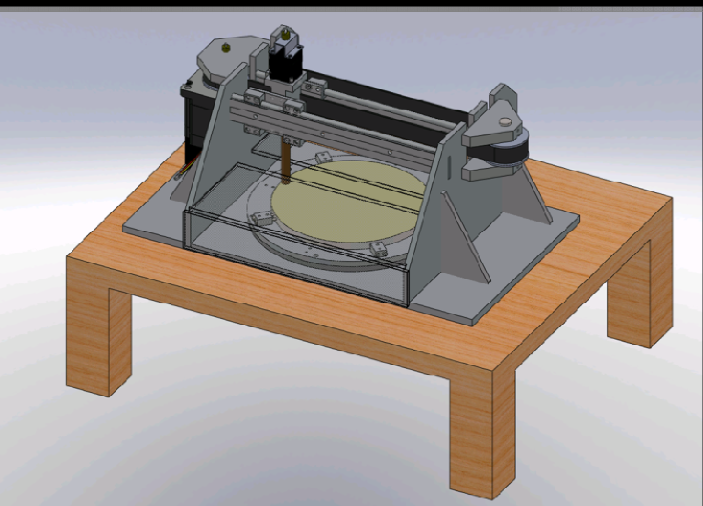
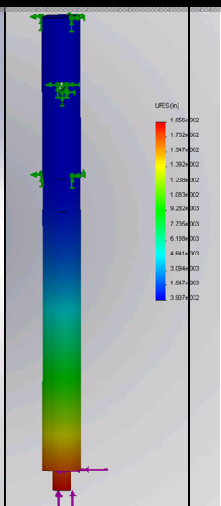
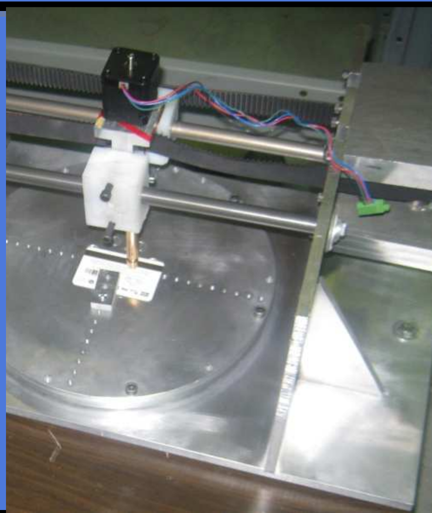
450
;ign and
nufacturing III
2009

ometer







pton

by

:reutzberg

resko)

:y of Michigan

nber, 2009

Contents

Executive Summary

Companies are often interested in reducing the friction and extending the lifetime of their products so that their products are viewed as more reliable, higher quality, and more efficient than competing products. This is often done by studying the friction and wear behavior of materials under different conditions through the use of tribometers and using the information gathered to determine the materials, lubrication, and/or the conditions needed to optimize the design. The tribometers available today are expensive and often only perform tests under one type of motion at a time. The goal of this project is to develop a less expensive tribometer that will not only perform standard test motions but will also give the user the ability to program a more complex path that the tribometer will follow so that the expected wear and friction in a design can be more accurately predicted.

In order to better understand the task at hand, our group has researched tribometers in depth. We have looked at current designs, relevant patents and current technology, and we have conducted interviews with various experts. From this research, we were able to determine corresponding specifications and requirements that we should meet. Once specifications were determined, we performed analyses to determine what functions are most important to the design of a tribometer and how the specifications and functions interact with one another. This was done so that the team could acquire a better understanding of the task at hand.

We then began generating different concepts that could potentially perform the required tasks. These concepts were directed only to one specific task at a time. Once our team knew the entire design space had been explored for each function, we picked some of the solutions most likely to work and compared them. Through the use of Pugh carts, the solutions that best met the sponsor's needs were determined. After examining how some concepts could impact others and what concepts would not work well with each other, the team created many complete tribometer concepts and evaluated them. Through the use of another Pugh chart, we selected a design that we believed to be the best. After numerous reiterations of getting feedback on this design, redesigning, and performing substantial engineering analysis, the team believes they have developed the design that best meets the sponsor's requirements.

The final design includes two stepper motors that move the pin and disk. One motor spins a gear which spins the disk while another motor drives a belt system that moves the pin along linear slides. Cross-wear testing can be done though the use of both motors. The normal load is applied through the use of a third motor which pushes the pin against the disk using a screw-drive and springs to accommodate small disk vibrations. This tribometer monitors the forces on the pin with strain gages and turns the screw drive accordingly to maintain a constant normal force. The tribometer keeps track of how far the pin moves in order to keep this force constant –allowing the tribometer to track the total wear of the pin and disk real time.

The team has manufactured a prototype that can be used to validate that the design works. The team had planned to validate their design but ran out of time due to a lot of circumstances that were outside of their control. The team recommends that the prototype be validated and improved upon in areas that are found to not work as predicted. We also recommend that an environmental system be design and built so that the tribometer can meet the ASTM standards required.

Abstract

Wear and friction reduce product lifetimes and increase the energy required in many processes. A tribometer is a device that tests the wear and friction behavior of components and materials. Two of the most common types of tribometers are the pin-on-disk, which places a pin on a rotating plate, and the linear reciprocating, which oscillates the pin back and forth. Current tribometers are expensive and limited to a single type of experiment. The goal of this project is to develop a cheap, compact tribometer that can perform a wide variety of tests in a repeatable and reliable fashion.

History and Background

Tribology deals with friction, lubrication, and wear associated with the interaction of moving surfaces. Humanity has been interested in these properties since prehistoric times. Friction, in fact, gave birth to civilization when someone realized that rubbing two sticks together could yield fire. The discovery of the wheel and the axle as well as the use of animal fat and plant-based lubricants followed, but friction itself wasn't studied until Leonardo da Vinci conceived the two basic laws in 1495. These laws were rediscovered by Guillaume Amontons in 1699 [1]. Charles August Coulomb would go on to refine Amontons's ideas and established the second law of friction in 1785. The Amontons-Coulomb Law states that frictional force is proportional to compressive force [2]. Isaac Newton would add the third law of friction when he stated that kinetic friction was independent of speed. But the understanding of why these laws are true did not occur until 1950 when Phillip Bowden and David Tabor determined that the area of contact between materials is limited to asperities on the surfaces [3].The contact area between asperities is only a small percentage of the total surface area. As the normal force increases, more asperities come into contact with each other which leads to an increase in the true contact area. The frictional force has been shown to be dependent on the increase of contact between asperities. Bowden and Tabor would go on to add that the adhesive interactions between asperities also affect the friction force. Today, the study of friction, lubrication and wear is an important aspect of countless professions and industries.

The devices which are used to perform many tests on the surface interactions between materials are called tribometers. Currently, the two most common types of tribometers are the pin-on-disk tribometer and linear reciprocation tribometer. Both of these tests involve moving a rounded pin with respect to a stationary flat plate or disk with which the pin is in perpendicular contact with. By applying a constant normal force to the pin and measuring the forces on the pin throughout the test, experimenters can determine how the coefficient of friction changes over time. By measuring the pin and disk, the wear of the materials as a function of time can also be determined. The difference in the two common tests mentioned come from the type of motion. The pin-on-disk device places a pin on a rotating plate while the linear reciprocation device has a pin which oscillates back and forth in a straight line. By varying the applied force, test speed, distance, and ambient conditions, along with the addition of lubrication, experimenters can evaluate their designs and make changes to their design in order to get better performance and longer life times.

Motivation

When testing on a material or component is performed both the linear and pin-on-disk tribometers are frequently used. This is because the wear and friction behavior of a material can differ depending on the path of the pin on these devices. Tests in which the path of the pin crosses in different directions yield further wear behavior. Today, thorough research of a specific material's wear behavior requires multiple test devices. A single tribometer with the ability to perform all of the necessary tests will save researchers time and money and will broaden and enhance our understanding of friction and wear behavior of materials. Our sponsor, Professor Gordon Krauss of the University of Michigan, performs research in the field of tribometry and has tasked us with designing and manufacturing a low-cost, multi-test tribometer to be used in his research.

Research

In depth research of both tribometry and tribometers was required because no member of our project group had any personal experience with how tribometers worked or why they were used. As part of our research, we sought out a wide range of resources on tribometers including several meetings with the project sponsor, patents, past and current designs, historical test data, and various test standards.

Interviews

In order to gain a better understanding of the requirements and to help develop specifications for the new tribometer, various face-to-face interviews were conducted. The interviews were used as one of the main sources of design requirements.

The first interview was conducted with the sponsor of the project, Gordon Krauss[4]. At this interview our group discussed what we had found in the various literature searches we conducted and how they compared to what Krauss wanted to see in a new tribometer design. Krauss listed of several specifications, and requirements of what was needed in his ideal tribometer. He also recommended that we determine several specifications based on current, and old designs. The interview was a good starting point for the project, and communication will continue to ensure we stay on budget and task.

A second interview was conducted on September 25, 2009 at the Ford Dearborn Proving Grounds with Dr. Arup Gangopadhyay, the technical leader of lubrication science for Ford's Chemical Engineering Department [14]. Gangopadhyay is responsible for determining ways to reduce friction and wear in cars, and has a laboratory with many tribometers to aid in his research. At this interview we were able to see working tribometers, including one that Ford custom built. We discussed what types of information Gangopadhyay is interested in acquiring when he performs tests and what type of conditions he usually tests under. He did not express any difficulties that he had had with the tribometers.

A third interview was conducted with John Baker at the University of Michigan [15]. Baker offered his help to ME 450 teams that would be using mechatronics in their projects. This interview was conducted to get Baker's advice on how to perform the motion control and collect the data that is required in our tribometer. He informed us of several different ways to control the motion ranging from stepper motors to DC motors with encoders. He also gave us advice on how to build our own force measuring device using strain gauges. Lastly, Baker advised us to split up the tasks required of our tribometer amongst ourselves and have one person become an expert on each part.

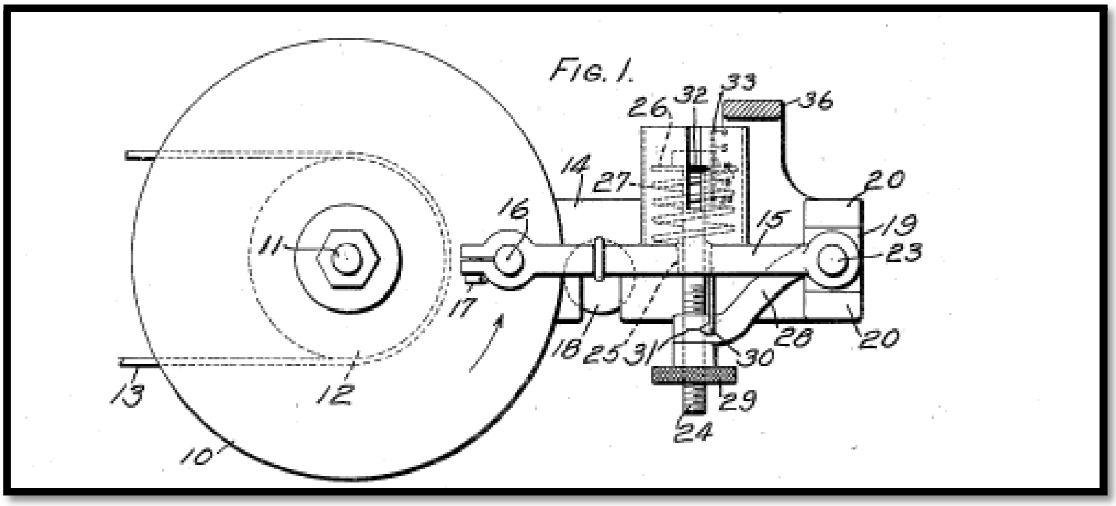
The most recent interview our team completed was with Sean Hickman, the Pinckney Community High School Robotics Teacher [16]. Hickman teaches a class in which his students often have to complete projects which require building a low cost device that can follow a reprogrammable path. He showed us several devices that used gantry type systems to achieve this variability. Some of these devices were run with rack and pinion while others were run with a ball screw type set up. These systems were also controlled in different manners, some used motor controllers that came with the motors being used while other systems were monitored with an encoder and controlled by a program logic controller (PLC).

Patents

The team also investigated previous designs and specifically patents of various applicable designs and subsystems. Using the website *patentstorm.us* [18], several patents were found that discussed technology which is related to the scope of this project.

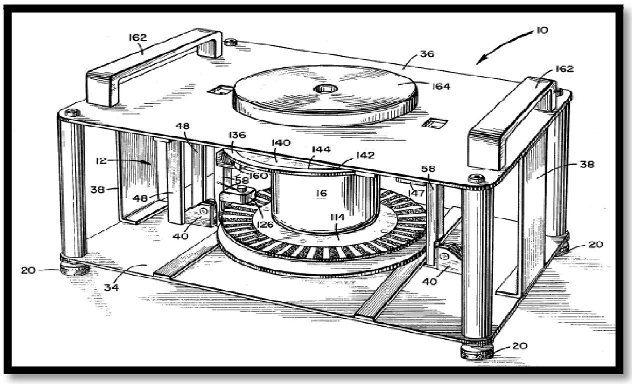
Patent 1534014[5] is the first wear testing device that is documented in the patents that we could find. Figure 1, page 5, shows the disk system, and specifically the spring and mechanical measurement system used to measure the frictional force.

Figure 1: Patent 1534014 showing mechanical measurement device[5]



Another interesting patent found is a pin on disk tribometer that illustrates a simple motor control using an encoder type device. The encoder motor control method may be something that can be incorporated in our design. This device also meets the need of being portable, and has handles built into the design (part 162). Patent 4051713 is illustrated in Figure 2, below.

