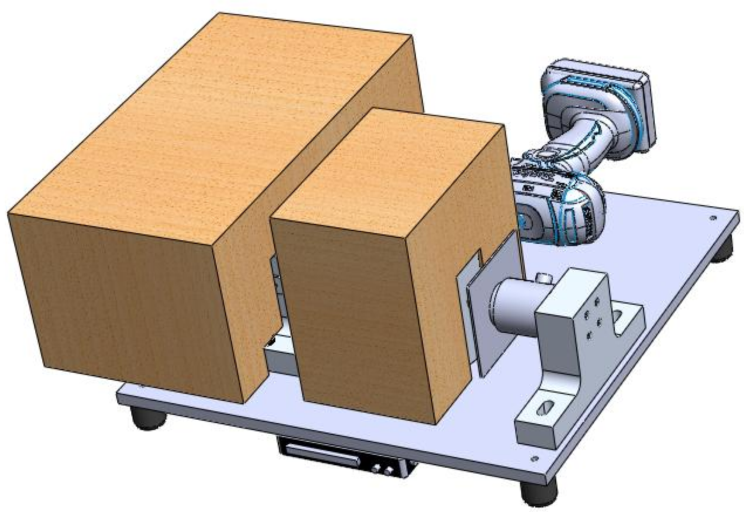


Figure 40. Drive Train Shielding and Thermal Chamber Placement

Chapter 5 – Product Realization

Manufacturing Processes Employed

Figure 41. Milled and CNC'ed Components

The project was manufactured using a combination of manual and CNC machines. Hand tools were used whenever possible in order to decrease the manufacturing time. All parts were designed to be easily manufactured, and off-the-shelf parts were purchased when possible. The majority of the manufacturing was done in the Cal Poly Machine Shops, with the CNC work being completed in an independent shop, free of charge.

Parts that required close tolerances or difficult contours were made using a CNC mill. These parts include the motor support, center bearing support, and the torque cell support, as seen in Figure 41 above. There was difficulty in getting the bearing cups aligned with one another. This dimension was critical in the reduction of output shaft vibrations.

The base plate and the pillow block supports were manufactured on a manual mill. The tolerances on these parts were not critical and they consisted of very basic shapes. They were designed and manufactured in such a way to reduce the number of machining setups and specialty tooling. All tapped parts were done by hand, with the exception of the CNC pieces.

Lathe parts were all done on manual lathes in the Cal Poly shops. It was difficult to meet the tolerance requirements with these machines, as much of the tooling needed was dull or unavailable.

Recommendations for Future

In order to meet the tight tolerance and surface finish requirements of the shaft seal tools, it is suggested that the part be ground down to size rather than turned on a lathe. The machines and lathe tools that we had available to us were not able to produce the quality of part that the seal is designed for. The surface finish on our lathe parts was inconsistent, and when we polished it down by hand, the tolerancing was lost.

Many of the parts that were made are difficult to service. The center bearing support bearings are pressed in, which makes them difficult to remove. If the part were redesigned, it would be beneficial to employ circlips on the output shaft, so that the shaft could be removed and then the bearings removed using a puller. There was also no indication of proper alignments for the shafts, with respect to one another. This problem could have been avoided by machining steps on each shaft so that they are positively aligned axially.

The pulleys that were used on the system are difficult to remove. Removal is necessary in order to meet the speed requirements and should be a quick process. Currently, the bushing for each pulley gets wedged into the sheave in order to clamp down on the shaft. To remove it, one my press off the bushing using the three retaining screws threaded into the removal holes. This is a lengthy process and future improvement could implement a quick change pulley system.

Chapter 6 – Design Verification

Executed Testing Descriptions

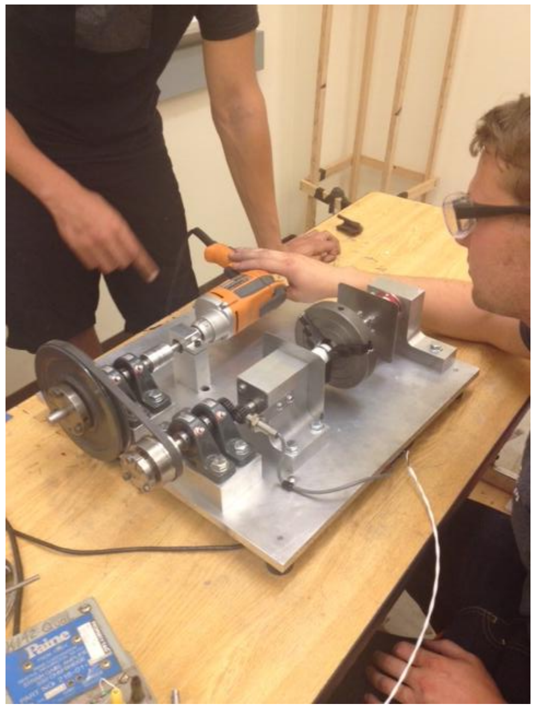


Figure 42. Testing Execution

From our analysis, we were able to find the first mode critical speeds for both the seal and bearing test setups using the Rayleigh method. However, we wanted to verify the first mode values. In order to do this, we tested our system, as seen in Figure 42 above, by varying the output speeds and watched for undesired vibration. A considerable change in the system's vibration indicated a critical speed. One notable critical speed, measured to be around 10,000 RPM for a ½" inner diameter bearing, was found while running this test, This critical speed occurred while ramping the system up to the full speed of 15,000 RPM. The system ran smoothly before and after reaching this critical speed, but vibrated much more at this speed. Performing critical speed tests allowed us to judge whether or not the system would remain stable at critical speeds. We found that the system became slightly unstable at this critical speed. However, unless the system is held constantly at this speed, the vibrations will only occur momentarily, which will not majorly affect the system's performance or measurements.

We also needed to verify that the selected drill motor could reach the speed values calculated and tabulated in Table 8 in Appendix E. The speed will be measured from the final 20 tooth gear

on the output shaft using the gear tooth hall effect sensor. According to our measurements, the system was able to reach the maximum speed required. However, the motor would heat up very quickly when it ran at high speeds. This problem consistently occurred when running the system at high speeds. In order to mitigate this problem, a more powerful motor or a motor with a stronger internal fan should have been used instead.

Excellent alignment of the drive train was imperative in order to accurately measure drag torques without vibration interference. Any parallel offset or angular offset misalignment would disturb the measurements. The system was tested many times to verify that the components were properly aligned. If any vibration occurred, it occurred consistently throughout the test run, at both high and low speeds. This meant that the system was imbalanced or misaligned somehow. We were able to visually pinpoint locations of imbalance and correct any misalignments present by slightly repositioning components. The base plate is also held up with rubber feet, which assist in isolating and damping vibrations in the system.

The sensors were calibrated and tested in order to ensure accurate measurements through the DAQ. The sensors were all calibrated by the manufacturers, but we performed calibration verification in case any offsets needed to be implemented. The hall effect sensor required no additional calibration. The torque sensor required much more calibration. The gain and the balance needed to be adjusted on the strain gage amplifier in order to obtain an adequate signal that was balanced around zero. However, the torque sensor signal still required an offset in the LabVIEW program in order to calculate the torque measurements accurately. The thermocouple module was already calibrated, so the output from the module into the DAQ was ready for reading.

Lastly, we performed testing on a ½" bearing to verify that the system was reading and measuring properly. The bearing was held in place with a Delrin attachment, so the inner race was spinning while the outer race was held with the 3-jaw chuck. Because the LabVIEW was not fully operation during this test, the torque and output speed data was only monitored in real time, not logged. The speed data read accurately as the speed of the drill motor was ramped up. The torque sensor reading increased as the speed increased, indicating that the drag torque was measurable. The bearing tested was relatively small, so we did not observe a large change in the torque measurement, but there was a visible change in the data.

Specification Verification

All of the specification requirements detailed in Appendix A have been met or exceeded according to the means of compliance indicated in Table 1.

Shielded wiring was provided by Parker Hannifin. The electrical wiring has been designed to be properly grounded. The thermal chamber has been designed and the insulation will be provided by Parker Hannifin. The test fixture is only operable by one person, as one computer or laptop will serve as the control station.

Various seal and bearing attachments have been created to accommodate the discrete seal sizes and continuous bearing sizes. Since all the attachments were not machined due to time constraints, we provided Parker Hannifin with the files and drawings necessary for them to manufacture these parts themselves.

All of the materials and components in the system are compatible with various greases and lubricants to be used by Parker Hannifin. Of course, the electronics, including the sensors and DAQ, should not interface with any greases or lubricants, so they have been placed in order to avoid such contact.

The drive train driven by the drill motor is capable of achieving speeds up to 15,000 RPM.

All the components can withstand the temperatures between -55 and 70 degrees Celsius, which are to be implemented with the Thermotron in Parker Hannifin's labs.

The seal shaft attachments have been designed to replicate the common shaft finish specified by Parker Hannifin. The specified finish can be found on the seal attachment drawings in Appendix B.

The maximum sampling rate of the DAQ is 40 MHz, which far exceeds the 100 Hz specification. The sampling rate used to accurately read the hall effect sensor output was 1,000 Hz, which also exceeds the 100 Hz minimum requirement.

Acceleration control is implemented based on user input on the drill motor. For example, if the operator speeds up the drill motor by pressing the trigger quickly, the acceleration will be larger.

The final base plate is 18" x 18", with a maximum height of about 10" without the thermal chamber, staying within the space limitations specified.

The fixture has been designed with rugged, long lifetime components so it will have a long lifespan.

All the electrical components that need external power from the wall, including the DAQ and drill motor, are compatible with the typical wall outlets.

The Thermotron interface has been integrated into the design of the thermal chamber per the specifications provided by Parker Hannifin.

Accurate and proper alignment was implemented in the manufacturing and assembly of the fixture in efforts to prevent or minimize vibration. Tolerances were maintained in order to help ensure this.

The fixture is capable of measuring the specified range of torque values with ease. However, the LabVIEW does still need to altered in order to properly log the data in a graph.

The entire system can safely be operated by one operator. There is only one active station at a time, as the operator will be unable to alter the system configuration while testing is occurring due to the safety covers and unable to test while the system is being altered, simply due to the fact that the system would not be able to operate properly if the configuration is not fully set.

All drawings and analysis have been provided in Appendices B and E, respectively. SolidWorks models and LabVIEW files have been provided to Parker Hannifin.

Chapter 7 – Conclusions & Recommendations

Our team has a few recommendations on actions that need to be taken and improvements that can be made to the final design. Firstly, the remainder of the seal and bearing attachments need to be machined. Due to time constraints and unforeseen roadblocks, we were unable to finish the fabrication of these parts. However, these parts are relatively simple to make, so we do not believe Parker Hannifin will have any issues making them. Second, the LabVIEW program created still needs to be fine tuned. The data is measured accurately in the system in real time, but the data is not accurately plotted for reasons still unknown to us. Next, the thermocouple module needs to be integrated into the system. A module and thermocouple that are compatible with our system have been selected and purchased, but have not been interfaced with the DAQ, again due to time constraints. Lastly, we recommend that the Ridgid drill motor be replaced with an industrial grade motor. Because of the additional friction introduced by the support bearings, the drill motor was unable to drive the system for very long before it would heat up. Once it heated up, it took a while to cool it back down enough to perform more testing. We believe that utilizing a more powerful motor with better means of internal cooling would work much better and allow for shorter breaks between tests.

This project has allowed our team to transform mere thoughts and ideas into a well-detailed design. We have faced many challenges in creating this design, especially with the large speed range that was required and the wide array of bearings and seals that needed to be accommodated. Also, staying within a designated budget, although quite difficult, led us to thinking outside of the box and developing new and unique solutions to the problem. Organizing the project in a subsystem oriented manner allowed us to develop valuable system integration skills. We are confident that this bearing and seal friction test machine will help Parker Hannifin pinpoint sources of drag and reduce the losses in their electromechanical actuators, thus improving Parker Hannifin's Aerospace Group as a whole.

References

Appendix A

Formal Engineering Specifications, Quality Function Deployment, Decision Matrices, Seal Sizing Information

Formal Engineering Specifications

1. The test fixture shall be safe to use with little to no chance of causing harm or damage to any person or thing.
    a. Shielded wiring, properly grounded electrical equipment, insulated thermal chamber, moving machinery guards, and a single control station will be used to implement this requirement.
2. The test fixture shall accommodate different seal sizes.
    a. Shaft size of ¼" to 3 ½" will be used and standard widths and outer diameters will be compatible.
    b. Specific width and outer diameter sizing provided by Parker Hannifin.
3. The test fixture shall accommodate different bearing sizes.
    a. Shaft size of ¼" to 2 ½" will be used and standard widths and outer diameters will be compatible.
    b. Compatible with a wide range of width and outer diameter sizes.
4. The test fixture shall accommodate the use of different greases and/or oils as lubricant during testing.
    a. All materials used in the design of the fixture will be compatible with the lubricants commonly used by Parker Hannifin.
5. The test fixture shall perform testing at various speeds.
    a. The range of speeds to be used is from 0 to 15,000 RPM.
6. The test fixture shall perform testing at various temperatures.
    a. The range of temperatures to be used is from -55 to 70°C.
7. The test fixture shall replicate the common shaft finishes used by Parker Hannifin.
    a. The shaft used will be ground, smoothed, and polished to standard specifications.
    b. Specific finish specifications provided by Parker Hannifin.
8. The test fixture shall have a good sampling rate for measurement.
    a. A minimum of a 100 Hertz sampling rate will be used.
9. The test fixture shall allow for appropriate acceleration of speed.
    a. The motor will have acceleration control features.
10. The test fixture shall be compact enough to fit on top of a desk.
    a. The size will be approximately 18"x18"x18".
11. The test fixture shall have a long lifespan and be rugged.
    a. The fixture will have a lifespan of at least 1000 sample runs for a duration of 5 minutes each.

Formal Engineering Specifications (Continued)

12. The test fixture shall be compatible with a typical wall outlet.
    a. A 115 Volt and 20 Amp circuit will power the fixture's electrical components.
    b. The maximum voltage is to be 220 Volts.
13. The test fixture shall have no electrical interference.
    a. All measurement signal wires will be shielded.
14. The test fixture shall interface with the Thermotron provided by Parker Hannifin.
    a. A full thermal chamber with attachments available for hookup to the Thermotron will be built around the test sample.
15. The test fixture shall take accurate measurements.
    a. Temperature and speed measurements will have a maximum error of ±2.0%.
    b. Torque measurements will have a maximum error of ±5.0%.
16. The test fixture shall be operated by one person.
    a. Only one control station will be installed.
17. The test fixture shall minimize vibration.
    a. Accurate, proper, and feasible alignment will be implemented into the design and manufacturing of the fixture.
18. The test fixture shall measure a range of torques.
    a. The range of torque values to be used is from 0 to 20 in-lb.
19. Our team shall provide all drawings and analysis for review and release.
    a. Drawings will be in PDF and STP formats.
    b. SolidWorks files, in addition, are to be provided.

Quality Function Deployment: House of Quality

The important features of the "House of Quality", shown on the following page, include the customer requirements list, engineering requirements list, and correlation sections. The far left side of the diagram lists all of the customer requirements that our team has gathered from Parker Hannifin. The top of the diagram lists the engineering requirements that our group has developed, many with help from Parker Hannifin, that will accomplish and fulfill all of the customer requirements. The correlation section is in between the customer and engineering requirements lists. This section allows us to determine if every possible pair of one customer requirement and one engineering requirement are correlated. The two can have a strong, medium, small or no correlation. The legend for the correlations used is included on the diagram attached.

Many of the engineering requirements can assist in fulfilling multiple customer requirements. If any requirements are correlated with many other requirements, it provides a visual representation of the requirement's importance. For, example, the customer requirement that the fixture be "Safe to Use" is correlated with many engineering requirements, indicating that this is a very important customer requirement to fulfill.

The diagram also provides a section to indicate weighting for each customer requirement. Each combination of customer requirements is compared against each other to determine which of the two are more important to accomplish. When all of the comparisons are made, the amount of times that the requirement was found to be more important than another requirement increases its weighting. Therefore, the higher the weighting for the requirement, the more important we found that requirement to be.

Also, the far right side lists the current solutions to the problem presented. In our case, the only solution was the informal testing done by Parker Hannifin that was unsafe and fairly inaccurate. As seen in the diagram, this current solution is not able to fulfill all of the customer and engineering requirements that Parker Hannifin desires.

Quality Function Deployment: House of Quality (Continued)

Friction Test Machine

Customer: Parker Hannifin Aerospace

Engineering Re uirements | Benchmarks

Customer Re uirements	Weighting (Total 100)	Shield Moving Machinery	Properly Ground Electrical Components	Accommodate Shaft Sizes from 1/4"-3 1/2"	Accommodate Standard Bearing Seal Widths	Accommodate Standard Bearing Seal Outer Diameters	Greases Compatible with Materials Used	0-15000 RPM Speed Range	-55-70° Celcius Temperature Range	Shafts Ground, Smoothed, and Polished To Standard Specification	Minimum 100 Hertz Sample Rate	Acceleration Control	Overall Dimensions of 18"x18"x18"	Minimum Total Lifespan of 1000 Sample Runs at 5 Minutes Each	Uses 115 Volt 20 Amp Wall Circuit (Maximum 220 Volt Wall Circu	Utilizes Shielding Wiring	Full Thermal Chamber Around Test Sample	Temperature and Speed Have Maximum ±2 Measurement Error	Tor ue Has Maximum ±5 Measurement Error	Single Control Station	Tor ue Range of 0-20 in-lb	Proper Alignment (No Need For Couplings)	Benchmarks	Hand-held Testing Performed by Parker	
Safe to Use	11.1	●	●						Δ						○		○			Δ					
Different Seal Sizes	3.9			●	●	●		○		○		Δ					○				○			○	
Different Bearing Sizes	8.5			●	●	●		○		○		Δ					○				○			○	
Different Greases	0.7						●	○									Δ							○	
Different Speeds	9.8	●						●				Δ									●			Δ	
Different Temperatures	7.2								●			Δ					●								
Common Shaft Finishes	1.3									●									○						
Good Sampling Rate	3.3							○			●														
Appropriate Acceleration	2.0	Δ						○				●													
Compact (Fit on a Desktop)	3.9			○	○	○							●				Δ								
Rugged/Long Lifespan	7.2						Δ							●								○			
Work with Standard Wall Outlets	7.2		●												●					Δ					
No Electrical Interference	4.6															●		●	●						
Interface with Thermo-tron	6.5	○							●								●								
Accurate Measurements	7.8															●		●	●			○			
Single Operator	0.7	Δ																		●					
Minimize Vibration	5.9	Δ						●				●										●			
Different Tor ues	8.5	○						●											○		●	Δ			
Units		-	V	in	in	in	-	rpm	°C	µ	Hz	Rad/s 2	in	Cycles	V,A	-	-			-	in-lb	in			
Benchmark				Δ	Δ	Δ							Δ												

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

Concept Down-Selection: Pugh Matrices

A Pugh Matrix compares how various generated ideas fulfill the selected criteria to a datum. The criteria selected come from the customer requirements that apply to that specific category of ideas. The datum is usually a competitor product benchmark. Since we have no benchmarks for each individual subsystem, we selected the datum to be the concept thought to be the "best" solution. A Pugh Matrix does not indicate the best solution, however. It, instead, compares the positive and negative qualities of each concept to the datum. It, thus, provides insight into which concepts do and do not satisfy the most important criteria.

Table 4. Pugh Matrix for Drive Train Mechanism During Preliminary Concept Generation

	Concept			
Criteria	Multiple Shift Gearbox	Manually Reversible Gearbox	Only Variable Speed Motor	Belt Drives
Safe to Use	S	S	-	D
Operates Safely in Speed Range	S	S	-	
Compact	-	S	+	
Minimal Losses	+	+	+	A
Operates Safely in Torque Range	S	S	-	
Easy to Operate	+	-	S	
Allows for Quick Changes in Speed and Torque	+	-	-	T
Inexpensive	-	-	S	
Easy to Assemble	+	-	S	
Easy to Maintain/Repair	-	S	S	U
Sum of +	4	1	2	
Sum of -	3	4	4	
Sum of S	3	5	4	M

<u>**Concept Down-Selection: Pugh Matrices (Continued)**</u>

Table 5. Pugh Matrix for Torque Sensor During Preliminary Concept Generation

	Concept					
	Load Cell			Reaction Torque Cell		
Criteria	Inside Thermotron	Outside Thermotron Inside Grip	Outside Thermotron Outside Grip	Inside Thermotron	Outside Thermotron Outside Grip	Outside Thermotron Inside Grip
Thermotron Interface	+	S	D	+	+	+
Measurement Accuracy	-	-		-	+	+
Weight	+	+		+	S	+
Bearing Compatibility	S	-	A	+	+	-
Seal Compatibility	S	-		+	+	-
Calibration	+	S		+	+	+
Adjustability	-	-	T	S	S	-
Manufacturing	+	-		S	S	-
Compact	-	S		+	S	S
Robust	S	-	U	+	+	+
Sum of +	4	1		7	6	5
Sum of -	3	6		2	0	4
Sum of S	3	3	M	1	4	1

Table 6. Pugh Matrix for Bearing/Seal Attachments During Preliminary Concept Generation

	Concept					
Criteria	Expandable Chuck	Two Cones	Threaded Shaft	Different Shafts	Stepped Shaft	Interchangeable Sleeves
Easy to Mount	S	-	-	-	-	D
Stable	S	+	-	+	-	
Accuracy of Tolerance	S	+	-	+	-	
Machinery Feasibility	S	-	-	S	-	A
Price	-	+	-	-	-	
Weight	S	-	-	-	-	
Life	S	-	-	-	-	T
Maintenance	-	-	-	-	+	
Design Time	S	-	-	S	-	
Practicality	S	+	+	-	+	U
Sum of +	0	4	1	2	2	
Sum of -	2	6	9	6	8	
Sum of S	8	0	0	2	0	M

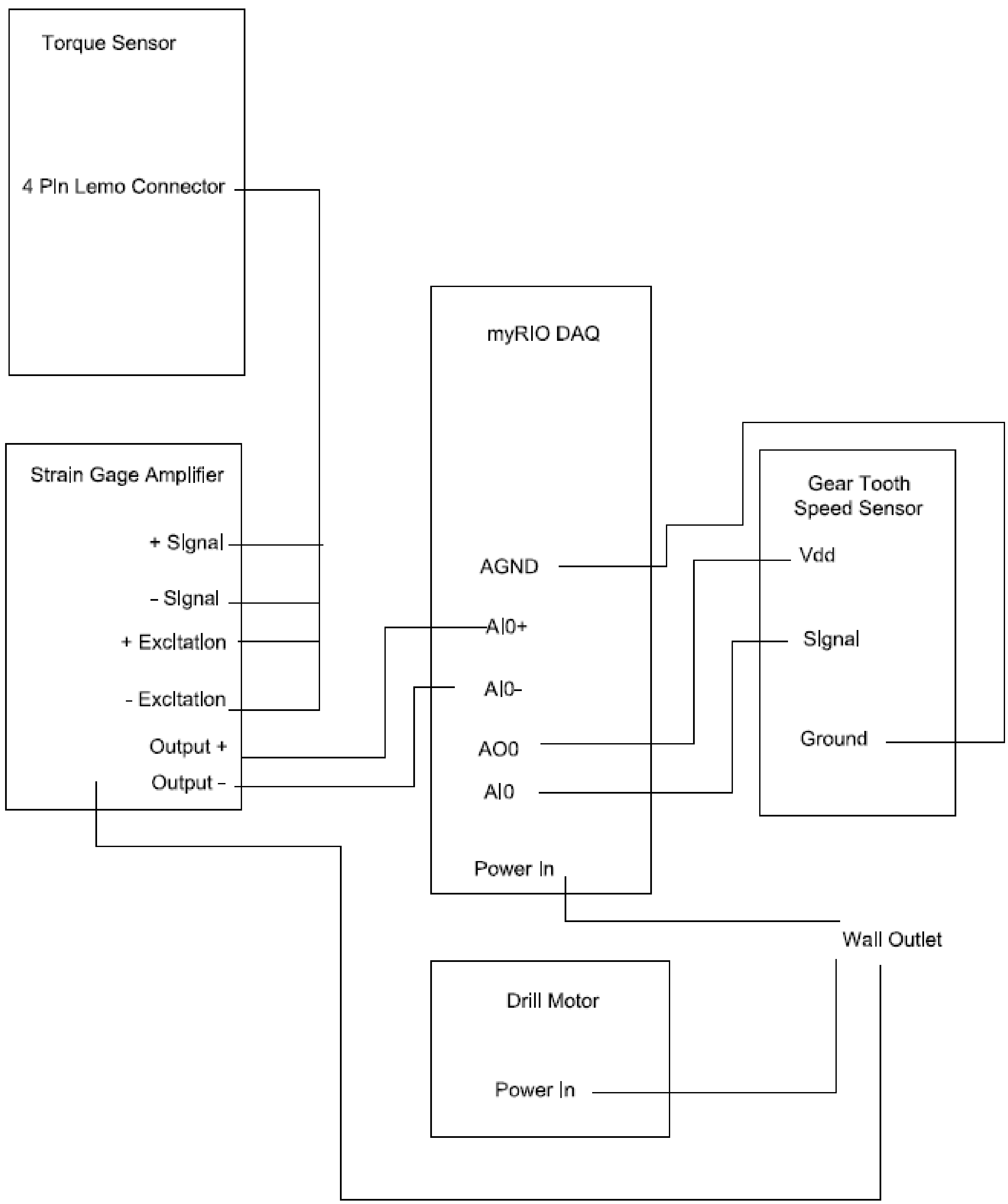
Table 7. List of Discrete Seal Sizes Provided by Parker Hannifin

Seal Nominal Sizes (in)		
Inner Diameter	Outer Diameter	Width
0.354	0.869	0.272
0.375	0.749	0.250
0.393	1.025	0.265
0.500	0.999	0.257
0.563	1.125	0.250
0.591	0.984	0.276
1.125	1.561	0.250
1.312	1.840	0.430
1.312	1.840	0.430
1.313	2.062	0.313
1.811	2.375	0.310
1.811	2.362	0.310
2.250	2.875	0.333
2.557	3.150	0.325
2.597	3.150	0.385
3.248	3.750	0.392

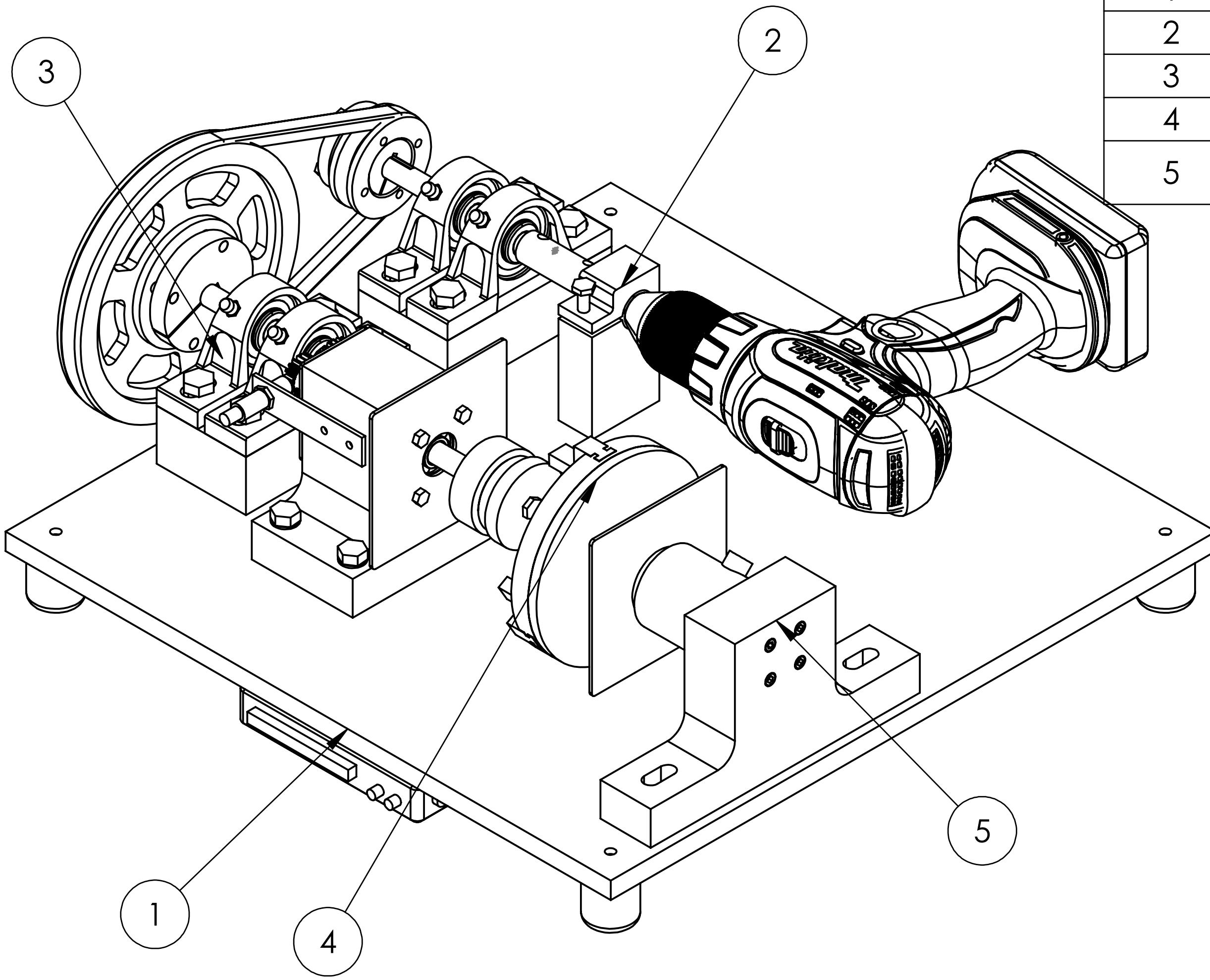
Appendix B

Drawing Packet

Wiring Diagram



ITEM NO.	PART NUMBER	PART NAME	QTY
1	CPSP14-01A	BODY ASSEMBLY	1
2	CPSP14-02A	MOTOR ASSEMBLY	1
3	CPSP14-03A	DRIVETRAIN ASSEMBLY	1
4	CPSP14-04A	TEST FIXTURES	1
5	CPSP14-05A	TORQUE CELL ASSEMBLY	1



NOTE: THERMOTRON COVER AND DRIVTRAIN SAFETY COVER NOT SHOWN

UNLESS OTHERWISE SPECIFIED:		NAME	DATE
DIMENSIONS ARE IN INCHES TOLERANCES: ONE PLACE ANGLE ±1.0° TWO PLACE ANGLE ±0.50° TWO PLACE DECIMAL ±0.010 THREE PLACE DECIMAL ±0.005	DRAWN	KYLE S.	6/12/14
	CHECKED		
	ENG APPR.		
	MFG APPR.		
INTERPRET GEOMETRIC TOLERANCING PER: ASME B46.1	Q.A.		
MATERIAL	COMMENTS:		
FINISH			
DO NOT SCALE DRAWING			

NEXT ASSY	USED ON
APPLICATION	

FRICTION TEST TEAM, PARKER

TITLE:

FULL ASSEMBLY

SIZE A | DWG. NO. CPSP14-00A | REV A

SCALE: 1:12 | WEIGHT: | SHEET 1 OF 1

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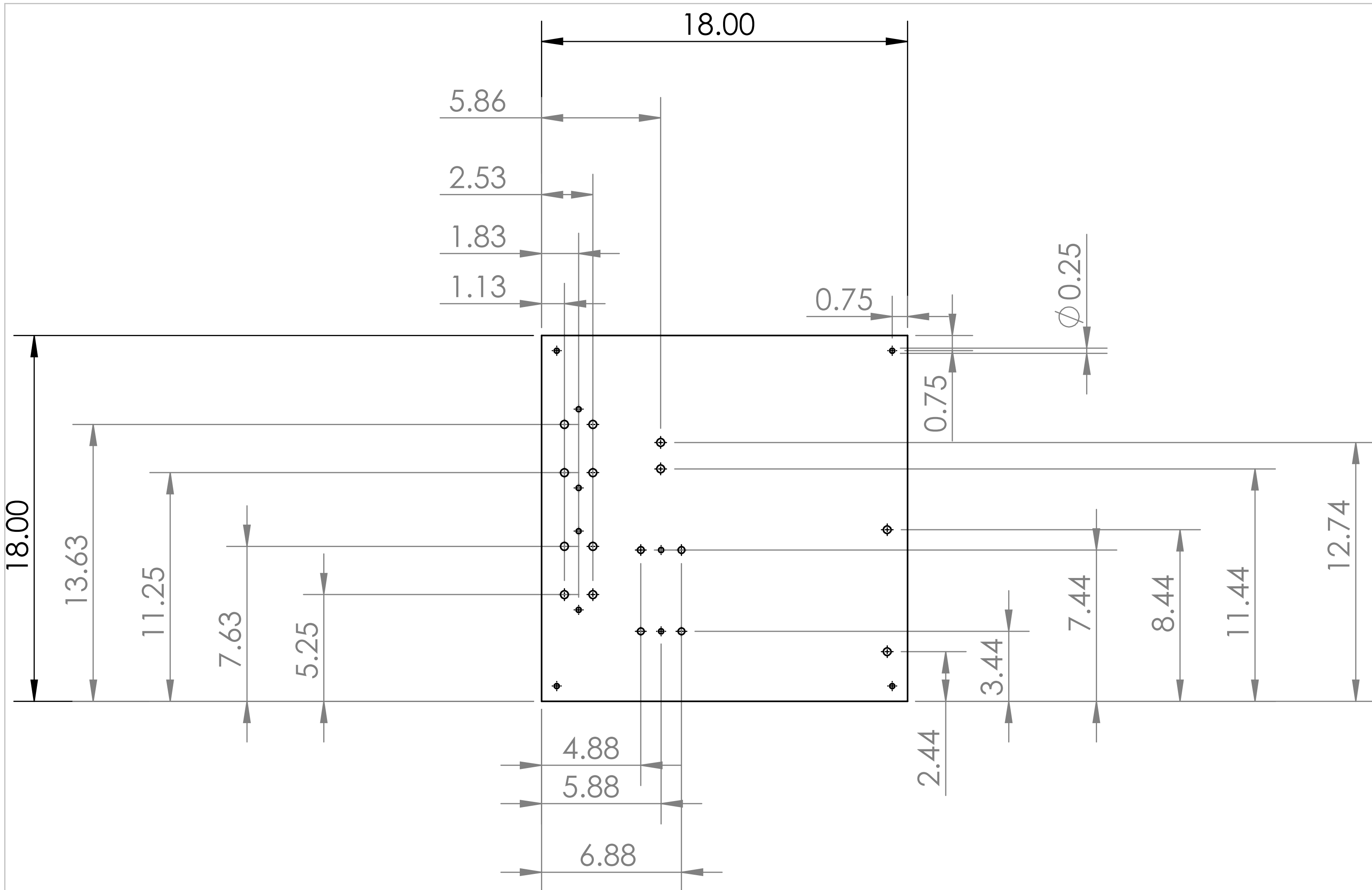

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APPLICATION	

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TOLERANCES:
ONE PLACE ANGLE ±1.0°
TWO PLACE ANGLE ±0.50°
TWO PLACE DECIMAL ±0.010
THREE PLACE DECIMAL ±0.005

INTERPRET GEOMETRIC TOLERANCING PER: ASME B46.1

MATERIAL
7075 ALUM

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	KYLE S.	2/13/14
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