Figure 2: Patent 4051713 Table Top and Encoder Design [6]

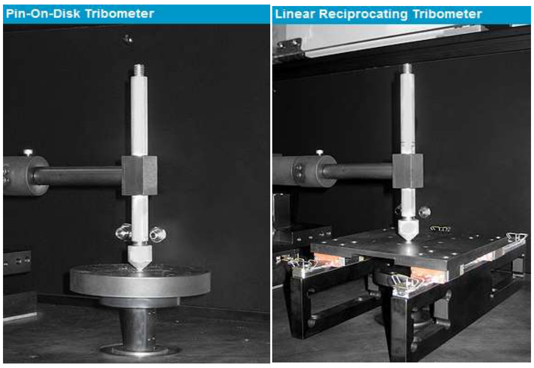


Current Designs

There are a few companies that currently manufacture tribometers, and we decided to conduct research to do benchmarking and reverse engineering. The two main companies researched were Nanovea and CETR because they deal with the same measurement magnitudes that our sponsor has requested. What we found are very expensive, very specific testing devices, that are not user friendly.

The Nanovea tribometer[7], seen in Figure 3(Pg.6), shows multiple test configurations that can be setup and torn down, but cost on the order of 10's of thousands of dollars. Nanovea tribometers meet all the standards by ASTM, and appear to have an excellent data acquisition, but we have no customer reviews on the tribometer to back up the statements on their websites. Nanovea has other features which add onto cost including extreme temperature, and vacuum chambers. We also tried to contact Nanovea via email and phone but no one has returned communication.

Figure 3: Nanovea Tribometer Pin-On-Disk and Linear Reciprocating Setup



CETR[8] currently has two products; a nano/ micro tribometer and a heavy-duty tribometer. The micro tribometer setup can be seen in Figure 4 and is of the same measurement magnitudes that our sponsor has requested. Their device can conduct: Pin on disk, disk on disk, ball on disk, plate on plate, reciprocating and fretting, uni-directional and multi-directional wear, are some of the typical test setups[8]. The problem is this device cost around $80,000 which does not meet our sponsors cost to function ratio. While our sponsor believes this is the most flexible triobometer on the market, he also informed us of some troubles he had switching out tooling and programming the machine to do what he wanted.

Figure 4: CETR Nano/Micro Tribometer Multi Test Station [8]

Standards

When looking at the current tribometers on the market, we realized their specification sheets all stated they met certain ASTM specifications for each test. We read each standard that was mentioned in these specification sheets. We found that two of these standards were within the scope of the project. This section outlines important information found within these standards.

ASTM G 99

This standard outlines the pin-on-disk apparatus, test methods, procedure, and what needs to be reported in order to meet the standard. This section will outline what we found to be relevant to the scope of this project.

This standard first describes the test method. The pin-on-disk test involves two specimen, a pin with a radius tip (typical r=1 to 10mm) and a flat circular disk (typical r= 15 mm to 50mm with thickness of 2 to 10mm and suface finish of .8µm arithmetic average or less). The pin is held perpendicular to the disk (±1°) and the machine causes one of the specimens to move so that the sliding path is a circle on the disk surface. The speed at which this happens is typically in the range of .3 to 3 rad/s and must be held constant (±1% of rated full load motor speed).

The pin is pressed onto the disk at a constant load during the test. This is typically done with attached weights but can be done by other means such as hydraulics or pneumatics. The standard states that if the pin is pressed against the disk by means of an arm or lever, that the pivot of the arm be "located in the plane of the wearing contact to avoid extraneous loading forces due to the sliding friction. The pin holder and arm must be of substantial construction to reduce vibration during the test."

The resulting wear from this is to be reported by measuring the dimensions of both specimens before and after the test (2.5 µm or better), or by measuring the weight of both specimens before and after the test (.1mg or better). The standard advises against monitoring the continuous wear depth by using data from position-sensing gages. The wear results must be reported with the corresponding speed, load, and sliding distance.

ASTM G 133

This standard outlines the linearly reciprocating apparatus, test method, procedure, and what needs to be reported in order to meet the standard. This section will outline what we found to be relevant to the scope of this project.

This standard states that this test involves two specimens, a flat specimen and a spherically ended specimen. The two specimens must move relative to each other in a "linear, back and forth sliding motion, under a prescribed set of conditions." These conditions are outlined in Figure 5(pg.8) below.

This standard requires that the relative humidity (±3%), temperature in degrees Celsious, and the statically applied loads (±2%) be measured. The standard also requires that a tension-compression load cell or similar force-sensing device be used to measure the friction forces during the test. This can be recorded using a strip-chart-recorder or a computerized data acquisition system. This standard also advises that the harmonic frequencies characteristic of the test machine be avoided.

Figure 5: ASTM G 133 Procedures for Linear Reciprocating Test

8.5.1 The two testing procedures are as follows.

8.5.1.1 *Procedure A*—Unlubricated wear testing at room temperature.

(*1*) Pin tip radius, 4.76 mm (3/16 in.),
(*2*) Normal force, 25.0 N,
(*3*) Stroke length, 10.0 mm,
(*4*) Oscillating frequency, 5.0 Hz,
(*5*) Test duration, 16 min 40 s (sliding distance 100 m),
(*6*) Ambient temperature, 22 ± 3°C,
(*7*) Relative humidity, 40 to 60 %, and
(*8*) Lubrication, none applied.

8.5.1.2 *Procedure B*—Lubricated wear testing at elevated temperature.

(*1*) Pin tip radius, 4.76 mm (3/16 in.),
(*2*) Normal force, 200.0 N,
(*3*) Stroke length, 10.0 mm,
(*4*) Oscillating frequency, 10.0 Hz,
(*5*) Test duration, 33 min 20 s (sliding distance 400 m),
(*6*) Temperature, 150 ± 2°C,
(*7*) Relative humidity, 40 to 60 %, and
(*8*) Lubrication, full immersion under the selected lubricant (see Note 2).

NOTE 2—This procedure requires full-immersion lubrication. If other methods, such a controlled drip feeding system, are used to simulate certain applications, the provisions of 8.6 will apply.

Test Performance/Results

Various procedures and sample results were studied as part of our literature review. This was done in order to see what our design could incorporate that couldease the task of the experimenter and to determine what kind of consistency between tests is expected.

Prior to the test performance, certain measures should be taken to ensure accurate results will be obtained. These measures include cleaning all parts and test specimens with lab tissues and different types of alcohol, the use of cotton gloves and tongs to avoid contamination, and the calibration of the force recorders with known weights. After the parts are assembled, the lubricant is heated to the required temperature. The temperature is measured by a double shielded thermocouple which relays the information to the heating system in order to maintain the desired temperature. Some tests proceeded to break in the ball or pin by starting with small normal forces and incrementally increasing them. This was done to prevent damage to the test specimens that may occur at the onset of testing when high loads are used [11, 12].

Below are sample results from tests performed according to ASTM Standard G 133 – Test Method for Linearly Reciprocating Ball-on-Flat Sliding Wear. Figure 6 shows typical values of the friction coefficient for Silicon Nitride while Figure 7 displays wear volume results. Eight tests resulted in an average coefficient of 0.81 with a standard deviation of 0.014 and an average wear volume of 0.543 mm^3 with a standard deviation of 0.189. These results will help give us a better idea of what to expect from our device. These figures indicate that the wear volume can be more variable than the friction coefficient and also provide us with some values we can use as benchmark comparisons if we were to test Silicon Nitride.

Fig. 6: ASTM Standard G 133 Sample Test Results

Friction Coefficient Results for Silicon Nitride Specimens Using Procedure A

NOTE—Coefficient of variation = ±1.8 %; 95 % confidence limits = 0.04.

	Laboratory A		Laboratory B	
Test Number	Friction Coefficient,μ	Deviation from Average, μ	Friction Coefficient,μ	Deviation from Average, μ
1	0.800	−0.005	0.715	−0.075
2	0.800	−0.005	0.854	0.064
3	0.803	−0.002	0.800	0.010
4	0.808	0.003	0.817	0.027
5	0.800	−0.005	0.742	−0.048
6	0.781	−0.024	0.811	0.021
7	0.820	0.015	0.716	−0.074
8	0.828	0.023	0.862	0.072
Average	0.81		0.79	
Standard deviation	0.014		0.058	
Coefficient of variation	±1.8%		±7.4%	
95% confidence limit	0.04		0.16	

Fig. 7: ASTM Standard G 133 Sample Test Results

Wear Volume Results for Silicon Nitride Specimens Using Procedure A

NOTE—Coefficient of variation = ±34.7 %; 95 % confidence limits = 0.53.

	Laboratory A		Laboratory B	
Test Number	Wear Volume, mm^3	Deviation from Average, mm^3	Wear Volume, mm^3	Deviation from Average, mm^3
1	0.746	0.203	0.394	0.020
2	0.611	0.068	0.513	0.139
3	0.507	−0.036	0.276	−0.098
4	0.635	0.092	0.325	−0.049
5	0.293	−0.250	0.427	0.053
6	0.229	−0.314	0.388	0.014
7	0.619	0.076	0.379	0.005
8	0.707	0.164	0.287	−0.087
Average	0.543		0.374	
Standard deviation	0.189		0.078	
Coefficient of variation	±34.7%		±20.8%	
95% confidence limit	0.53		0.22	

It is important to note that the results shown above indicate that results from tribometer to tribometer do not agree completely. These results indicate that measured quantities may vary significantly from test to test. In order to understand why these values vary so often, it is imperative that the tribometer that tests are being run on is calibrated and characterized correctly.

Control Systems and Data Acquisition

This research looked at different ways to control the entire system and ways to receive the data from required tests the specifications ask for. Controllers and data acquisition could be separate in the case of stepper motors, or could be integrated together if DC motors were used. Motors all need some sort of amplification to take a given signal and turn it into a bigger power to drive the motor. Controllers send data to amplifiers via protocols such as pulse width modulation (PWM) or just pulses in the case of a stepper motor. National Instruments sells systems that can act as both a controller and a data acquisition device, but we will need a nearly real time sampling rate to ensure we maintain certain specifications.

The use of electronics is also common place when acquiring data. There are also old fashioned ways of making measurements mechanical, like the patents previously mentioned, but we believe these devices will take up an unnecessary amount of table space and will make it difficult for the user to analyze the data. Because the University of Michigan provides access to many National Instrument hardware and software, National Instruments will be one of the first choices in data acquisition systems (DAQ). Sampling sensors, then saving that data to computers has become the standard practice when making measurements because of the flexibility of a modern computer according to John Baker. He also recommended National Instruments because of LabVIEW which makes interfacing a user interface easy with acquiring various data. [15]

Stepper motors were also mentioned as a good way to create the motion needed for this project. Stepper motors have their own version of amplifiers, wiring, and control systems. The motor hooks up to a stepper amp, which then hooks up to a stepper controller. The controller interfaces with a computer that has software preprogrammed to import path files (such as a .DXF).

Project Requirements and Engineering Specifications

The main goal of the Multi-Test Tribometer is to be able to perform rotational, linear reciprocating, and custom combination movement patterns; however, there are many more requirements which the device must achieve in order to have successful tests. The tribometer must maintain a constant normal force, determine the coefficient of friction between the pin and disk, measure wear of the pin and disk, maintain constant temperature and humidity levels, measure electrical contact resistance, and allow tests to be conducted in a lubricant bath.

A big step in the design process is to translate the products requirements and customer needs to engineering targets. Each product requirement must be analyzed to determine how they can be expressed in a quantifiable manner. In this manner engineering parameters can be formed to help us further understand and attack the problem at hand. The next step is deducing these engineering parameters into specific target values. A couple of questions must be considered during this process. What is the relative importance of each project requirement, and how will its importance affect the target value? How will our target values compare with those of competitive products? Dissecting these questions one can get a better look into how the target values can be determined.

Sponsor Requirements

Our sponsor, Gordon Krauss, outlined certain product requirements that he would like our product to meet. First, the tribometer must take and store real time data of the normal and frictional forces, the electrical contact resistance of the ball bearing, and the amount of wear resulting on the ball bearing and test material. The amount of cycles and total distance of the test must also be determined. A constant temperature and humidity of the testing atmosphere must be maintained. Access to the testing area should be provided and all tests should be performed in a safe manner.

Engineering Specifications

Once we obtained some the basic requirements from our sponsor, we were able to conduct some research to evaluate more specific engineering targets. Shown in Table 1 is the complete list of specifications that were determined from demands of our sponsor, ASTM standards, and competitor products.

Table 1: Engineering Specifications

Specifications	Values
Linear Testing Speed m/s (ft/s)	.01+-1* (.03-3.28)
+Rotational Testing Speed rad/s (RPM)	.3-20π (2.9-600)
*Measurement Resolution of distance and force (%)	1
*Control Precision mm (in.)	1 (.04)
*Test Specimen Size Diameter mm (in.)	25.4-254 (1–10)
+Test Specimen diameter resolution mm (in.)	.025 (.001)
*Volume cm (in.)	60.96x60.96x91.44 (24x24x36)
*Normal Load Range N (lb)	.1-200 (.02-45)
+Friction Coefficient Measurement	.01-1
*Stroke Length cm (in.)	0-25.4 (0-10)
*Data Acquisition Frequency (kHz)	20
*Operating Temperature Range °C (°F)	0-150 (32-302)
*Temperature Resolution °C (°F)	5 (5)
*Relative Humidity Range (% water)	0-100
+Relative Humidity Resolution(%)	3
°Electrical Contact Resistance Range (Ohms)	0-1000
°Electrical Resistance Resolution(ohms)	1
*Ball Sizes mm (in.)	1.59, 3.18, 6.35 (1/16 , 1/8, 1/4)
+Wear range mm (in.)	0-3.18 (0-1/8)
+Wear resolution in (μm)	9.8 x 10^{-5} (2.5)
*Total Cycles	0-1M
*Distance Measurement in (mm)	±.04 (±1)
+Perpendicularity of the Disk to the pin (degrees)	± 1

*Sponsor requested

+ASTM standard

°Competitor specification

Quality Function Deployment (QFD):

To analyze the specifications determined for the tribometer, our team used a Quality Function Deployment (QFD) chart. This chart allowed us to assign weights to requirements and correlate how the specifications related to these requirements. By multiplying the number weighted to a requirement and the number rating of how much a specification helps to meet the said requirement, we are able to determine what specifications are most critical to meeting the optimum design. We can use this chart later when comparing designs. The completed QFD chart is in Appendix B.