FRICTION TEST TEAM, PARKER

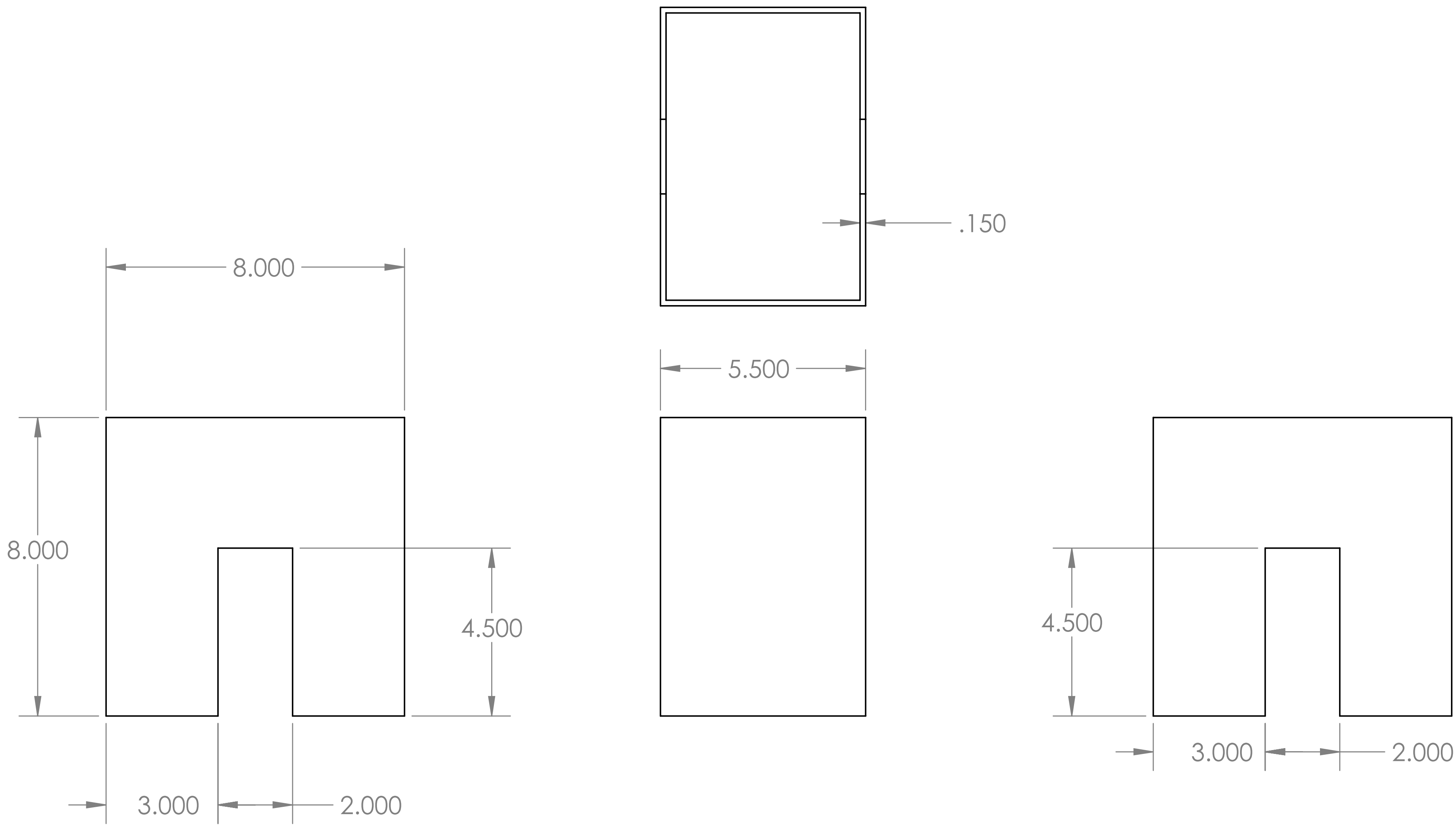
TITLE:

BASEPLATE

SIZE	DWG. NO.	REV
A	CPSP14-01-01	A

SCALE: 1:8 WEIGHT: SHEET 1 OF 1

5 4 3 2 1



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	CHECKED	M.NADRI	06/09/14	THERMOTRON
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER: ASME B46.1	Q.A.			
MATERIAL PLYWOOD	COMMENTS:			SIZE A · DWG. NO. CPSP14-01-02 · REV A
FINISH				
DO NOT SCALE DRAWING				SCALE: 1:1 · WEIGHT: · SHEET 1 OF 1

NEXT ASSY	USED ON
APPLICATION	

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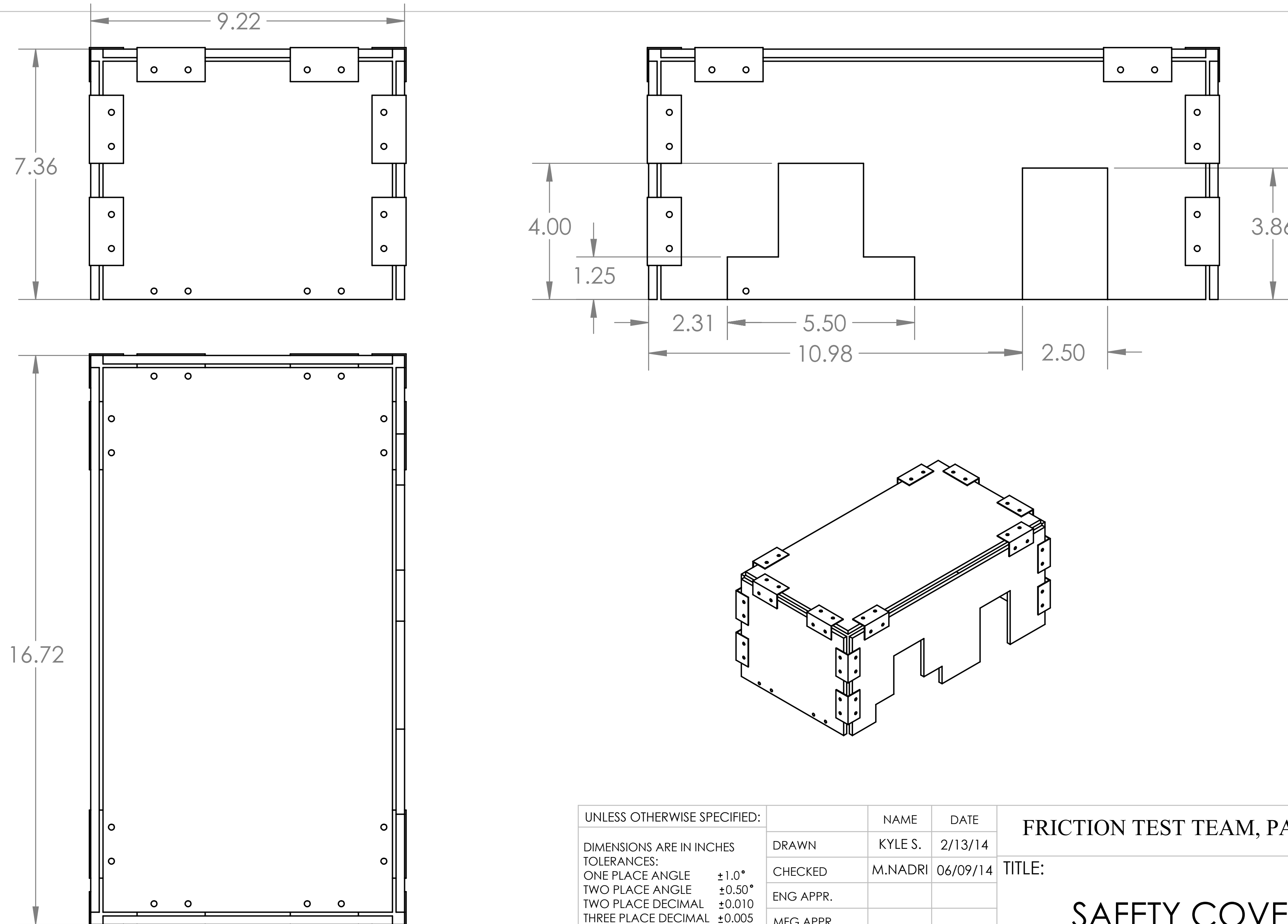

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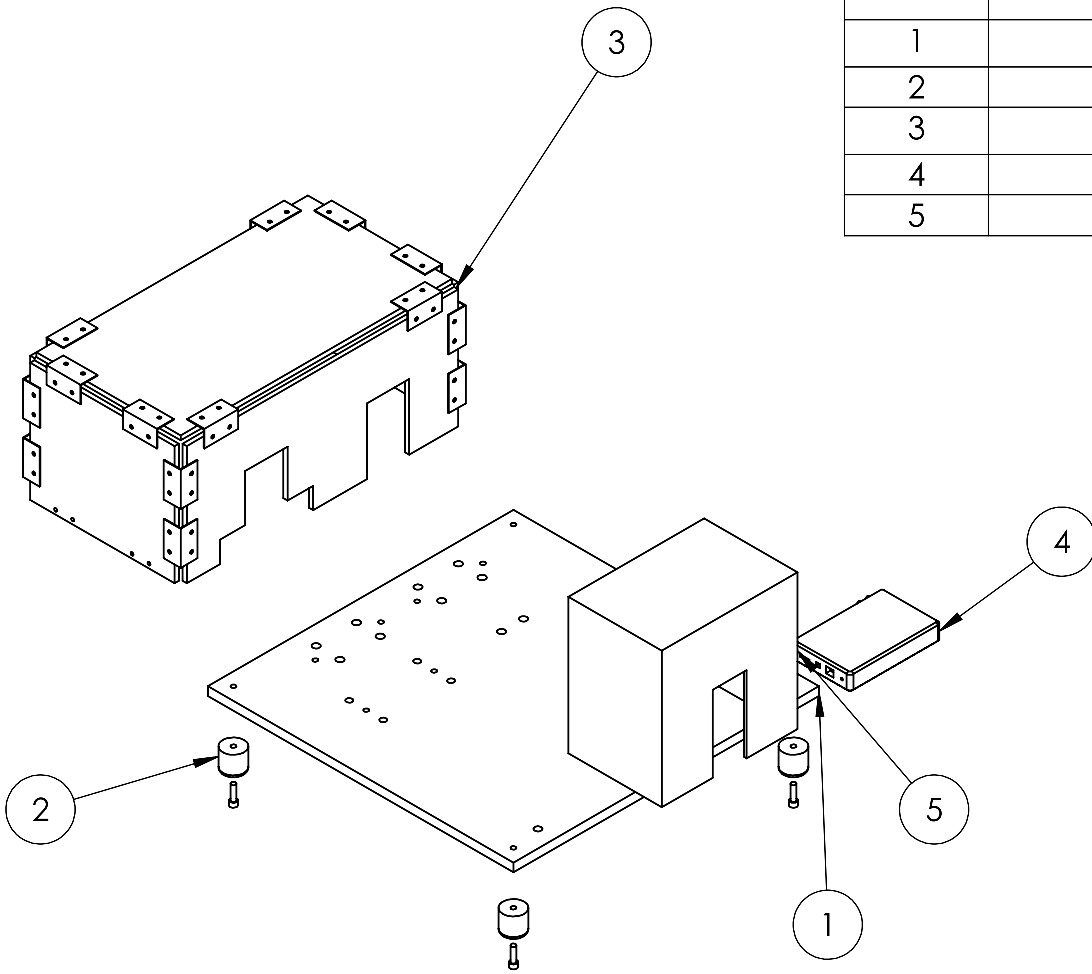
	CHECKED	M.NADRI	06/09/14	SAFETY COVER
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER: ASME B46.1	Q.A.			
MATERIAL	COMMENTS:			SIZE A / DWG. NO. CPSP14-01-03 / REV A
FINISH				
DO NOT SCALE DRAWING				SCALE: 1:1 / WEIGHT: / SHEET 1 OF 1

NEXT ASSY	USED ON
APPLICATION	

5 4 3 2 1

ITEM NO.	PART NUMBER	PART NAME	QTY.
1	CPSP14-01-01	Baseplate	1
2		Rubber Feet	4
3	CPSP14-03-01-03	Safety Guard	1
4		DAQ	1
5	CPSP14-01-02	Thermotron	1



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	CHECKED			BODY ASSEMBLY
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER: ASME B46.1	Q.A.			
MATERIAL	COMMENTS:			SIZE A · DWG. NO. CPSP14-01A · REV A
FINISH				
DO NOT SCALE DRAWING				SCALE: 1:12 · WEIGHT: · SHEET 1 OF 1

NEXT ASSY	USED ON
APPLICATION	

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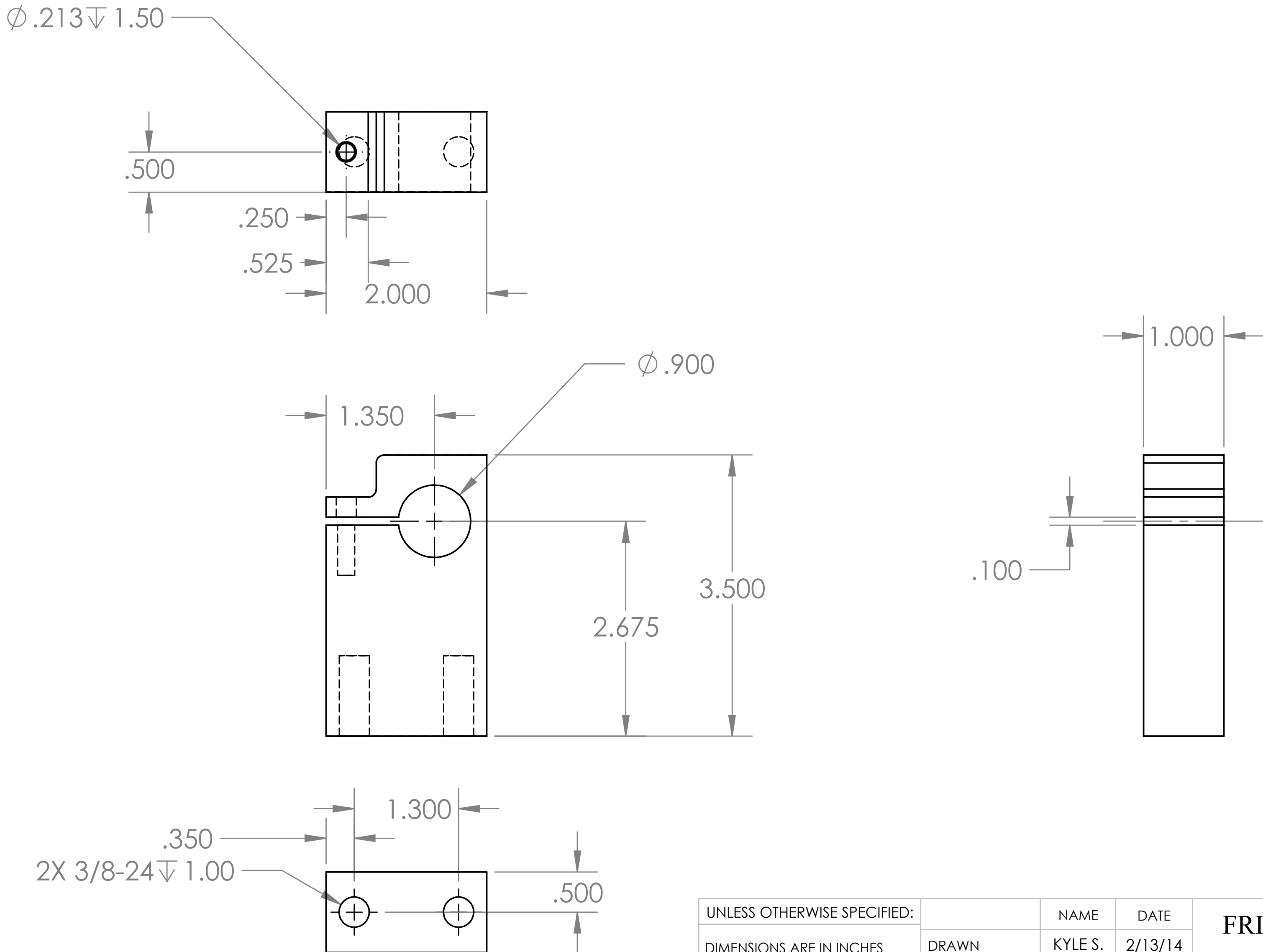

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DIMENSIONS ARE IN INCHES TOLERANCES: ONE PLACE ANGLE ±1.0° TWO PLACE ANGLE ±0.50° TWO PLACE DECIMAL ±0.010 THREE PLACE DECIMAL ±0.005	DRAWN	KYLE S.	2/13/14	TITLE:
	CHECKED	M.NADRI	06/09/14	DRILL CLAMP
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER: ASME B46.1	Q.A.			
MATERIAL 6061 - T6	COMMENTS:			SIZE A · DWG. NO. CPSP14-02-01 · REV A
FINISH				
DO NOT SCALE DRAWING				SCALE: 1:1 · WEIGHT: · SHEET 1 OF 1

NEXT ASSY	USED ON
CPSP14-02A	
APPLICATION	

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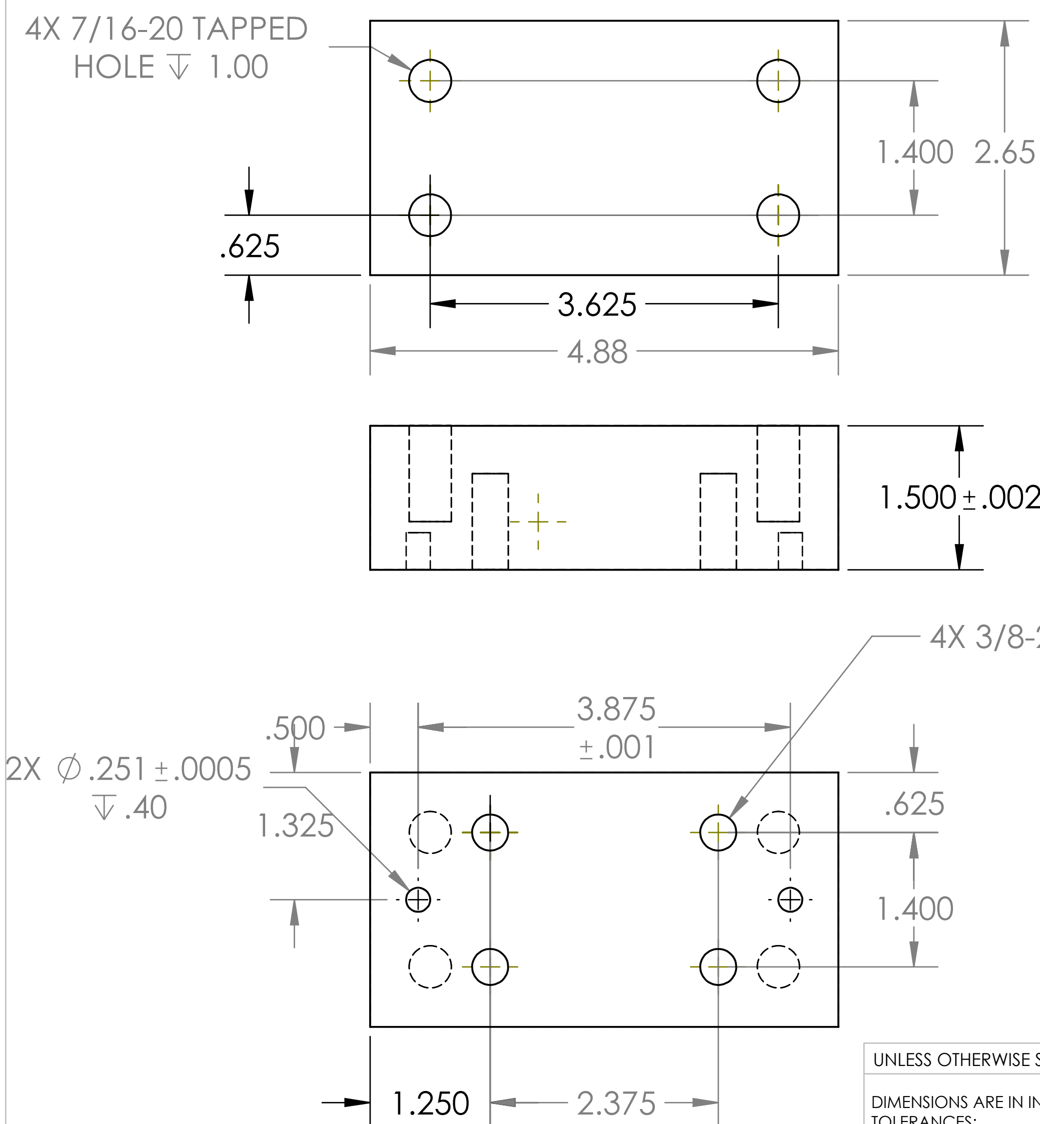

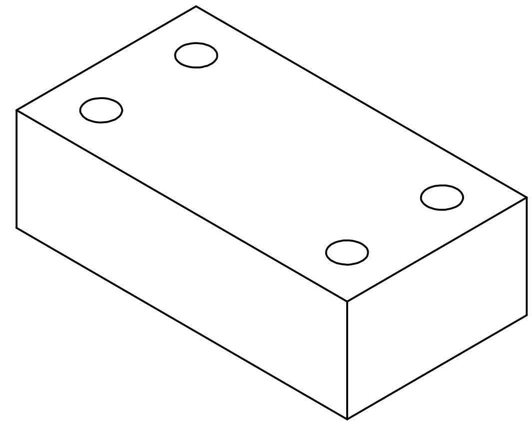

NOTE: DWG. NO. CPSP14-03-02 DOES NOT INCLUDE FEATURE 1

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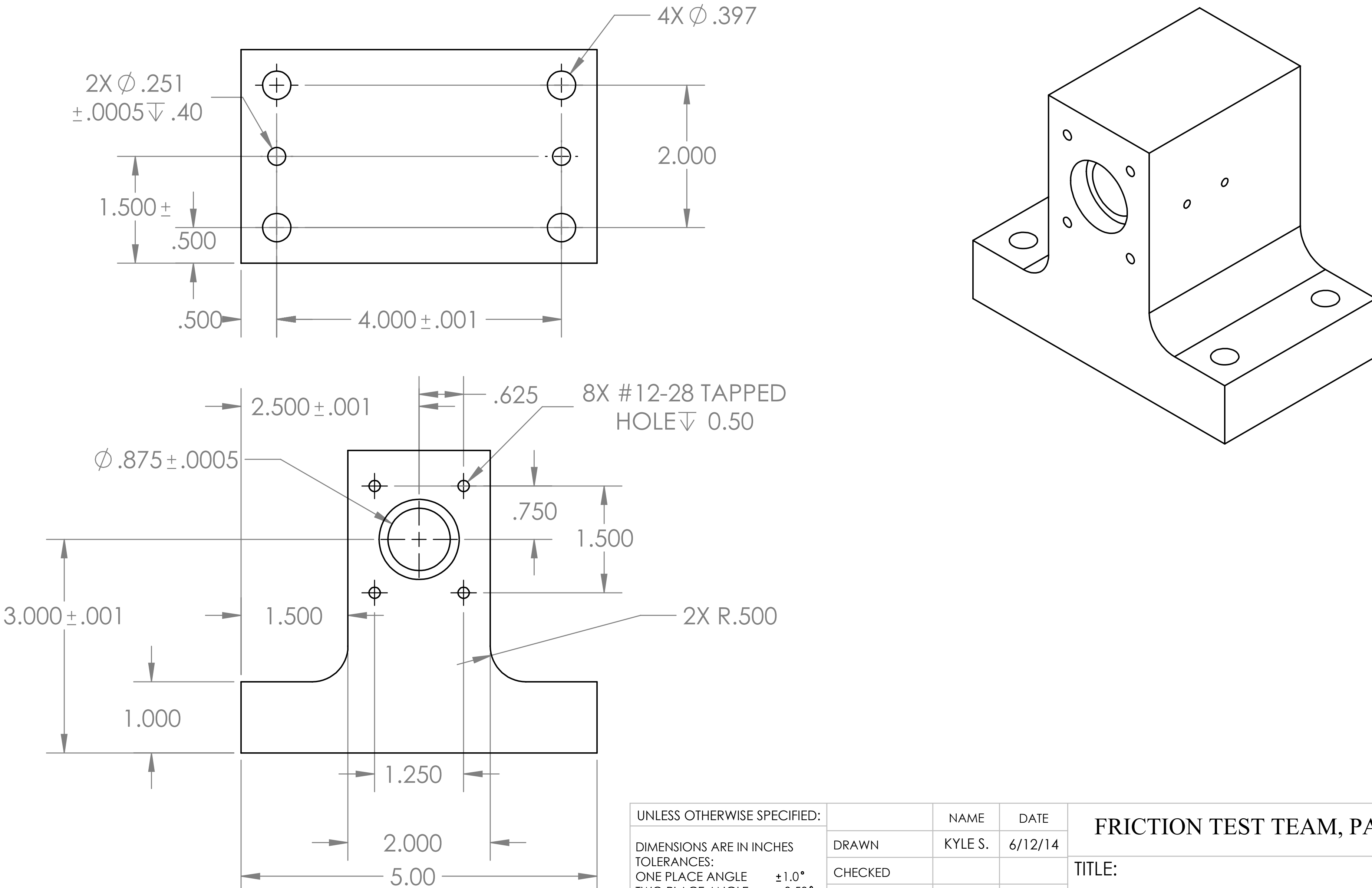
	CHECKED	M.NADRI	06/09/14	PILLOW BLOCK RISER
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER: ASME B46.1	Q.A.			
MATERIAL 6061 - T6	COMMENTS:			SIZE A · DWG. NO. CPSP14-03-01/02 · REV A
FINISH				
DO NOT SCALE DRAWING				SCALE: 1:2 · WEIGHT: · SHEET 1 OF 1

NEXT ASSY: CPSP14-03A	USED ON
APPLICATION	

5 4 3 2 1



UNLESS OTHERWISE SPECIFIED:		NAME	DATE
DIMENSIONS ARE IN INCHES TOLERANCES: ONE PLACE ANGLE ±1.0° TWO PLACE ANGLE ±0.50° TWO PLACE DECIMAL ±0.010 THREE PLACE DECIMAL ±0.005	DRAWN	KYLE S.	6/12/14
	CHECKED		
	ENG APPR.		
	MFG APPR.		
INTERPRET GEOMETRIC TOLERANCING PER: ASME B46.1	Q.A.		
MATERIAL 6061 - T6	COMMENTS:		
FINISH			
DO NOT SCALE DRAWING			

FRICTION TEST TEAM, PARKER

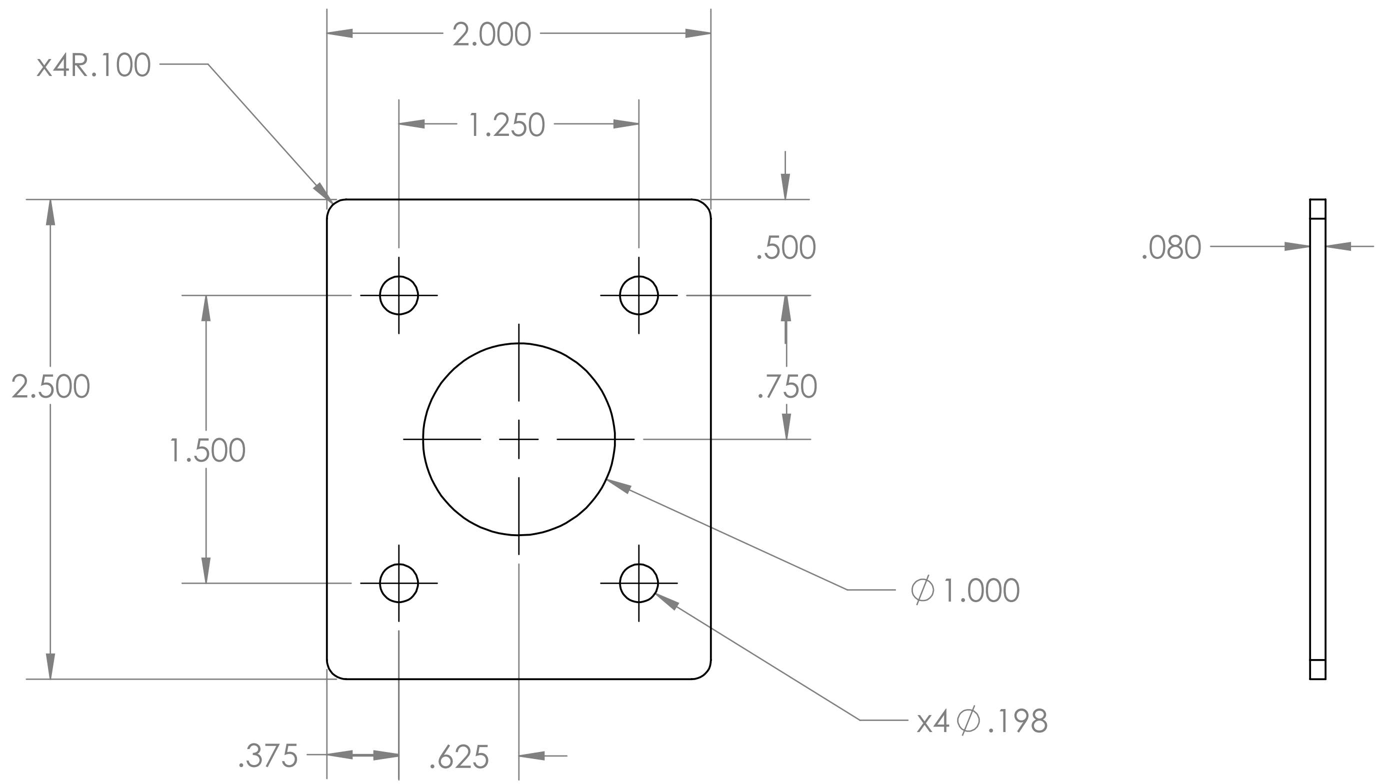
TITLE:

CENTER BEARING SUPPORT

SIZE	DWG. NO.	REV
A	CPSP14-03-03	A
SCALE: 1:2	WEIGHT:	SHEET 1 OF 1

NEXT ASSY: CPSP14-03A	USED ON
APPLICATION	

5 4 3 2 1



UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
ONE PLACE ANGLE ±1.0°
TWO PLACE ANGLE ±0.50°
TWO PLACE DECIMAL ±0.010
THREE PLACE DECIMAL ±0.005

INTERPRET GEOMETRIC TOLERANCING PER: ASME B46.1

MATERIAL 6061 - T6

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	KYLE S.	2/13/14
CHECKED	M.NADRI	06/09/14
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

FRICTION TEST TEAM, PARKER

TITLE:

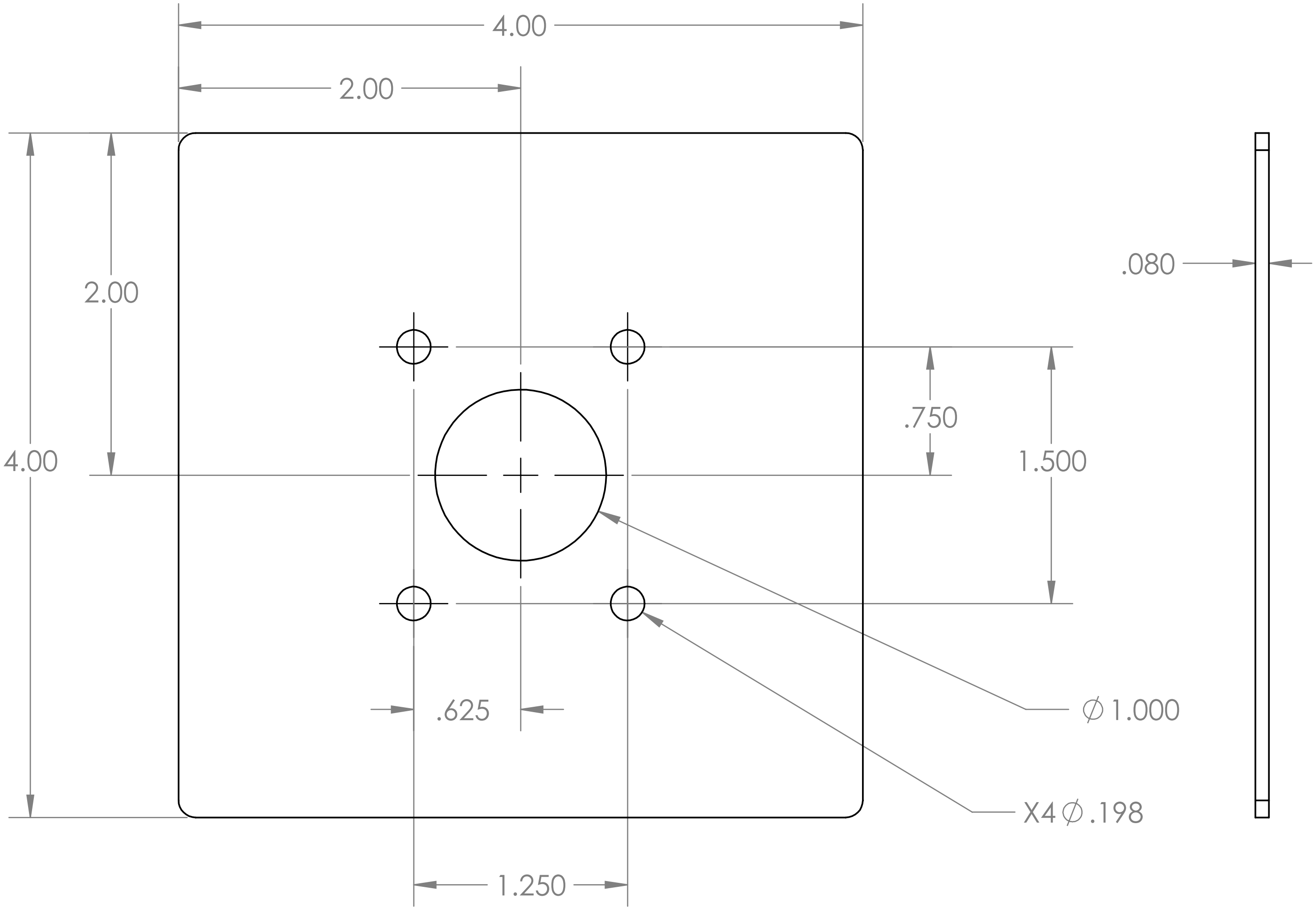
SM BEARING PLATE

SIZE A | DWG. NO. CPSP14-03-05 | REV A

SCALE: 1:1 | WEIGHT: | SHEET 1 OF 1

NEXT ASSY: CPSP14-03A	USED ON
APPLICATION	

5 4 3 2 1



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DIMENSIONS ARE IN INCHES TOLERANCES: ONE PLACE ANGLE ±1.0° TWO PLACE ANGLE ±0.50° TWO PLACE DECIMAL ±0.010 THREE PLACE DECIMAL ±0.005	DRAWN	KYLE S.	2/13/14	TITLE:
	CHECKED	M.NADRI	06/09/14	LRG BEARING PLATE
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	MFG APPR.			
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MATERIAL 6061 - T6	COMMENTS:			SIZE A · DWG. NO. CPSP14-03-06 · REV A
FINISH				
DO NOT SCALE DRAWING				SCALE: 1:1 · WEIGHT: · SHEET 1 OF 1

NEXT ASSY	USED ON
CPSP14-03A	
APPLICATION	

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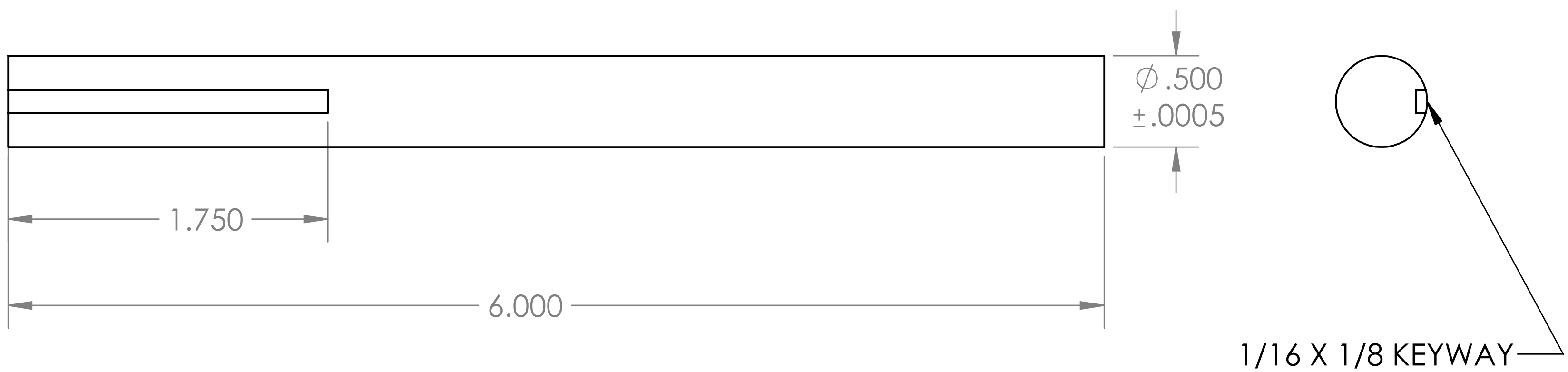