We used the results from the QFD to determine what specifications we should focus most of our time on while developing the alpha design. These include the ability to perform both test methods, minimizing the cost wherever possible, the ability to measure the distance of wear accurately, measuring the forces accurately, and the ability to control the speed of the tribometer to within the specifications of 1 m/s ±1%.

The QFD chart also allows us to view tradeoffs of specifications. For example, we can note on the QFD that if the tribometer's vibrations increase the measurement resolution of disk wear will decrease. By seeing how certain designs to perform a function may positively or negatively interact with other functions, we can get a better idea of how to select the best design.

The QFD also compares current competitor's specifications to our prototype's. The performance of many of the functions on the prototype are either comparable to or exceeding our competitors product. Some of the performance levels of our product do not compare well the competitors. However, considering that our alpha design will be able to perform both test methods, we feel we have a large advantage over current products.

Concept Generation

The first step we took towards generating concepts was to develop a functional decomposition for the Multi-Test Tribometer. The functional decomposition can be seen in Appendix D. Creating a functional decomposition allowed us to establish and assess exactly what tasks and capabilities our device will need to be able to accomplish. The functional decomposition also made clear that the tribometer's functions could be performed by separate components or modules. There would be a component for applying the normal force, a component to perform rotational motion, a component to measure the wear, and so on. The final assembly of all of these parts would then constitute the Multi-Test Tribometer.

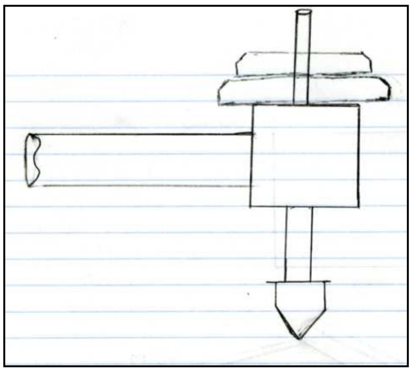
Instead of generating concepts for the finished tribometer assembly, we decided to initially focus on generating multiple concepts for each component and then analyzing which designs would work well together. Each member of our team came up with multiple concepts for each module of the tribometer and then presented their ideas to the group. Individual brainstorming was done prior to group brainstorming to prevent any single person's ideas from dominating the discussion or excessively influencing the group's train of thought. We then had a team brainstorm in which everyone's ideas were written down and discussed as a group. The team brainstorm led to additional designs and concept generation. A complete list of component designs can be found in Appendix E.1. To outline the process used for concept generation, we have included a detailed write up showing how this was done for the function of applying normal force. This is shown in the following section.

Concepts for the Application of a Normal Force

The tribometer must be able to apply a constant normal force between the pin and the disk. The following ideas were generated to address this requirement.

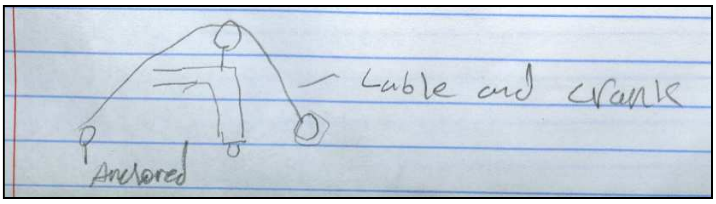
Dead Weight

The pin will be set against the disk using a lever arm and will be free to move in the vertical direction. Calibrated weights will then be placed directly above the pin thus creating a constant normal force of the pin against the disk. The pull of gravity will constantly pull the weight down and the force will not change due to wear in the pin or disk.



Cable with Crank

A cable will be positioned in a grooved slot directly above the pin. The cable will be anchored at one end and attached to a crank at the other. When the crank is turned, the cable will be pulled taught and will force the pin downwards. A motor can be attached to the crank to maintain a constant normal force as wear occurs. This will require feedback from the normal force sensor.

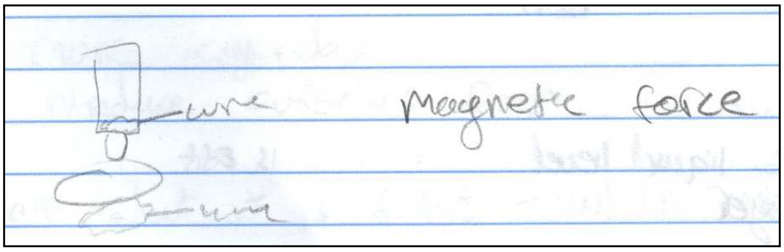


Screw

Similar to the cable with crank concept. The shaft that contains the pin will be threaded on the inside. The pin will screw into the shaft until it touches the disk. From this point on, as the pin is screwed in more, it is forced against the disk thus creating a normal force. A motor with a worm gear will be attached to the pin to perform the action and ensure that a constant normal force is obtained. This will require feedback from the normal force sensor.

Magnets

A magnet placed under the disk will pull the pin against the disk due to magnetic properties of the pin. This will create a normal force against the disk.



Hydraulic / Pneumatic

A hydraulic or pneumatic system will be used to force the pin against the disk using liquid or air pressure.

Spring

A spring will be placed under the disk, and will be compressed to a known length. The pin will be positioned so it is just touching the disk, with no normal force occurring between the two. The spring will then be released and will attempt to reach its uncompressed length, thus forcing the disk against the pin and creating a normal force.

Concept Selection

Pugh charts were then made for each separate module to compare the different designs we generated and to aid us in selecting the best options. These can be found in Appendix E.2. We compiled a list of selection criteria appropriate for our project and then individually scored each design on how well it satisfied the selection goals. If a design satisfied a criterion very well it was given more points, and if it didn't satisfy the criterion it was given less points. Through this process we were able to assign a rank to each design. We then discussed our individual rankings as a group and talked about any noticeable differences in points before settling on a final design ranking for each component. This process was performed for each module. After all the scores had been tallied we organized the design components into a flow chart, located in Appendix F, which allowed us to more easily view how the component designs would work together and which would be more difficult to integrate into an assembled product. Some designs could be eliminated immediately either due to infeasibility of the design (ex: if we would need to manufacture something on a small scale that would be impossible in the shops available to us and we could not just purchase a product on the market) , cost (ex: the cost of a concept was well over the budget given to the team) , or for being too complex for the scope of our project. We generated concepts for the assembled tribometer using different component-design combinations that we thought would work well together. Two base concepts - an inexpensive, simple assembly and a very expensive, highly accurate assembly – were initially generated to give us a better idea of the range of finished products our individual component designs would allow. Multiple assembled concepts were generated that sought to balance cost against an effective device that fulfilled all of the requirements. These assembly concepts were then ranked using the Pugh chart found in Appendix G. Several concepts are displayed and described below.

Table 2: Application of Normal Force - Concepts

	Concept A	Concept B	Concept C	Concept D "Alpha Des."	Concept E "Awesome"	Concept F "Cheap"
Normal Force Application	Dead Weight	Cable w/ crank	Dead Weight	Screw w/ Worm Drive	Pneumatic	Dead Weight
Force Measurement	Multi load cells	Strain Gauge	Multi load cells	Strain Gauges / Load Cell	6-axis load cell	Strain gauge
Wear Measurement	LVDT	LVDT	LVDT	Worm Drive Encoder	White light inferometer	None
Motion Control	Pin all motion	Pin all motion	Pin Recip. Disk Rotate	Pin Recip. Disk Rotate	Pin Recip. Disk Rotate	Pin all motion
Temperature Control	TEC	TEC	TEC	TEC	TEC	None
Humidity Control	Silica Gel	Silica Gel	Humidifier	Humidifier/ Silica Gel	Humidifier / Dehumidifier	None
Disk Holder	Chuck	Screws	Screws	Screws	Chuck	Screws
Pin Holder	Screw / Washer	Screw / Washer	Screw / Washer	Screw / Washer	Screw Sizes	Screw / Washer
Electronic Controls	PCI	PCI	PCI	USB	CRIO	USB
Motors	Stepper	DC	Stepper	DC	DC	DC
Pugh Score	325	330	334	378	327	340

Concept A

This design does not use any of the more complex components and would therefore be easy to use and manufacture. However, its simplicity also results in low reliability and accuracy. All-pin motion would make programming various motion paths easier, but may not achieve the rotational speed requirements. Using dead weights adds to the size and weight of the assembly. Moving a pin with dead weight would be unwieldy and potentially dangerous.

Concept B
Concept B uses the cable with crank method will be lighter than using dead weights. Compared to the screw and worm drive, this method is similar but requires an LVDT to measure the wear. This adds cost and weight. The pin-all motion will again be more versatile and using DC motors instead of steppers will weigh more, but will be more capable of meeting the speed requirements.

Concept C
Concept C is nearly identical to Concept A, but with pin-reciprocating and rotating disk motion rather than all-pin. This results in a less versatile, but safer and more durable design.

Concept D
Our alpha design uses the screw system with an encoder on a worm drive. This method will apply the normal force and simultaneously measure wear. It is much lighter and easier to move the pin compared to using dead weights. Using pin-reciprocating and rotating disk motion will allow the device to meet the speed requirements but will result in less flexible movement compared to all-pin motion for moving the pin in complex paths. A National Instruments USB DAQ will be used with a LabView GUI.

Concept E
If cost was not an issue, this is the device we would make. The components would be mostly bought already made, and our only task would be to assemble the parts. It would be very reliable and highly accurate as well as versatile. However the price would be far too high for our budget and therefore is not feasible.

Concept F
Concept F is the cheapest assembly of components possible. We would manufacture our own force sensors using strain gauges and would control the motion using a USB. This concept would not achieve many of the requirements such as temperature and humidity control, and wear measurement. It would also not be very reliable or accurate.

Alpha Selection
The rankings obtained using the Pugh charts gave us a more detailed illustration of which components would work well together, which wouldn't, and which would have no effect on one another. Several trade-offs were looked at including the motion of the overall system. Pin-all motion was originally the idea that was going to be focused on, but after looking at the mass of the motors and the complexities of controlling such a system, we needed another option. Going back to the Pugh chart, we looked at the next best option which was a pin reciprocating, and a disk spinning. We could accomplish this motion with a simpler mechanism and control system utilizing a 4-bar linear slide system and the spinning disk. Programming this system for the two standard tests will be very simple, but programming for a combined test will be more difficult. The sponsor seemed to favor this trade off, so the choice was made to switch to a pin reciprocating and disk spinning design.

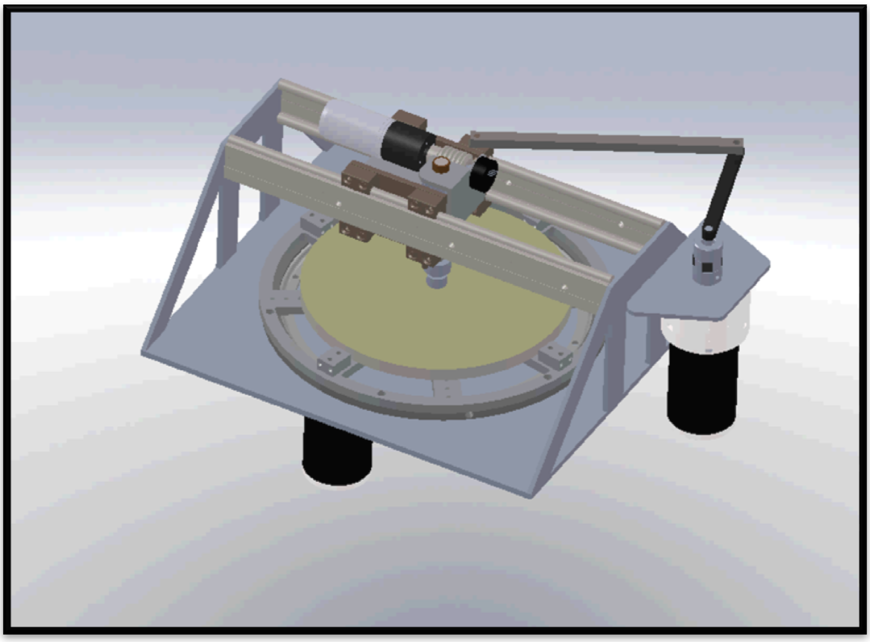
Another trade off looked at was the strain gauges vs. buying force transducers. The force transducers give us an easier way to measure force, but are bulky and expensive. On the other hand mounting strain gauges to the pin will be cheap, but hard to accomplish and a lot of math is involved. Due to the goal of being extremely low budget, we selected using strain gauges. This will be a difficult task and one person, Brian Kirby, will be in charge of force sensing for the project.

Further group discussion and brainstorming led to the selection of an alpha design that took into account all of the previously established module rankings and inter-component relationships that had been generated. This alpha design was selected because it meets the highest priority requirements and has a relatively low cost compared to system performance. It is also optimized based on trade-offs between the different options which were generated. This design was also chosen because we feel comfortable that we can construct and program a working device based on our team's current engineering capabilities and level of knowledge.

Alpha Design

After completing the concept selection, talking to the project sponsor, and performing some preliminary calculations, we believe we have a good initial design to move forward with. The design attempts to incorporate all of what the sponsor has requested, however some of the requests have required some relaxation in the specifications. The design accomplishes the main goal of creating a variable path tribometer, at a low cost, and which can measure accurately and precisely. The final render of the alpha design can be seen in Figure 8. The design was broken up into three main sub assemblies: motion generation, the pin gantry, and environmental control.

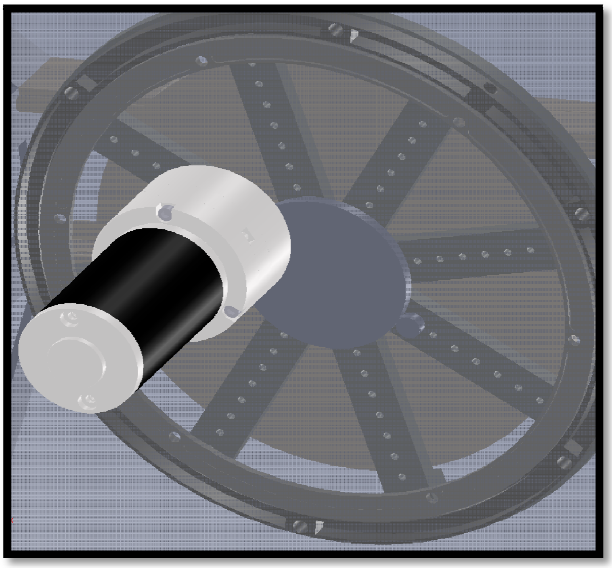
Figure 8: Alpha Design



Motion Control

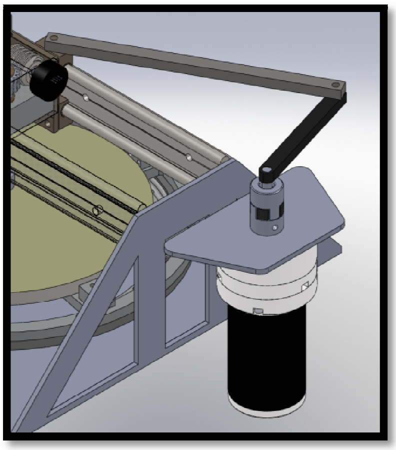
In order to generate the motion needed for the two different "standard" tests, and the variable tests the sponsor wants we will be using two motors to generate two separate motions, a spinning motion and a reciprocating motion. The disk will be mounted on a giant bearing mounted to the base of the table, which will then be geared to an electric motor to create the spinning motion. This replicates the standard pin on disk spin test which has the disk spinning and the pin loaded. The motor is mounted below the table, and drives the disk via a gearing system. The rotational system can be seen in Figure 9.

Figure 9: Rotational Drive Gearing System



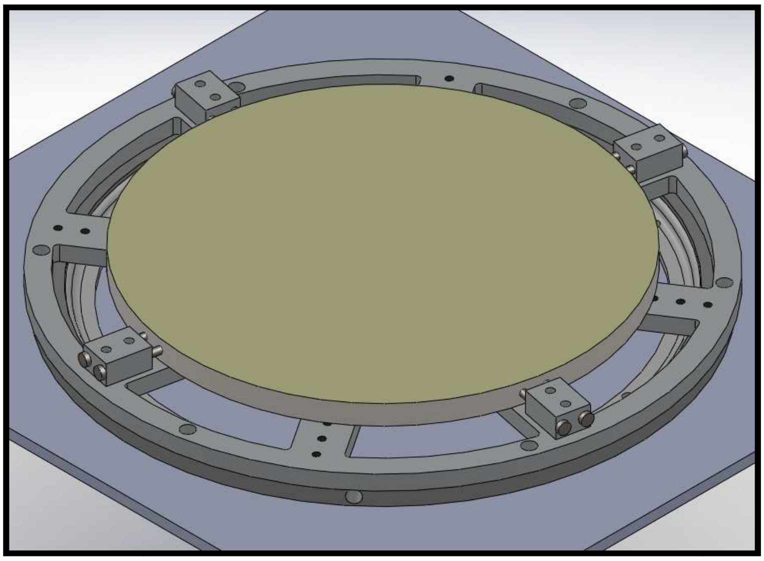
The pin will be mounted on a linear bearing system above the disk to generate a reciprocating motion. An electric motor with a slider crank system will drive the reciprocating pin. Figure 10 shows how the motion is driven and generated using the slider crank system. The system was selected because it simplifies motor movement by incorporating a back and forth motion while only spinning in one direction. A method utilizing a belt drive was also considered, but would require the motor to frequently change directions while also having to overcome the added inertia due to the weight of the applied normal force. In order to reach speeds of 1 m/s on the smallest sample size, accelerations of nearly 16 g's would be required. To achieve constant linear velocity, the motor will have to actively change its rotational velocity throughout the test. To incorporate different sized samples, the user will have to manually adjust the arm length with a nut a bolt to ensure they don't have too long of a stroke length. Simply combining the two different motions will be easier and cheaper to control the system movement for the two standard tests. The variable tests will be harder to program, but it will still be possible to generate variable paths for various testing.

Figure 10: The Linear Drive Crank System



The sample is held down with an adjustable chuck type device. Figure 11 shows how the sample will be held down. This is a very easy to manufacture, cheap, and light weight system, but will be difficult to ensure who balanced the sample is on the sample disk. In order to ensure the sample is centered, a centering pin will be placed in the middle of the chuck device and a small hole will be drilled at the center of the sample. Vibration damping feet for the bearing will also be incorporated.

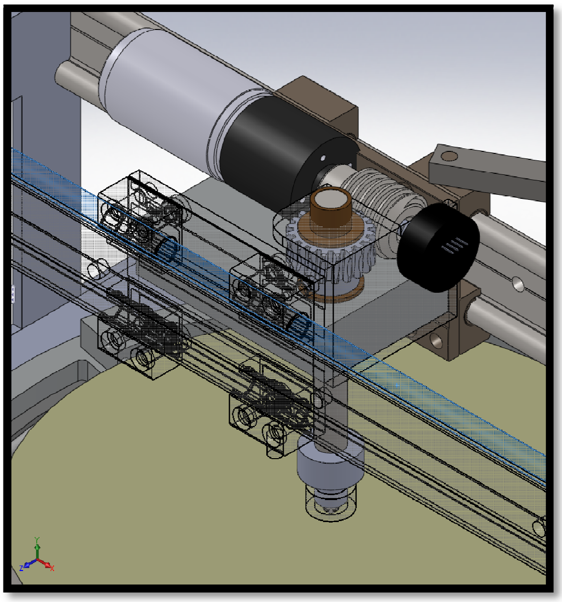
Figure 11: Sample Clamping Method



Pin Gantry

The heart and soul of the wear system is the pin system on our variable machine which can be seen in Figure 12.

Figure 12: The Pin Gantry Assembly



The pin gantry will apply the normal load that can change on the fly, and also be equipped with all of the measurement transducers to find the normal force applied, the frictional force, and a simple wear calculation. To apply the normal force to the sample a small motor will be mounted to a worm drive on the gantry turn a vertical screw which is the pin. This will force the pin down onto the sample, giving feedback from the normal force transducer and allowing us to adjust the normal force on the fly.

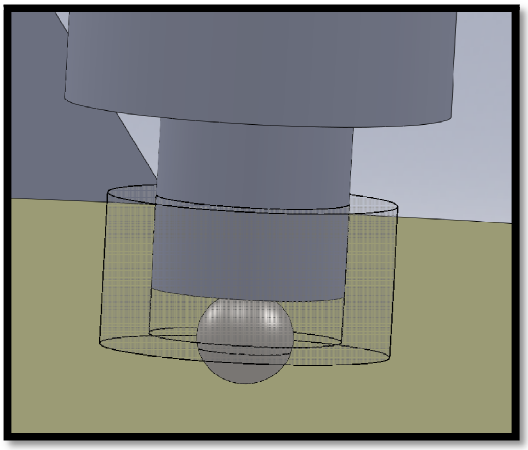
As the pin wears the motor will have to automatically lower the pin more to keep the normal force constant, by measuring the number of turns the motor has moved will give us a good total wear estimate of the track. An encoder will be mounted directly from the motor to measure the wear. The sponsor has informed us that having a feedback loop may cause the system to go crazy because the wear track is very “bumpy”. To combat this problem we plan to add a spring into the pin system to smooth out bumps, and also correct the normal force at a lower sampling rate which will smooth out feedback.

The friction force will be measured by strain gauges mounted to the sides of the pin. When the pin moves and deflects causing strain the x-y forces can then be calculated from the strain in the pin giving the force due to friction. Having all of those force transducers will lead to the end goal of measuring the friction coefficient between the pin and the disk. To sense the small strains with higher accuracy, the team will amplify the signal using a wheatstone bridge. Instead of multiple resistors, the team will use identical strain gauges that will not be strained. By doing this, the team will not have to account for the change in the resistance of the strain gauge due to temperature changes.

Electrical resistance between the ball and the disk will be measured by a resistance sensor in the pin, attached to the ball. This sensor will also be attached to the ground. By measuring the electrical resistance, data pertaining to lubrication effectiveness and wear can be acquired.

The pin is not consists of a shaft with a ball bearing at the end. The sponsor has asked for variable ball diameter sizing (1/4”, 1/8”, 1/16”). To accommodate this, the design will have a screw-on tip at the end that can be interchanged depending on what ball diameter is desired. To change out the ball, the linear gantry moves to the far end of the limit where the ball can be changed out without interference from the sample. This feature can be seen in Figure 13.

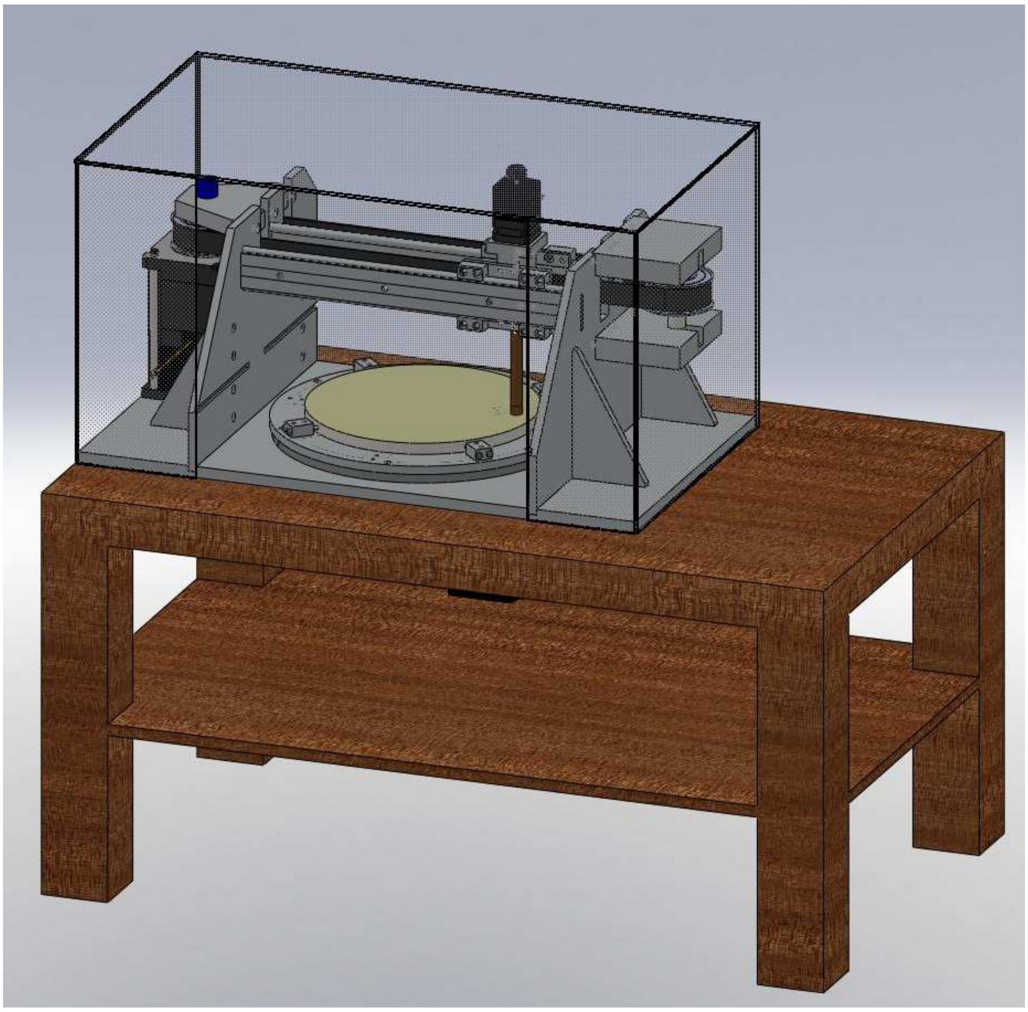
Figure 13: The ball holder mechanism



Engineering Analysis

This section consists of detailed analyses and design changes that were performed after DR2. In the following sections we will describe the processes and equations we used to analyze and improve upon our Alpha Design, the rationale behind any design changes that were made, and specific fabrication and validation plans we will use to manufacture and verify our device. This section is split into four subsystems that each contains the processes mentioned above. These subsystems are Motion Control, Pin Gantry System, Force Measurement, and Environmental Control. Each team member was responsible for the overall design and analysis of a subsystem. This was done to better utilize the team's time and resources. Team meetings were held to discuss integration of the subsystems and an assembled CAD model was made to verify that incorporation of the subsystems was possible and accurate. A segment detailing the integration of these four subsystems and an overall project plan is presented at the end of this section. Figure 14 shows the final design.

Figure 14: Final Design



Motion System

After more in depth analysis of the initial alpha design concept, a pin that performs all motion with an X-Y gantry, we realized that there were several questions regarding certain aspects of the design. The first major problem was the control of the system. We initially thought that the use of linear motion in both the X and Y directions would be a simple task. However, finding a method to control where exactly the pin would be located is proved difficult. There were off the self systems which used stepper motors, but these would be extremely difficult to integrate in a feedback system needed to ensure that we were getting the correct speeds and accelerations. The greatest downfall with the pin all-motion system is that the leading edge of the ball bearing that touches the sample will have a constantly changing leading edge, which is not traditionally how the tests are conducted. For all of these reasons we chose the next best motion system which consists of a spinning disk, and a linearly reciprocating pin.