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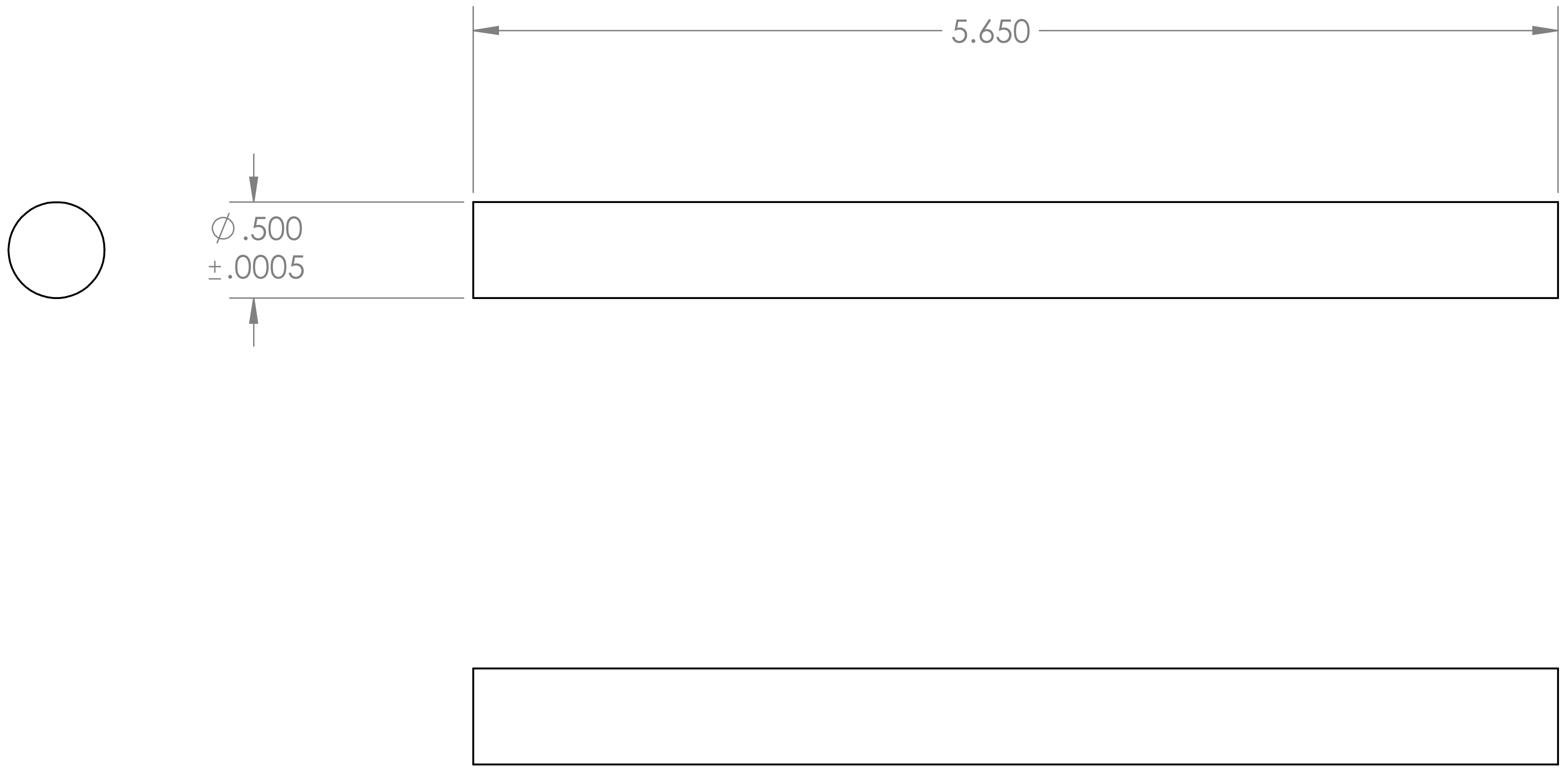
	CHECKED	M.NADRI	06/09/14	INPUT DRIVE SHAFT
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER: ASME B46.1	Q.A.			
MATERIAL 303 SS	COMMENTS:			SIZE A / DWG. NO. CPSP14-03-07 / REV A
FINISH				
DO NOT SCALE DRAWING				SCALE: 1:4 / WEIGHT: / SHEET 1 OF 1

NEXT ASSY	USED ON
CPSP14-03A	
APPLICATION	

5 4 3 2 1



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DIMENSIONS ARE IN INCHES TOLERANCES: ONE PLACE ANGLE ±1.0° TWO PLACE ANGLE ±0.50° TWO PLACE DECIMAL ±0.010 THREE PLACE DECIMAL ±0.005	DRAWN	KYLE S.	2/13/14	TITLE:
	CHECKED	M.NADRI	06/09/14	MID DRIVE SHAFT
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER: ASME B46.1	Q.A.			
MATERIAL 303 SS	COMMENTS:			SIZE A / DWG. NO. CPSP14-03-08 / REV A
FINISH				
DO NOT SCALE DRAWING				SCALE: 1:1 / WEIGHT: / SHEET 1 OF 1

NEXT ASSY: CPSP14-03A	USED ON
APPLICATION	

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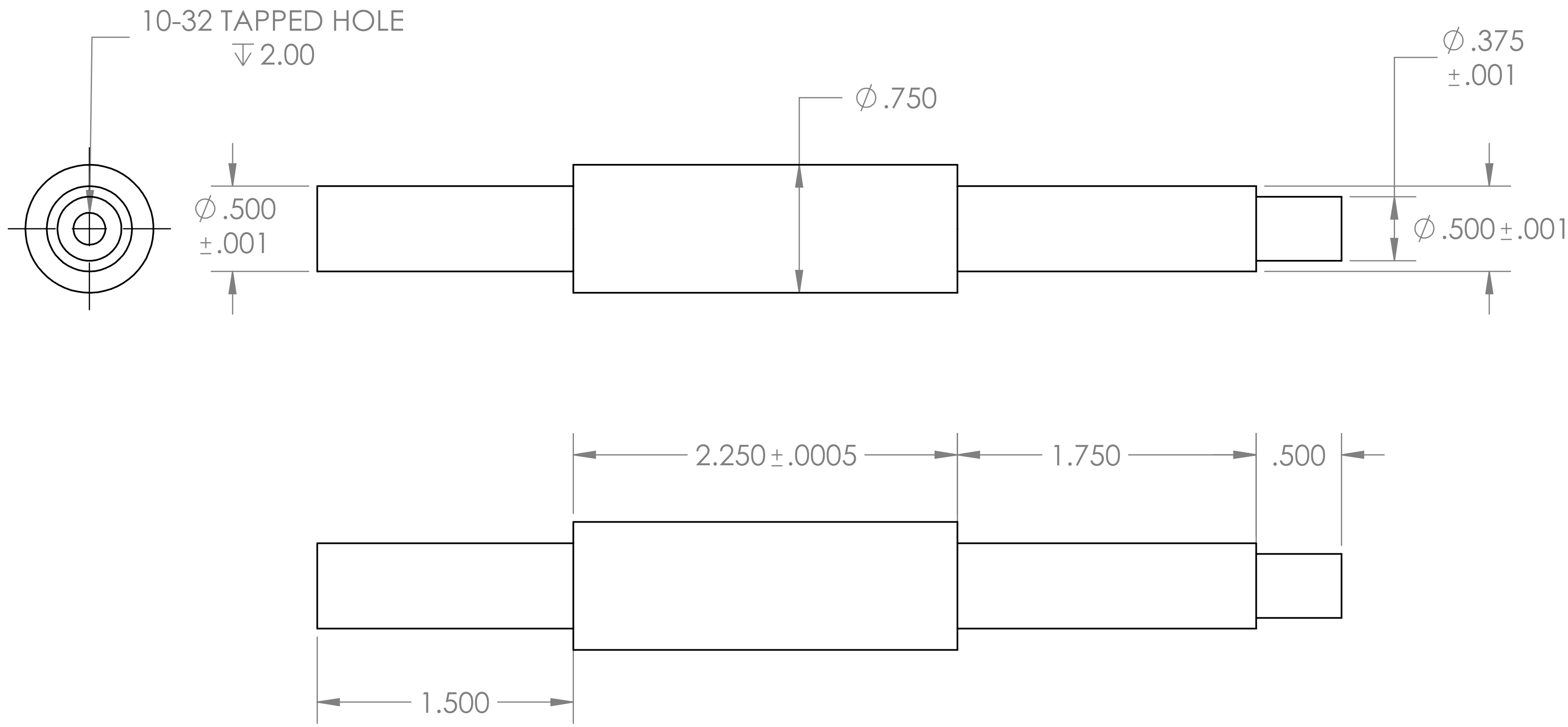

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DIMENSIONS ARE IN INCHES TOLERANCES: ONE PLACE ANGLE ±1.0° TWO PLACE ANGLE ±0.50° TWO PLACE DECIMAL ±0.010 THREE PLACE DECIMAL ±0.005	DRAWN	KYLE S.	2/13/14	TITLE:
	CHECKED	M.NADRI	06/09/14	OUTPUT DRIVE SHAFT
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	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER: ASME B46.1	Q.A.			
MATERIAL 303 SS	COMMENTS:			SIZE A · DWG. NO. CPSP14-03-09 · REV A
FINISH				
DO NOT SCALE DRAWING				SCALE: 1:2 · WEIGHT: · SHEET 1 OF 1

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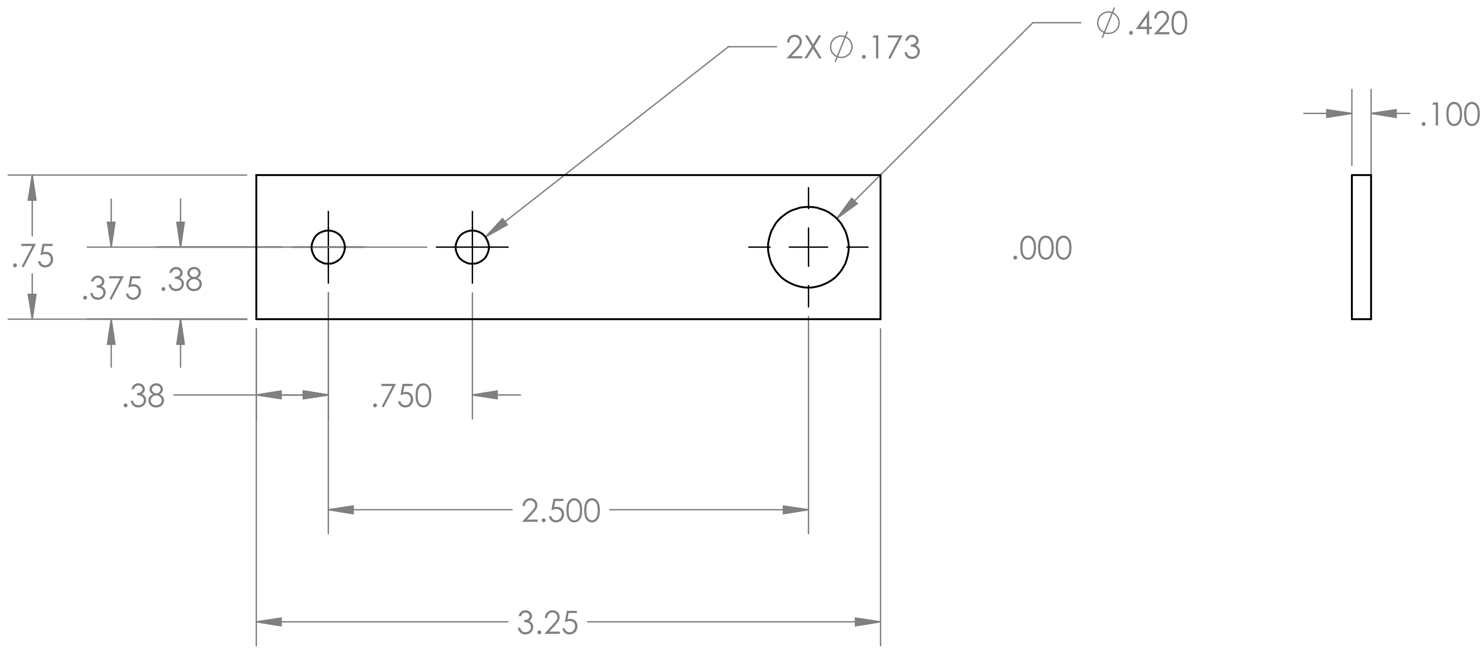

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	CHECKED	M.NADRI	06/09/14	SENSOR BRACKET
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER: ASME B46.1	Q.A.			
MATERIAL 4140	COMMENTS:			SIZE A — DWG. NO. CPSP14-03-10 — REV A
FINISH				
DO NOT SCALE DRAWING				SCALE: 1:1 — WEIGHT: — SHEET 1 OF 1

NEXT ASSY: CPSP14-03A	USED ON
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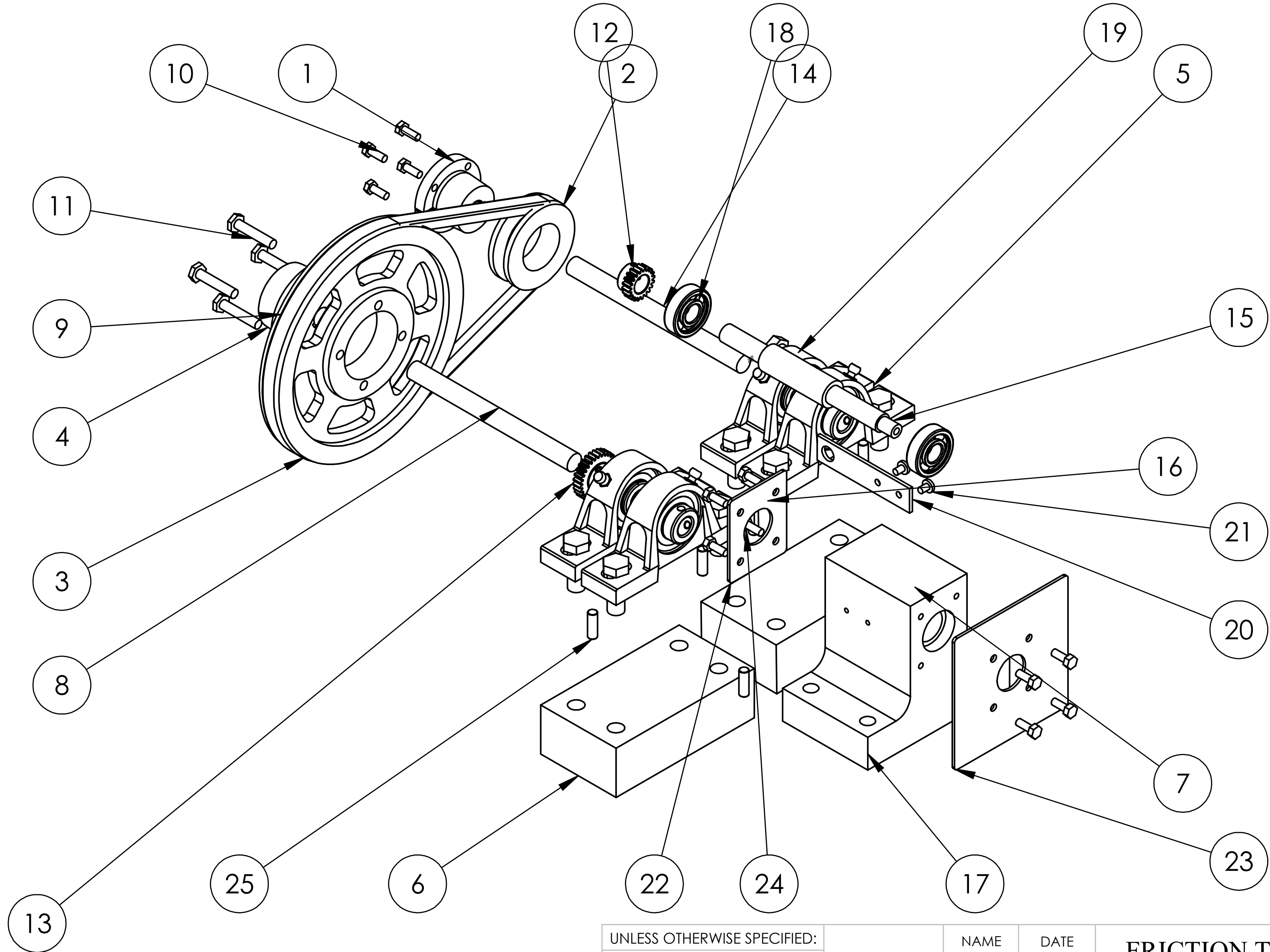

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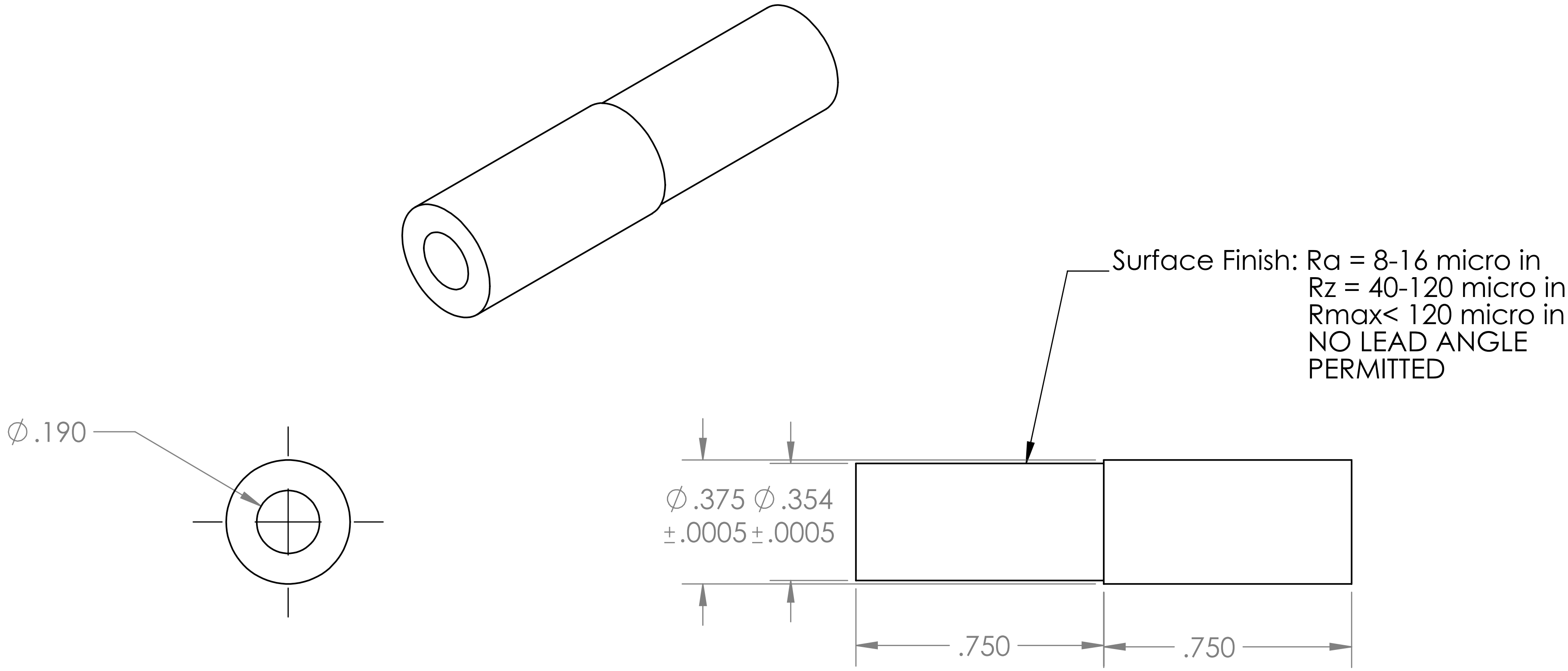
DIMENSIONS ARE IN INCHES TOLERANCES: ONE PLACE ANGLE ±1.0° TWO PLACE ANGLE ±0.50° TWO PLACE DECIMAL ±0.010 THREE PLACE DECIMAL ±0.005	DRAWN	KYLE S.	2/13/14	TITLE:
	CHECKED			DRIVETRAIN ASSEMBLY
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER: ASME B46.1	Q.A.			
MATERIAL	COMMENTS:			SIZE A — DWG. NO. CPSP14-03A — REV A
FINISH				
DO NOT SCALE DRAWING				SCALE: 1:8 — WEIGHT: — SHEET 1 OF 2

NEXT ASSY	USED ON
APPLICATION	

5 4 3 2 1

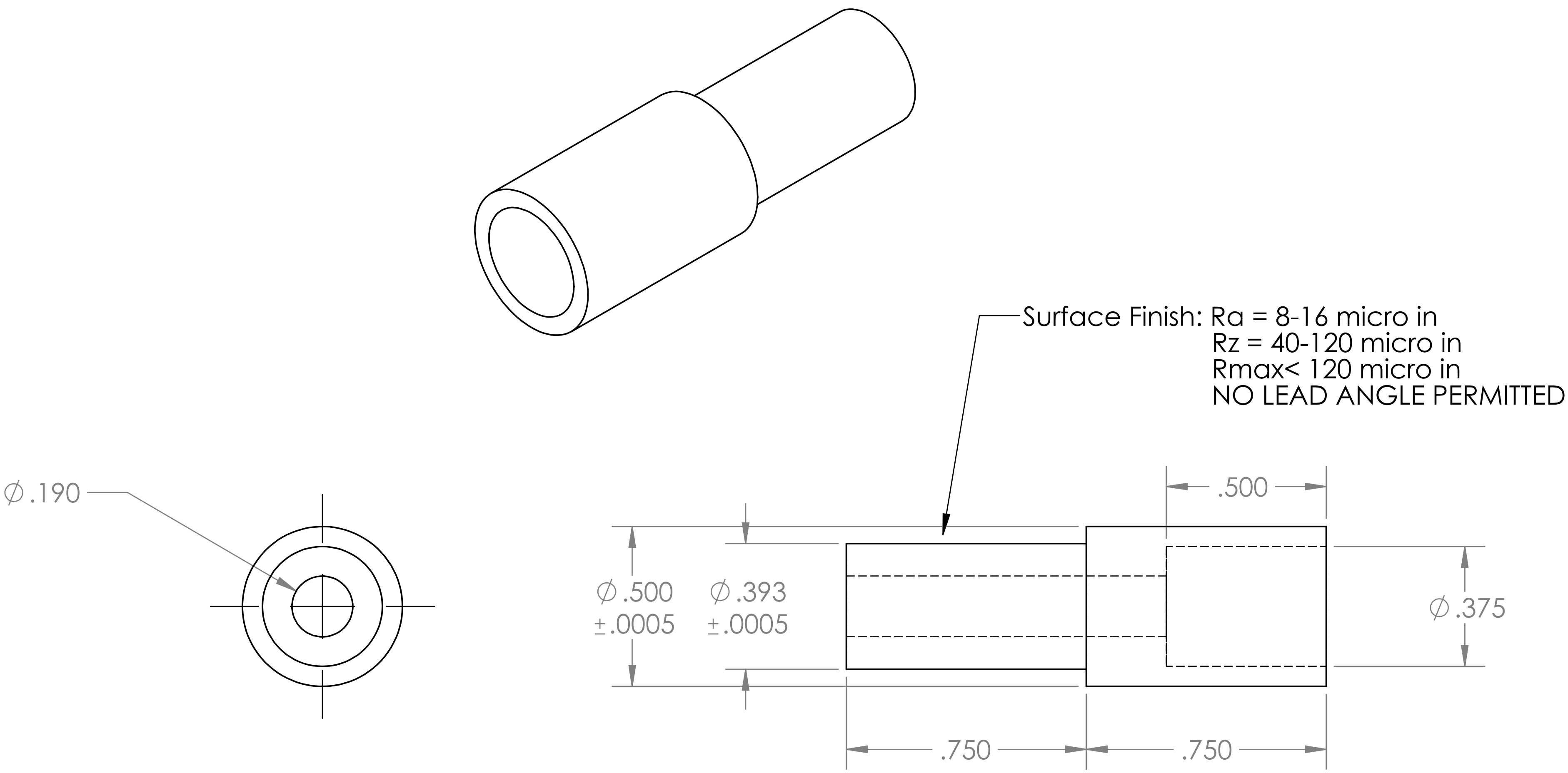
ITEM NO.	PART NUMBER	PART NAME	QTY.
1		JA Bushing	1
2		Powerwedge 2.35 Sheave	1
3		Powerwedge 6.5 Sheave	1
4		SH Bushing	1
5		Browning .5 Pillow Block	4
6	CPSP14-03-02	Riser	1
7	CPSP14-03-01	Input Riser	1
8	CPSP14-03-08	Intermediate Shaft	1
9		Belt	1
10		Small Pulley Bolt	4
11		Large Pulley Bolt	4
12		20TGear	1
13		30TGear	1
14	CPSP14-03-07	Input Shaft	1
15	CPSP14-03-09	Output Shaft	1
16		7-16x1.25 Hex Head	8
17	CPSP14-03-03	Center Support	1
18		Center Support Bearing	2
19		3/8 Hex Head x1.5"	1
20		Speed Sensor Bracket	1
21		#8 Machine Screw	2
22	CPSP14-03-05	Small Retaining Plate	1
23	CPSP14-03-06	Large Retaining Plate	1
24		#12x.5" bolt	8
25		0.250" Dowel Pin x .75"	6



UNLESS OTHERWISE SPECIFIED:		NAME	DATE	FRICTION TEST TEAM, PARKER
DIMENSIONS ARE IN INCHES TOLERANCES: ONE PLACE ANGLE ±1.0° TWO PLACE ANGLE ±0.50° TWO PLACE DECIMAL ±0.010 THREE PLACE DECIMAL ±0.005	DRAWN	KYLE S.	2/13/14	TITLE:
	CHECKED	M.NADRI	06/09/14	SEAL TOOL #1
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER: ASME B46.1	Q.A.			
MATERIAL 303 SS	COMMENTS:			SIZE A · DWG. NO. CPSP14-04-01 · REV A
FINISH				
DO NOT SCALE DRAWING				SCALE: 2:1 · WEIGHT: · SHEET 1 OF 1

NEXT ASSY	USED ON
CPSP14-03A	
APPLICATION	

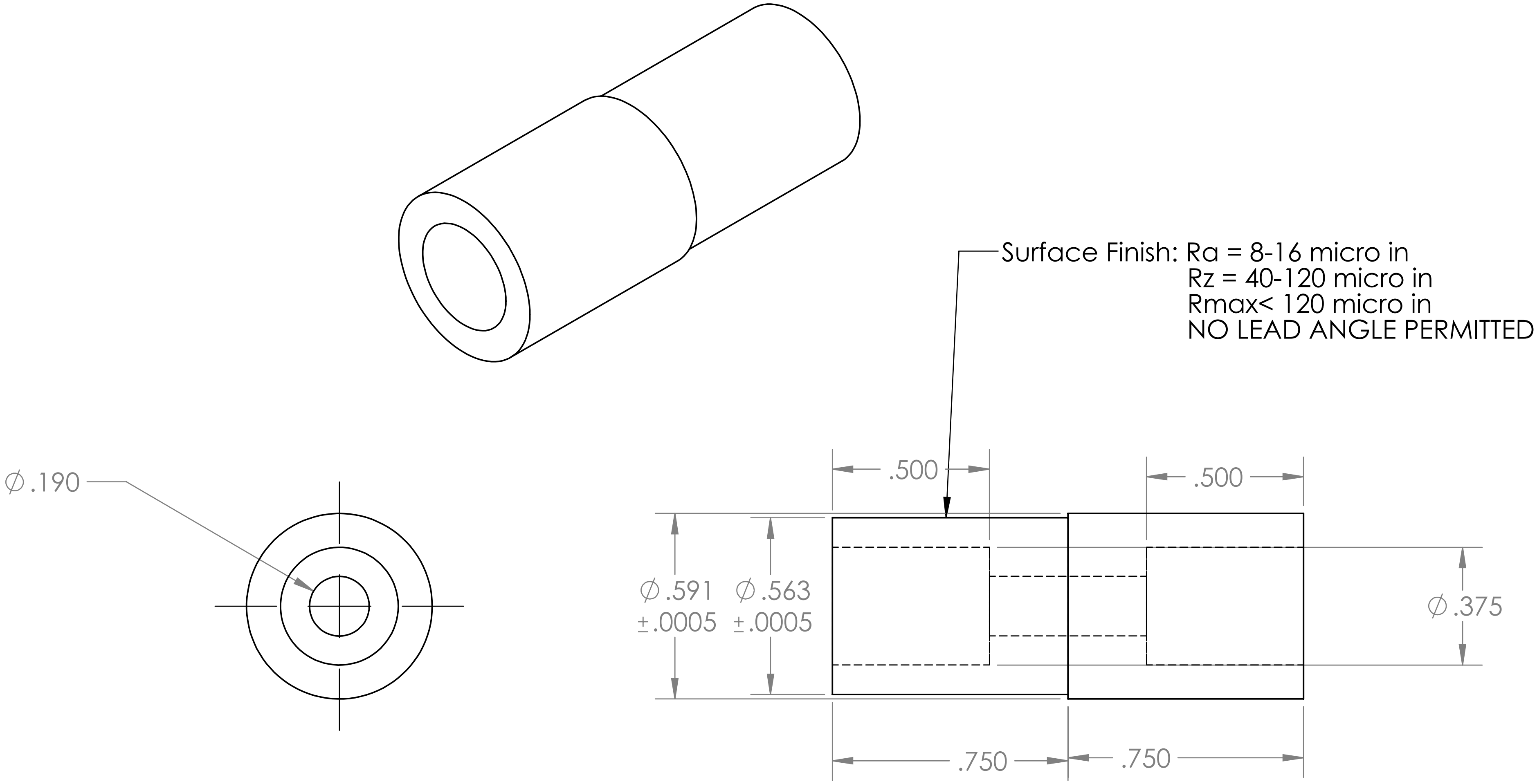
5 4 3 2 1



UNLESS OTHERWISE SPECIFIED:		NAME	DATE	FRICTION TEST TEAM, PARKER
DIMENSIONS ARE IN INCHES TOLERANCES: ONE PLACE ANGLE ±1.0° TWO PLACE ANGLE ±0.50° TWO PLACE DECIMAL ±0.010 THREE PLACE DECIMAL ±0.005	DRAWN	KYLE S.	2/13/14	TITLE:
	CHECKED	M.NADRI	06/09/14	SEAL TOOL #2
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER: ASME B46.1	Q.A.			
MATERIAL 303 SS	COMMENTS:			SIZE A · DWG. NO. CPSP14-04-02 · REV A
FINISH				
DO NOT SCALE DRAWING				SCALE: 2:1 · WEIGHT: · SHEET 1 OF 1

NEXT ASSY: CPSP14-03A	USED ON
APPLICATION	

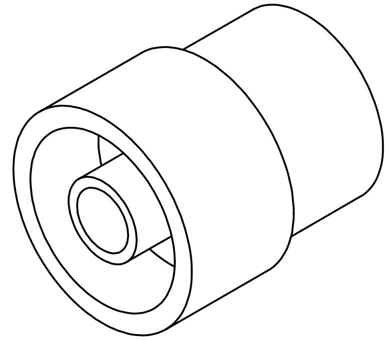
5 4 3 2 1



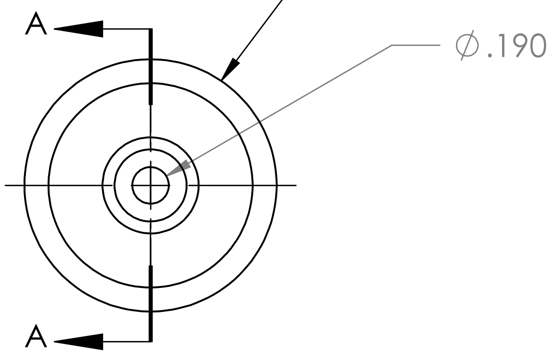
UNLESS OTHERWISE SPECIFIED:		NAME	DATE	FRICTION TEST TEAM, PARKER
DIMENSIONS ARE IN INCHES TOLERANCES: ONE PLACE ANGLE ±1.0° TWO PLACE ANGLE ±0.50° TWO PLACE DECIMAL ±0.010 THREE PLACE DECIMAL ±0.005	DRAWN	KYLE S.	2/13/14	TITLE:
	CHECKED	M.NADRI	06/09/14	SEAL TOOL #3
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER: ASME B46.1	Q.A.			
MATERIAL 303 SS	COMMENTS:			SIZE A	DWG. NO. CPSP14-04-03	REV A
FINISH				
DO NOT SCALE DRAWING				SCALE: 2:1	WEIGHT:	SHEET 1 OF 1

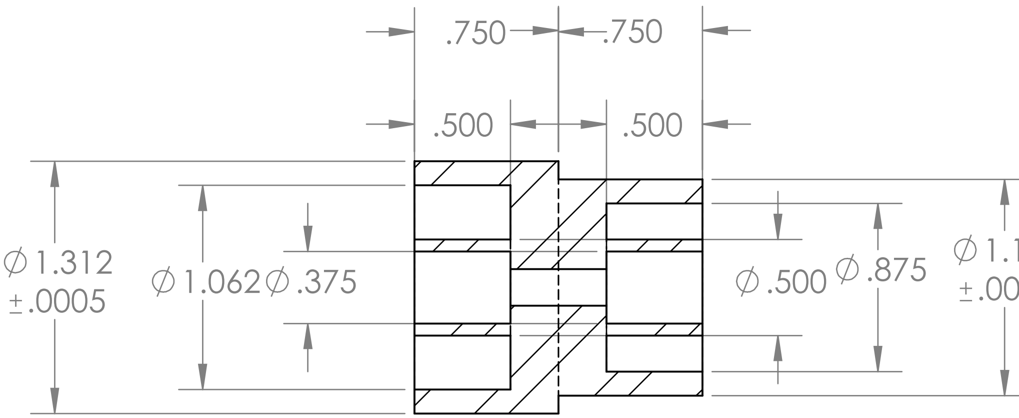
NEXT ASSY	USED ON
APPLICATION	

5 4 3 2 1



Surface Finish: Ra = 8-16 micro in
Rz = 40-120 micro in
Rmax< 120 micro in
NO LEAD ANGLE PERMITTED





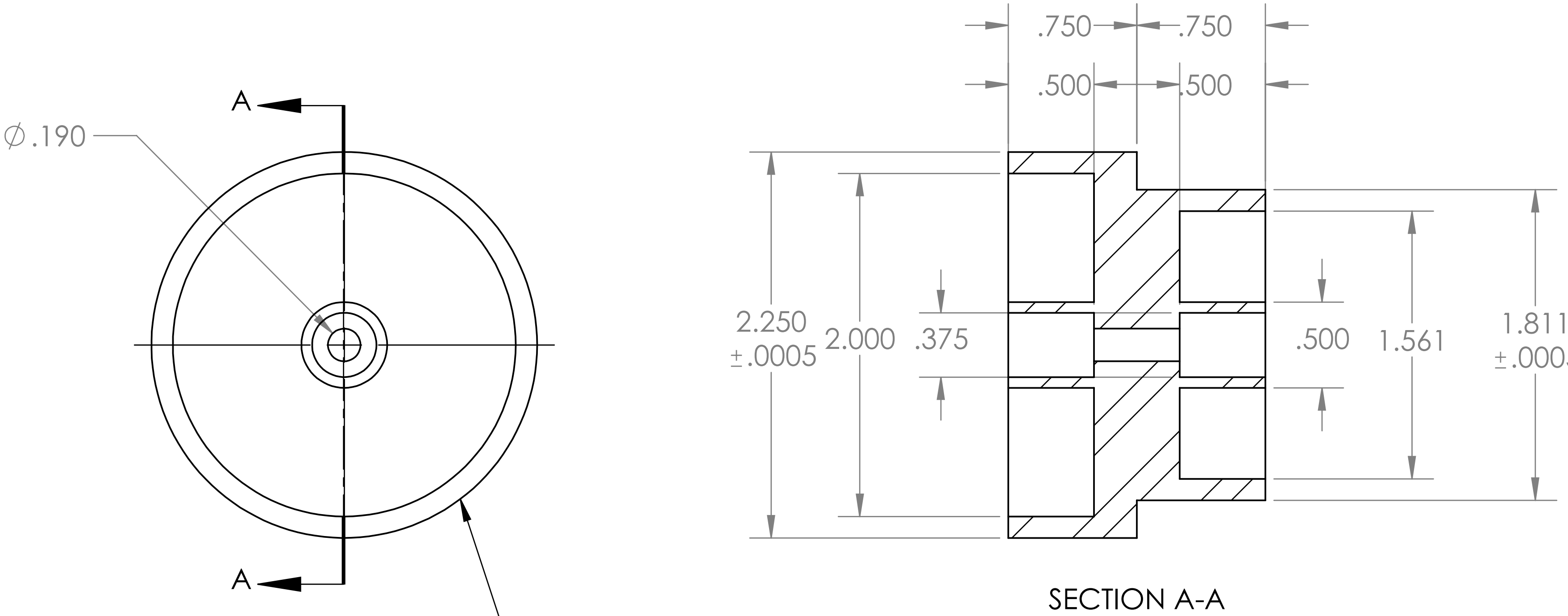
SECTION A-A

UNLESS OTHERWISE SPECIFIED:		NAME	DATE	FRICTION TEST TEAM, PARKER
DIMENSIONS ARE IN INCHES TOLERANCES: ONE PLACE ANGLE ±1.0° TWO PLACE ANGLE ±0.50° TWO PLACE DECIMAL ±0.010 THREE PLACE DECIMAL ±0.005	DRAWN	KYLE S.	2/13/14	TITLE:
	CHECKED	M.NADRI	06/09/14	SEAL TOOL #4
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER: ASME B46.1	Q.A.			
MATERIAL 303 SS	COMMENTS:			SIZE A	DWG. NO. CPSP14-04-04	REV A
FINISH				
DO NOT SCALE DRAWING				SCALE: 1:1	WEIGHT:	SHEET 1 OF 1