The final design uses two motors to create the two separate motions of the disk spinning and the pin reciprocating across the disk. Using this setup keeps the leading edge of the sample constant throughout the standard linear reciprocating tests, and a pin on disk test, but the leading edge will be varying along a combined motion test. This will have to be addressed by another team, because we do not have time to study the effects of this. The sponsor has requested the leading edge to be constant at the start and end points. By using this motion selection we will be able to perform the rotational test, and potentially the linear test, much easier because no complex motion control is needed. The motor must simply be turned on and set to the desired speed to perform the test. The rotational mechanism is a motor geared to a sample clamp system which is mounted to a bearing. The gear system was chosen because it can transfer high loads and torques well at high speeds. Designing the linear reciprocating motion was more difficult.

Initially, we attempted to copy several linear reciprocating tribometers seen during preliminary research and use a four bar linkage that utilizes a slider to create a linear motion. The control scheme would be just as easy as the rotational system because you simply turn on the motor, it rotates, and the linkage creates the linear motion needed. While this idea seemed simple at first, we discovered that there are several downsides to this setup. One problem is the linkage creates an elliptical velocity profile due to the complex motion of the linkage. This would need to be corrected to be a constant velocity profile. The linkage works by only allowing the x- velocity vector from an arm attached to a motor to be used. At the edges where there is essentially no x-velocity vector, the motor is very inefficient and could cause problems if a variable test was run in the end regions. After further discussion with the sponsor, we decided that the negatives of a linkage far outweighed the nice simplicity of a control system.

The linear motion system that was chosen will utilize a timing belt that is attached to the motor via a timing pulley. The control system will be somewhat more complex and will involve a calibration system of limit switches to ensure that the linear slide does not run off the track. The pulley system is able to apply full torque anywhere along its velocity path and it is not constricted by a linkage for its motion. The velocity path is also easier to control because it will now begin linear and the user could more easily program a variable speed test. Although this system strays from some of the traditional tribometers we have seen in research, a pulley system is used in other similar machines such as a laser cutter. Finding the right size belt to transfer the heavy loads that will be seen will be vital to the success of this design.

Motor Selection

After selecting which methods for motion generation were the best, research on all of the different types of motors was conducted to compare the benefits and downfalls of each type of motor. The two types of motors selected to be analyzed in full detail were stepper motors and direct current (DC) motors because of their wide use in similar applications, their ease of control, and their relatively low cost compared to other types of motors. Concurrently, simple math models were generated to look at the worst case scenarios the motors would face to generate further specifications specifically for the motors in the tribometer. Two different models were made, one for each type of motion. After looking at all the options a final selection for stepper motors from keling technology inc was made.

Motion Models

Using basic static and dynamic physics equations, simple math models were generated for both the linear and rotational tests. These models were used to develop the specifications for the motors needed, such as max torque, nominal torque, minimum speed, etc.

Linear Test Math Model: From the initial specifications we knew that a normal force would be applied and that this would translate to a longitudinal force based on the coefficient of friction using eq. 1. For the motor, the worst case scenario would consist of having to travel at the max load (200 N) with a very high coefficient of friction (1) at the max specified speed of 1 m/s. There are multiple factors present including the constant force on the pin, the time it takes to takes to accelerate up to speed, and the force on the pin caused by the acceleration.

$$Frictional\ Force = \mu(Coefficent\ of\ Friction) * Normal\ Force\ (N) \qquad \text{Eq. 1}$$

After pluging into eq. 1, the maximum longitudinal force will be 200 N. Plugging the force into Eq. 2combined with the radius of the pulley attached to the motor will give the motor torque required without a gearbox.

$$Torque = Force \times radius \qquad \text{Eq. 2}$$

A gearbox may not be needed, but it was factored into the optimization. The gearbox is simply a ratio that will require higher RPM’s and lower torque from the motor. A gearbox is used to make sure the motor is moving at the nominal speed and torque for continuous use. The equation used to determine the torque from a given gear ratio is Eq. 3.

$$\tau_2 = \frac{gear\ radius_1}{gear\ radius_2}(Input\ Torque) \qquad \text{Eq. 3}$$

After determining the nominal torque, the max torque at start up needed to be found to ensure that the motor would have enough torque to begin moving the gantry system. The force needed for acceleration comes from Newton’s second law of motion which leads to eq. 4. The mass of the gantry was assumed to be 3kg.

$$\Sigma F = mass * acceleration \qquad \text{Eq. 4}$$

Acceleration was assumed to be constant and was calculated based off of the travel distance at constant speed required by the sponsor. The specification set out by the sponsor was for 80% of the wear track to be at a constant speed. So that would mean 10% could be used for acceleration at each end, which is where the most force will be needed. The gantry would have to accelerate up to 1m/s and the smallest diameter sample we would see is 2 inches, giving a total of .2inches to accelerate to 1 m/s. The acceleration was found using eq. 5.

$$acceleration = \frac{(Final\ Velocity)^2}{2*Distance} \qquad \text{Eq.5}$$

The force from acceleration and the force due to friction combine to give the highest torque that needs to be provided by the motor, which, using eq. 5, equates to a torque of 25Nm. After the gantry accelerates to the max speed, it will only be affected by the friction force and will require only the nominal torque to continue moving.

$$v = \omega r \qquad \text{Eq. 6}$$

The rotational calculations utilize many of the same equations and ideas from the linear reciprocating motor selection study. The major roadblock found while calculating the rotational motor requirements was the 600 RPM requirement found in the specifications. After talking with the sponsor, and reanalyzing the standards set forth by ASTM, we could not understand how they got to “600 r/min”. After discussing this with the sponsor, he felt it was best to keep the specification, but to make sure that we can hit 1m/s rotationally, and then discussing the upper limits of rotational testing.

For the worst case nominal torque needed, the same frictional force on 200N could be applied at the radius of the largest sample, which is five inches. The worst case nominal torque the motor will experience is 25Nm.

The highest torque seen will be due to accelerating the inertia of the sample disk and holding system and the friction force associated with a moment arm. The motor must overcome this torque to reach the correct speed and maintain it.

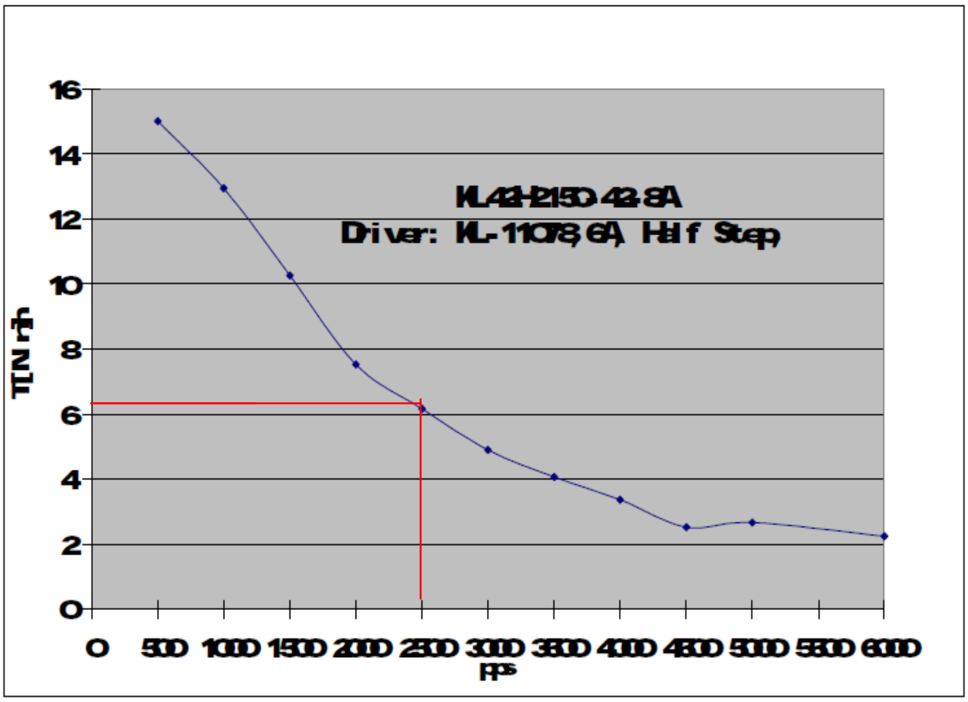
$$Torque = interia(I) * angular\ acceleration(\alpha) \qquad \text{Eq. 6b}$$

Designing for optimal torque is important because it ensures that the motor will work at its highest power output. The manufacturer of the motor will give a torque vs. rotational speed curve that will be essential in making the correct motor selection. A motor should be able to overcome the maximum torque required and it should also operate constantly at or to the right of the maximum power output point. The maximum power output is at half of the max rpm, and half of the max torque. Figure 15 illustrates a motor torque curve of the motor that we choose with the optimal torque point shown.

Final Motor Selection

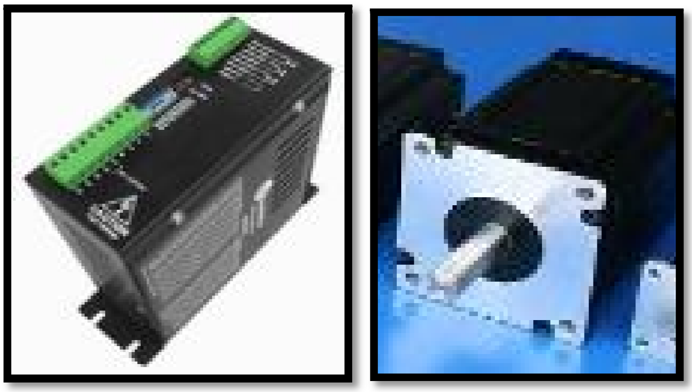
The same motor was selected for the two separate motions in order to keep simplicity and uniformity throughout. The motors we selected are essentially the biggest off-the-shelf stepper motors we could buy. The National Electrical Manufacturers Association(NEMA) size 42 motors are made by Keling Technology Inc. based out of Chicago, IL. The motors will be paired with a Keling-made KL-11078 Micro-stepping Driver to generate the motion. These motor controllers can plug directly into the wall and can perform micro-stepping. Micro-stepping is a method of increasing the resolution of a stepper motor to make it more smooth and precise by having more steps per revolution.

Figure 15: The torque curve for the motor with the optimal torque shown.



To verify that these motors will work, Keling provided a motor torque curve with half step(.9 degrees), which utilizes the same motor driver we plan to purchase operating at an assumed 110 VAC. The important aspect of picking out motors is to make sure that the highest power that the motor can output is being used at the same time it encounters the highest loads. Based on the calculations from previous sections we were sure that these would be operated at their or below the nominal loading. Figure 16 shows a picture of the proposed motor and control from Keling Technology.

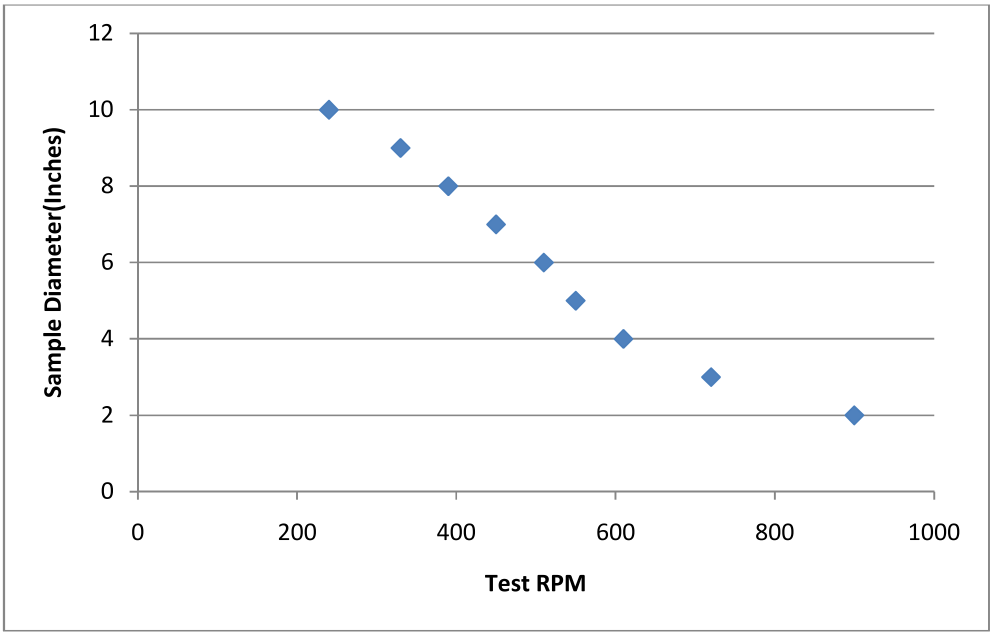
Figure 16: The motor and driver system.



System Limitations

Although the motors we have chosen are the biggest possible motors that anyone can buy that are not custom, they still fall short in some aspects of the design. After doing the above calculations, we believe that it is not possible to achieve certain speeds of rotational testing depending on the sample size. This is due to the limited motor torque, at such speeds. Since we are testing high frictional forces, there will be a constant torque on the motor, on a pretty substantial moment arm. Without going into too much detail with the problem, Figure 17 shows the sample size vs the speed that we can test at maximum loading conditions (200N normal force, with coefficient of friction of 1).