NEXT ASSY	USED ON
APPLICATION	

PROPRIETARY AND CONFIDENTIAL

5 4 3 2 1

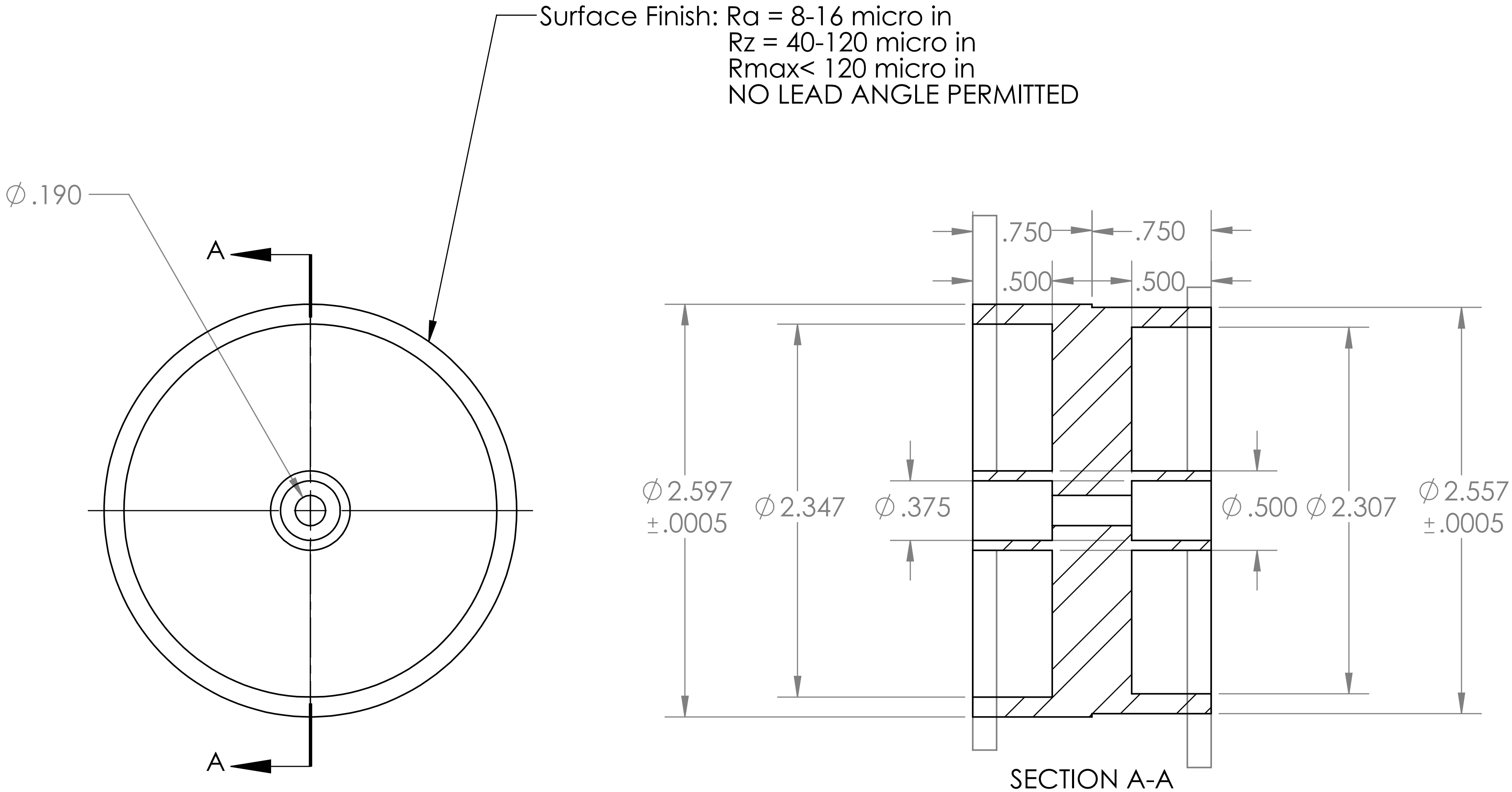


Surface Finish: Ra = 8-16 micro in
Rz = 40-120 micro in
Rmax< 120 micro in
NO LEAD ANGLE PERMITTED

UNLESS OTHERWISE SPECIFIED:		NAME	DATE	FRICTION TEST TEAM, PARKER
DIMENSIONS ARE IN INCHES TOLERANCES: ONE PLACE ANGLE ±1.0° TWO PLACE ANGLE ±0.50° TWO PLACE DECIMAL ±0.010 THREE PLACE DECIMAL ±0.005	DRAWN	KYLE S.	2/13/14	TITLE:
	CHECKED	M.NADRI	06/09/14	SEAL TOOL #5
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER: ASME B46.1	Q.A.			
MATERIAL 303 SS	COMMENTS:			SIZE A — DWG. NO. CPSP14-04-05 — REV A
FINISH				
DO NOT SCALE DRAWING				SCALE: 1:1 — WEIGHT: — SHEET 1 OF 1

NEXT ASSY	USED ON
APPLICATION	

5 4 3 2 1



UNLESS OTHERWISE SPECIFIED:		NAME	DATE
DIMENSIONS ARE IN INCHES TOLERANCES: ONE PLACE ANGLE ±1.0° TWO PLACE ANGLE ±0.50° TWO PLACE DECIMAL ±0.010 THREE PLACE DECIMAL ±0.005	DRAWN	KYLE S.	2/13/14
	CHECKED	M.NADRI	06/09/14
	ENG APPR.		
	MFG APPR.		
INTERPRET GEOMETRIC TOLERANCING PER: ASME B46.1	Q.A.		
MATERIAL 303 SS	COMMENTS:		
FINISH			
DO NOT SCALE DRAWING			

FRICTION TEST TEAM, PARKER

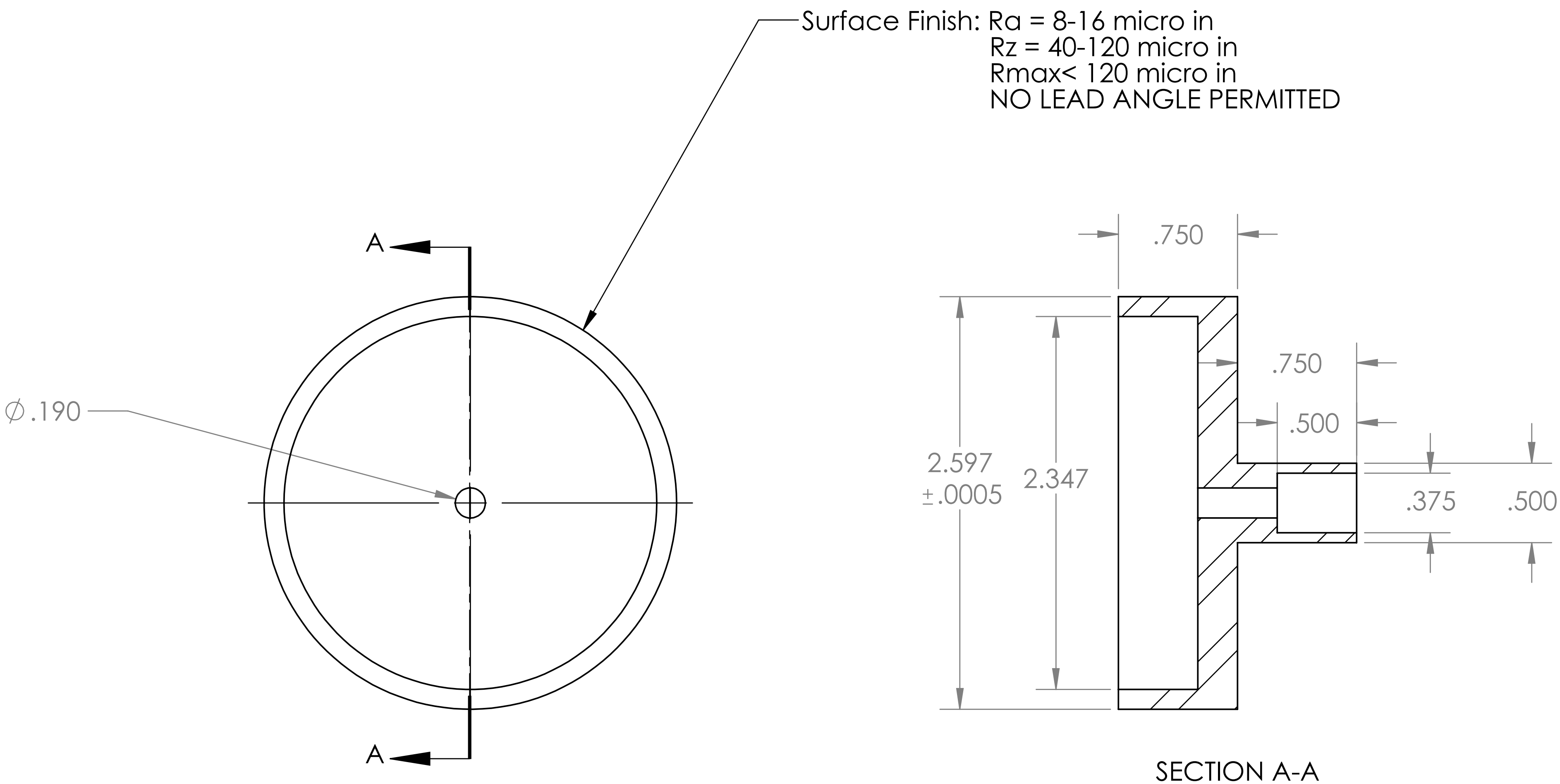
TITLE:

SEAL TOOL #6

SIZE	DWG. NO.	REV
A	CPSP14-04-06	A
SCALE: 1:1	WEIGHT:	SHEET 1 OF 1

NEXT ASSY	USED ON
APPLICATION	

5 4 3 2 1



UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
ONE PLACE ANGLE ±1.0°
TWO PLACE ANGLE ±0.50°
TWO PLACE DECIMAL ±0.010
THREE PLACE DECIMAL ±0.005

INTERPRET GEOMETRIC TOLERANCING PER: ASME B46.1

MATERIAL
303 SS

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	KYLE S.	2/13/14
CHECKED	M.NADRI	06/09/14
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

FRICTION TEST TEAM, PARKER

TITLE:

SEAL TOOL #7

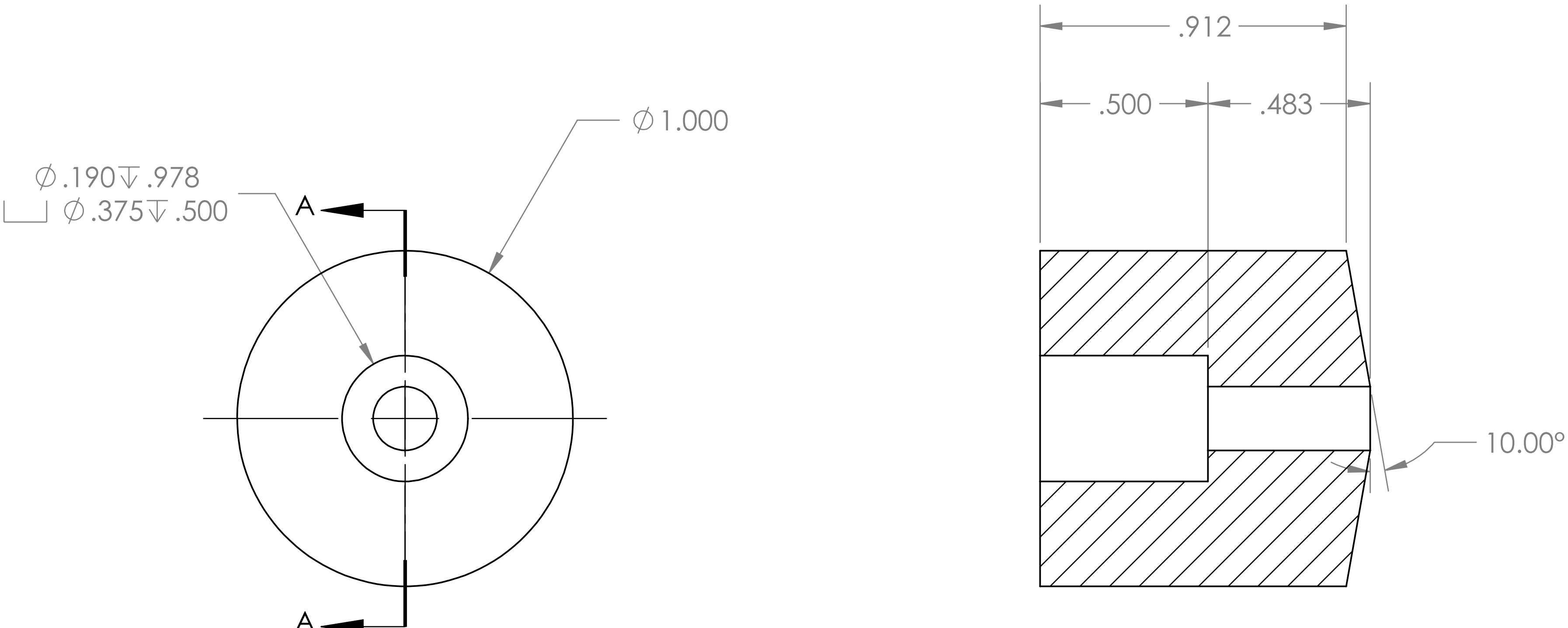
SIZE A | DWG. NO. CPSP14-04-07 | REV A

SCALE: 1:1 | WEIGHT: | SHEET 1 OF 1

NEXT ASSY | USED ON

APPLICATION

5 4 3 2 1



SECTION A-A

UNLESS OTHERWISE SPECIFIED:		NAME	DATE
DIMENSIONS ARE IN INCHES TOLERANCES: ONE PLACE ANGLE ±1.0° TWO PLACE ANGLE ±0.50° TWO PLACE DECIMAL ±0.010 THREE PLACE DECIMAL ±0.005	DRAWN	KYLE S.	2/13/14
	CHECKED	M.NADRI	06/09/14
	ENG APPR.		
	MFG APPR.		
INTERPRET GEOMETRIC TOLERANCING PER: ASME B46.1	Q.A.		
MATERIAL DELRIN	COMMENTS:		
FINISH			
DO NOT SCALE DRAWING			

FRICTION TEST TEAM, PARKER

TITLE:

SM BEARING TOOL 1

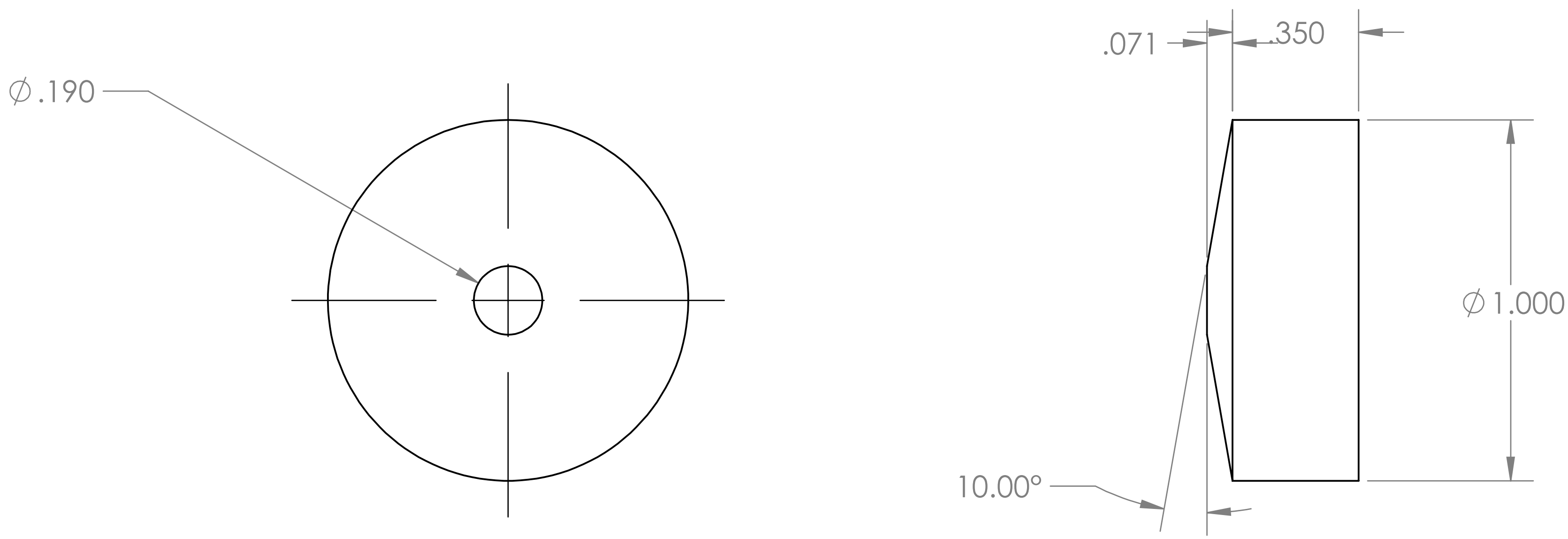
SIZE A | DWG. NO. CPSP14-04-08 | REV A

SCALE: 2:1 | WEIGHT: | SHEET 1 OF 1

NEXT ASSY | USED ON

APPLICATION

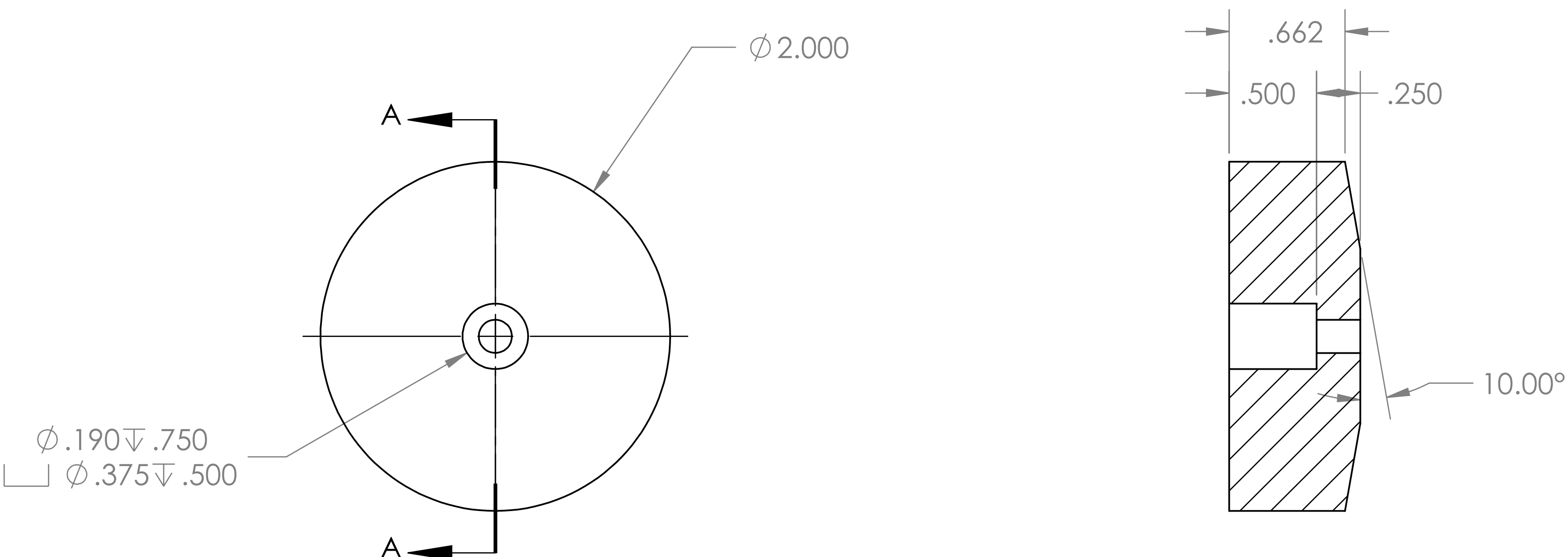
5 4 3 2 1



UNLESS OTHERWISE SPECIFIED:		NAME	DATE	FRICTION TEST TEAM, PARKER
DIMENSIONS ARE IN INCHES TOLERANCES: ONE PLACE ANGLE ±1.0° TWO PLACE ANGLE ±0.50° TWO PLACE DECIMAL ±0.010 THREE PLACE DECIMAL ±0.005	DRAWN	KYLE S.	2/13/14	TITLE:
	CHECKED	M.NADRI	06/09/14	SM BEARING TOOL 2
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER: ASME B46.1	Q.A.			
MATERIAL DELRIN	COMMENTS:			SIZE A · DWG. NO. CPSP14-04-09 · REV A
FINISH				
DO NOT SCALE DRAWING				SCALE: 2:1 · WEIGHT: · SHEET 1 OF 1

NEXT ASSY	USED ON
APPLICATION	

5 4 3 2 1



SECTION A-A

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
ONE PLACE ANGLE ±1.0°
TWO PLACE ANGLE ±0.50°
TWO PLACE DECIMAL ±0.010
THREE PLACE DECIMAL ±0.005

INTERPRET GEOMETRIC TOLERANCING PER: ASME B46.1

MATERIAL
DELRIN

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	KYLE S.	2/13/14
CHECKED	M.NADRI	06/09/14
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

FRICTION TEST TEAM, PARKER

TITLE:

MED BEARING TOOL 1

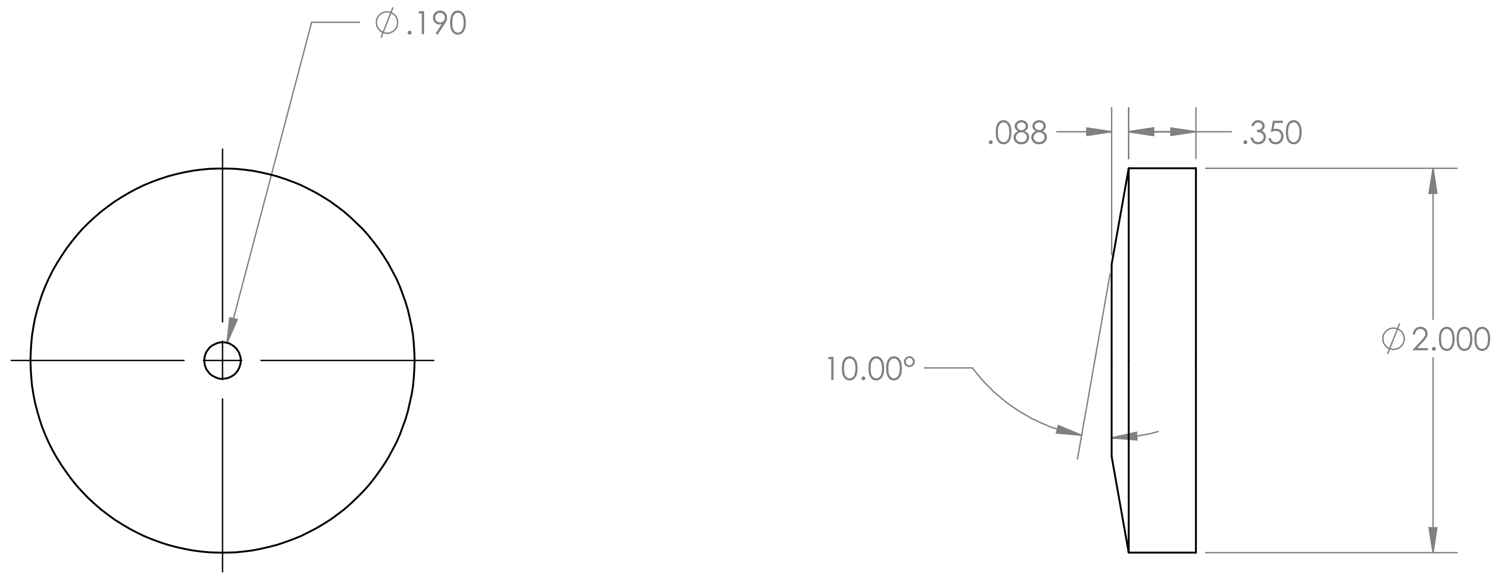
SIZE A | DWG. NO. CPSP14-04-10 | REV A

SCALE: 1:1 | WEIGHT: | SHEET 1 OF 1

NEXT ASSY | USED ON

APPLICATION

5 4 3 2 1



UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
ONE PLACE ANGLE ±1.0°
TWO PLACE ANGLE ±0.50°
TWO PLACE DECIMAL ±0.010
THREE PLACE DECIMAL ±0.005

INTERPRET GEOMETRIC TOLERANCING PER: ASME B46.1

MATERIAL
DELRIN

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	KYLE S.	2/13/14
CHECKED	M.NADRI	06/09/14
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

FRICTION TEST TEAM, PARKER

TITLE:

MED BEARING TOOL 2

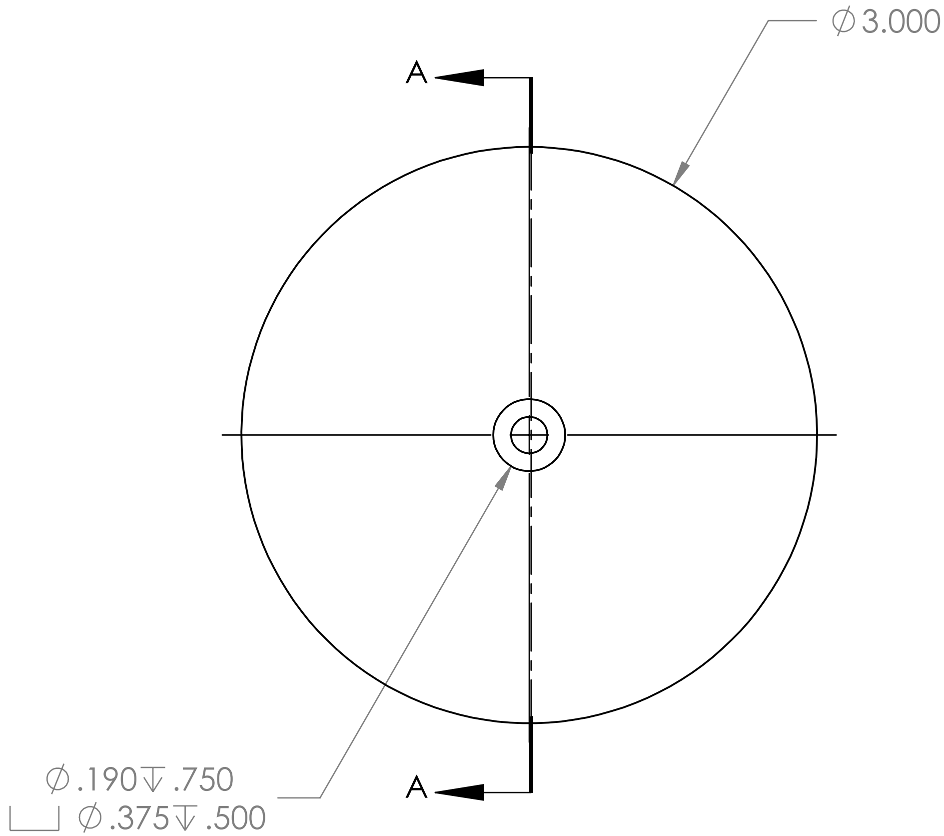
SIZE A | DWG. NO. CPSP14-04-11 | REV A

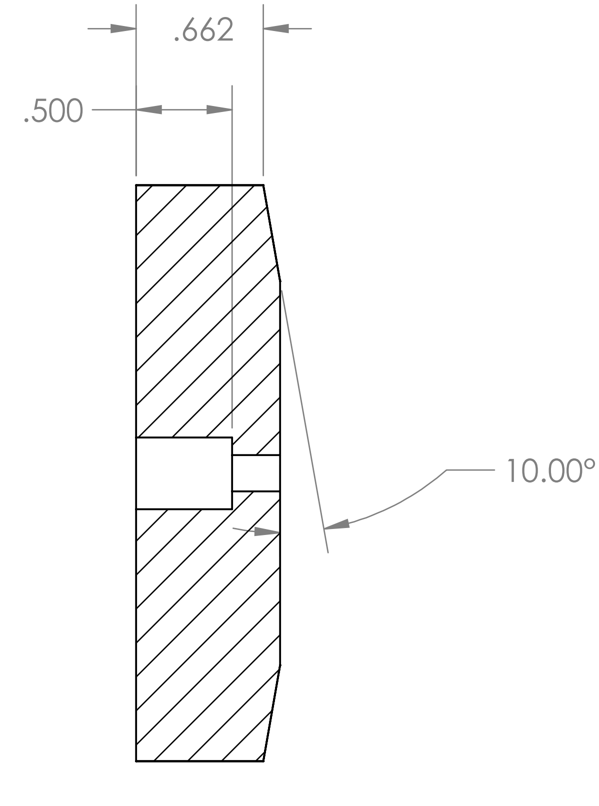
SCALE: 1:1 | WEIGHT: | SHEET 1 OF 1

NEXT ASSY | USED ON

APPLICATION

5 4 3 2 1





SECTION A-A

UNLESS OTHERWISE SPECIFIED:		NAME	DATE	FRICTION TEST TEAM, PARKER
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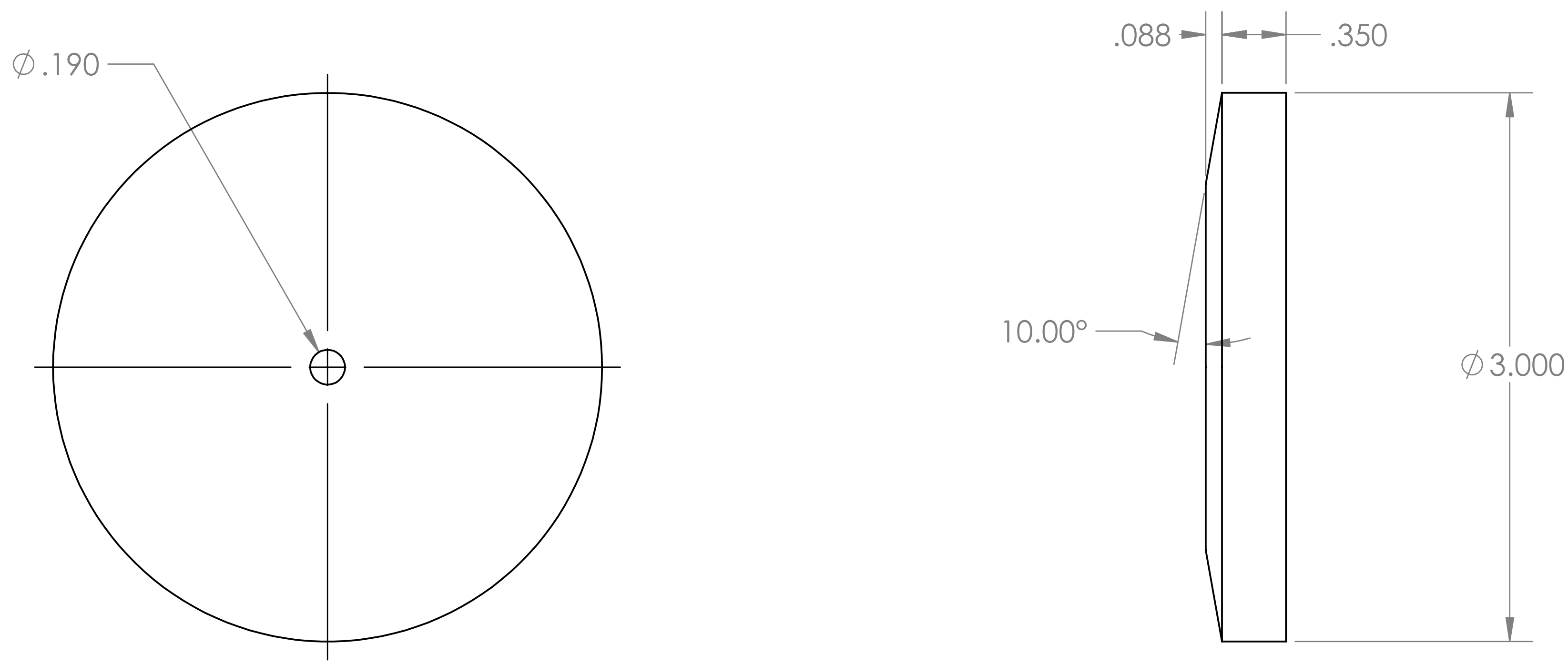
	CHECKED	M.NADRI	06/06/14	LRG BEARING TOOL 1
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER: ASME B46.1	Q.A.			
MATERIAL DELRIN	COMMENTS:			SIZE A / DWG. NO. CPSP14-04-12 / REV A
FINISH				
DO NOT SCALE DRAWING				SCALE: 1:1 / WEIGHT: / SHEET 1 OF 1

NEXT ASSY	USED ON
APPLICATION	

5 4 3 2 1



UNLESS OTHERWISE SPECIFIED:		NAME	DATE	FRICTION TEST TEAM, PARKER
DIMENSIONS ARE IN INCHES TOLERANCES: ONE PLACE ANGLE ±1.0° TWO PLACE ANGLE ±0.50° TWO PLACE DECIMAL ±0.010 THREE PLACE DECIMAL ±0.005	DRAWN	KYLE S.	2/13/14	TITLE:
	CHECKED	M.NADRI	06/09/14	LRG BEARING TOOL 2
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER: ASME B46.1	Q.A.			
MATERIAL DELRIN	COMMENTS:			SIZE A · DWG. NO. CPSP14-04-13 · REV A
FINISH				
DO NOT SCALE DRAWING				SCALE: 1:1 · WEIGHT: · SHEET 1 OF 1

NEXT ASSY	USED ON
APPLICATION	

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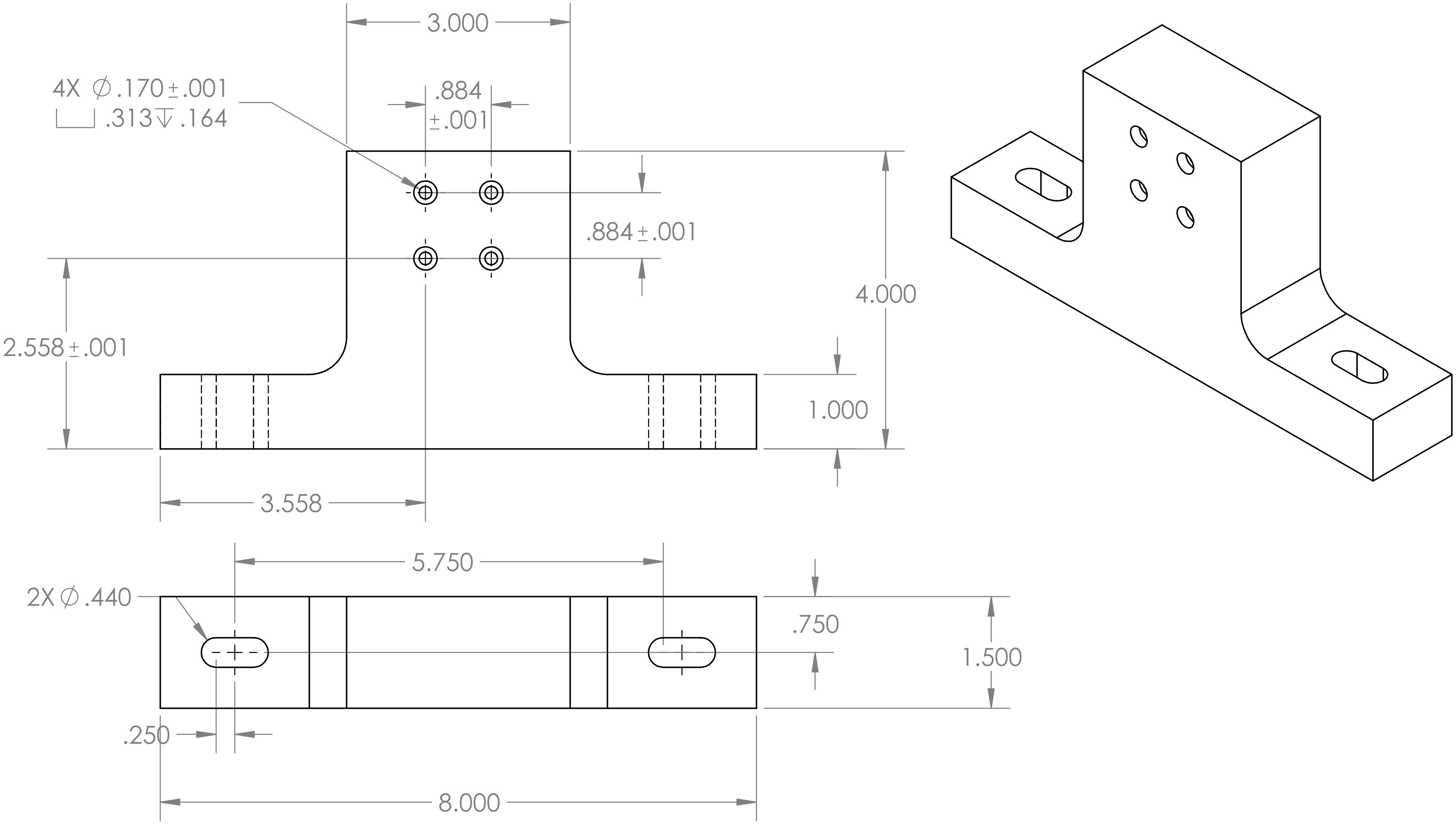


UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
ONE PLACE ANGLE ±1.0°
TWO PLACE ANGLE ±0.50°
TWO PLACE DECIMAL ±0.010
THREE PLACE DECIMAL ±0.005

INTERPRET GEOMETRIC TOLERANCING PER: ASME B46.1

MATERIAL
6061 - T6

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	KYLE S.	2/13/14
CHECKED	M.NADRI	06/09/14
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

FRICTION TEST TEAM, PARKER

TITLE:

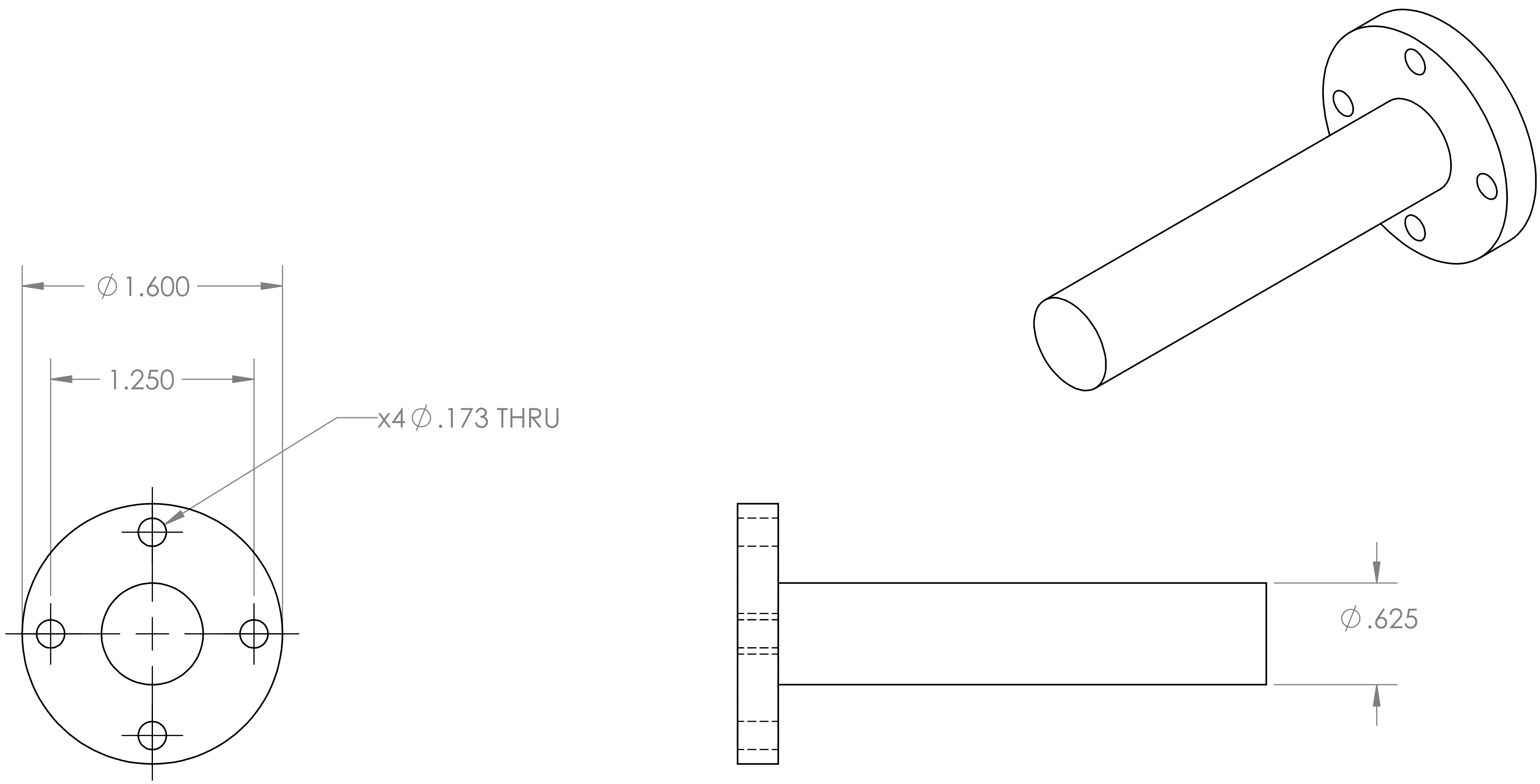
TORQUE CELL MOUNT

SIZE A — DWG. NO. CPSP14-05-01 — REV A

SCALE: 1:2 WEIGHT: SHEET 1 OF 1

NEXT ASSY	USED ON
CPSP14-05A	
APPLICATION	

5 4 3 2 1



UNLESS OTHERWISE SPECIFIED:		NAME	DATE	FRICTION TEST TEAM, PARKER
DIMENSIONS ARE IN INCHES TOLERANCES: ONE PLACE ANGLE ±1.0° TWO PLACE ANGLE ±0.50° TWO PLACE DECIMAL ±0.010 THREE PLACE DECIMAL ±0.005	DRAWN	KYLE S.	2/13/14	TITLE:
	CHECKED	M.NADRI	06/09/14	TORQUE COUPLER
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER: ASME B46.1	Q.A.			
MATERIAL 6061 - T6	COMMENTS:			SIZE A — DWG. NO. CPSP14-05-02 — REV A
FINISH				
DO NOT SCALE DRAWING				SCALE: 1:2 — WEIGHT: — SHEET 1 OF 1

NEXT ASSY: CPSP14-05A — USED ON

APPLICATION

PROPRIETARY AND CONFIDENTIAL

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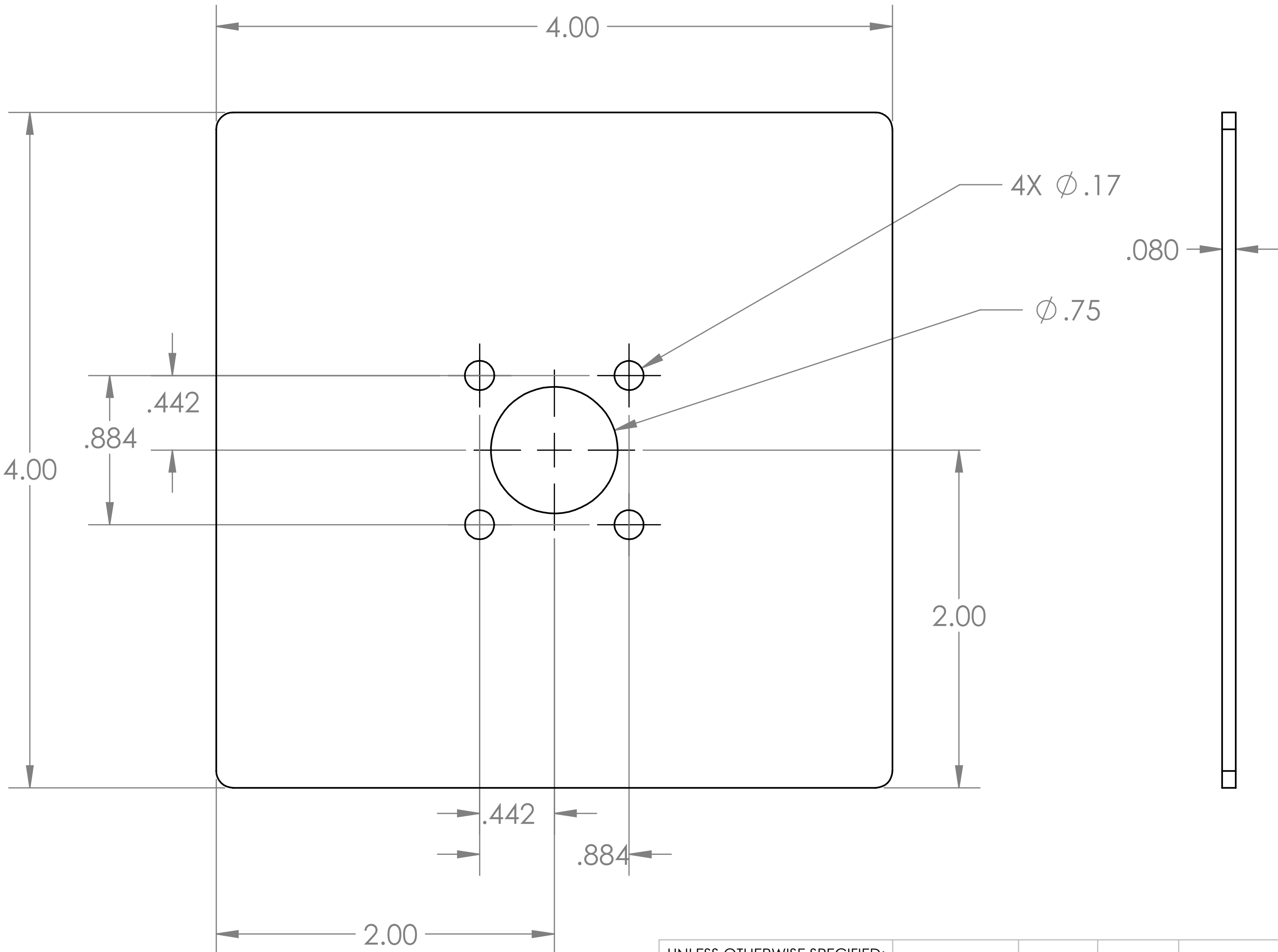
5 4 3 2 1



UNLESS OTHERWISE SPECIFIED:		NAME	DATE
DIMENSIONS ARE IN INCHES TOLERANCES: ONE PLACE ANGLE ±1.0° TWO PLACE ANGLE ±0.50° TWO PLACE DECIMAL ±0.010 THREE PLACE DECIMAL ±0.005	DRAWN	KYLE S.	2/13/14
	CHECKED		
	ENG APPR.		
	MFG APPR.		
INTERPRET GEOMETRIC TOLERANCING PER: ASME B46.1	Q.A.		
MATERIAL 6061 - T6	COMMENTS:		
FINISH			
DO NOT SCALE DRAWING			

FRICTION TEST TEAM, PARKER

TITLE:

TORQUE GUARD

SIZE A | DWG. NO. CPSP14-05-04 | REV A

SCALE: 1:1 | WEIGHT: | SHEET 1 OF 1

NEXT ASSY: CPSP14-03A	USED ON
APPLICATION	

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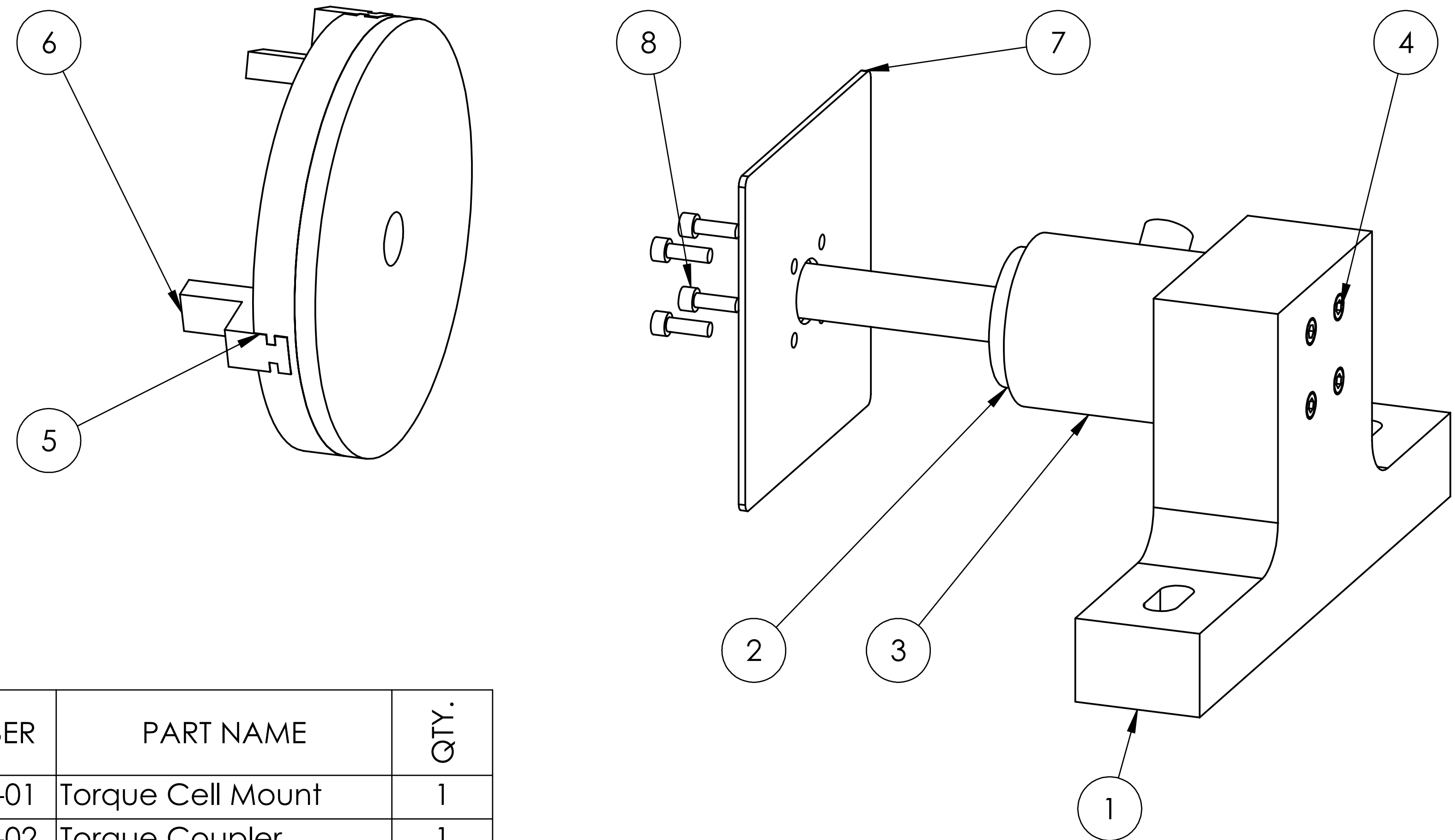


ITEM NO.	PART NUMBER	PART NAME	QTY.
1	CPSO14-05-01	Torque Cell Mount	1
2	CPSO14-05-02	Torque Coupler	1
3		Futek TFF400	1
4		#8-32 CapScrew x1.6"	4
5		Grizzly 5" 3-Jaw Chuck	1
6		Chuck Jaws	3
7	CPSO14-05-04	TorqueCell Guard	1
8		#8-32 CapScrew x.5"	4

NEXT ASSY: CPSP14-00A | USED ON

APPLICATION

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
ONE PLACE ANGLE ±1.0°
TWO PLACE ANGLE ±0.50°
TWO PLACE DECIMAL ±0.010
THREE PLACE DECIMAL ±0.005

INTERPRET GEOMETRIC TOLERANCING PER: ASME B46.1

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	KYLE S.	2/13/14
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

FRICTION TEST TEAM, PARKER

TITLE:

TORQUE ASSEM

SIZE A | DWG. NO. CPSP14-05A | REV A

SCALE: 1:4 | WEIGHT: | SHEET 1 OF 1

5 4 3 2 1

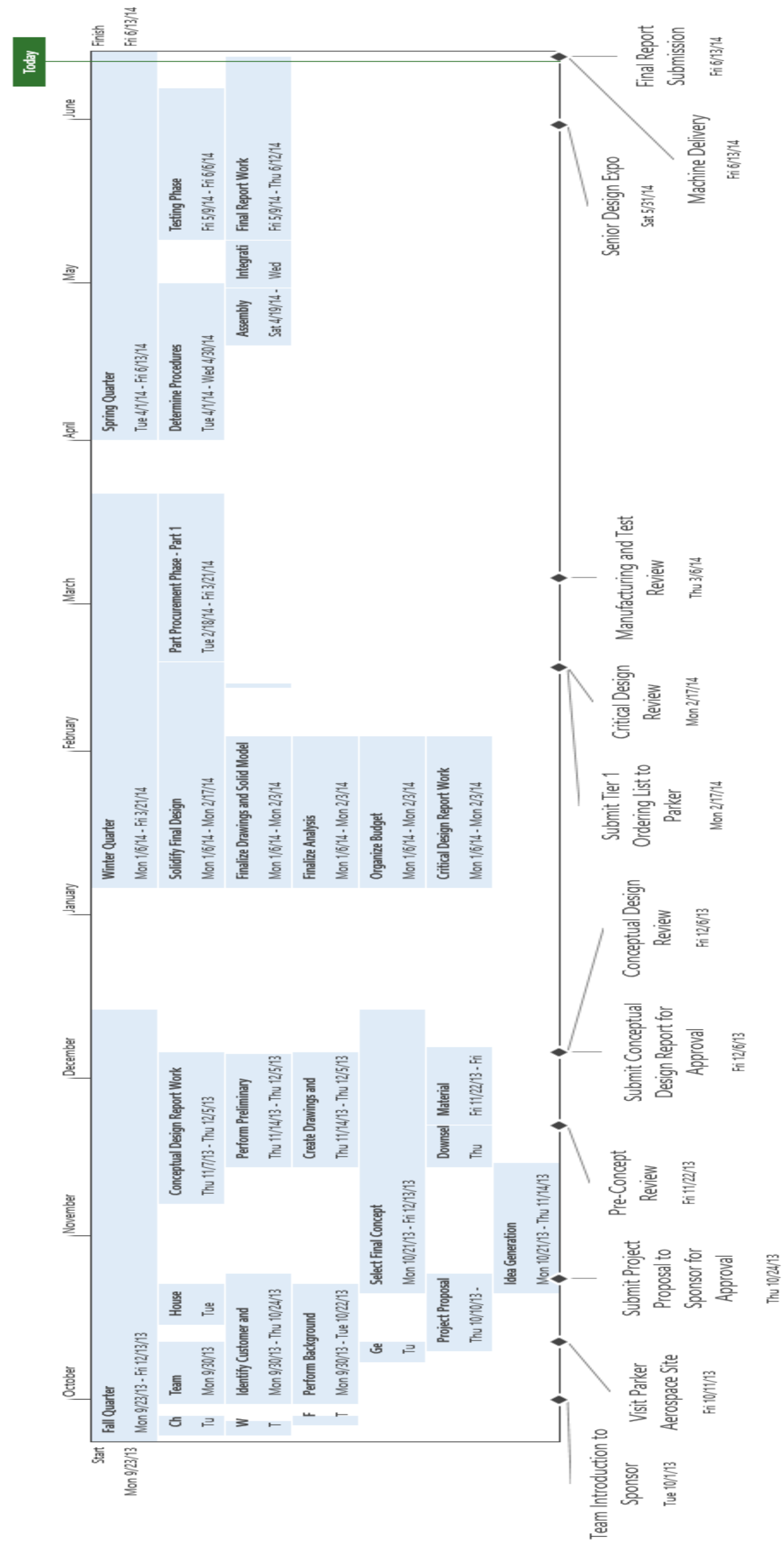
Appendix C

Vendor Information and Cost Breakdown

List #	Item	Qty	Unit Cost	Ordered From	Part Number	Tax	Shipping	Total Cost
1	Browning VPS-208 Pillow Blocks	4	$ 40.21	State Motors & Control	767633	$ -	$ -	$ 160.84
2	3/4" Diameter, 1' Long 303 Stainless Steel Rod	1	$ 15.75	McMaster-Carr	8984K91	$ 2.43	$ 6.37	$ 24.55
3	0.050" Thick, 6"x12" 4130 Alloy Steel Sheet	1	$ 16.62	McMaster-Carr	4459T143	$ -	$ -	$ 16.62
4	1/2" Diameter, 2' Long 303 Stainless Steel Rod	1	$ 14.64	McMaster-Carr	8984K83	$ 2.43	$ 10.98	$ 28.05
5	3/4" Diameter, 1' Long 303 Stainless Steel Rod	1	$ 15.75	McMaster-Carr	8984K91	$ -	$ -	$ 15.75
6	30 Tooth Hard Spur Gear	1	$ 43.59	Stock Drive Products	S10C9Z-024H030	$ -	$ -	$ 43.59
7	20 Tooth Hard Spur Gear	1	$ 37.40	Stock Drive Products	S10C9Z-024H020	$ -	$ -	$ 37.40
8	Delrin 3" Diameter Rod, 2' Long	1	$ 130.55	Professional Plastics, Inc.	RDELNA3.000	$ -	$ -	$ 130.55
9	Browning 6.5" Power-Wedge QD Pulley 1-3V65	1	$ 51.23	State Motors & Control	13V650SH	$ -	$ -	$ 51.23
10	Browning 2.35" Power-Wedge QD Pulley 1-3V23	1	$ 24.22	State Motors & Control	13V235JA	$ -	$ -	$ 24.22
11	Browning 1/2" SH QD Bushing with 1/8"x1/16" Keyway	1	$ 18.17	State Motors & Control	1077528	$ -	$ -	$ 18.17
12	Browning 1/2" JA QD Bushing with 1/8"x1/16" Keyway	1	$ 13.29	State Motors & Control	1074640	$ -	$ -	$ 13.29
13	Browning Power-Wedge Single Groove Cog Belt, 26.5" Long	1	$ 10.12	State Motors & Control	3001039	$ -	$ -	$ 10.12
14	6061 T651 Plate 0.5"x18"x18"	1	$ 82.62	Future Alloys	N/A	$19.39	$ 29.15	$ 131.16
15	7075 T6511 Rod 1.75" Diam., 12" Long	1	$ 20.58	Future Alloys	N/A	$ -	$ -	$ 20.58
16	7075 T651 Plate 1.75"x2.75"x12"	1	$ 41.23	Future Alloys	N/A	$ -	$ -	$ 41.23
17	7075 T651 Plate 1.5"x4"x10"	1	$ 42.84	Future Alloys	N/A	$ -	$ -	$ 42.84
18	7075 T651 Plate 3.25"x4.5"x6"	1	$ 62.65	Future Alloys	N/A	$ -	$ -	$ 62.65
19	7075 T651 Plate 1"x2"x6"	1	$ 8.57	Future Alloys	N/A	$ -	$ -	$ 8.57
20	National Instruments myRIO-1900 DAQ	1	$ 249.00	National Instruments	782693-01	$ -	$ 18.79	$ 267.79
21	Hamlin Hall Effect Magnetic Sensor	1	$ 18.30	Newark Element 14	55075-00-02-A	$ -	$ -	$ 18.30
22	Mini Shielded Ball Bearings, 0.5" Bore	4	$ 9.79	Grainger	1ZEH5	$ 3.14	$ 10.48	$ 52.78
23	Futek TFF400 160 in-oz Reaction Torque Sensor	1	$950.00	Futek	FSH02592	$ -	$202.41	$ 1,152.41
24	Futek 15' Long 4 Pin Lemo Cable	1	$ 100.00	Futek	FSH01790	$ -	$ -	$ 100.00
25	Grizzly 5" 3-Jaw Wood Chuck, 5/8" Unthreaded	1	$ 54.95	Grainger	H8036	$ -	$ 11.95	$ 66.90
26	Ridgid Heavy-Duty 6.5-Amp VSR Drywall Screwdriver	1	$ 21.63	Home Depot	-	$ -	$ -	$ 21.63
27	Misc. Hardware	1	$ 66.06	Home Depot	-	$ -	$ -	$ 66.06
28	Round Rubber Bumper, 1 1/4" Diam., 1 1/4" High w/Washer	1	$ 10.14	McMaster-Carr	9540K808	$ 0.76	$ 5.46	$ 16.36
29	MAX31855 Thermocouple Amplifier	1	$ 29.90	Adafruit	269	$ -	$ 10.71	$ 40.61
30	Aluminum Helical Flexible Shaft Coupling, 12MM x 1/2" Diam.	1	$ 65.99	McMaster-Carr	2464K6	$ -	$ 10.41	$ 76.40
							Total	**$ 2,760.65**

Appendix D

Gantt Chart



Today
Start
Mon 9/23/13
Finish
Fri 6/13/14
October
November
December
January
February
March
April
May
June
Fall Quarter
Mon 9/23/13 - Fri 12/13/13
Winter Quarter
Mon 1/6/14 - Fri 3/21/14
Spring Quarter
Tue 4/1/14 - Fri 6/13/14
Ch
Tu
Team
Mon 9/30/13
House
Tue
Conceptual Design Report Work
Thu 11/7/13 - Thu 12/5/13
Solidify Final Design
Mon 1/6/14 - Mon 2/17/14
Part Procurement Phase - Part 1
Tue 2/18/14 - Fri 3/21/14
Determine Procedures
Tue 4/1/14 - Wed 4/30/14
Testing Phase
Fri 5/9/14 - Fri 6/6/14
W
T
Identify Customer and
Mon 9/30/13 - Thu 10/24/13
Perform Preliminary
Thu 11/14/13 - Thu 12/5/13
Finalize Drawings and Solid Model
Mon 1/6/14 - Mon 2/3/14
Assembly
Sat 4/19/14 -
Integrati
Wed
Final Report Work
Fri 5/9/14 - Thu 6/12/14
F
T
Perform Background
Mon 9/30/13 - Tue 10/22/13
Create Drawings and
Thu 11/14/13 - Thu 12/5/13
Finalize Analysis
Mon 1/6/14 - Mon 2/3/14
Ge
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Select Final Concept
Mon 10/21/13 - Fri 12/13/13
Organize Budget
Mon 1/6/14 - Mon 2/3/14
Project Proposal
Thu 10/10/13 -
Downsel
Thu
Material
Fri 11/22/13 - Fri
Critical Design Report Work
Mon 1/6/14 - Mon 2/3/14
Idea Generation
Mon 10/21/13 - Thu 11/14/13
Team Introduction to Sponsor
Tue 10/1/13
Visit Parker Aerospace Site
Fri 10/11/13
Submit Project Proposal to Sponsor for Approval
Thu 10/24/13
Pre-Concept Review
Fri 11/22/13
Submit Conceptual Design Report for Approval
Fri 12/6/13
Conceptual Design Review
Fri 12/6/13
Submit Tier 1 Ordering List to Parker
Mon 2/17/14
Critical Design Review
Mon 2/17/14
Manufacturing and Test Review
Thu 3/6/14
Senior Design Expo
Sat 5/31/14
Machine Delivery
Fri 6/13/14
Final Report Submission
Fri 6/13/14

WBS	Task Name	% Complete	Hours Spent	Start	Finish
1	**Fall Quarter**	**100%**	**210**	**Mon 9/23/13**	**Fri 12/13/13**
1.1	**Choose Project**	**100%**	**8**	**Tue 9/24/13**	**Fri 9/27/13**
1.1.1	Watch Project Presentations	100%	6	Tue 9/24/13	Thu 9/26/13
1.1.2	Fill Out Project Preference Form	100%	2	Thu 9/26/13	Fri 9/27/13
1.2	**Team Organization**	**100%**	**3**	**Mon 9/30/13**	**Fri 10/11/13**
1.2.1	Team Introduction to Sponsor	100%	1	Tue 10/1/13	Tue 10/1/13
1.2.2	Generate Team Contract	100%	2	Tue 10/8/13	Fri 10/11/13
1.3	**Identify Customer and Engineering Requirements**	**100%**	**61**	**Mon 9/30/13**	**Thu 10/24/13**
1.3.1	Perform Background Research	100%	20	Mon 9/30/13	Tue 10/22/13
1.3.2	Visit Parker Aerospace Site	100%	10	Fri 10/11/13	Fri 10/11/13
1.3.3	House of Quality	100%	10	Tue 10/15/13	Tue 10/22/13
1.3.4	Project Proposal Work	100%	20	Thu 10/10/13	Thu 10/24/13
1.3.5	Submit Project Proposal to Sponsor for Approval	100%	1	Thu 10/24/13	Thu 10/24/13
1.4	**Select Final Concept**	**100%**	**138**	**Mon 10/21/13**	**Fri 12/13/13**
1.4.1	Idea Generation	100%	40	Mon 10/21/13	Thu 11/14/13
1.4.2	Perform Preliminary Analysis	100%	20	Thu 11/14/13	Thu 12/5/13
1.4.3	Create Drawings and Models	100%	20	Thu 11/14/13	Thu 12/5/13
1.4.4	Down-selection Process	100%	15	Thu 11/14/13	Thu 11/21/13
1.4.5	Pre-Concept Review	100%	1	Fri 11/22/13	Fri 11/22/13
1.4.6	Material Selection	100%	20	Fri 11/22/13	Fri 12/6/13
1.4.7	Conceptual Design Report Work	100%	20	Thu 11/7/13	Thu 12/5/13
1.4.8	Submit Conceptual Design Report for Approval	100%	1	Fri 12/6/13	Fri 12/6/13
1.4.9	Conceptual Design Review	100%	1	Fri 12/6/13	Fri 12/6/13
2	**Winter Quarter**	**100%**	**115**	**Mon 1/6/14**	**Fri 3/21/14**
2.1	**Solidify Final Design**	**100%**	**115**	**Mon 1/6/14**	**Mon 2/17/14**
2.1.1	Finalize Drawings and Solid Model	100%	50	Mon 1/6/14	Mon 2/3/14
2.1.2	Finalize Analysis	100%	30	Mon 1/6/14	Mon 2/3/14
2.1.3	Organize Budget	100%	10	Mon 1/6/14	Mon 2/3/14
2.1.4	Critical Design Report Work	100%	20	Mon 1/6/14	Mon 2/3/14
2.1.5	Submit Critical Design Report for Approval	100%	1	Thu 2/13/14	Thu 2/13/14
2.1.6	Critical Design Review	100%	4	Mon 2/17/14	Mon 2/17/14
2.2	**Part Procurement Phase - Part 1**	**100%**	**25**	**Tue 2/18/14**	**Fri 3/21/14**
2.2.1	Submit Tier 1 Ordering List to Parker	100%	5	Mon 2/17/14	Mon 2/17/14
2.2.2	Manufacturing and Test Review	100%	10	Thu 3/6/14	Thu 3/6/14
2.2.3	Project Update Memo Submission	100%	5	Thu 3/13/14	Thu 3/13/14
2.2.4	Submit Tier 2 Ordering List to Parker	100%	5	Thu 3/20/14	Thu 3/20/14
3	**Spring Quarter**	**94%**	**248**	**Tue 4/1/14**	**Fri 6/13/14**
3.1	**Part Procurement Phase - Part 2**	**100%**	**15**	**Thu 4/3/14**	**Fri 4/4/14**
3.1.1	Submit Tier 3 Ordering List to Parker	100%	5	Thu 4/3/14	Thu 4/3/14
3.1.2	Purchase All Small Hardware	100%	10	Fri 4/4/14	Fri 4/4/14
3.2	Fabricate Parts	92%	75	Tue 4/1/14	Fri 4/18/14
3.3	Assembly Phase	100%	30	Sat 4/19/14	Tue 4/29/14
3.4	Integration Phase	90%	40	Wed 4/30/14	Thu 5/8/14
3.5	Determine Procedures	100%	10	Tue 4/1/14	Wed 4/30/14
3.6	Testing Phase	80%	40	Fri 5/9/14	Fri 6/6/14
3.7	Final Report Work	100%	30	Fri 5/9/14	Thu 6/12/14
3.8	Senior Design Expo	100%	3	Sat 5/31/14	Sat 5/31/14
3.9	Final Report Submission	100%	2	Fri 6/13/14	Fri 6/13/14
3.10	Machine Delivery	100%	3	Fri 6/13/14	Fri 6/13/14

Appendix E

Detailed Supporting Analysis

Calculations for Verification of Motor in Conceptual Design

Objective: We need to determine if the motor selected for the conceptual design will be able to achieve the rotational speeds and torque values that we expect to see. Parker Hannifin has told our team to expect torque values of 1 in-lb at 15,000 RPM and 20 in-lb at 200 RPM. We have found a motor on the market that we believe will achieve these values.

Inputs: Baldor Brushed AC Motor
0.25 Horsepower
3,450 RPM Maximum Speed

Assumptions: Absolute maximum horsepower of 0.25 HP and speed of 3,450 RPM

Analysis:

$$Maximum\ Torque\ Output, T = \frac{63{,}025\ (Horsepower)}{Rotational\ Speed\ in\ RPM} = \frac{63{,}025\ (0.25\ HP)}{3{,}450\ RPM} = 4.57\ in - lb$$

Using basic gear and torque ratio laws, this motor will output a torque value of 1.05 in-lb at 15,000 RPM and 22.84 in-lb at 630 RPM. This is calculated using the gear ratios of 1:4.31 and 5:1, respectively.

Conclusion: We will need to step the motor speed down from 630 RPM to 200 RPM using a speed controller. The rotational speed and torque values are higher than the required values, which will ensure that we can overcome the friction losses from the support bearings and shafts. The ratios of 1:4.31 and 5:1 are reasonable enough to use a V-belt. We could also choose to use gears.

Calculation of Gear Tooth Frequency for Sensor Selection

Objective: In order to select an appropriate Hall Effect gear tooth sensor for the system, we need to determine the frequency at which the gear teeth will pass the sensor for the maximum speed condition of 15,000 RPM.

Inputs: Maximum final gear speed of 15,000 RPM

Assumptions: Constant speeds, absolute maximum of 15,000 RPM

Analysis:

$$Maximum\ Speed = 15{,}000\ RPM$$

Converting from speed to frequency:

$$\left(15{,}000\ \frac{revolutions}{minute}\right)\left(20\ \frac{gear\ teeth}{revolution}\right)\left(\frac{minute}{60\ seconds}\right)\left(2\,\frac{pulses}{gear\ tooth}\right) = 10{,}000\ Hertz$$

Conclusion: Because each gear tooth will create two pulses in the sensor output signal, the maximum frequency (switching speed) that the sensor will need to handle is 10,000 Hertz. An appropriate sensor can be selected by reviewing the specifications of various Hall Effect gear tooth sensors.

Calculation of Basic Power Needs for Electronics

Objective: We need to verify that the input and output signals of the DAQ from the sensors are not exceeding the power limitations of the DAQ.

Inputs:
Torque Sensor Max Possible Input: 10 V at 2 mA
Torque Sensor Max Output: 10 V at 2 mA
Gear Tooth Sensor Max Possibly Input: 10 V at 2 mA
Gear Tooth Sensor Max Output: 0.6 V at 20 mA
Thermocouple Max Output: 3 V
FPGA Max Power Rating: 10 mW
Analog Input/Output Max Power Rating: 20 mW

Assumptions: Negligible resistance

Analysis:

$$P_{in-TS} = 10\,Volts * 2\,mA = 20\,mW$$
$$P_{out-TS} = 10\,Volts * 2\,mA = 20mW$$
$$P_{in-GTS} = 10\,Volts * 2\,mA = 20\,mW$$
$$P_{out-GTS} = 0.6\,Volts * 20\,mA = 12\,mW$$
$$P_{out-T} = 3\,Volts * 0\,A = 0\,W$$

Conclusion: Since the torque sensor and thermocouple signals will provide analog inputs to the DAQ and be powered by the analog output pins, it can be seen that their input and output maximum power values are below the acceptable limit. The gear tooth sensor will be analog powered and the output will go into the FPGA pins in the DAQ. The input power is acceptable, but the output power is not. However, the maximum power rating for the gear tooth sensor is based on a 25 Volt input, which we will not be using. Therefore, with an input voltage that the DAQ can provide, the power output from the sensor should be acceptable.

<u>Calculation to Verify Ridgid Motor Meets Performance Requirements</u>

Motor Selection:

The rotational speed range required by Parker is 0-15000 RPM, meaning that we would need to find a motor that spins at a high RPM. Also, this motor should have a speed controller. The best motor that we could get for an unbeatable price of \$20 is a drill motor that has the following specifications:

$$V_m = 15V \ , \ I = 6.5Amp$$

$$\Rightarrow P_m = V_m . I = 115V \times 6.5Amp = 747.5W \simeq 1hp$$

$$\boxed{P_m = 1hP} \quad \text{at} \quad w_{max} = 4000 \text{ RPm} \qquad \left(0 \leqslant w < 4000 \text{RPM}\right)$$

The calculated torque is:

$$T_{max} = \frac{hP \times 63025}{w} \qquad \left(\text{where hp is the horse power} = 1hp \text{ and } w = 4000 \text{ RPM}\right)$$

$$\boxed{T_{max} = 15.75 \text{ Lb.in}}$$

We want a max rotational speed of 15000 RPM, also a minimum torque of 20Lb.in at low rotational speeds.

⟹ We need an easy to make gearbox, thus a swapable Vbelt-pulleys system for a first stage reduction with two spur gears (Second Stage) were selected to do this task.

High RPM Power Transmission: $w_{max} = 15000$ RPM

First stage: 6.50" : 2.35"

Second stage: 1.50 : 1

⟹ total of 4.148 gear ratio

Low RPM Power Transmission $T_{min} = 20Lb.in$

First stage: 2.35" : 6.50"

Second stage: 1.50 : 1

⟹ total of .54 gear ratio

Calculation to Verify Ridgid Drill Motor Meets Performance Requirements (Continued)

However, the Pulleys had an upper limitation since $V_{max} = 6500$ ft/min

Using this formula $n_{max} = \frac{12 V_{max}}{\pi d}$, we were able to approximate the maximum rotational speed that the small and large pulley can withstand.

⟹ Small pulley ω_{max} : **10557 RPM**

Large pulley ω_{max} : **3816.8 RPM**

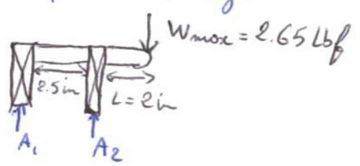
Remarks:

We provided a spreadsheet that has the test torques and rotational speeds range (See attachment)

Calculation for Support Ball Bearing Selection

Ball Bearing Selection for the output shaft

We made sure that the selected bearings would be suitable for our design. Thus, we calculated the bearing rating C_{10} and picked bearings that have that adequate rating.