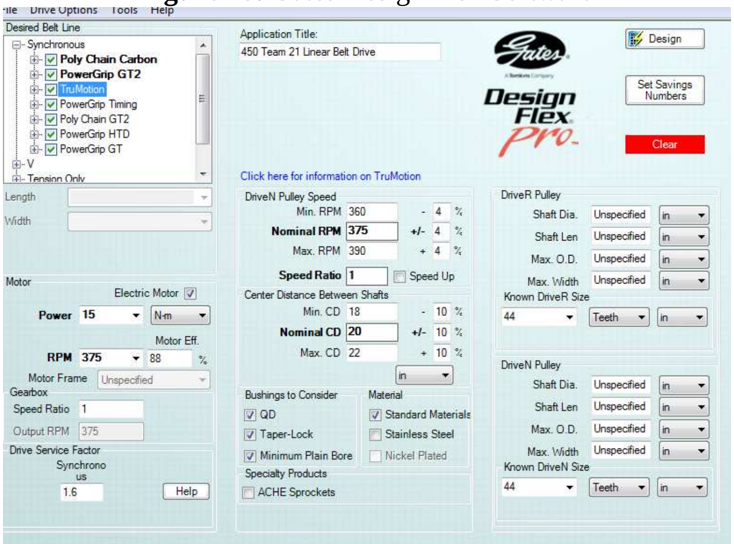
Figure 17: Sample Diamter is Inversly Proportional to Test Rotational Test



Pulley System

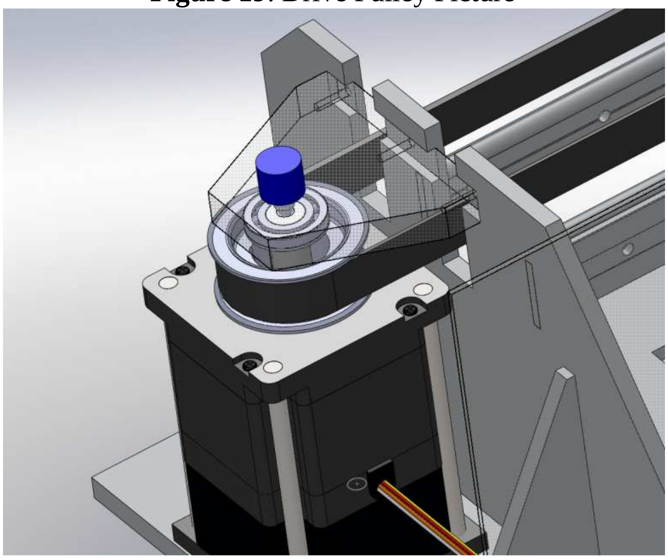
As previously mentioned, a timing pulley system will be used to move the pin gantry back and forth thus creating the linear motion that is required for the tribometer. We located a prominent belt supplier, Gates, who has developed custom belt design software for their products. After talking with Chuck Wilstead, an applications engineer at Gates, he recommended their Design Flex Pro software which is free from their website. After plugging in the torque, belt length, rpm, and other factors of our system, the Design Flex software recommended a 5mm pitch PowerGrip GT2 series with a width of 25mm. Figure 17 shows the summary from the Design Flex software program. From the math performed earlier, we found that we needed a 2.75" diameter pulley which would then be run around to another pulley of the same size, and finally clamped to the linear gantry. We found a 2.757 pitch diameter pulley from motion Industries which will run along a 1300mm long Gates belt. The system is 5mm pitch and will need to be tensioned between 58 and 64 pounds to get the correct power transmission it was designed for.

Figure 18: Gates Design Flex Software



The drive pulley will be manufactured to make use of the already keyed drive shaft of the motor. A bearing will be attached to the free side of the motor to ensure that the drive shaft does not observe any side loads from the movement or tension in the belt. Figure 19 shows how the drive pulley will be set up, and Appendix H and Appendix I show the final print and the manufacturing plan for the drive pulley, respectively.

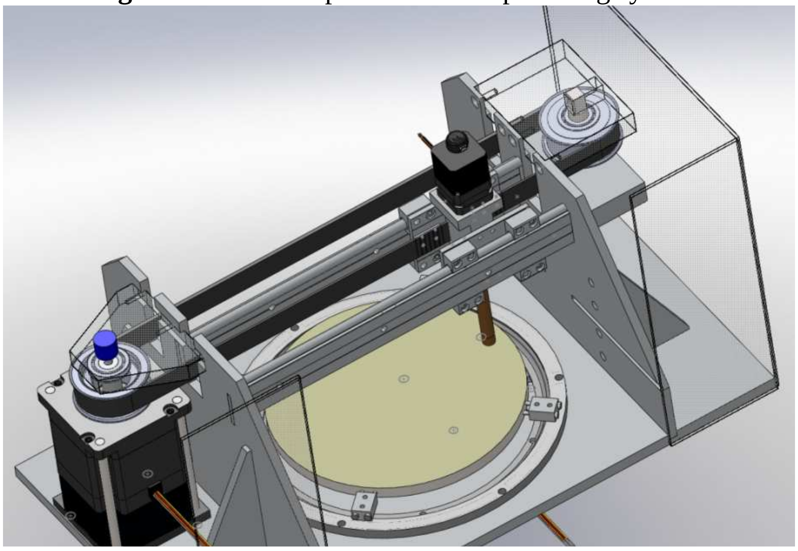
Figure 19: Drive Pulley Picture



On the opposite side, the same pulley will be used as an idler pulley to keep the system tensioned and working correctly. The pulley will be lathed to fit 2 .75” bore ball bearings to keep the idler aligned and spinning correctly. The shaft that the idler pulley rides on will be mounted to the side wall opposite of the motor. We will be required to keep tight tolerances on all parts to ensure correct alignment. The final print and manufacturing plan can be seen in Appendices H and I.

The belt will be clamped onto the linear bearing system using a system that sandwiches the belt between two plates that mesh with the teeth of the belt. The clamping system is what allows the gantry to move back and forth with the belt. It can be seen below along with the complete pulley system in Figure 20.

Figure 20: The complete linear reciprocating system



Gear System

The rotational system is very different from the linear system, and it will see much higher loads due to the 10 inch sample size disk which will create higher sustained torques on the system. To compensate for this, the stepper motor will be geared. According to the above math models, a 5:1 gear ratio would be the best option for torque and speed. The next step was to figure out the optimum pitch and size required for the gears to achieve a 5:1 ratio with the torques the gears would be subjected to. The biggest driving factor for the gearing was to make sure that it could be purchased off the self. We could not find a convenient gear design selection program like we did for the belt system, so we had to perform our own analysis and calculations.

The first selection to be made was the pressure angle, which comes in 3 choices (14.5°, 20°, 25°). A 20 degree pressure angle was chosen because of its wide availability when compared with 25, and its ability to transmit higher torques then 14.5. The next selection was to determine the strength of the gear and to make sure that it can handle the loads it will undergo. After reading several books online and in the library, one book, Gear Design and Production, referenced the Lewis Equation for gears (Pg. 23). Eq 7 is the Lewis equation which relates bending stress on the teeth to the load applied, speed and form factors, as well as the width of the gear.

$$W(Safe\ Tooth\ Load) = \frac{Material\ Strength * Width * Service\ Factor}{Pitch} \qquad \text{(Eq. 7)}$$

An online calculator was found that could provide optimum gear pitch, number of teeth and ratio. These values were partnered closely with looking at which gears were available off the self at various suppliers including: Motion Industries, McMaster Carr, Browning, and Boston Gear. Based on what is available commercially and the Lewis equation, a set of steel alloy gears from Browning will create a 3.33 gear ratio. The 16 pitch, .75 inch width gears were the smallest gears that could handle the loads seen from the motor.

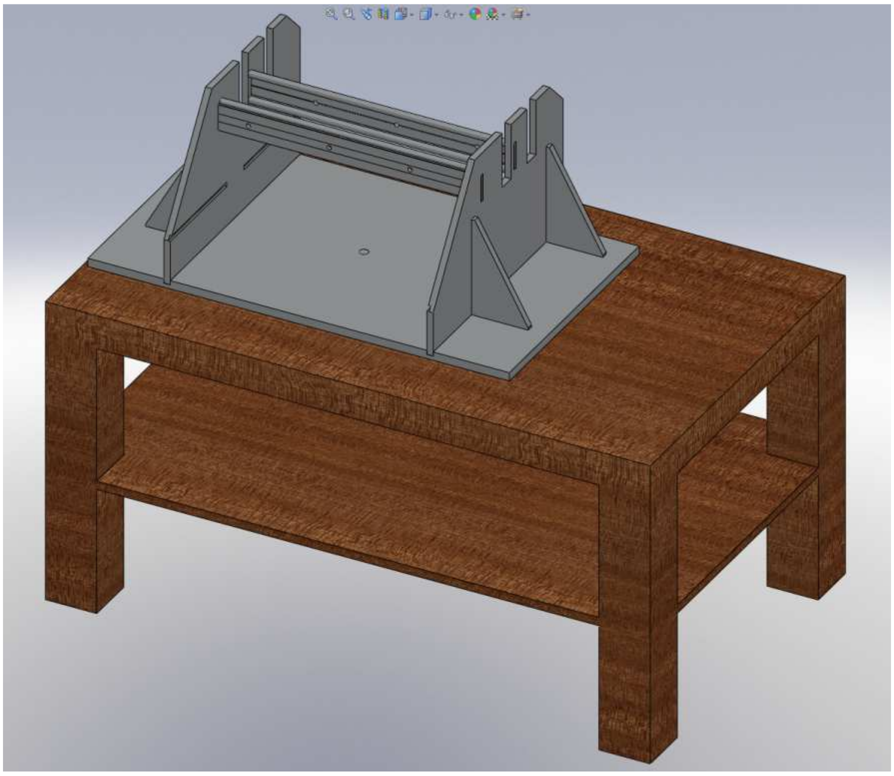
Predicted Performance

The current system, even with the best off the shelf components, will not be able to meet all specifications all the time. This downfall will only be seen in the rotational aspect in which the bigger sample sizes are used. This is because the torque on the motor is extremely high due to the large moment caused by friction far from the center of rotation.

Chassis

After the motors were selected and the sample clamping method system was finalized, the chassis system could be designed to hold everything together in the correct alignment. The main skeleton of the tribometer will be a professional looking coffee table from IKEA on which the main test bed will rest. All of the support electronics will rest on the shelf below. The chassis is made up of a base plate and two vertical sides that are reinforced by four side-supports. All of the pieces interlock with slots and are then screwed into place. The slots ensure precision and strength of the entire system. Figure 21 shows the interlocking chassis system on the coffee table. The print of the table can be seen in Appendix I. An in-depth description of the material selection process for the chassis sides can be found in Appendix C [27].

Figure 21: The chassis system on the IKEA coffee table.



Bearing Selection

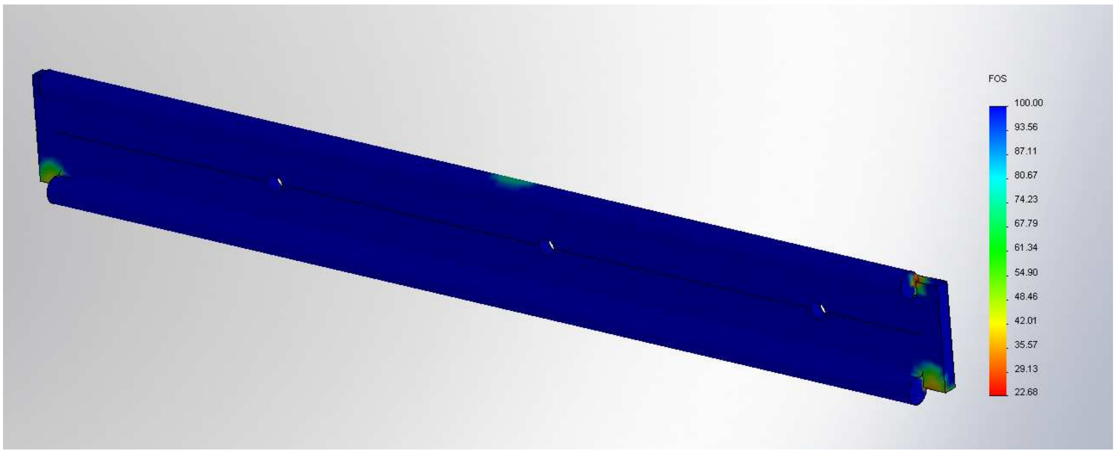
Bearings are mechanisms that constrain a motion between two moving parts. Several bearings are required in our design to constrain linear and rotational motion. Bearings are needed to reduce friction and keep parts in precise alignment. They were used to keep the sample in a rotational configuration and to keep the linear pulley system aligned properly.

Linear Bearing

The linear bearing system will come from Igus, the manufacturer of DryLin® products which uses an anodized aluminum and special plastic to produce linear bearings. The max speed it can attain is 49 fps (15 m/s) which is much greater than our 1 m/s specification. The two gantry system will only weigh .08 lbs (32 g) and it will arrive pre-built, ready to use out of the box. Igus also has the Young Engineers Support (YES) program in which they give students free samples for engineering projects, so their materials are free.

We also had to make sure the loading could be handled by the bearing rails and the carriage system. The gantry system will utilize 2 parallel linear bearings to split the loads and minimize moments. The 200N normal force will act on all 6 axes which will cause moments around 2 axes because the frictional force is acting on a .5 inch moment arm. This will result in a 22.5 inch-pound moment acting on the carriage. It is rated to 50 pounds, so we have a safety factor of about 2 on this design. We would like to thank Igus for their support with the tribometer prototype project. Figure 22 shows an FEA analysis of the linear bearing system with a max loading of 200 N applied at a single point at the center. This was used as a quick check to make sure the linear bearing system would hold up under max loading.

Figure 22: Safety Factor of 22 for the linear bearing system yield

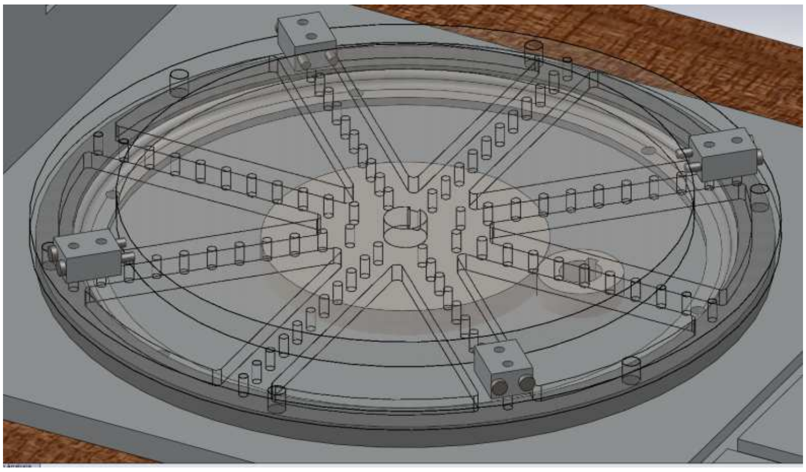


The linear bearing will have to be slightly modified to fit into the puzzle chassis. The ends of the given length of slide material will be milled into a box which will fit into designated slots on the vertical sides of the tribometer. The moving bearing part of the slide will also have holes drilled into it for the belt clamp and pin gantry system, which will be mounted to the slide cars that ride on the slide material. The final prints and manufacturing plan for the linear bearing system can be seen in appendix H and I respectively.

Rotational Bearing

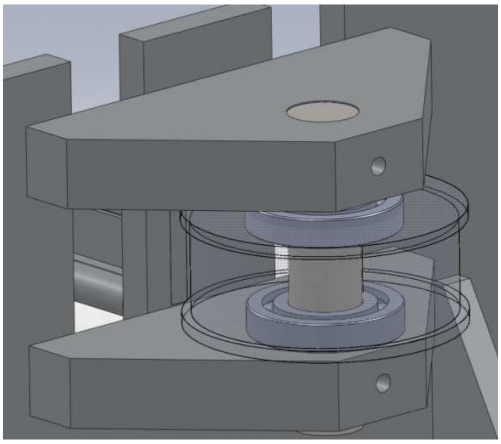
The specimen restraint will be mounted with ¼”-20 screws to the rotational bearing, a large-diameter bearing purchased from McMaster Carr. The bearing is as wide as possible in order to spread out loads and help dampen vibrations caused by the sample wearing or being off balance. The ball bearing is rated to a 175 pound load capacity, well above the 45 pounds that it will see due to the normal force from the pin on the sample. The bearing does not have a rotational speed specification, so it will require testing to determine if it can reach high rotational speeds. If it cannot reach high rotational speeds at the specifications level, we will have to replace the bearing. The technical drawing provided by McMaster Carr can be seen in appendix I. Figure 23 shows the rotational bearing mounted to the base plate and with the specimen restraint on top of it.

Figure 23: Large Diameter Turntable mounted into the tribometer assembly



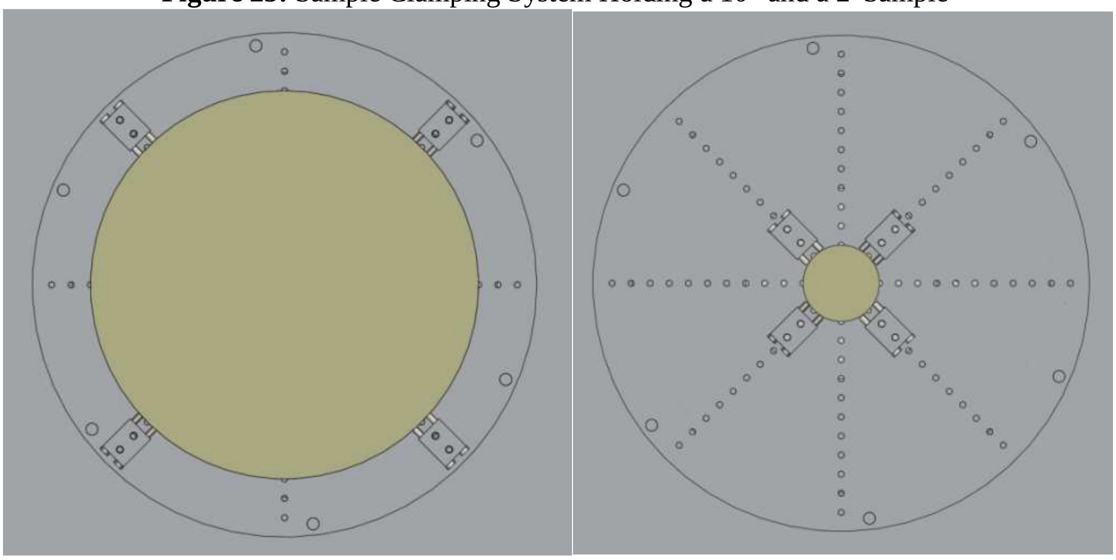
The idler pulley also needed a bearing to constrain its rotational motion and to reduce as much friction as possible. The bearing will see side loads occurring from the constant tension in the belt system and the acceleration and deceleration of the pin gantry system on the linear slide system. These forces will equal a maximum of about 300 pounds. There will be no thrust loads seen by the bearings as long as the belt is lined up properly. The bearings chosen for the idler are rated to 7500 RPM and a load of 1500 for the maximum specifications, so they are a good choice for the system. The technical drawing of these bearings can be seen in Appendix H and the bearings can be seen in the idler pulley below in Figure 24.

Figure 24: The idler bearings inside the idler pulley



Specimen Restraint
We did not foresee any problems with the specimen restraint system from the alpha design, so it has been finalized. The main piece will be made out 6061 aluminum, and will be made using a CNC mill due to the complexity and numerous tapped 10-32 holes. The 10-32 holes are spaced ½" apart and extend out to the radius of the main mount at which they are spaced 45 degrees apart. The top is now one solid piece so that lubricant cannot leak through, but the back has been pocketed to reduce weight to aid angular acceleration. In the middle is a pin, which will correspond to a hole drilled into the direct center of the sample to aid in balancing the sample. The sample can be directly clamped down using the 10-32 holes with screws, or the user can use an adjustable chuck mechanism. The chuck parts screw down into the 10-32 holes, then additional 10-32 will come out laterally to clamp the sample in place. This is a simpler and much more lightweight design than a full chuck system. A lubricant bath tub can also be bolted onto the specimen restraint. The restraint can hold any size or shape sample up to about a 10" diameter. Figure 25 shows the system holding a 10" diameter, and a 2" diameter sample.

Figure 25: Sample Clamping System Holding a 10" and a 2"Sample



Sensing Systems-
Stepper motors are open loop control systems, but to verify that they are operating properly we have decided to install sensors as a secondary backup, for validation, and as a safety precaution.

Light Encoder – Originally the idea presented in the alpha design was to use a rotational encoder mounted to a gear system to help measure rotational speed. An easier method we discovered is to hook up a light sensor which acts like an encoder based on the level of reflection it sees. A reflective light sensor will be mounted on the underside of the chassis and will look up at the rotational bearing which will have a simple black and white pattern painted on it. When the light sensor sees the white, it will read high, and when it sees black it will read low. This will send pulses to the DAQ system to let us know the distance we have traveled, and to ensure the stepper motors have not skipped a step.

Potentiometer- A 10K ohm, three turn potentiometer from Vishay will be purchased through Digikey to offer an absolute position for the linear slide at any given time. Potentiometers beat out most other forms of position sensing because they do not reset after power is cycled. The potentiometer will be mounted directly to the drive pulley and will theoretically only turn about 1.5 times. It will be used to validate speed of the linear system, but also as a safety net for the limits of the linear system. The potentiometer will ensure the linear system stays on the sample at all time, even if the stepper system skips a step.

Programming

The programming for the linear and rotational tests will be fairly straight forward. The main reason for the selection of a stepper system is that when we send the stepper drivers a correctly shaped pulse of the correct amplitude, the motor will move one step. Sending more pulses results in the motor moving more. Needed programming will consist of writing a function that sends out pulses based on the user input for speed and distance that was entered before the test begins. The linear system will also require additional user input which tells the width of the sample so that the computer can calculate limits for the stops. The computer will also utilize the potentiometer as a safety device to ensure that the pin is in the correct limits in case the stepper system skips a step.

We will use LabView 2009 from National Instruments partnered with a borrowed USB DAQ 6008 card for testing purposes. We will then buy a USB 6009 card to get the correct analog input rates. Using the drivers from NI, DAQ express VI's, we will be able to easily read in the analog signals and save them into a spreadsheet.

In order to conduct variable and custom tests we will have to develop a new form of control most likely based on a polar coordinate system. Within the current time frame, we will probably not have time to create such a system, so we will try to program in an "hour glass" shape which could be used to do cross-wear pattern testing. The custom shape will utilize aspects from both the linear and rotational control to create the test profiles needed.

Validation

After the system is completed, it will need to be validated to ensure that it has met the specifications that we have set for it. The first step will be to look at the feedback given to us using the encoder and potentiometer that will be mounted to the system. The sensors will output raw data that we can turn into speed and acceleration of both the linear system and rotational system that we can then compare to the numbers given in the specifications.

Another way to validate the system that is completely external from the tribometer system is to use a video camera with a high speed film to create a video of the system as it operates. If a mark is put on the disk or linear gantry and a set distance is marked off in the background, you can directly view how far the mark moved over a time frame in the video. This can give speed and acceleration completely separate from the system. Both ways of validation will be used to compare to the specifications for speed and acceleration.

Manufacturing Plan

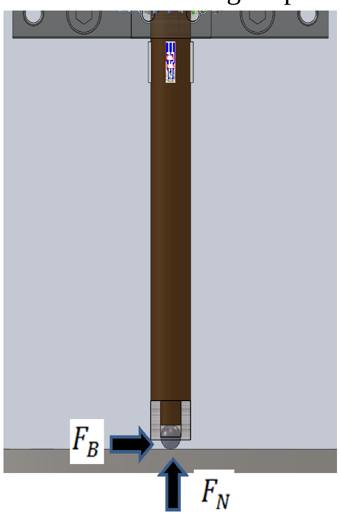
All of the materials that require shipping were ordered on November 19, 2009, and will be received sometime within the next week. In the mean time, we will go out and purchase the materials to make the chassis system locally, and start machining and building it. We have been to the undergraduate shop to discuss capabilities and times that are available, and we may take up the opportunity to use other shops for increased machining capabilities such as CNC.

After the other materials are received we can begin finishing the chassis plates with the proper mounting holes. We will wait to put in the mounting holes, in order to verify that the CAD models online match the actual physical part that we receive from the supplier. We will do this for the motors, the large rotational bearing, and the smaller bearings and pulleys. The sample clamp piece will have to be CNCed at an outside shop to acquire a round shape out of a square piece of stock. After the main chassis system is in place with the motors mounted down, we can finish up the mounting of the gears and pulleys. This will include boring and keying the hole to mate the drive components with the motors. Finally, everything can be test fit and mounted. Once it all fits, we can test the motion system with first the linear control, then the rotational control, just to see if it moves. After we are certain that it works, we can finish installing all the other tribometer systems. All of the prints and manufacturing steps for each part are identified with prints and charts in appendix I and H respectively.

Pin Gantry System

The purpose of this section is to outline how the pin gantry system will apply a fixed normal force between the pin and disk, determine the amount of wear, and measure electrical resistance. The integration of the pin gantry system with the force measurement and motion control systems will also be covered. The main function of the pin gantry system is to force the pin against the disk by means of a spring and screw-drive mechanism attached to a stepper motor. A closed loop control system connecting the motor to the force sensors will allow the system to maintain a constant normal force despite the effects of wear on the pin and disk. Below is a picture showing the forces acting on the pin shaft. The normal force between the pin and the disk, F_N, is set by the user while the bending force due to friction, F_B, is a result of the normal force and coefficient of friction between the ball bearing and the disk material.

Figure 26: Forces acting on pin shaft



Material Selection and Geometry Analysis

Before we could determine the specific parameters of the pin shaft, we had to first select what material it would be made out of. The material selection process was driven by the environmental conditions the shaft will be required to perform under as well as the specification that the pin cannot deflect more than 1° from vertical during any test. The deflection will be greatest when a test is performed with the highest normal force applied, 200N, and a coefficient of friction of 1. Due to the high temperatures seen by the shaft only metals were analyzed as possible options.

The engineering material selection program CES[27] was used to select various materials that could be used in our application. Steel was the first metal that was analyzed due to its high availability, high strength, and machinability. However, the shaft will be exposed to a relative humidity of up to 100% and steel is not corrosion resistant, so it was eliminated. Stainless steel was evaluated next because it is a corrosion resistant alloy of steel. Various alloys and grades of stainless steel were analyzed and Stainless Steel 303 was initially selected for the pin shaft. Upon further inspection, it was realized that only cold drawn 303 would be able to withstand the stresses seen at the highest loads. This variant was not readily available to purchase so additional alloys were identified as alternatives. However, the stainless steels that were available to purchase and strong enough to withstand the stresses in our application were either very difficult or impossible to machine. Threading, cutting, and drilling holes in the pin shaft would require special carbide bits and would be a highly tedious task. The difficulties of machining these stainless steels led us to search for other possibilities. Bronze was investigated, and eventually a high-strength bronze alloy that is corrosion resistant and easily machinable was selected. This alloy is known as Bronze 544. The material selection process for the shaft is described in detail in Appendix C.

After we selected Bronze 544 we could determine the geometry of the pin shaft. The geometry was driven by the conditions that the shaft cannot deflect more than 1° from vertical and that the shaft should never yield due to stresses induced by the normal and frictional forces applied at the tip. A rod of circular cross-section was chosen over a rectangular or square cross-section to eliminate stress concentrations that would be present in the corners of the beam. A rod of circular cross-section will also be easier to thread at one end in order to attach the ball mount. Initial analysis of the shaft led us to change the design of the pin gantry system in DR2. Instead of having the spring at the bottom near the ball, we moved the spring to the top, closer to the motor. This eliminated the need of having a long hollow shaft to hold the spring and the threaded rod, which allows us to use a solid rod. The new design also greatly decreases the length of the threaded rod. We feel that this new design is much more robust because it involves fewer parts with more simple geometries. These simpler geometries can be more accurately modeled using beam bending equations to predict what the resultant stresses and displacements of the shaft will be.