Performing bending moment analysis, we could find the reactions on the bearings.

$$\circlearrowleft^{+} M_A = 0 \Leftrightarrow -W(l+L) + A_1 l = 0$$

$$\Rightarrow \boxed{A_1 = 4.77\ Lbf}$$

$$\oplus \uparrow \Sigma F_y = 0 \quad \Rightarrow \boxed{A_2 = -2.12\ Lbf}$$

We need to pick the bearing that has the highest C_{10} $\Rightarrow$ Bearing A_1, because it has higher loads.

$$C_{10} = F_D \left(\frac{\mathcal{L}_D n_D 60}{10^6}\right)^{1/3} = 4.77\ Lbf \left(\frac{15000\ RPM \times 5000\ h \times 60}{10^6}\right)$$

$$\boxed{C_{10} = 78.75\ Lbf}$$

Assume

- Ball Bearing
- Test would run at a duration of 5000 h
- Rated to 10^6 cycle at max conditions

Remark: The picked bearing is a bearing that is under the worst condition, thus the same bearing could be applied to the whole system

Calculation to Determine Output Shaft Deflection Behavior

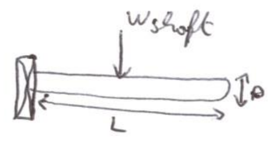
Deflection Analysis:

we made sure that the maximum loaded shaft deflection does not exceed the allowable one specified by Shigley.

our shaft specifications are:

$D = 12.7\ mm$; $L = 0.0508\ m$; $I = 1.01 \times 10^{-3}\ m^4$;

$E_{steel} = 200 \times 10^9\ N/m^2$; $\rho_{steel} = 8050\ Kg/m^3$



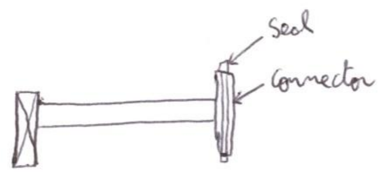
$$W_{shaft} = mg = \rho_{shaft} V g = \frac{\pi D^2}{4} \times L \times \rho_{shaft} \times g = \frac{\pi}{4}(12.7 \times 10^{-3})^2 m^2 \times 0.0508\,m \times 9.81\,m/s^2 \times 8050\,\frac{Kg}{m^3}$$

$$\Rightarrow \boxed{W_{shaft} = 0.5081\ N}$$

Our Design is split into two designs; the seals and the bearings that have different maximum loads.

Seals Design: Max Seal connector Deflection

The deflection caused by the connector and Seals can be modeled by this equation:

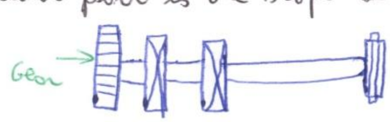


δ_s: Max connector + Seal weight $\simeq 4.5\ N$

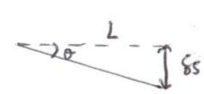
$$\Rightarrow \delta_s = \frac{W_s L^3}{3EI} = \frac{4.5\,N \times (0.0508\,m)^3}{3 \times 200 \times 10^9\,N/m^2 \times 1.01 \times 10^{-3}\,m^2}$$

$$\Rightarrow \boxed{\delta_s = 9.76 \times 10^{-7}\ m}$$

The most important part is the slope because a gear would be mounted to this shaft



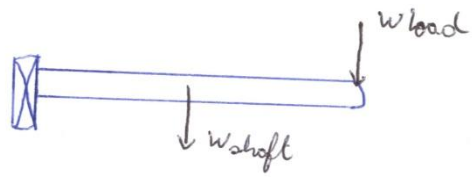
The slope is: $\tan\theta_{max} = \frac{\delta_s}{L} \Rightarrow \theta_{max} = 1.92 \times 10^{-5\,\circ}$



Calculation of Critical Speed for Cantilevered Output Shaft

Critical speed for a cantilever case:

We want to let Parker know about the critical speeds range of our output shaft that is cantilevered. Thus, we had to find the shaft deflections causing "the first mode of vibration" (Bending). The other modes would be found using the shake table (Testing plan)



Shaft Deflection:

$$\delta_{shaft} = \frac{WL^3}{8EI} = 0.508\ N$$

Load Deflection:

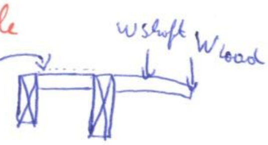
$$\delta_{load} = \frac{WL^3}{3EI}$$

(δ_{load} is variable because our loads are variable)

$\Rightarrow$ The total deflection $\delta_{total} = \delta_{shaft} + \delta_{load}$

Assume:

- Cones and attachment modeled as point loads
- Center of mass of shaft is in the middle
- Ignoring the following deflection



Using this first mode critical shaft speed formula, we could generate the range of critical speeds under different loads:

$$N_c = \frac{30}{\pi}\sqrt{\frac{g}{\delta_{total}}}$$

$$\Rightarrow \theta_{max} = 3.35 \times 10^{-7} \text{ rad} < 0.003 \quad \text{Allowable by Shigley}$$

Bearing Design: Max Bearings + Cones deflection:

Since Bearings are going to be mounted to two cones made of plastic, and our $D_{max} = 2.5''$, we would round it to $D_{max} = 3''$ to consider the cones weight $W_B = 5.56\ N$ $(D_{max} = 0.077)$

$$\delta_b = \frac{W_b L^3}{3EI} \Rightarrow \boxed{\delta_b = 1.203 \times 10^{-6}\ m}$$

$$\Rightarrow \theta_{max} = \tan^{-1}\left(\frac{\delta_b}{L}\right) = 4.13 \times 10^{-8} \text{ rad} < 0.003 \text{ rad allowed by Shigley}$$

Remark:

We need to make sure that the shaft + Bearings + Gears are well aligned to avoid further deflections.

<u>Calculation Spreadsheet to Determine Output Speeds and Torques for Drive Train Ratios</u>

Table 8. Calculated Output Speeds and Torques at Various Drive Train Locations

Input Speed (RPM)	Input Torque (in-lb)	High Speed Gear Ratio		Low Speed Gear Ratio		High Speed Final Drive		Low Speed Final Drive	
		Large Pulley Speed (RPM)	Small Pulley Speed (RPM)	Small Pulley Speed (RPM)	Large Pulley Speed (RPM)	Spur Gears Ratio 1:1.50 (RPM)	Spur Gear 1:1.50 Torque (in-lb)	Spur Gears Ratio 1.50:1 (RPM)	Spur Gear 1.50:1 Torque (in-lb)
100	630.25	100	276.59	100	36.15	414.89	151.91	149.99	2614.81
500	126.05	500	1382.95	500	180.77	2074.43	30.38	749.97	522.96
1000	63.03	1000	2765.90	1000	361.53	4148.85	15.19	1499.93	261.48
2000	31.51	2000	5531.80	2000	723.06	8297.70	7.60	2999.87	130.74
3000	21.01	3000	8297.70	3000	1084.59	12446.55	5.06	4499.80	87.16
3500	18.01	3500	9680.65	3500	1265.36	14520.98	4.34	5249.77	74.71
3600	17.51	3600	9957.24	3600	1301.51	14935.86	4.22	5399.76	72.63
3700	17.03	3700	10233.83	3700	1337.66	15350.75	4.11	5549.75	70.67
3800	16.59	3800	10510.42	3800	1373.81	15765.63	4.00	5699.75	68.81
3900	16.16	**3900**	**10787.01**	3900	1409.97	16180.52	3.90	5849.74	67.05
4000	15.76	**4000**	**11063.60**	4000	1446.12	16595.40	3.80	5999.73	65.37

The cells highlighted red correspond the points where the small pulley speed is exceeding its maximum value. The input speeds that produce this result will be avoided.

Calculation Spreadsheet to Determine Output Shaft Critical Speeds & Deflections

Table 9. Calculated Seal and Bearing Output Shaft Critical Speeds and Deflections

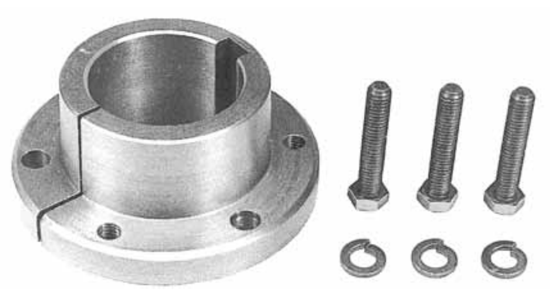
Seal Apparatus		Bearing Apparatus								
Outer Diameter (in)	Weight (lb)	Outer Diameter (in)	Weight (lb)	Shaft Deflection (m)	Bearing Deflection (m)	Seal Deflection (m)	Total Bearing Deflection (m)	Total Seal Deflection (m)	Critical Speed of Bearing Apparatus (RPM)	Critical Speed of Seal Apparatus (RPM)
0.354	0.044	0.2362	0.2900	4.12E-08	2.27E-06	4.23E-07	2.83E-06	4.65E-07	17774	43881
0.375	0.044	0.3150	0.3400	4.12E-08	3.03E-06	4.23E-07	3.31E-06	4.65E-07	16433	43881
0.393	0.068	0.3543	0.3700	4.12E-08	3.41E-06	6.54E-07	3.60E-06	6.96E-07	15760	35863
0.500	0.198	0.5000	0.4100	4.12E-08	4.81E-06	1.91E-06	3.99E-06	1.95E-06	14980	21438
0.563	0.102	0.6250	0.4400	4.12E-08	6.01E-06	9.81E-07	4.28E-06	1.02E-06	14466	29576
0.591	0.102	0.7500	0.4700	4.12E-08	7.22E-06	9.81E-07	4.56E-06	1.02E-06	14001	29576
1.125	0.350	1.0000	0.5300	4.12E-08	9.62E-06	3.37E-06	5.14E-06	3.41E-06	13191	16199
1.312	0.350	1.1250	1.1200	4.12E-08	1.08E-05	3.37E-06	1.08E-05	3.41E-06	9093	16199
1.313	0.429	1.2500	1.4500	4.12E-08	1.20E-05	4.13E-06	1.40E-05	4.17E-06	7995	14648
1.811	0.825	1.3780	1.6300	4.12E-08	1.33E-05	7.94E-06	1.57E-05	7.98E-06	7542	10588
2.250	0.825	1.5000	1.7400	4.12E-08	1.44E-05	7.94E-06	1.68E-05	7.98E-06	7301	10588
2.557	1.168	1.7500	1.8900	4.12E-08	1.68E-05	1.12E-05	1.82E-05	1.13E-05	7006	8905
2.597	1.168	2.0000	2.1500	4.12E-08	1.92E-05	1.12E-05	2.07E-05	1.13E-05	6569	8905
3.248	1.500	2.1250	2.2600	4.12E-08	2.04E-05	1.44E-05	2.18E-05	1.45E-05	6408	7861
		2.5000	2.3200	4.12E-08	2.41E-05		2.24E-05		6324	
		2.7500	2.4100	4.12E-08	2.65E-05		2.32E-05		6205	
		3.0000	2.6500	4.12E-08	2.89E-05		2.55E-05		5918	

The highlighted cells correspond to the speeds that our system is capable of achieving. Since these critical speeds are within the possible output speed range, these speeds will be avoided.

Appendix F

Vendor Supplied Component Specifications and Data Sheets

A



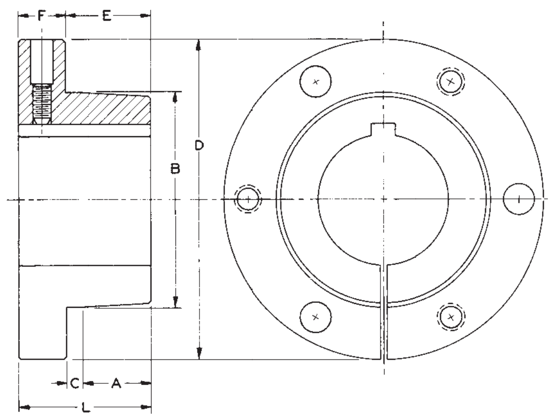
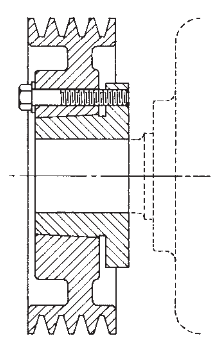


Table No. 1

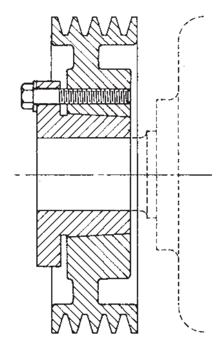
Bushing Specifications

Bushing	Bore Range	Dimensions D	L	A	B	C	E	F	Cap screws No.	Size	Bolt Circle	Torque Capacity In-Lbs	Wrench Torque Ft.-Lbs	Average Weight Lbs.
JA	1/2" - 1 1/4"	2"	.89"	.37"	1.375"	3/16"	.56"	.33"	3	10 - 24 X 1	1 21/32"	1000	6	.8
SH	1/2 - 1 5/8	2 11/16	1.20	.53	1.871	7/32	.75	.45	3	1/4 - 20 X 1 3/8	2 1/4	3500	9	1.0
SDS	1/2 - 2	3 1/8	1.33	.66	2.1875	7/32	.88	.45	3	1/4-20 X 1 3/8	2 11/16	5000	9	1.2
SD	1/2 - 2	3 1/8	1.83	1.16	2.1875	7/32	1.38	.45	3	1/4-20 x 1 7/8	2 11/16	5000	9	1.5
SK	1/2 - 2 5/8	3 7/8	1.75	.97	2.8125	7/32	1.19	.56	3	5/16 - 18 X 2	3 5/16	7000	15	2.0
SF	1/2 - 2 15/16	4 5/8	1.995	1.22	3.125	7/32	1.44	.56	3	3/8 - 16 X 2	3 7/8	11000	30	3.5
E	7/8 - 3 1/2	6	2.64	1.60	3.834	9/32	1.88	.76	3	1/2 - 13 X 2 3/4	5	20000	60	9.0
F	1 - 4	6 5/8	3.61	2.41	4.438	11/32	2.75	.88	3	9/16 -12 x 3 5/8	5 5/8	30000	75	14
J	1 1/2 - 4 1/2	7 1/4	4.48	3.19	5.148	5/16	3.50	1.00	3	5/8-11 X 4 1/2	6 1/4	45000	135	22
M	2 - 5 1/2	9	6.73	5.16	6.494	11/32	5.50	1.25	4	3/4-10 X 6 3/4	7 7/8	85000	225	51
N	2 7/16 - 5 7/8	10	8.11	6.07	6.992	9/16	6.63	1.50	4	7/8-9 X 8	8 1/2	150000	300	66
P	2 15/16 - 7	11 3/4	9.38	7.00	8.242	5/8	7.63	1.77	4	1 - 8 x 9 1/2	10	250000	450	122

NOTE — All Bushings shown except JA have setscrew over keyway.



MOUNT EITHER WAY



BUSHING FLANGE TOWARD MACHINE OR MOTOR

1. Align tapped holes in bushing flange with drilled holes in sheave hub.
2. Insert capscrews through drilled holes in sheave hub and thread loosely into tapped holes in bushing flange.
3. Position assembly on shaft and tighten capscrews progressively and uniformly.

TO REMOVE

1. Remove capscrews and thread into tapped holes in sheave hub. Tighten progressively until bushing is free from sheave taper.
2. Remove assembly from shaft.

BUSHING FLANGE AWAY FROM MACHINE OR MOTOR

1. Align drilled holes in bushing flange with tapped holes in sheave hub.
2. Insert capscrews through drilled holes in bushing flange and thread loosely into tapped holes in sheave hub.
3. Position assembly on shaft and tighten capscrews progressively and uniformly.

TO REMOVE

1. Remove capscrews and thread into tapped holes in bushing flange. Tighten progressively until bushing is free from sheave taper.
2. Remove assembly from shaft.

CAPSCREWS ARE ALWAYS ACCESSIBLE FROM THE OUTSIDE

A

Table No. 1

Stock Inch Bore Bushings

Stock Bores	Keyseat	JA	SH	SDS	SD	SK	SF	E	F	J
1/2	1/8 X 1/16	X	X	X	X	X	X	—	—	—
9/16	1/8 X 1/16	X	X	X	X	X	X	—	—	—
5/8	3/16 X 3/32	X	X	X	X	X	X	—	—	—
1/16	3/16 X 3/32	X	X	X	X	X	X	—	—	—
3/4	3/16 X 3/32	X	X	X	X	X	X	—	—	—
13/16	3/16 X 3/32	X	X	X	X	X	X	—	—	—
7/8	3/16 X 3/32	X	X	X	X	X	X	X	—	—
15/16	1/4 X 1/8	X	X	X	X	X	X	X	—	—
1	1/4 X 1/8	X	X	X	X	X	X	X	X	—
1 1/16	1/4 X 1/8	S	X	X	X	X	X	X	X	—
1 1/8	1/4 X 1/8	S	X	X	X	X	X	X	X	—
1 3/16	1/4 X 1/8	S	X	X	X	X	X	X	X	—
1 1/4	1/4 X 1/8	S	X	X	X	X	X	X	X	—
1 5/16	5/16 X 5/32	—	X	X	X	X	X	X	X	—
1 15/16*	3/8 X 3/16	—	X	X	X	X	X	X	X	—
1 3/8	5/16 X 5/32	—	X	X	X	X	X	X	X	—
1 3/8*	3/8 X 3/16	—	X	X	X	X	X	X	X	—
1 7/16	3/8 X 3/16	—	S	X	X	X	X	X	X	—
1 1/2	3/8 X 3/16	—	S	X	X	X	X	X	X	X
1 9/16	3/8 X 3/16	—	S	X	X	X	X	X	X	X
1 5/8	3/8 X 3/16	—	S	X	X	X	X	X	X	X
1 11/16	3/8 X 3/16	—	—	S	S	X	X	X	X	X
		SDS	**SD**	**SK**	**SF**	**E**	**F**	**J**	**M**	**N**
1 3/4	3/8 X 3/16	S	S	X	X	X	X	X	—	—
1 13/16	1/2 X 1/4	S	S	X	X	X	X	X	—	—
1 7/8	1/2 X 1/4	S	S	X	X	X	X	X	—	—
1 15/16	1/2 X 1/4	S	S	X	X	X	X	X	—	—
2	1/2 X 1/4	N	N	X	X	X	X	X	X	—
2 1/16	1/2 X 1/4	—	—	X	X	X	X	X	X	—
2 1/8	1/2 X 1/4	—	—	X	X	X	X	X	X	—
2 3/16	1/2 X 1/4	—	—	S	X	X	X	X	X	—
2 1/4	1/2 X 1/4	—	—	S	X	X	X	X	X	—
2 5/16	5/8 X 5/16	—	—	S	S	X	X	X	X	—
2 3/8	5/8 X 5/16	—	—	S	S	X	X	X	X	—
2 7/16	5/8 X 5/16	—	—	S	S	X	X	X	X	X
2 1/2	5/8 X 5/16	—	—	S	S	X	X	X	X	X
2 9/16	5/8 X 5/16	—	—	N	S	X	X	X	X	X
2 5/8	5/8 X 5/16	—	—	N	S	X	X	X	X	X
		SF	**E**	**F**	**J**	**M**	**N**	**P**		
2 11/16	5/8 X 5/16	S	X	X	X	X	X	—		
2 3/4	5/8 X 5/16	S	X	X	X	X	X	—		
2 13/16	3/4 X 3/8	S	X	X	X	X	X	—		
2 7/8	3/4 X 3/8	S	X	X	X	X	X	—		
2 15/16	3/4 X 3/8	S	S	X	X	X	X	X		
3	3/4 X 3/8	—	S	X	X	X	X	X		
3 1/16	3/4 X 3/8	—	S	X	X	X	X	X		
3 1/8	3/4 X 3/8	—	S	X	X	X	X	X		
3 3/16	3/4 X 3/8	—	S	X	X	X	X	X		
3 1/4	3/4 X 3/8	—	S	X	X	X	X	X		
3 5/16	7/8 X 7/16	—	S	S	X	X	X	X		
3 3/8	7/8 X 7/16	—	S	S	X	X	X	X		
3 7/16	7/8 X 7/16	—	S	S	X	X	X	X		
3 1/2	7/8 X 7/16	—	S	S	X	X	X	X		
3 9/16	7/8 X 7/16	—	—	S	X	X	X	X		
3 5/8	7/8 X 7/16	—	—	S	X	X	X	X		
3 11/16	7/8 X 7/16	—	—	S	X	X	X	X		
3 3/4	7/8 X 7/16	—	—	S	X	X	X	X		
3 13/16	7/8 X 7/16	—	—	S	X	X	X	X		
3 7/8	1 X 1/2	—	—	S	S	X	X	X		
3 15/16	1 X 1/2	—	—	S	S	X	X	X		
4	1 X 1/2	—	—	N	S	X	X	X		
		J	**M**	**N**	**P**					
4 1/16	1 X 1/2	S	X	X	X					
4 1/8	1 X 1/2	S	X	X	X					
4 3/16	1 X 1/2	S	X	X	X					
4 1/4	1 X 1/2	S	X	X	X					
4 5/16	1 X 1/2	S	X	X	X					
4 3/8	1 X 1/2	S	X	X	X					
4 7/16	1 X 1/2	S	X	X	X					
4 1/2	1 X 1/2	S	X	X	X					
4 9/16	1 X 1/2	S	X	X	X					
4 5/8	1 1/4 X 5/8	—	X	X	X					
4 11/16	1 1/4 X 5/8	—	X	X	X					
4 3/4	1 1/4 X 5/8	—	S	X	X					
4 13/16	1 1/4 X 5/8	—	S	X	X					
4 7/8	1 1/4 X 5/8	—	S	X	X					
4 15/16	1 1/4 X 5/8	—	S	X	X					
5	1 1/4 X 5/8	—	S	X	X					
5 1/16	1 1/4 X 5/8	—	S	X	X					
5 1/8	1 1/4 X 5/8	—	S	S	X					
5 3/16	1 1/4 X 5/8	—	S	S	X					
5 1/4	1 1/4 X 5/8	—	S	S	X					
5 5/16	1 1/4 X 5/8	—	S	S	X					
5 3/8	1 1/4 X 5/8	—	S	S	X					
5 7/16	1 1/4 X 5/8	—	S	S	X					
5 1/2	1 1/4 X 5/8	—	S	S	X					

Stock Inch Bore Bushings

Table No. 1 (Cont.)

Stock Bores	Keyseat	N	P
5 9/16	1 1/2 X 3/4	S	X
5 5/8	1 1/2 X 3/4	S	X
5 11/16	1 1/2 X 3/4	S	X
5 3/4	1 1/2 X 3/4	S	X
5 13/16	1 1/2 X 3/4	S	X
5 7/8	1 1/2 X 3/4	S	S
5 15/16	1 1/2 X 3/4	—	S
6	1 1/2 X 3/4	—	S
6 1/16	1 1/2 X 3/4	—	S
6 1/8	1 1/2 X 3/4	—	S
6 3/16	1 1/2 X 3/4	—	S
6 1/4	1 1/2 X 3/4	—	S
6 5/16	1 1/2 X 3/4	—	S
6 3/8	1 1/2 X 3/4	—	S
6 7/16	1 1/2 X 3/4	—	S
6 1/2	1 1/2 X 3/4	—	S
6 9/16	1 3/4 X 3/4	—	S
6 5/8	1 3/4 X 3/4	—	S
6 11/16	1 3/4 X 3/4	—	S
6 3/4	1 3/4 X 3/4	—	S
6 13/16	1 3/4 X 3/4	—	S
6 7/8	1 3/4 X 3/4	—	S
6 15/16	1 3/4 X 3/4	—	S
7	1 3/4 X 3/4	—	S

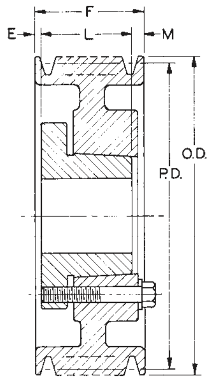
Table No. 2

Stock Millimeter Bore Bushings

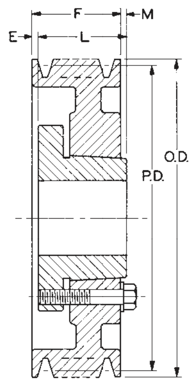
Stock Bores	Keyseat (Millimeters)	SH	SDS	SD	SK	SF	E	F	J
24	8 X 3.5	X	X	X	X	—	—	—	—
25	8 X 3.5	X	X	X	X	—	—	—	—
28	8 X 3.5	X	X	X	X	X	—	—	—
30	8 X 3.5	X	X	X	X	X	—	—	—
32	10 X 4	X	X	X	X	X	—	—	—
35	10 X 4	X	X	X	X	X	—	—	—
38	10 X 4	—	X	X	X	X	X	—	—
40	12 X 4	—	X	X	X	X	X	—	—
42	12 X 4	—	X	X	X	X	X	—	—
45	14 X 4.5	—	—	—	—	X	X	X	—
48	14 X 4.5	—	S	—	X	X	X	X	—
50	14 X 4.5	—	—	—	X	X	X	X	X
55	16 X 5	—	—	—	X	X	X	X	X
60	18 X 5.5	—	—	—	—	X	X	X	X
65	18 X 5.5	—	—	—	—	—	X	X	X
70	20 X 6	—	—	—	—	—	X	X	X
75	20 X 6	—	—	—	—	—	X	X	X
80	22 X 7	—	—	—	—	—	—	X	X
85	22 X 7	—	—	—	—	—	—	X	X
90	25 X 7	—	—	—	—	—	—	X	X
95	25 X 7	—	—	—	—	—	—	—	X
100	28 X 8	—	—	—	—	—	—	—	X

*Bushings with 5/16 X 5/32" Keyway will be shipped unless the 3/8 x 3/16 Keyway is specified on the order.

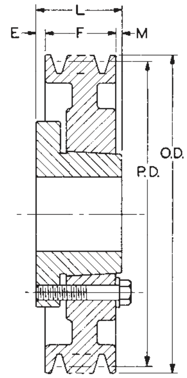
X = Stock Bore with Standard Keyway
N = Stock Bore with No Keyway
S = Stock Bore with Shallow Keyway, Rectangular Key is furnished to fit standard keyseat.



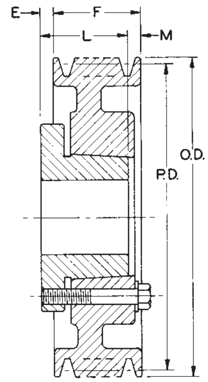
TYPE A



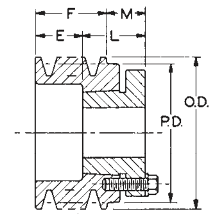
TYPE B



TYPE C



TYPE D



TYPE E

Table No. 1

Specifications

Part No.	Bushing	Bore Range	Type *	P.D. "3V" Belts	O.D.	E	L	M	Wt. Less Bushing
1 Groove, F = 11/16"									
13V220JA	JA	1/2" - 1 1/4"	E-1	2.15"	2.20"	5/8"	1"	15/16"	.4
13V235JA	JA	1/2 - 1 1/4	E-1	2.30	2.35	19/32	1	29/32	.4
13V250JA	JA	1/2 - 1 1/4	E-1	2.45	2.50	19/32	1	29/32	.5
13V265JA	JA	1/2 - 1 1/4	D-1	2.60	2.65	3/8	1	1/16	.6
13V280JA	JA	1/2 - 1 1/4	D-1	2.75	2.80	3/8	1	1/16	.7
13V300JA	JA	1/2 - 1 1/4	D-1	2.95	3.00	3/8	1	1/16	.8
13V315JA	JA	1/2 - 1 1/4	D-1	3.10	3.15	3/8	1	1/16	.9
13V335JA	JA	1/2 - 1 1/4	D-1	3.30	3.35	3/8	1	1/16	1.0
13V365SH	SH	1/2 - 1 5/8	D-1	3.60	3.65	21/32	1 5/16	1/32	1.3
13V412SH	SH	1/2 - 1 5/8	D-1	4.07	4.12	21/32	1 5/16	1/32	1.8
13V450SH	SH	1/2 - 1 5/8	D-1	4.45	4.50	21/32	1 5/16	1/32	2.1
13V475SH	SH	1/2 - 1 5/8	D-1	4.70	4.75	21/32	1 5/16	1/32	2.4
13V500SH	SH	1/2 - 1 5/8	D-1	4.95	5.00	21/32	1 5/16	1/32	2.7
13V530SH	SH	1/2 - 1 5/8	D-1	5.25	5.30	21/32	1 5/16	1/32	2.9
13V560SH	SH	1/2 - 1 5/8	D-2	5.55	5.60	21/32	1 5/16	1/32	3.0
13V600SH	SH	1/2 - 1 5/8	D-2	5.95	6.00	21/32	1 5/16	1/32	3.2
13V650SH	SH	1/2 - 1 5/8	D-2	6.45	6.50	21/32	1 5/16	1/32	4.2
13V690SH	SH	1/2 - 1 5/8	D-3	6.85	6.90	21/32	1 5/16	1/32	4.4
13V800SDS	SDS	1/2 - 2	D-3	7.95	8.00	21/32	1 5/16	1/32	5.8
13V1060SDS	SDS	1/2 - 2	D-3	10.55	10.60	21/32	1 5/16	1/32	7.9
13V1400SK	SK	1/2 - 2 5/8	C-3	13.95	14.00	25/32	1 15/16	15/32	14.8
13V1900SK	SK	1/2 - 2 5/8	C-3	18.95	19.00	25/32	1 15/16	15/32	24.0
2 GROOVES, F = 1 3/32"									
23V220JA	JA	1/2 - 1 1/4	E-1	2.15	2.20	1 1/32	1	15/16	.6
23V235JA	JA	1/2 - 1 1/4	E-1	2.30	2.35	1	1	29/32	.6
23V250JA	JA	1/2 - 1 1/4	E-1	2.45	2.50	1	1	29/32	.7
23V265JA	JA	1/2 - 1 1/4	D-1	2.60	2.65	3/8	1	15/32	.8
23V280JA	JA	1/2 - 1 1/4	D-1	2.75	2.80	3/8	1	15/32	.9
23V300JA	JA	1/2 - 1 1/4	D-1	2.95	3.00	3/8	1	15/32	1.2
23V315JA	JA	1/2 - 1 1/4	D-1	3.10	3.15	3/8	1	15/32	1.2
23V335SH	SH	1/2 - 1 5/8	D-1	3.30	3.35	17/32	1 5/16	5/16	1.3
23V365SH	SH	1/2 - 1 5/8	D-1	3.60	3.65	17/32	1 5/16	5/16	1.5
23V412SH	SH	1/2 - 1 5/8	D-1	4.07	4.12	11/32	1 5/16	1/8	2.2
23V450SH	SH	1/2 - 1 5/8	D-1	4.45	4.50	11/32	1 5/16	1/8	2.7
23V475SH	SH	1/2 - 1 5/8	D-1	4.70	4.75	11/32	1 5/16	1/8	3.1
23V500SH	SH	1/2 - 1 5/8	D-1	4.95	5.00	11/32	1 5/16	1/8	3.4
23V530SH	SH	1/2 - 1 5/8	D-1	5.25	5.30	11/32	1 5/16	1/8	3.9
23V560SH	SH	1/2 - 1 5/8	D-2	5.55	5.60	11/32	1 5/16	1/8	4.0
23V600SH	SH	1/2 - 1 5/8	D-2	5.95	6.00	11/32	1 5/16	1/8	4.4
23V650SDS	SDS	1/2 - 2	D-2	6.45	6.50	11/32	1 5/16	1/8	6.0
23V690SDS	SDS	1/2 - 2	D-2	6.85	6.90	11/32	1 5/16	1/8	7.5
23V800SDS	SDS	1/2 - 2	D-3	7.95	8.00	11/32	1 5/16	1/8	8.5
23V1060SK	SK	1/2 - 2 5/8	C-3	10.55	10.60	17/32	1 15/16	5/16	12.5
23V1400SK	SK	1/2 - 2 5/8	C-3	13.95	14.00	17/32	1 15/16	5/16	19.5
23V1900SK	SK	1/2 - 2 5/8	C-3	18.95	19.00	17/32	1 15/16	5/16	27.0
23V2500SF	SF	1/2 - 2 15/16	C-3	24.95	25.00	19/32	2 1/16	3/8	38.0

***Suffix on Type indicates: 1 = Solid Construction, 2 = Web Construction, 3 = Arm Construction.**

Hardened Spur Gears - 24 Pitch

- **AGMA Q10**
- **20° PRESSURE ANGLE**
- **1/4 & 3/8 FACE**
- **1/2 BORE**

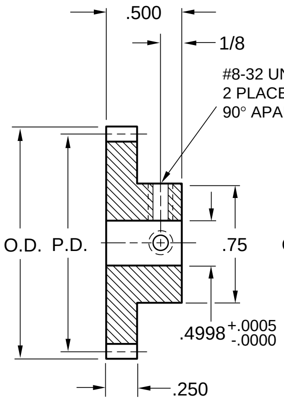


Fig. 1

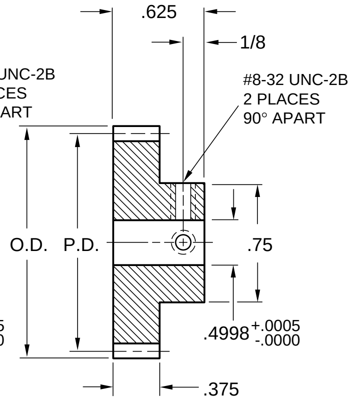
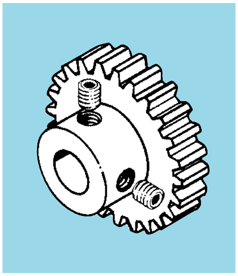


Fig. 2



MATERIAL: Alloy Steel
Surface Hardened to $R_C \geq 55$

Catalog Number		No. of Teeth	P.D.	O.D.
.250 Face Fig. 1	.375 Face Fig. 2			
S10A9Z-024H018	**S10C9Z-024H018**	18	0.7500	0.833
S10A9Z-024H020	**S10C9Z-024H020**	20	0.8333	0.917
S10A9Z-024H021	**S10C9Z-024H021**	21	0.8750	0.958
S10A9Z-024H024	**S10C9Z-024H024**	24	1.0000	1.083
S10A9Z-024H030	**S10C9Z-024H030**	30	1.2500	1.333
S10A9Z-024H036	**S10C9Z-024H036**	36	1.5000	1.583
S10A9Z-024H042	**S10C9Z-024H042**	42	1.7500	1.833
S10A9Z-024H048	**S10C9Z-024H048**	48	2.0000	2.083
S10A9Z-024H054	**S10C9Z-024H054**	54	2.2500	2.333
S10A9Z-024H060	**S10C9Z-024H060**	60	2.5000	2.583
S10A9Z-024H072	**S10C9Z-024H072**	72	3.0000	3.083
S10A9Z-024H084	**S10C9Z-024H084**	84	3.5000	3.583
S10A9Z-024H096	**S10C9Z-024H096**	96	4.0000	4.083

Available on special order: Different number of teeth, bore size or face width.
See technical page **1**-15 for application hints.

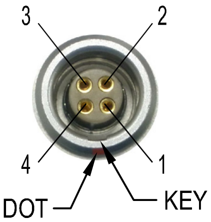
FUTEK MODEL TFF400

FLANGE TO FLANGE REACTION TORQUE SENSOR WITH THRU HOLE CENTER

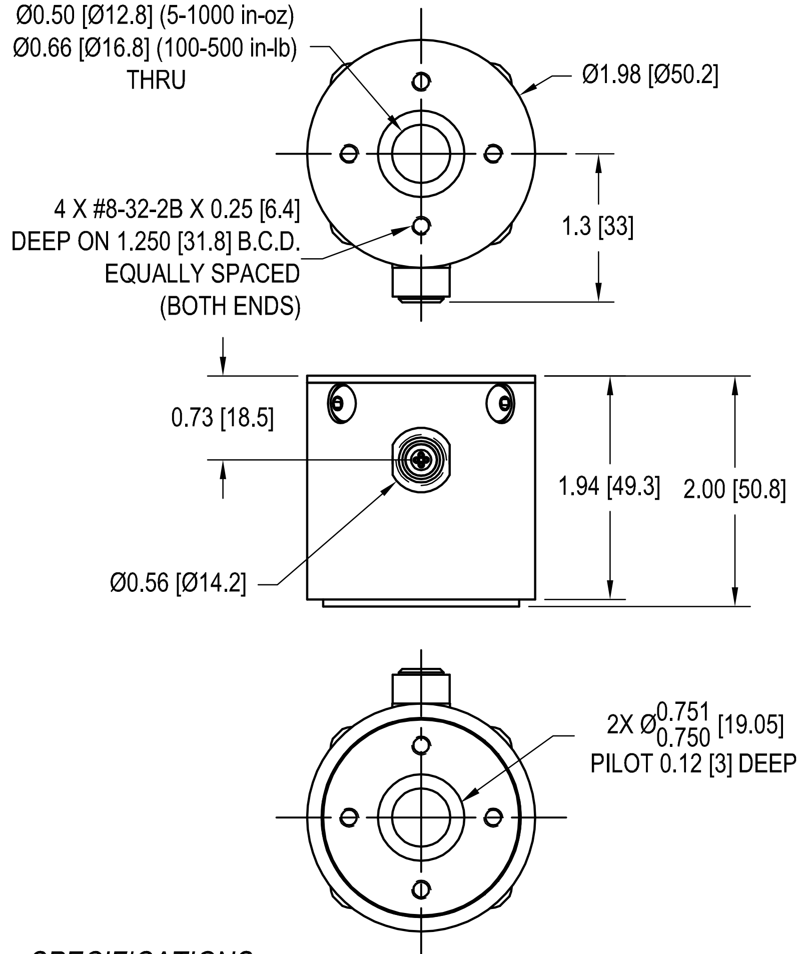
Drawing Number: FI1251-D

INCH [mm] | R.O.= Rated Output

WIRING CODE (CC4)			
+Excitation	-Excitation	+Signal	-Signal
PIN 1	PIN 4	PIN 2	PIN 3



NOTE: UNIT WILL HAVE RED ANODIZED COVER AFTER MANUFACTURE DATE OF 01/15/2010



ITEM#	CAPACITY in-oz	Nm	TORSIONAL STIFFNESS in-oz/rad
FSH02587	5*	0.04	325
FSH02588	10*	0.08	650
FSH02589	20*	0.15	1600
FSH02590	50*	0.37	3500
FSH02592	160*	1.2	11000
FSH02593	400*	3.0	30000
FSH02594	1000	7.5	71000
FSH02595	100 in-lb	12	77000 in-lb/rad
FSH02596	200 in-lb	24	95000 in-lb/rad
FSH02597	500 in-lb	60	199000 in-lb/rad

* WITH OVERLOAD PROTECTION
FOR HIGHER CAPACITIES REFER TO MODELS TFF600-750 AND TDF600-675

SPECIFICATIONS:

RATED OUTPUT	*1 mV/V nom. (5 in-oz)* *2 mV/V nom. (10 in-oz to 500 in-lb)*
SAFE OVERLOAD	*300% (5 to 400 in-oz) 150% (1000 in-oz to 500 in-lb)*
ZERO BALANCE	*±1% of R.O. (10 in-oz to 500 in-lb)* *±2% of R.O. (5 in-oz)*
EXCITATION (VDC OR VAC)	*18 MAX*
BRIDGE RESISTANCE	*350 Ω nom. (5 to 1000 in-oz)* *700 Ω nom. (100 to 500 in-lb)*
NONLINEARITY	*±0.2% of R.O.*
HYSTERESIS	*±0.2% of R.O.*
NONREPEATABILITY	*±0.05% of R.O.*
TEMP. SHIFT ZERO	*±0.002% of R.O./°F [0.0036% of R.O./°C]*
TEMP. SHIFT SPAN	*±0.002% of LOAD/°F [0.0036% of LOAD/°C]*
COMPENSATED TEMP.	*60 to 160°F [15 to 72°C]*
OPERATING TEMP.	*-60 to 200°F [-50 to 93°C]*
WEIGHT	*9 oz [250 g]*
MATERIAL	*ALUMINUM*

CONNECTOR: 4 Pin LEMO Receptacle (EGG.OB.304.CLL)

ACCESSORIES AND RELATED INSTRUMENTS AVAILABLE

CALIBRATION (STD)	*5 pt. CW; 60.4KΩ SHUNT CAL. VALUE (10 to 1000 in-oz)* *100KΩ SHUNT CAL. VALUE (5 in-oz, 100 to 500 in-lb)*
CALIBRATION (OPTIONAL)	*5pt. CCW*
CALIBRATION TEST EXCITATION	*10 VDC*

IRVINE, CA 92618 USA
1-800-23-FUTEK (38835)
INTERNET:
http://www.futek.com

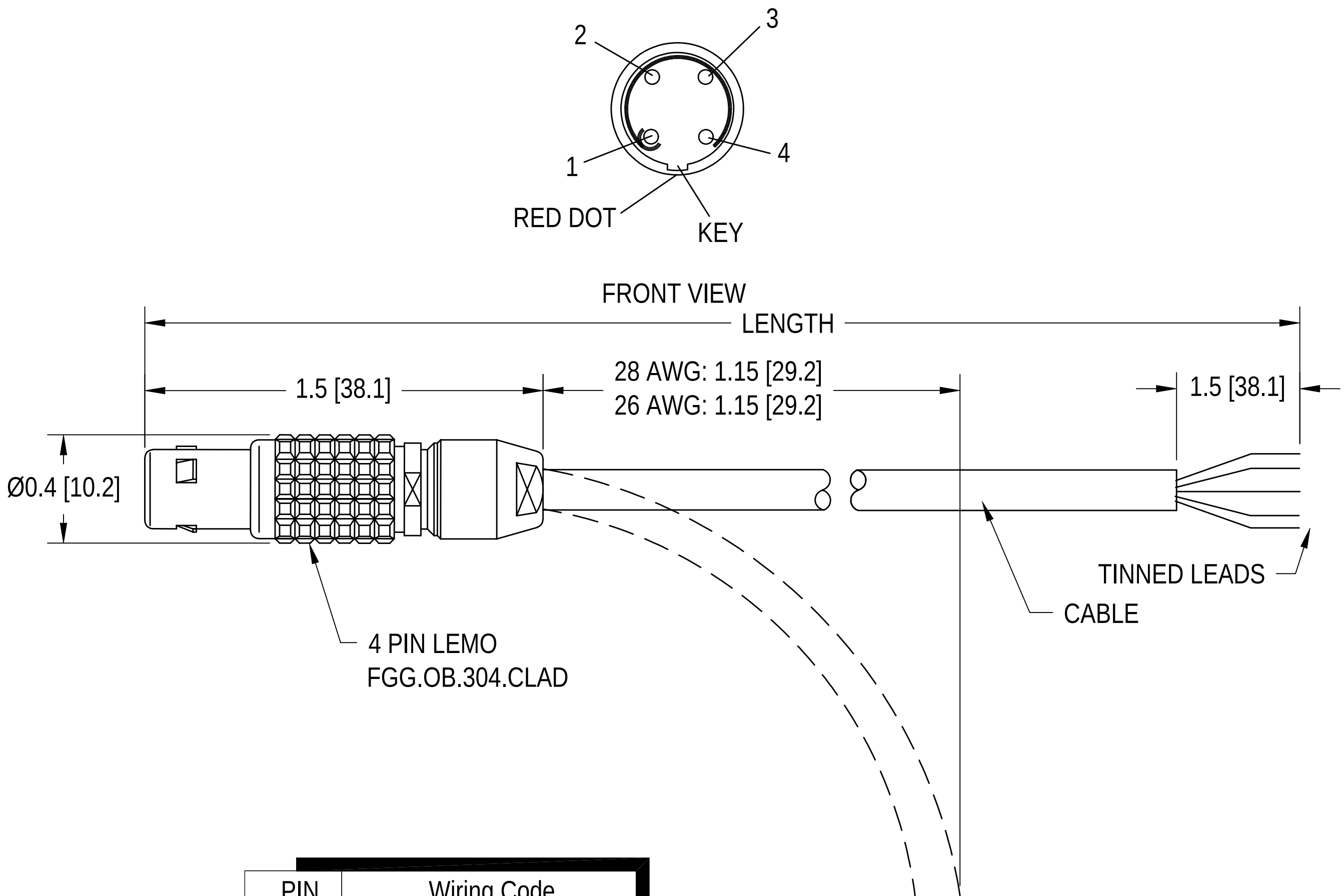
FUTEK MODEL ZCC940

Drawing Number: FI1101-B

INCH [mm]

4 Pin Lemo To Cable Assembly

FOR FUTEK MODELS WITH CC4 CONNECTION



PIN	Wiring Code
1	RED (+EXCITATION)
2	GREEN (+SIGNAL)
3	WHITE (-SIGNAL)
4	BLACK (-EXCITATION)
	SHIELD (FLOATING)

ITEM #	LENGTH	CABLE	SPECIFICATIONS
FSH01789	5 ft	28 AWG, 4 CONDUCTOR BRAIDED SHIELDED CLEAR PVC DIAMETER: 0.115 [2.9] CABLE MINIMUM BEND RADIUS: 1.115 [29.2]	TEMP: -40° to 176°F -40° to 80°C IP RATING IP50
FSH00173	10 ft		
FSH01790	15 ft		
FSH01787	20 ft		
FSH01785	30 ft		
FSH01791	50 ft		
FSH02114	100 ft		
FSH02115	5 ft	26 AWG, 4 CONDUCTOR BRAIDED SHIELDED TEFLON DIAMETER: 0.115 [2.9] CABLE MINIMUM BEND RADIUS: 1.15 [29.2]	
FSH01486	10 ft		
FSH02441	15 ft		
FSH02116	20 ft		
FSH02117	30 ft		
FSH02118	50 ft		
FSH02119	100 ft		

FUTEK
ADVANCED SENSOR TECHNOLOGY, INC.

CAGE CODE # 1X8M6

10 THOMAS
IRVINE, CA 92618 USA
1-800-23-FUTEK (38835)

INTERNET:
http://www.futek.com

HAMLIN

www.hamlin.com

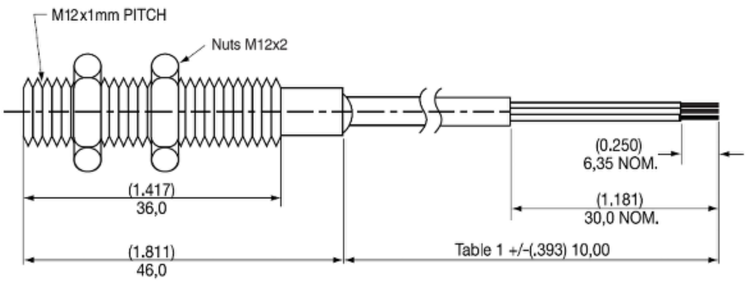
55075 Hall Effect Geartooth Speed Sensor

Features
- Rotary position geartooth sensor
- Stainless steel M12 threaded barrel
- Electronic protection against severe and automotive environments
- Self adjusting magnetic range
- 3 wire (voltage output)
- EMC Protection (consult Hamlin)
- Short circuit/reverse voltage protection

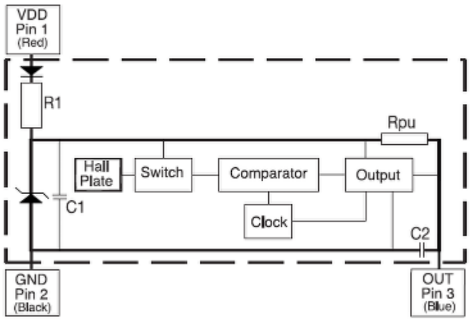
Benefits
- Long life; up to 20 billion operations
- High speed operation
- No chopper delay
- Unaffected by harsh environments
- Rotary orientation not critical
- On-chip 10 bit A/D converter
- Customer selection of cable length and connector type

Applications
- Geartooth sensor
- Camshaft sensor
- Linear encoder
- Rotary encoder

DIMENSIONS (in) mm



BLOCK DIAGRAM

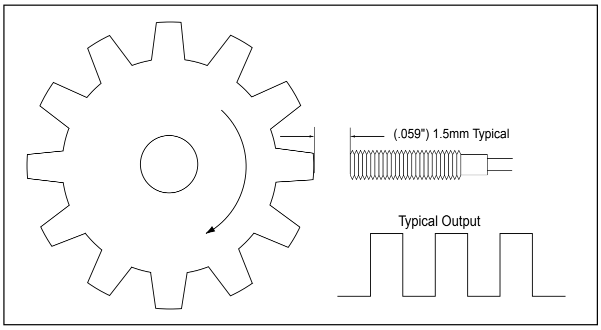


SPECIFICATIONS

Hall Type			Digital Switch 3 Wire (Voltage Output)
Supply Voltage (Note1)	Maximum Operation Overvoltage Protection	Vdc Vdc Vdc - max.	-25 to +25.2 4.75 to 25.2 27
Output High Voltage		Vdc - min.	VDD - 2 (sinking output with internal pull-up resistor)
Output Low Voltage		Vdc - max.	0.6 @20mA
Output Current (continuously on)		mA - max.	20
Current Consumption		mA - min. mA - max.	1 10.5
Switching Speed		KHz-max.	15
Temperature	Operating Storage	°C °C	-40 to +85 -65 to +85

Note 1: As long as Tj (Junction Temperature) is not exceeded.

APPLICATION EXAMPLE - Geartooth Sensor



CUSTOMER OPTIONS - Cable Length and Terminations

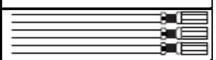
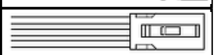
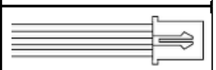
Table 1

Cable Length Options:
(Cable Type 24 AWG 7/32
PVC 105°C Double Insulated

Select Option	Cable Length (in) mm
01	(3.94) 100
02	(11.81) 300
03	(19.69) 500
04	(29.53) 750
05	(39.37) 1000

Table 2

Termination Options:

Select Option	Description	
A	Tinned leads	
C	6.35mm fastons	
D	AMP MTE 2,54mm pitch	
E	Jst xhp 2,5mm pitch	

ORDERING INFORMATION

55075 – 00 – XX – X

Series 55075

Cable Length — Table 1

Termination — Table 2

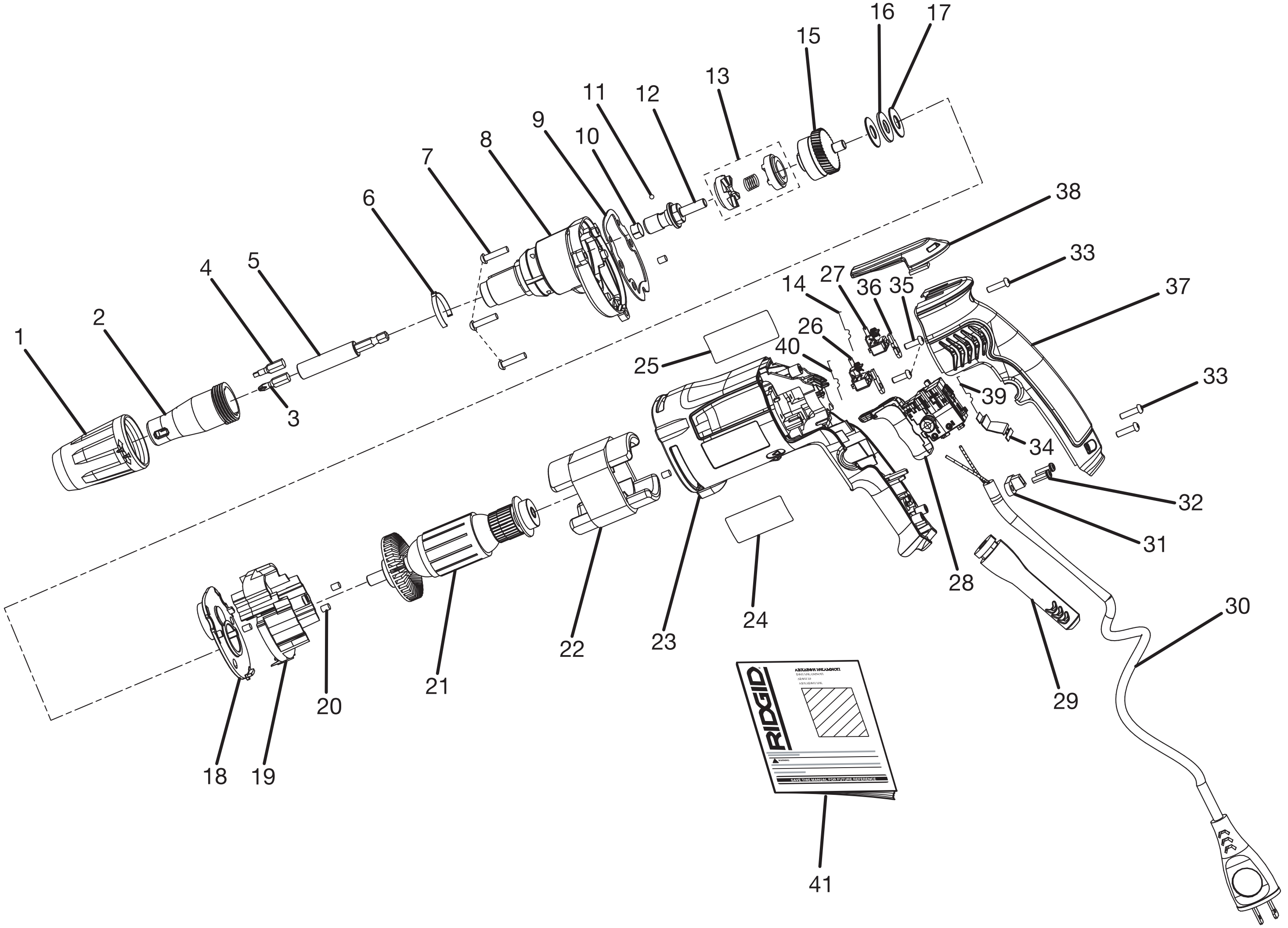
Hamlin USA Tel: +1 920 648 3000 • Fax: +1 920 648 3001 • Email: sales.us@hamlin.com
Hamlin Europe Tel: +44 (0)1603 257700 • Fax: +44 (0)1603 257702 • Email: sales.uk@hamlin.com
Hamlin China Tel: +86 (0) 512 69365 800 • Fax: +86 (0) 512 69365 811 • Email: sales.cn@hamlin.com

DRGNo: 55075XXXXX REV : AG Date: 9/6/12 DCR:B9016

RIDGID R6000-1

KEY	P/N	QTY
1	590888002	1
2	300600007	1
3	678051002	1
4	678051001	1
5	671220001	1
6	671218003	1
7	660065002	3
8	300208039	1
9	900884001	1
10	671221001	1
11	6861701	1
12	620248001	1
13	300357009	1
14	290221001	1
15	300003057	1
16	6906601	1
17	6906501	2
18	300598006	1
19	590889001	1
20	560712001	5
21	200214004	1
22	740487001	1
23	590890001	1
24	940304054	1
25	940976125	1
26	290069057	1
27	290069061	1
28	760339002	1
29	560968001	1
30	730387003	1
31	512360001	1
32	6620801	2
33	660024006	3
34	900943001	1
35	660208011	2
36	631115001	2
37	590892001	1
38	590893001	1
39	290102004	1
40	290061059	1
41	987000143	1

04-17-07
(REV:00)



⚠ **WARNING:** The assembly shown represents an important part of the Double Insulated System. To avoid the possibility of alteration or damage to the System, service should be performed by your nearest authorized service center.

⚠ **AVERTISSEMENT :** L'ensemble illustré représente une partie importante du système à isolation double. Afin d'éviter la possibilité de modification ou de dommage au système, la réparation ne doit être effectuée que par votre centre de réparation centre commercial autorisé.

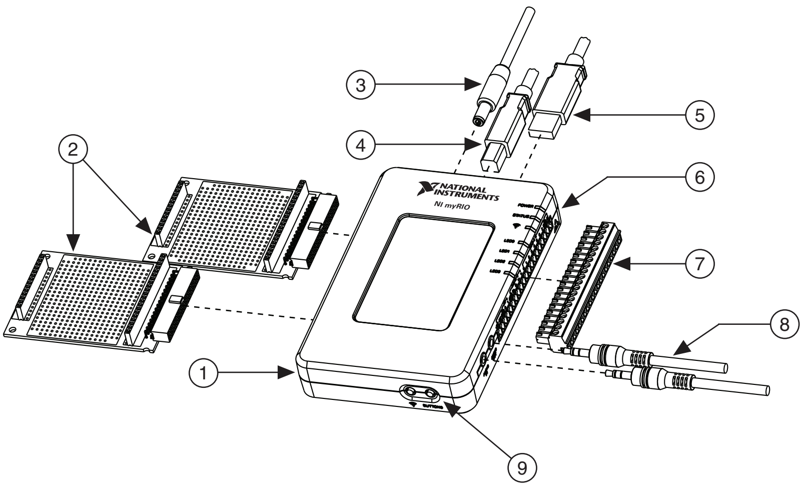
⚠ **ADVERTENCIA:** El conjunto que se ilustra representa una parte importante del Sistema de Aislamiento Doble. Para evitar la posibilidad de alteración o daño del sistema, la reparación debe ser efectuada en su centro de servicio autorizado.

USER GUIDE AND SPECIFICATIONS

NI myRIO-1900

The National Instruments myRIO-1900 is a portable reconfigurable I/O (RIO) device that students can use to design control, robotics, and mechatronics systems. This document contains pinouts, connectivity information, dimensions, mounting instructions, and specifications for the NI myRIO-1900.

Figure 1. NI myRIO-1900



1 NI myRIO-1900
2 myRIO Expansion Port (MXP) Breakouts (One Included in Kit)
3 Power Input Cable
4 USB Device Cable
5 USB Host Cable (Not Included in Kit)
6 LEDs
7 Mini System Port (MSP) Screw-Terminal Connector
8 Audio In/Out Cables (One Included in Kit)
9 Button0

Safety Information

Caution Do not operate the hardware in a manner not specified in this document and in the user documentation. Misuse of the hardware can result in a hazard. You can compromise the safety protection if the hardware is damaged in any way. If the hardware is damaged, return it to National Instruments for repair.

Clean the hardware with a soft, nonmetallic brush. Make sure that the hardware is completely dry and free from contaminants before returning it to service.

Electromagnetic Compatibility Guidelines

This product was tested and complies with the regulatory requirements and limits for electromagnetic compatibility (EMC) stated in the product specifications. These requirements and limits provide reasonable protection against harmful interference when the product is operated in the intended operational electromagnetic environment.

This product is intended for use in commercial locations. There is no guarantee that harmful interference will not occur in a particular installation or when the product is connected to a test object. To minimize interference with radio and television reception and prevent unacceptable performance degradation, install and use this product in strict accordance with the instructions in the product documentation.

Furthermore, any modifications to the product not expressly approved by National Instruments could void your authority to operate it under your local regulatory rules.

Caution This product was tested for EMC compliance using myRIO application software. The maximum length for USB cables is 2.0 m (6.6 ft), and the maximum length for signal wires is 30.0 cm (11.8 in.).

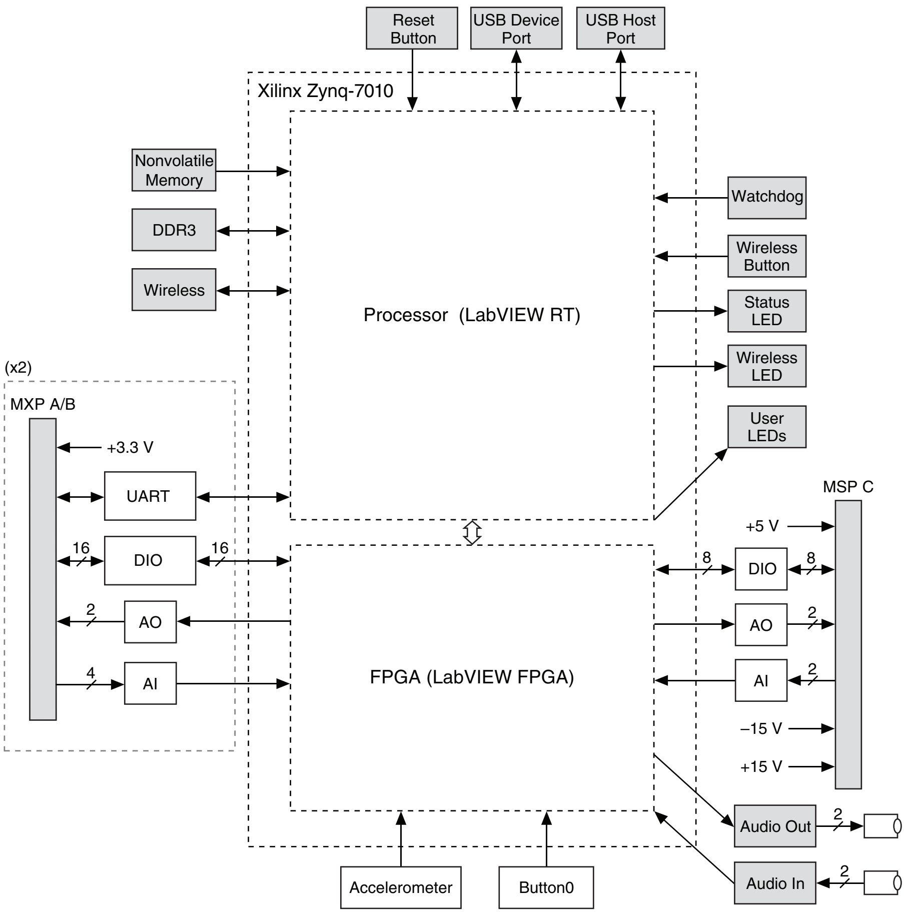
Caution The mounting keyholes on the back of the NI myRIO-1900 are sensitive to electrostatic discharge (ESD). When handling the device, be careful not to touch inside the keyholes.

Hardware Overview

The NI myRIO-1900 provides analog input (AI), analog output (AO), digital input and output (DIO), audio, and power output in a compact embedded device. The NI myRIO-1900 connects to a host computer over USB and wireless 802.11b,g,n.

The following figure shows the arrangement and functions of NI myRIO-1900 components.

Figure 2. NI myRIO-1900 Hardware Block Diagram



Connector Pinouts

NI myRIO-1900 Expansion Port (MXP) connectors A and B carry identical sets of signals. The signals are distinguished in software by the connector name, as in `ConnectorA/DIO1` and `ConnectorB/DIO1`. Refer to the software documentation for information about configuring and using signals. The following figure and table show the signals on MXP connectors A and B. Note that some pins carry secondary functions as well as primary functions.

Figure 3. Primary/Secondary Signals on MXP Connectors A and B

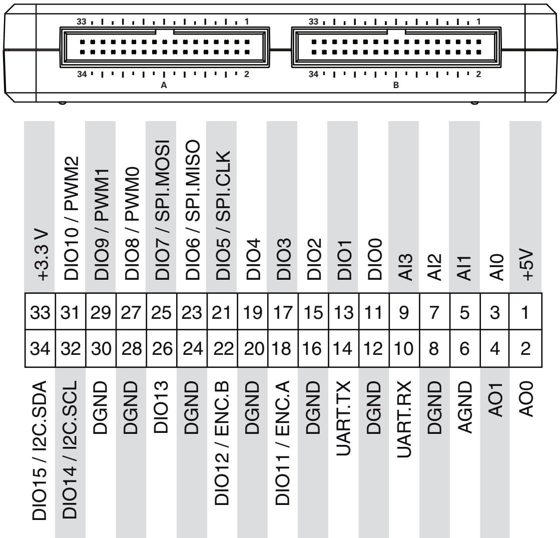


Table 1. Descriptions of Signals on MXP Connectors A and B

Signal Name	Reference	Direction	Description
+5V	DGND	Output	+5 V power output.
AI <0..3>	AGND	Input	0-5 V, referenced, single-ended analog input channels. Refer to the *Analog Input Channels* section for more information.
AO <0..1>	AGND	Output	0-5 V referenced, single-ended analog output. Refer to the *Analog Output Channels* section for more information.
AGND	N/A	N/A	Reference for analog input and output.
+3.3V	DGND	Output	+3.3 V power output.
DIO <0..15>	DGND	Input or Output	General-purpose digital lines with 3.3 V output, 3.3 V/5 V-compatible input. Refer to the *DIO Lines* section for more information.
UART.RX	DGND	Input	UART receive input. UART lines are electrically identical to DIO lines.
UART.TX	DGND	Output	UART transmit output. UART lines are electrically identical to DIO lines.
DGND	N/A	N/A	Reference for digital signals, +5 V, and +3.3 V.

The following figure and table show the signals on Mini System Port (MSP) connector C. Note that some pins carry secondary functions as well as primary functions.

Figure 4. Primary/Secondary Signals on MSP Connector C

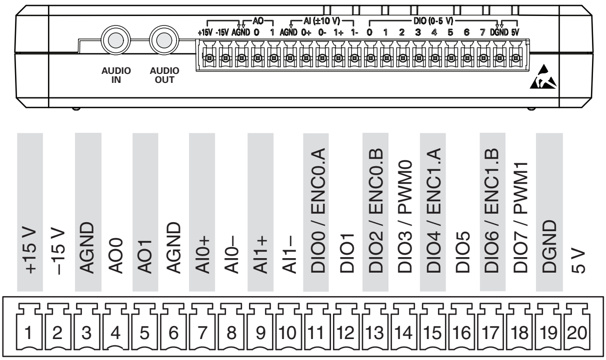


Table 2. Descriptions of Signals on MSP Connector C

Signal Name	Reference	Direction	Description
+15V/-15V	AGND	Output	+15 V/-15 V power output.
AI0+/AI0-; AI1+/AI1-	AGND	Input	±10 V, differential analog input channels. Refer to the *Analog Input Channels* section for more information.
AO <0..1>	AGND	Output	±10 V referenced, single-ended analog output channels. Refer to the *Analog Output Channels* section for more information.
AGND	N/A	N/A	Reference for analog input and output and +15 V/-15 V power output.
+5V	DGND	Output	+5 V power output.
DIO <0..7>	DGND	Input or Output	General-purpose digital lines with 3.3 V output, 3.3 V/5 V-compatible input. Refer to the *DIO Lines* section for more information.
DGND	N/A	N/A	Reference for digital lines and +5 V power output.

Table 3. Descriptions of Signals on Audio Connectors

Signal Name	Reference	Direction	Description
AUDIO IN	N/A	Input	Left and right audio inputs on stereo connector.
AUDIO OUT	N/A	Output	Left and right audio outputs on stereo connector.

Analog Input Channels

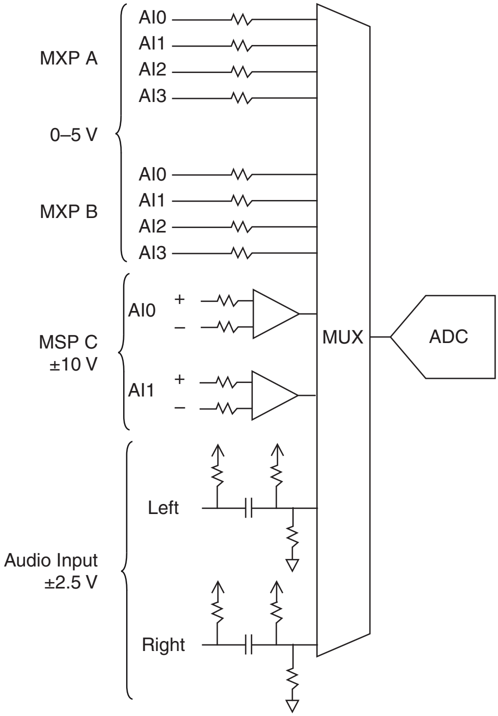
The NI myRIO-1900 has analog input channels on myRIO Expansion Port (MXP) connectors A and B, Mini System Port (MSP) connector C, and a stereo audio input connector. The analog inputs are multiplexed to a single analog-to-digital converter (ADC) that samples all channels.

MXP connectors A and B have four single-ended analog input channels per connector, AI0-AI3, which you can use to measure 0-5 V signals. MSP connector C has two high-impedance, differential analog input channels, AI0 and AI1, which you can use to measure signals up to ±10 V. The audio inputs are left and right stereo line-level inputs with a ±2.5 V full-scale range.

Note For important information about improving measurement accuracy by reducing noise, go to `ni.com/info` and enter the Info Code `analogwiring`.

Figure 5 shows the analog input topology of the NI myRIO-1900.

Figure 5. NI myRIO-1900 Analog Input Circuitry



Analog Output Channels

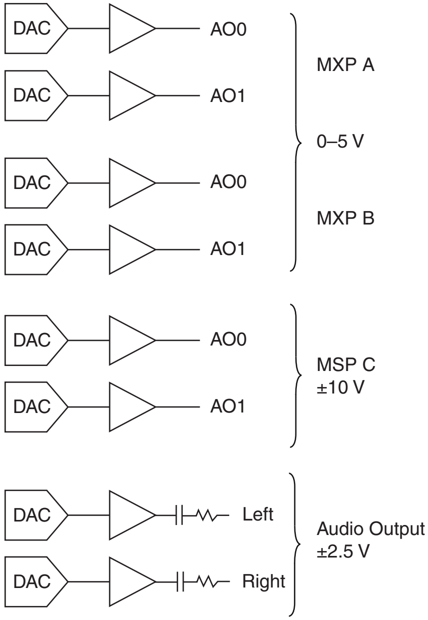
The NI myRIO-1900 has analog output channels on myRIO Expansion Port (MXP) connectors A and B, Mini System Port (MSP) connector C, and a stereo audio output connector. Each analog output channel has a dedicated digital-to-analog converter (DAC), so they can all update simultaneously. The DACs for the analog output channels are controlled by two serial communication buses from the FPGA. MXP connectors A and B share one bus, and MSP connector C and the audio outputs share a second bus. Therefore, the maximum update rate is specified as an aggregate figure in the *Analog Output* section of the *Specifications*.

MXP connectors A and B have two analog output channels per connector, AO0 and AO1, which you can use to generate 0-5 V signals. MSP connector C has two analog output channels, AO0 and AO1, which you can use to generate signals up to ±10 V. The audio outputs are left and right stereo line-level outputs capable of driving headphones.

Caution Before using headphones to listen to the audio output of the NI myRIO-1900, ensure that the audio output is at a safe level. Listening to audio signals at a high volume may result in permanent hearing loss.

Figure 6 shows the analog output topology of the NI myRIO-1900.

Figure 6. NI myRIO-1900 Analog Output Circuitry



Accelerometer

The NI myRIO-1900 contains a three-axis accelerometer. The accelerometer samples each axis continuously and updates a readable register with the result. Refer to the *Accelerometer* section of the *Specifications* for the accelerometer sample rates.

Converting Raw Data Values to Voltage

You can use the following equations to convert raw data values to volts:

$$V = Raw\ Data\ Value * LSB\ Weight$$

$$LSB\ Weight = Nominal\ Range \div 2^{ADC\ Resolution}$$

where *Raw Data Value* is the value returned by the FPGA I/O Node,

LSB Weight is the value in volts of the increment between data values,

Nominal Range is the absolute value in volts of the full, peak-to-peak nominal range of the channel,

and *ADC Resolution* is the resolution of the ADC in bits. (*ADC Resolution* = 12.)

- For AI and AO channels on the MXP connectors,

$$LSB\ Weight = 5\text{ V} \div 2^{12} = 1.221\text{ mV}$$

$$Maximum\ reading = 4095 * 1.221\text{ mV} = 4.999\text{ V}$$

- For AI and AO channels on the MSP connectors,

$$LSB\ Weight = 20\text{ V} \div 2^{12} = 4.883\text{ mV}$$

$$Maximum\ Positive\ Reading = +2047 * 4.883\text{ mV} = 9.995\text{ V}$$

$$Maximum\ Negative\ Reading = -2048 * 4.883\text{ mV} = -10.000\text{ V}$$

- For Audio In/Out,

$$LSB\ Weight = 5\text{ V} \div 2^{12} = 1.221\text{ mV}$$

$$Maximum\ Positive\ Reading = +2047 * 1.221\text{ mV} = 2.499\text{ V}$$

$$Maximum\ Negative\ Reading = -2048 * 1.221\text{ mV} = -2.500\text{ V}$$

- For the accelerometer,

$$LSB\ Weight = 16\text{ g} \div 2^{12} = 3.906\text{ mg}$$

$$Maximum\ Positive\ Reading = +2047 * 3.906\text{ mg} = +7.996\text{ g}$$

$$Maximum\ Negative\ Reading = -2048 * 3.906\text{ mg} = -8.000\text{ g}$$

DIO Lines

The NI myRIO-1900 has 3.3 V general-purpose DIO lines on the MXP and MSP connectors. MXP connectors A and B have 16 DIO lines per connector. On the MXP connectors, each DIO line from 0 to 13 has a 40 kΩ pullup resistor to 3.3 V, and DIO lines 14 and 15 have 2.2 kΩ pullup resistors to 3.3 V. MSP connector C has eight DIO lines. Each MSP DIO line has a 40 kΩ pulldown resistor to ground. DGND is the reference for all the DIO lines. You can program all the lines individually as inputs or outputs. Secondary digital functions include Serial Peripheral

Interface Bus (SPI), I2C, pulse-width modulation (PWM), and quadrature encoder input. Refer to the NI myRIO software documentation for information about configuring the DIO lines.

Figure 7. DIO Lines <13..0> on MXP Connector A or B

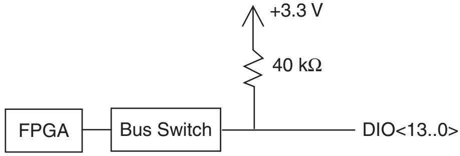


Figure 8. DIO Lines <15..14> on MXP Connector A or B

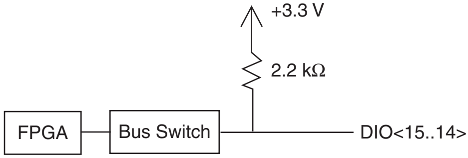
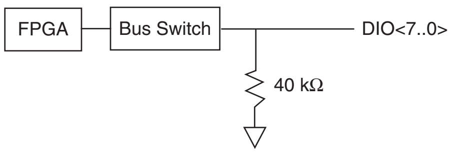


Figure 9. DIO Lines <7..0> on MSP Connector C



When a DIO line is floating, it floats in the direction of the pull resistor. A DIO line may be floating in any of the following conditions:

- when the myRIO device is starting up
- when the line is configured as an input
- when the myRIO device is powering down

You can add a stronger resistor to a DIO line to cause it to float in the opposite direction.

UART Lines

The NI myRIO-1900 has one UART receive input line and one UART transmit ouput line on each MXP connector. The UART lines are electrically identical to DIO lines 0 to 13 on the MXP connectors. Like those lines, UART.RX and UART.TX have 40 kΩ pullup resistors to 3.3 V. Use LabVIEW Real-Time to read and write over the UART lines.

Using the Reset Button

Pressing and releasing the Reset button restarts the processor and the FPGA.

Pressing and holding the Reset button for 5 seconds, then releasing it, restarts the processor and the FPGA and forces the NI myRIO-1900 into safe mode. In safe mode, the NI myRIO-1900 launches only the services necessary for updating configuration and installing software.

When the NI myRIO-1900 is in safe mode, you can communicate with it by using the UART lines on MXP connector A. You need the following items to communicate with the myRIO device over UART:

- USB-to-TTL serial UART converter cable (for example, part number TTL-232RG-VSW3V3-WE from FTD Chip)
- Serial-port terminal program configured with the following settings:
  - 115,200 bits per second
  - Eight data bits
  - No parity
  - One stop bit
  - No flow control

Using the Wireless Button and LED

For information about using the Wireless button, go to `ni.com/info` and enter the Info Code `myriowirelessbutton`.

For information about using the Wireless LED, go to `ni.com/info` and enter the Info Code `myriowirelessled`.

Using Button0

Button0 produces a logic TRUE when depressed and a logic FALSE when not depressed. Button0 is not debounced.

Understanding LED Indications

Power LED

The Power LED is lit while the NI myRIO-1900 is powered on. This LED indicates that the power supply connected to the device is adequate.

Status LED

The Status LED is off during normal operation. The NI myRIO-1900 runs a power-on self test (POST) when you apply power to the device. During the POST, the Power and Status LEDs turn on. When the Status LED turns off, the POST is complete. The NI myRIO-1900 indicates specific error conditions by flashing the Status LED a certain number of times every few seconds, as shown in Table 4.

Table 4. Status LED Indications

Number of Flashes Every Few Seconds	Indication
2	The device has detected an error in its software. This usually occurs when an attempt to upgrade the software is interrupted. Reinstall software on the device.
3	The device is in safe mode.
4	The software has crashed twice without rebooting or cycling power between crashes. This usually occurs when the device runs out of memory. Review your RT VI and check the memory usage. Modify the VI as necessary to solve the memory usage issue.
Continuously flashing or solid	The device has detected an unrecoverable error. Contact National Instruments.

LEDs 0-3

You can use LEDs 0-3 to help debug your application or easily retrieve application status. Logic TRUE turns an LED on and logic FALSE turns an LED off.

Using the USB Host Port

The NI myRIO-1900 USB host port supports Web cameras that conform to the USB Video Device Class (UVC) protocol as well as machine vision cameras that conform to the USB3 Vision standard and are USB 2.0 backward compatible. The NI myRIO-1900 USB host port also supports Basler ace USB3 cameras.

The NI myRIO-1900 USB host port also supports USB Flash drives and USB-to-IDE adapters formatted with FAT16 and FAT32 file systems. LabVIEW usually maps USB devices to the `/U`, `/V`, `/W`, or `/X` drive, starting with the `/U` drive if it is available.

NI myRIO-1900 Physical Dimensions

Figure 10. NI myRIO-1900 Dimensions, Front

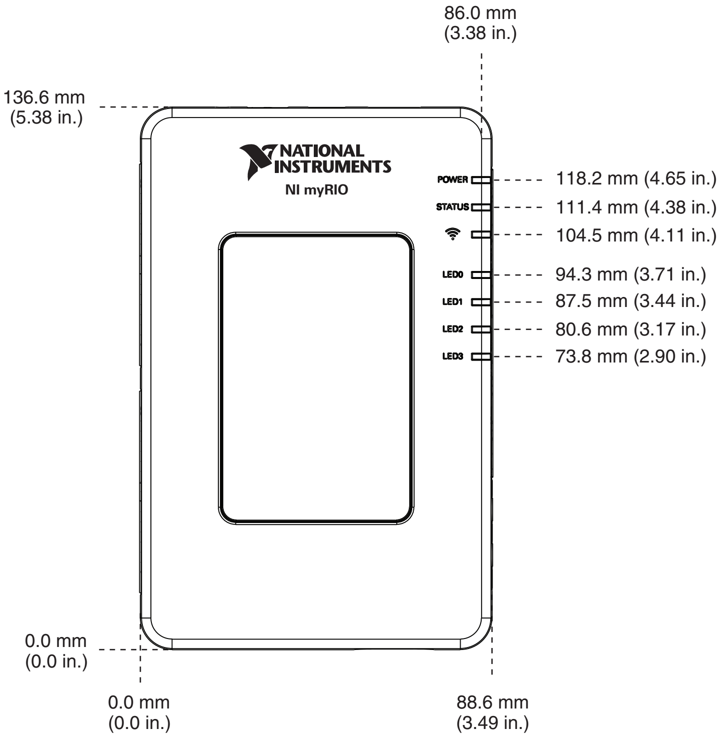


Figure 11. NI myRIO-1900 Dimensions, Back

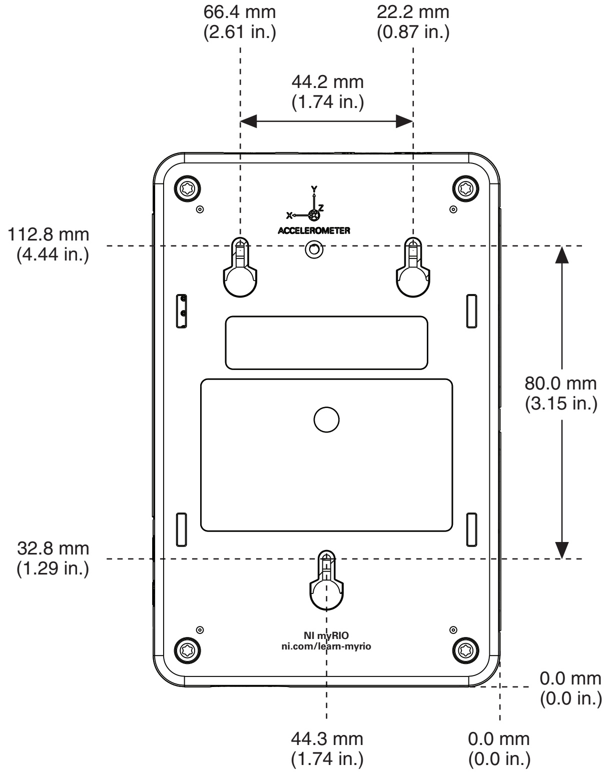
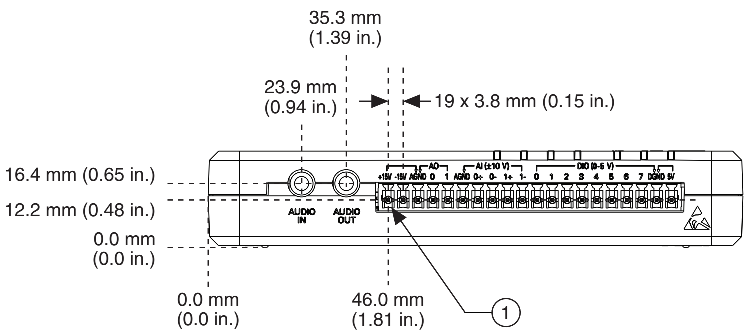
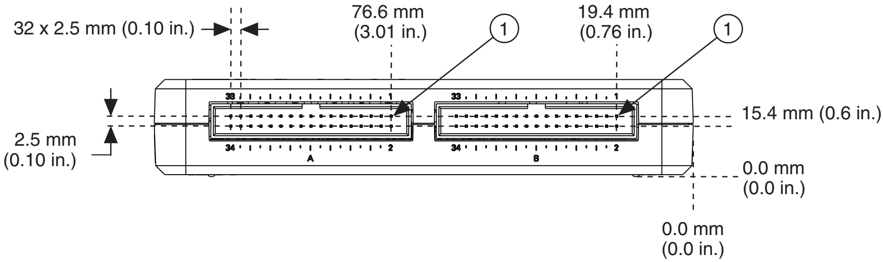


Figure 12. NI myRIO-1900 Dimensions, MSP Side



1 Pin 1

Figure 13. NI myRIO-1900 Dimensions, MXP Side



1 Pin 1

Figure 14. NI myRIO-1900 Dimensions, I/O End

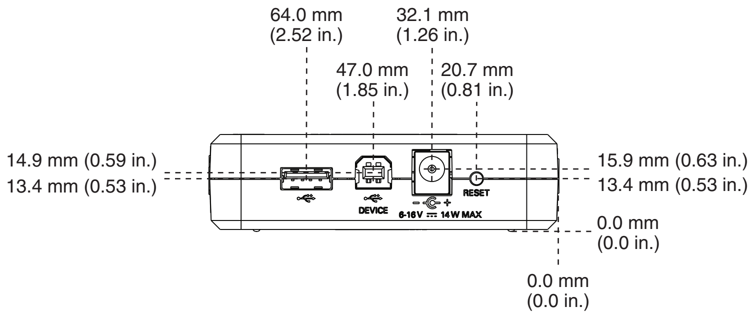
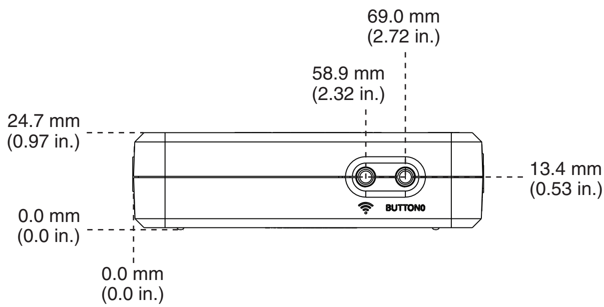


Figure 15. NI myRIO-1900 Dimensions, User End

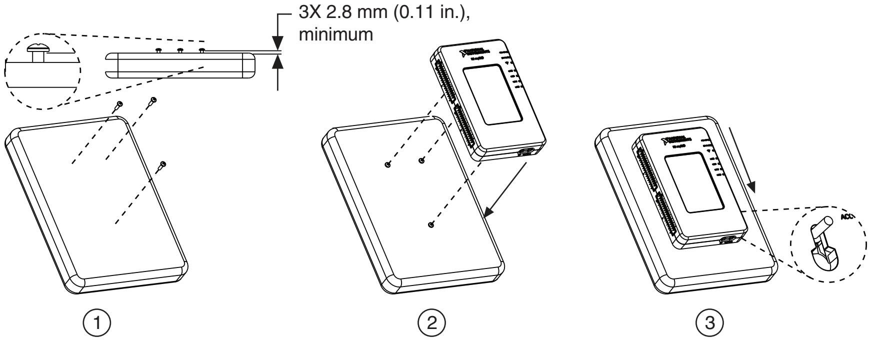


Mounting the NI myRIO-1900

Mounting the NI myRIO-1900 Using the Key Holes

You can use the provided key holes on NI myRIO-1900 to mount the device on a flat surface. Install the NI myRIO-1900 as shown in Figure 16. Use Unified #4 or ISO M3 screws to mount the NI myRIO-1900 using the key holes. Panhead screws are suitable for use with the NI myRIO-1900 key holes.

Figure 16. Mounting the NI myRIO-1900 Using the Key Holes

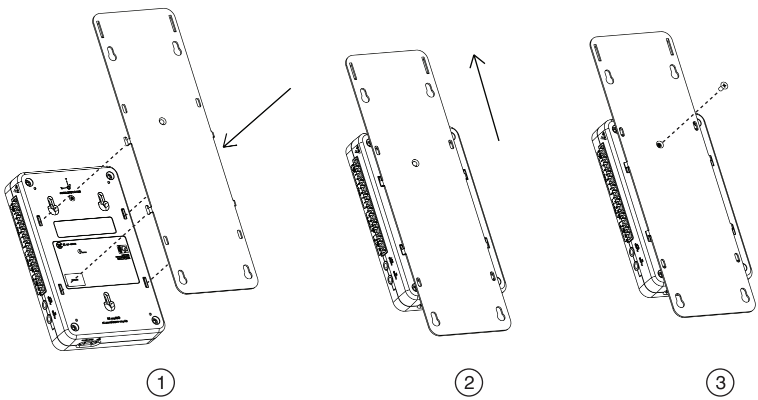


1 Install three Unified #4 or M3 screws in the flat surface using the key hole dimensions of the NI myRIO-1900 as a guide. Refer to Figure 11 for NI myRIO-1900 key hole dimensions. Leave a minimum spacing of 2.8 mm (0.11 in.) between the flat surface and the screw heads.
2 Place the NI myRIO-1900 on the screw heads.
3 Slide the NI myRIO-1900 down to secure the key holes on the screw heads.

Mounting the NI myRIO-1900 Using the Panel Mounting Kit

You can use the Panel Mounting Kit for NI myRIO-1900 to mount the device on a flat surface such as a panel or wall. Install the panel mounting kit on the NI myRIO-1900 as shown in Figure 17.

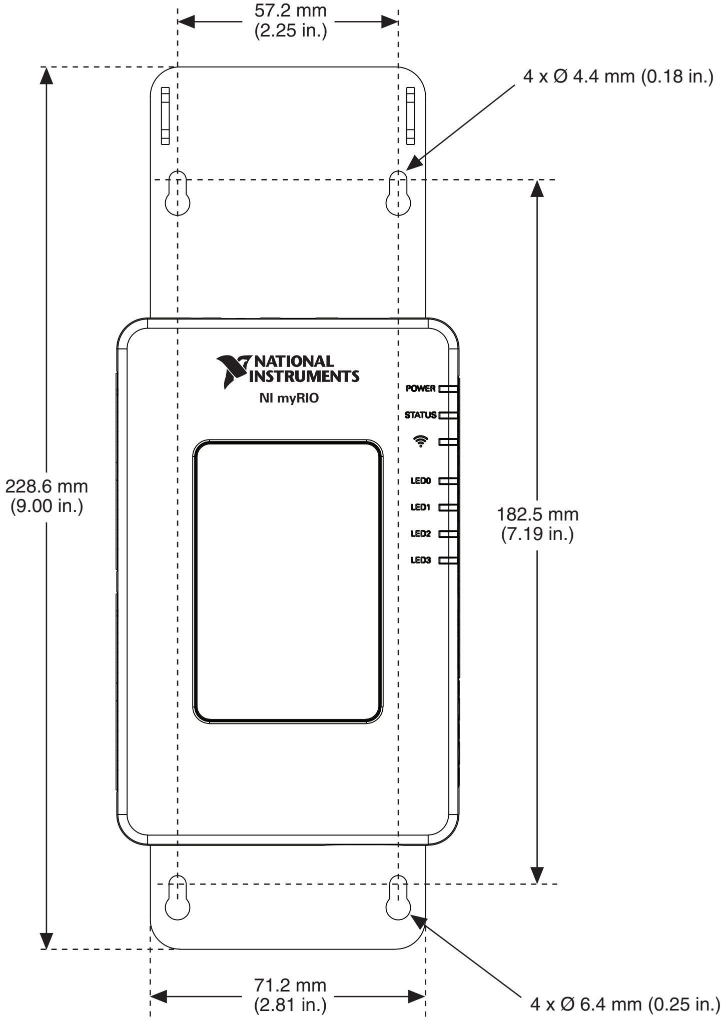
Figure 17. Installing the Panel Mounting Kit on the NI myRIO-1900



1 Place the panel on the back of the NI myRIO-1900
2 Slide the panel up to line up the screw holes on the panel and the NI myRIO-1900.
3 Secure the panel to the NI myRIO-1900. You must use the included 4-40 × 1/4 in. screw to attach the panel mounting kit to the NI myRIO-1900. Tighten the screw to 0.76 N · m (6.7 lb · in.) of torque. Do not exceed 0.87 N · m (7.7 lb · in.) of torque.

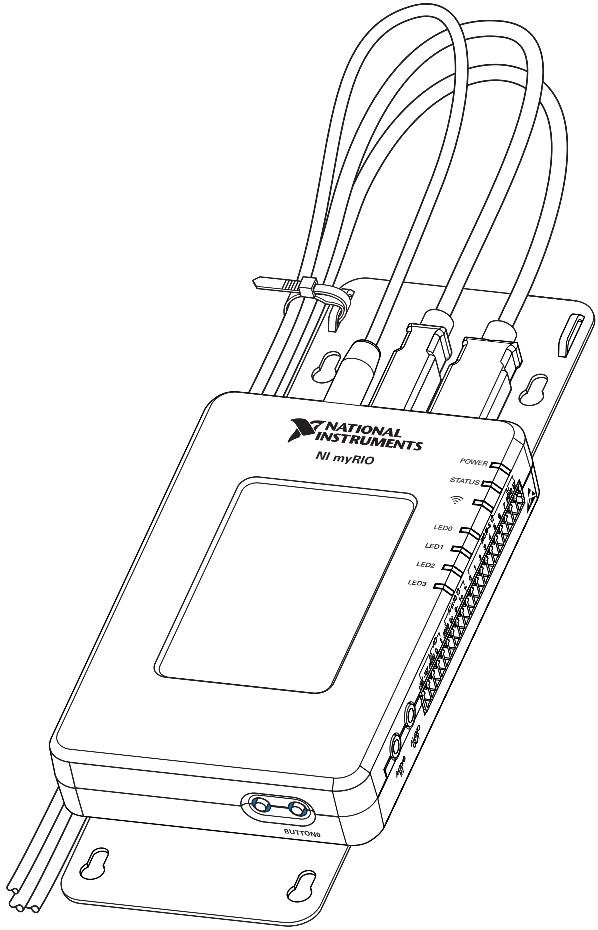
Fasten the panel mounting kit to the panel or wall using screws appropriate for the surface. The following figure shows the dimensions of the NI myRIO-1900 with the panel mounting kit installed.

Figure 18. Dimensions of NI myRIO-1900 with Panel Mounting Kit



Use a cable tie to secure the power and USB cables to the panel mounting kit as shown in Figure 19.

Figure 19. Securing the Power and USB Cables to the Panel Mounting Kit



Cables and Accessories

Table 5. Accessories Available from NI

Accessory	Description	NI Part Number
Power supply	Power supply for NI myRIO-1900	723403-01
MXP breakouts	Set of five MXP breakout boards for NI myRIO-1900	782696-01
MSP connector	MSP replacement connector plug for NI myRIO-1900	765788-01
Panel mounting kit	Panel mounting kit for NI myRIO-1900	783091-01

Specifications

The following specifications are typical for the 0 to 40 °C operating tempreature range unless otherwise noted.

Processor

Processor type .... Xilinx Z-7010

Processor speed .... 667 MHz

Processor cores 2

Memory

Nonvolatile memory .... 256 MB

DDR3 memory .... 512 MB

- DDR3 clock frequency .... 533 MHz
- DDR3 data bus width .... 16 bits

For information about the lifespan of the nonvolatile memory and about best practices for using nonvolatile memory, go to `ni.com/info` and enter the Info Code `SSDBP`.

FPGA

FPGA type .... Xilinx Z-7010

Wireless Characteristics

Radio mode .... IEEE 802.11 b,g,n

Frequency band .... ISM 2.4 GHz

Channel width .... 20 MHz

Channels ... USA 1-11, International 1-13

TX power ... +10 dBm max (10 mW)

Outdoor range ... Up to 150 m (line of sight)

Antenna directivity ... Omnidirectional

Security ... WPA, WPA2, WPA2-Enterprise

USB Ports

USB host port ... USB 2.0 Hi-Speed

USB device port ... USB 2.0 Hi-Speed

Analog Input

Aggregate sample rate ... 500 kS/s

Resolution ... 12 bits

Overvoltage protection ... ±16 V

MXP connectors

- Configuration ... Four single-ended channels per connector
- Input impedance ... >500 kΩ acquiring at 500 kS/s
1 MΩ powered on and idle
4.7 kΩ powered off
- Recommended source impedance ... 3 kΩ or less
- Nominal range ... 0 V to +5 V
- Absolute accuracy ... ±50 mV
- Bandwidth ... >300 kHz

MSP connector

- Configuration ... Two differential channels
- Input impedance ... Up to 100 nA leakage powered on;
4.7 kΩ powered off
- Nominal range ... ±10 V
- Working voltage (signal + common mode) ... ±10 V of AGND
- Absolute accuracy ... ±200 mV
- Bandwidth ... 20 kHz minimum, >50 kHz typical

Audio input

- Configuration ... One stereo input consisting of two AC-coupled, single-ended channels
- Input impedance ... 10 kΩ at DC
- Nominal range ... ±2.5 V
- Bandwidth ... 2 Hz to >20 kHz

Analog Output

Aggregate maximum update rates

- All AO channels on MXP connectors....... 345 kS/s
- All AO channels on MSP connector and audio output channels......................... 345 kS/s

Resolution .... 12 bits

Overload protection .... ±16 V

Startup voltage .... 0 V after FPGA initialization

MXP connectors

- Configuration .... Two single-ended channels per connector
- Range .... 0 V to +5 V
- Absolute accuracy.................................... 50 mV
- Current drive .... 3 mA
- Slew rate .... 0.3 V/µs

MSP connector

- Configuration .... Two single-ended channels
- Range .... ±10 V
- Absolute accuracy.................................... ±200 mV
- Current drive .... 2 mA
- Slew rate .... 2 V/µs

Audio output

- Configuration .... One stereo output consisting of two AC-coupled, single-ended channels
- Output impedance .... 100 Ω in series with 22 µF
- Bandwidth .... 70 Hz to >50 kHz into 32 Ω load; 2 Hz to >50 kHz into high-impedance load

Digital I/O

Number of lines

- MXP connectors .... 2 ports of 16 DIO lines (one port per connector); one UART.RX and one UART.TX line per connector
- MSP connector.. 1 port of 8 DIO lines

Direction control .... Each DIO line individually programmable as input or output

Logic level .... 5 V compatible LVTTL input; 3.3 V LVTTL output

Input logic levels

- Input low voltage, V_{IL} .... 0 V min; 0.8 V max
- Input high voltage, V_{IH} .... 2.0 V min; 5.25 V max

Output logic levels

- Output high voltage, V_{OH} sourcing 4 mA .... 2.4 V min; 3.465 V max
- Output low voltage, V_{OL} sinking 4 mA .... 0 V min; 0.4 V max

Minimum pulse width.... 20 ns

Maximum frequencies for secondary digital functions

- SPI .... 4 MHz
- PWM.... 100 kHz
- Quadrature encoder input .... 100 kHz
- I^2C.... 400 kHz

UART lines

- Maximum baud rate.... 230,400 bps
- Data bits.... 5, 6, 7, 8
- Stop bits 1, 2
- Parity.... Odd, Even, Mark, Space
- Flow control.... XON/XOFF

Accelerometer

Number of axes.... 3

Range .... ±8 g

Resolution.... 12 bits

Sample rate .... 800 S/s

Noise.... 3.9 mg_{rms} typical at 25 °C

Power Output

+5 V power output

- Output voltage .... 4.75 V to 5.25 V
- Maximum current on each connector .... 100 mA

+3.3 V power output

- Output voltage .... 3.0 V to 3.6 V
- Maximum current on each connector .... 150 mA

+15 power output

Output voltage....................................+15 V to +16 V

Maximum current ....32 mA (16 mA during startup)

-15 V power output

Output voltage....................................-15 V to -16 V

Maximum current ....32 mA (16 mA during startup)

Maximum combined power from +15 V
and -15 V power output ....500 mW

Power Requirements

NI myRIO-1900 requires a power supply connected to the power connector.

Caution You must use either the power supply provided in the shipping kit, or another UL Listed ITE power supply marked *LPS*, with the NI myRIO-1900.

Power supply voltage range............................6-16 VDC

Maximum power consumption ....14 W

Typical idle power consumption.......................2.6 W

Environmental

To meet these specifications, you must operate the NI myRIO-1900 with the window facing away from the mounting surface and ensure that there is at least 1 in. of clearance in front of the window during use.

Ambient temperature near device
(IEC 60068-2-1, IEC 600682-2).......................0 to 40 °C

Storage temperature
(IEC 60068-2-1, IEC 600682-2).......................-20 to 70 °C

Operating humidity (IEC 60068-2-56) ....10 to 90% RH, noncondensing

Storage humidity (IEC 60068-2-56) ....10 to 90% RH, noncondensing

Maximum altitude..2,000 m

Pollution Degree (IEC 60664) 2

Indoor use only.

Physical Characteristics

Weight ....193 g (6.8 oz)

Safety

Safety Standards

This product is designed to meet the requirements of the following standards of safety for electrical equipment for measurement, control, and laboratory use:

- IEC 61010-1, EN 61010-1
- UL 61010-1, CSA 61010-1

Note For UL and other safety certifications, refer to the product label or the *Online Product Certification* section.

Caution Using the NI myRIO-1900 in a manner not described in this document may impair the protection the NI myRIO-1900 provides.

Hazardous Locations

The NI myRIO-1900 is not certified for use in hazardous locations.

Electromagnetic Compatibility

This product meets the requirements of the following EMC standards for electrical equipment for measurement, control, and laboratory use:

- EN 61326-1 (IEC 61326-1): Class A emissions; Basic immunity
- EN 55022 (CISPR 22): Group 1, Class A emissions
- EN 55011 (CISPR 11): Group 1, Class A emissions
- AS/NZS CISPR 11: Group 1, Class A emissions
- AS/NZS CISPR 22: Group 1, Class A emissions
- FCC 47 CFR Part 15B: Class A emissions
- ICES-001: Class A emissions

Note For EMC declarations and certifications, refer to the *Online Product Certification* section.

CE Compliance CE

This product meets the essential requirements of applicable European Directives as follows:

- 2006/95/EC; Low-Voltage Directive (safety)
- 2004/108/EC; Electromagnetic Compatibility Directive (EMC)
- 1999/5/EC; Radio and Telecommunications Terminal Equipment Directive (R&TTE)

Online Product Certification

To obtain product certifications and the Declaration of Conformity (DoC) for this product, visit `ni.com/certification`, search by model number or product line, and click the appropriate link in the Certification column.

Environmental Management

NI is committed to designing and manufacturing products in an environmentally responsible manner. NI recognizes that eliminating certain hazardous substances from our products is beneficial to the environment and to NI customers.

For additional environmental information, refer to the *Minimize Our Environmental Impact* web page at `ni.com/environment`. This page contains the environmental regulations and directives with which NI complies, as well as other environmental information not included in this document.

Waste Electrical and Electronic Equipment (WEEE)

EU Customers At the end of the product life cycle, all products *must* be sent to a WEEE recycling center. For more information about WEEE recycling centers, National Instruments WEEE initiatives, and compliance with WEEE Directive 2002/96/EC on Waste and Electronic Equipment, visit `ni.com/environment/weee`.

电子信息产品污染控制管理办法 （中国 RoHS）

中国客户 National Instruments 符合中国电子信息产品中限制使用某些有害物质指令 (RoHS)。关于 National Instruments 中国 RoHS 合规性信息，请登录 `ni.com/environment/rohs_china`。 (For information about China RoHS compliance, go to `ni.com/environment/rohs_china`.)

Regulatory Information

United States

FCC Radio Exposure

The radiated output power of this device is below the FCC radio frequency exposure limits. Nevertheless, this device should be used in such a manner that the potential for human contact during normal operation is minimized. This device has been evaluated for and shown compliant with the FCC RF Exposure limits under mobile exposure conditions (antennas are greater than 20 cm from a person’s body). This device cannot be co-located with any other transmitter unless approved by FCC.

This product does not contain any user serviceable components. Any unauthorized product changes or modifications will invalidate the warranty and all applicable regulatory certifications and approvals.

FCC Interference Statement

This equipment generates, uses, and can radiate radio frequency energy and, if not installed and used in accordance with the manufacturer's instruction manual, may cause interference with radio and television reception. This equipment has been tested and found to comply with the limits for a Class B digital device pursuant to Part 15 of the FCC Rules.

This device complies with Part 15 of the FCC Rules. Operation is subject to the following two conditions:

1. This device may not cause harmful interference.
2. This device must accept any interference received, including interference that may cause undesired operation.

This wireless adapter generates, uses, and can radiate radio frequency energy. If the wireless adapter is not installed and used in accordance with the instructions, the wireless adapter may cause harmful interference to radio communications. There is no guarantee, however, that such interference will not occur in a particular installation. If this wireless adapter does cause harmful interference to radio or television reception (which can be determined by turning the equipment off and on), the user is encouraged to try to correct the interference by taking one or more of the following measures:

- Reorient or relocate the receiving antenna of the equipment experiencing the interference.
- Increase the distance between the wireless adapter and the equipment experiencing the interference.
- Connect the equipment to an outlet on a circuit different from which the receiver is connected.
- Consult the dealer or an experienced radio/TV technician for help.

Canada

Industry Canada (IC) Notices

This product complies with Industry Canada RSS-210.

This device complies with Industry Canada license-exempt RSS standard(s). Operation is subject to the following two conditions: (1) this device may not cause interference, and (2) this device must accept any interference, including interference that may cause undesired operation of the device.

Under Industry Canada regulations, the radio transmitter(s) in this device may only operate using an antenna of a type and maximum (or lesser) gain approved for the transmitter by Industry Canada. To reduce potential radio interference to other users, the antenna type and its gain should be so chosen that the equivalent isotropically radiated power (e.i.r.p.) is not more than that necessary for successful communication.

Avis d’Industry Canada (IC)

Cet appareil est conforme aux norme RSS210 d’Industrie Canada.

Cet appareil est conforme aux normes d'exemption de licence RSS d’Industry Canada. Son fonctionnement est soumis aux deux conditions suivantes : (1) cet appareil ne doit pas causer d’interférence et (2) cet appareil doit accepter toute interférence, notamment les interférences qui peuvent affecter son fonctionnement.

Conformément aux réglementations d'Industry Canada, les émetteurs radio de cet appareil ne peuvent fonctionner qu’à l’aide d'une antenne dont le type et le gain maximal (ou minimal) pour ces émetteurs – transmetteurs sont approuvés par Industry Canada. Pour réduire le risque d’interférence éventuelle pour les autres utilisateurs, le type et le gain de l’antenne doivent être choisis de manière à ce que la puissance isotrope rayonnée équivalente (p.i.r.e.) minimale nécessaire à une bonne communication soit fournie.

EU Regulatory Statements

cs	Česky [Czech]	*National Instruments* tímto prohlašuje, _e tento NI myRIO-1900 je ve shodě se základními po_adavky a dalšími příslušnými ustanoveními směrnice 1999/5/ES.
da	Dansk [Danish]	Undertegnede *National Instruments* erklćrer herved, at fřlgende udstyr NI cDAQ-9191 overholder de vćsentlige krav og řvrige relevante krav i direktiv 1999/5/EF.
de	Deutsch [German]	Hiermit erklärt *National Instruments*, dass sich das Gerät NI myRIO-1900 in Übereinstimmung mit den grundlegenden Anforderungen und den übrigen einschlägigen Bestimmungen der Richtlinie 1999/5/EG befindet.
et	Eesti [Estonian]	Käesolevaga kinnitab*National Instruments* seadme NI myRIO-1900 vastavust direktiivi 1999/5/EÜ põhinõuetele ja nimetatud direktiivist tulenevatele teistele asjakohastele sätetele.
en	English	Hereby, *National Instruments*, declares that this NI myRIO-1900 is in compliance with the essential requirements and other relevant provisions of Directive 1999/5/EC.
es	Español [Spanish]	Por medio de la presente *National Instruments* declara que el NI myRIO-1900 cumple con los requisitos esenciales y cualesquiera otras disposiciones aplicables o exigibles de la Directiva 1999/5/CE.
el	Ελληνική [Greek]	ΜΕ ΤΗΝ ΠΑΡΟΥΣΑ *National Instruments* ΔΗΛΩΝΕΙ ΟΤΙ NI myRIO-1900 ΣΥΜΜΟΡΦΩΝΕΤΑΙ ΠΡΟΣ ΤΙΣ ΟΥΣΙΩΔΕΙΣ ΑΠΑΙΤΗΣΕΙΣ ΚΑΙ ΤΙΣ ΛΟΙΠΕΣ ΣΧΕΤΙΚΕΣ ΔΙΑΤΑΞΕΙΣ ΤΗΣ ΟΔΗΓΙΑΣ 1999/5/ΕΚ.
fr	Français [French]	Par la présente *National Instruments* déclare que l'appareil NI myRIO-1900 est conforme aux exigences essentielles et aux autres dispositions pertinentes de la directive 1999/5/CE.
it	Italiano [Italian]	Con la presente *National Instruments* dichiara che questo NI myRIO-1900 è conforme ai requisiti essenziali ed alle altre disposizioni pertinenti stabilite dalla direttiva 1999/5/CE.
lv	Latviski [Latvian]	Ar šo *National Instruments* deklarē, ka NI myRIO-1900 atbilst Direktīvas 1999/5/EK būtiskajām prasībām un citiem ar to saistītajiem noteikumiem.
lt	Lietuvių [Lithuanian]	Šiuo *National Instruments* deklaruoja, kad šis NI myRIO-1900 atitinka esminius reikalavimus ir kitas 1999/5/EB Direktyvos nuostatas.
nl	Nederlands [Dutch]	Hierbij verklaart *National Instruments* dat het toestel NI myRIO-1900 in overeenstemming is met de essentiële eisen en de andere relevante bepalingen van richtlijn 1999/5/EG.
mt	Malti [Maltese]	Hawnhekk, *National Instruments*, jiddikjara li dan NI myRIO-1900 jikkonforma mal-htigijiet essenzjali u ma provvedimenti ohrajn relevanti li hemm fid-Dirrettiva 1999/5/EC.
hu	Magyar [Hungarian]	Alulírott, *National Instruments* nyilatkozom, hogy a NI myRIO-1900 megfelel a vonatkozó alapvetõ követelményeknek és az 1999/5/EC irányelv egyéb elõírásainak.

Polski [Polish]	Niniejszym *National Instruments*. oświadcza, że NI myRIO-1900 jest zgodny z zasadniczymi wymogami oraz pozostałymi stosownymi postanowieniami Dyrektywy 1999/5/EC.
Português [Portuguese]	*National Instruments* declara que este NI myRIO-1900 está conforme com os requisitos essenciais e outras disposições da Directiva 1999/5/CE.
Slovensko [Slovenian]	*National Instruments* izjavlja, da je ta NI myRIO-1900 v skladu z bistvenimi zahtevami in ostalimi relevantnimi določili direktive 1999/5/ES.
Slovensky [Slovak]	*National Instruments* týmto vyhlasuje, _e NI myRIO-1900 spĺňa základné po_iadavky a všetky príslušné ustanovenia Smernice 1999/5/ES.
Suomi [Finnish]	*National Instruments* vakuuttaa täten että NI myRIO-1900 tyyppinen laite on direktiivin 1999/5/EY oleellisten vaatimusten ja sitä koskevien direktiivin muiden ehtojen mukainen.
Svenska [Swedish]	Härmed intygar *National Instruments* att denna NI myRIO-1900 står I överensstämmelse med de väsentliga egenskapskrav och övriga relevanta bestämmelser som framgår av direktiv 1999/5/EG.
Íslenska [Icelandic]	Hér með lýsir *National Instruments* yfir því að NI myRIO-1900 er í samræmi við grunnkröfur og aðrar kröfur, sem gerðar eru í tilskipun 1999/5/EC.
Norsk [Norwegian]	*National Instruments* erklærer herved at utstyret NI myRIO-1900 er i samsvar med de grunnleggende krav og øvrige relevante krav i direktiv 1999/5/EF.

Note Refer to the Declaration of Conformity (DoC) for this product for any additional regulatory compliance information. To obtain the DoC for this product, visit `ni.com/certification`, search by model number or product line, and click the appropriate link in the Certification column.

Singapore

Complies with
IDA Standards
DA105692

Taiwan R.O.C.

低功率電波輻性電機管理辦法
第十二條經型式認證合格之低功率射頻電機，非經許可，公司、商號或使用者均不得擅自變更頻率、加大功率或變更原設計之特性及功能。
第十四條低功率射頻電機之使用不得影響飛航安全及干擾合法通信；經發現有干擾現象時，應立即停用，並改善至無干擾時方得繼續使用。
前項合法通信，指依電信規定作業之無線電信。低功率射頻電機須忍受合法通信或工業、科學及醫療用電波輻射性電機設備之干擾。

Mexico

La operación de este equipo está sujeta a las siguientes dos condiciones:
1) es posible que este equipo o dispositivo no cause interferencia perjudicial y
2) este equipo debe aceptar cualquier interferencia, incluyendo la que pueda causar su propia operación no deseada.

Brazil

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Brasil-Aviso da Anatel

Este equipamento opera em caráter secundário, isto é, não tem direito a proteção contra interferência prejudicial, mesmo de estações do mesmo tipo, e não pode causar interferência a sistemas operando em caráter primário.

Warranty

For customers other than private individual users in the EU: The NI myRIO-1900 is warranted against defects in materials and workmanship for a period of one year from the date of shipment, as evidenced by receipts or other documentation. National Instruments will, at its option, repair or replace equipment that proves to be defective during the warranty period. This warranty includes parts and labor.

For private individual users in the EU: Based on your statutory rights, National Instruments will—through its distributor—cure defects in materials and workmanship within two years from delivery.

Where to Go for Support

The National Instruments Web site is your complete resource for technical support. At ni.com/support you have access to everything from troubleshooting and application development self-help resources to email and phone assistance from NI Application Engineers.

A Declaration of Conformity (DoC) is our claim of compliance with the Council of the European Communities using the manufacturer's declaration of conformity. This system affords the user protection for electromagnetic compatibility (EMC) and product safety. You can obtain the DoC for your product by visiting ni.com/certification. If your product supports calibration, you can obtain the calibration certificate for your product at ni.com/calibration.

National Instruments corporate headquarters is located at 11500 North Mopac Expressway, Austin, Texas, 78759-3504. National Instruments also has offices located around the world to help address your support needs. For telephone support in the United States, create your service request at ni.com/support and follow the calling instructions or dial 512 795 8248. For telephone support outside the United States, visit the Worldwide Offices section of ni.com/niglobal to access the branch office Web sites, which provide up-to-date contact information, support phone numbers, email addresses, and current events.

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