The deflection of the shaft was analyzed first. A safety factor of two was placed on the deflection because we cannot assume that the pin will be completely perpendicular to the disk to begin with. The shaft was then simulated as a cantilevered beam with a uniform diameter and the maximum normal force and frictional coefficient applied. The constraint shown in equation 8was derived. In this equation, L is the length of the shaft, D is the diameter and E is the Young's modulus of the shaft (E of 544 is approximately 15 MPSI).

$$\tan^{-1}\frac{2877.55L^2}{3E\pi D^4} < .5° \qquad \text{(Eq. 8)}$$

If we take the length to be the 4 in., as in the design currently, and plug into Equation 8, we can determine the minimum allowed diameter of 0.44 in. Other factors such as spring diameters led us to select a shaft diameter of 0.5 in. When 0.5 in. is plugged into eq. 8 for *D*, and 4 in. is used for *L*, a deflection of .3° is observed. This results in a safety factor of over 3 against deflecting more that 1° from vertical which will be sufficient to compensate for preliminary alignment errors. Because bronze becomes less stiff as temperature increases, we also analyzed the deflection at the maximum temperature of 150°C. At this temperature the Young's modulus for Bronze 544 is 14.4 MPSI, and the deflection increases slightly to .31°. Therefore, we are confident that the shaft will not deflect more than 1°, even at the most extreme temperature.

Using these dimensions, the maximum stress from the normal force (Equation 9 where σ_N is the stress due to the normal force, F_N) and the maximum bending stress (Equation 10 where σ_B is the bending stress due to the frictional force, F_F and *r* is the radius of the shaft) from the lateral friction force, can be determined. The maximum stress that will be seen by the pin is 15 KPSI. This gives a safety factor of 3 versus the yield stress of bronze 544 (60 KPSI).

$$\sigma_N = \frac{4F_N}{\pi D^2} \qquad \text{(Eq. 9)}$$

$$\sigma_B = \frac{4LF_F}{\pi r^3} \qquad \text{(Eq. 10)}$$

The chassis that holds all of the Pin Gantry System parts and interfaces with the Linear Motion Control system will be made out of Aluminum 661. Aluminum was selected because it is a strong, lightweight material that is easily machinable. Initial fabrication plans and methods for all machined parts are discussed on Pg. 44.

Spring Selection

In our design, the spring transmits the force from the screw-drive onto the pin shaft in the following manner: as the motor spins the threaded rod, it screws downward and compresses the spring. The compressed spring applies a force against the pin shaft according to the following equation in which *k* is the spring constant and *x* is the distance the spring has been compressed.

$$F_s = kx \qquad \text{(Eq. 11)}$$

Since we know the spring constant from the manufacturer's information, we can compress the spring a known amount using the screw-drive and thus achieve a known normal force on the pin shaft.

Not only does the spring transmit the force against the pin shaft, it also must absorb minor deflections in disk flatness and smoothness. Small irregularities on the surface of the disk are likely to occur, and these will force the pin up and down as the pin moves across the disk's surface. The maximum deflection that the spring must compensate for can be calculated from the specification that the disk must be flat within 1° of horizontal. The calculation is shown below.

$$y = R\sin(\theta) \qquad \text{(Eq. 12)}$$

The maximum vertical deflection, *y*, is found using the maximum radius of the disk, 5 inches, and the maximum horizontal offset of the disk, 1°. This results in a deflection of 0.087 inches. Therefore, the spring must be able to compensate for both a positive or negative deflection of 0.09 inches at any time. Based on these assumptions regarding the disk's flatness, the smallest force a spring can transmit is found by inserting 0.09 inches for x in Eq. 12. If the spring is initially compressed 0.09 inches, it can compensate for the maximum deflections in both the positive and negative directions. The spring will also be able to compensate for surface asperities present on the disk. If the material on the disk is rough, the spring will be able to move up and down over the changes in the surface, thus preventing the motor from attempting to constantly correct the height of the pin due to surface roughness.

Different types of springs were looked at for application in our device. Initially, wave springs were investigated as a possible choice due to a recommendation from our sponsor. Wave springs can transmit the same forces as conventional compression springs, but are more compact and require less compression to do so. Using wave springs instead of compression springs will minimize the space, and therefore the amount of material required for the pin gantry system. Springs were chosen based on their ability to fit within the ½ inch bore of the pin gantry system, and the minimum and maximum forces they could apply based on their spring constants. We found that multiple springs would have to be used because one spring can not apply all loads from 0.1 – 200N based on our assumption that the spring must be initially compressed 0.09 inches. A table of the chosen springs and their respective load ranges is found below.

Table 3: Spring Selection

Load (lbs)	Manufacturer	Part Number	Spring Force Constant (lbs/in)
14.5 - 60	Smalley	CS050-M9 (stacked)	145
4.3 - 15	Smalley	CS050-H9	45
1.2 - 5	Smalley	CS050-L9	12
.183 - 1.28	McMaster Carr	1986K11	1.83

To perform a test at a certain load, the appropriate spring should be chosen and inserted into the device. The springs will be able to be interchanged fairly easily by removing the pin shaft from the chassis and placing the desired spring within the bore.

Motor

The motor assembly consists of a stepper motor with a driver and an encoder connected to a threaded rod to form a screw-drive system. A stepper motor was chosen over a DC motor because a stepper motor with a micro-stepping driver would give us the necessary control we require and would be easier to program. Stepper motors are programmed to travel a certain number of pulses input by the user, while a DC motor would require additional math be performed based on the geometry of the motor and the rotational speed. A ½ inch diameter steel threaded rod with 20 threads per inch was initially chosen for the screw because it fit within the dimensions of the bore hole and could comfortably transmit a force of 200N. Based on the screw geometry, the minimum mated contact length needed between the screw and the chassis was calculated. The tensile stress area of the screw, A_t, was found based on the following equation in which D is the diameter of the screw and p is the thread pitch.

$$- \quad \text{(Eq. 13)}$$

With D = 0.5 and p = 1/20. A_t was found to be 0.161 in^2. This could then be used to find the required length of thread engagement, L_e.

$$L_e = \frac{2A_t}{.5\pi(D - .64952p)} \qquad \text{(Eq. 14)}$$

The minimum thread length required is 0.44 inches.

The next step was determining how much torque is required to screw the threaded rod down onto a maximum force of 200N. This torque value would then be used to select an appropriate stepper motor. A simplified equation to find the torque, T, based on the diameter, D, and maximum force, F, is shown below.

$$T = .2DF \qquad \text{(Eq. 15)}$$

The value .2 is a coefficient for the thread condition of the steel. At ½ inch diameter and 200N of force, the torque required is 72 oz-in. We have therefore selected a stepper motor with a holding torque of 100 oz-in for a factor of safety of 1.39.

Our device is required to be able to control the amount of linear translation of the screw to 1 micron. A micron is equivalent to 39.37 x 10^{-6} inches. To achieve this amount of control, the stepper motor needs a driver capable of 1270 pulses/revolution. This number was found based on the fact that one revolution of the motor, and therefore screw, would yield .05 inches of vertical translation. This is found from the knowledge that there are 20 threads per inch, so one turn is 1/20 of an inch vertical translation. To achieve 1 micron of translation, the screw must be turned 7.87 x 10^{-4} revolutions. Therefore, one pulse per 7.87 x 10^{-4} revolutions equals 1270 pulses/rev. The driver and motor we have selected are capable of up to 1600 pulses/rev which satisfies this requirement.

To measure the wear of the ball and disk, an encoder on the motor will keep track of how far the motor has rotated. As the ball wears, the normal force between the pin and disk will decrease. In order to maintain the force, the motor will turn the screw to keep the spring compressed a fixed amount. By measuring how much the motor rotates, we can find the vertical distance the screw has travelled which will equal the amount of wear that has occurred.

Electrical Resistance

Our device is required to be able to measure electrical resistance from 0.1 – 1000 ohms between the pin and the disk. This will be achieved by attaching one wire to the pin and one to the disk, applying a current between the two, and measuring the voltage drop between the two. The resistance can then be found using Ohm's Law, seen below.

$$R = \frac{V}{I} \qquad \text{(Eq. 16)}$$

The electrical resistance will be measured throughout the test and can be used to observe when lubricants or coated metals wear out or change. When a lubricant or coating is present, the resistivity between the pin and the disk will be very high. As the coating wears down, the resistance between the ball and disk will decrease. If the resistance is measured throughout the test, the moment the coating or lubricant wears out can be found and used to see how much time or distance travelled the coating or lubricant can be used.

Initial Fabrication Plan

This section details the fabrication plans for each part in the Pin Gantry System. Detailed drawings of the parts can be found in Appendix H while step-by-step manufacturing plans including tool choice and machine feeds and speeds can be found in Appendix I.

Bronze Shaft

The shaft will be a 12 inch long circular rod of Bronze 544 purchased from McMaster Carr. Initially, the shaft will be cut to the required length of 6 inches. Holes for the shoulder bolts will then be drilled 5.125 inches from the bottom. Finally, the bottom .25 inches of the shaft will be threaded at 5/16" diameter with 18 threads per inch. The drawing and manufacturing steps for the shaft can be found in Appendices H and I, respectively.

Ball Holder

Three ball holders, one for each size ball will be manufactured from Bronze 544. Each ball holder will be .406 inches long and will have a threaded bore of 5/16 – 18. Holes will be drilled into the bottom of the ball holder. The hole for the ¼" diameter ball will be .24 inches in diameter, the hole for the 1/8" diameter ball will be .115 inches in diameter, and the hole for the 1/16" ball will be .0525 inches in diameter. The drawing and manufacturing steps for the ball holders can be found in Appendices H and I, respectively.

Aluminum Chassis

The chassis will be made from Aluminum 6061. This part will be complicated and special care will be taken to get it right on the first try. All machining will be done on the mill. The drawing and manufacturing steps for the chassis can be found in Appendices H and I, respectively..

Threaded Rod

The threaded rod will be made from Grade B-8 18-8 Stainless Steel and will have holes drilled along one side and down the center. The drawing and manufacturing steps for the threaded rod can be Appendices H and I, respectively.

Cost of Subsystem

The total cost of the Pin Gantry System is estimated to be $300.00. The part by part cost breakdown can be seen in Table 4 below. The cost of the wave springs is $0.00 because Smalley will samples at no cost.

Table 4 : Cost of Pin Gantry System

G-1	Normal Force Motor	NEMA 17 Bipolar Stepper Motor	Keling Technology, Inc	KL17H247-168-4B	1	$20.00	$20.00
G-2	Microstepping Driver	Microstepping Driver	Keling Technology, Inc	KL- 4020	1	$39.95	$39.95
G-3	Power Supply	Stepper Motor Power Supply	Keling Technology, Inc	KL-320-36 9.6A 110V/220V	1	59.95	$59.95
G-4	Bronze Shaft	Hi-Strength Bearing-Grade Bronze (Alloy 544) .5" Dia, 1' length	Mcmaster Carr	8971K711	1	$16.82	$16.82
G-5	Wave Spring 1	Spring Loads 4.3 - 15 lbs	Smalley	CS050-H9	1	$0.00	$0.00
G-6	Wave Spring 2	Spring Loads 1.2 - 5 lbs	Smalley	CS050-L9	1	$0.00	$0.00
G-7	Wave Spring 3	Spring Loads 14.5 - 60 lbs	Smalley	custom	1	$0.00	$0.00
G-8	Spring	Spring Loads .183 - 1.26 lbs	Mcmaster Carr	1986K11	1	$10.15	$10.15
G-10	Ball Bearing 1	1/16" Steel Ball Bearing	Mcmaster Carr	96455K71	1	$4.34	$4.34
G-11	Ball Bearing 2	1/8" Steel Ball Bearing	Mcmaster Carr	96455K49	1	$1.98	$1.98
G-12	Ball Bearing 3	1/4" Steel Ball Bearing	Mcmaster Carr	96455K52	1	$3.11	$3.11
G-13	Encoder	E4P OEM Mini Optical Kit Encoder	US Digital	E4P-360-197-D-D-D-3	1	$32.25	$32.25
G-14	Encoder Cable	4-Pin Micro / Unterminated 4-Wire Discrete Cable	US Digital	CA-MIC4-W4-NC-2	1	$7.30	$7.30
G-15	shoulder bolts	3/16 Shoulder bolt	Mcmaster Carr	93996A718	2	$3.99	$7.98
G-16	Steel Threaded rod	Grade B-8 18-8 Stainless Steel 1/2"-20 Thread, 1' Length	Mcmaster Carr	91187A560	1	$8.62	$8.62
						TOTAL	$203.83

Validation Approach

Before our device is ready to be used each variable that is being controlled must be validated for accuracy. For the pin gantry system, the variables that require validation include the wear measurement system and the electrical resistance system. The alignment of the shaft must also be verified. The applied force will be verified using the force sensing system which is described in detail in the next section of this report.

The wear measurement system will be validated by observing exactly how much the screw travels vertically when the stepper motor rotates 1 pulse. The observation of this distance will be done using a microscope and a micrometer. Theoretically, since there are 1600 pulses/rev and one revolution results in 0.05 inches of vertical translation, 1 pulse will result in 0.05/1600 inches of translation or 3.125×10^{-5} inches. However, in a real world application this value is likely to be different due to energy lost throughout the process of turning the motor to turning the screw to compressing the spring to applying a force to the shaft. Therefore, multiple tests will be performed to find the average vertical translation output by various pulse amounts.

The electrical resistance system can be validated using multiple resistors of known resistance. The resistors will range in value from 0.1 – 1000 ohms because that is the specification range of electrical resistance that our device must be able to measure. The resistors will be placed between the wires attached to the pin and disk and the resistance measured by the ohmmeter will be observed. The measured resistance will be compared to the actual value of the resistors and the accuracy of our electrical resistance system can be verified.

The vertical alignment of the pin shaft is required to be perpendicular to the horizontal disk within ±1°. To verify that the shaft meets this specification, we will use an inclinometer to accurately measure the angle the pin shaft makes with the horizontal. If the pin shaft does not meet the alignment specifications, adjustments will be made until the shaft is within 1° of vertical.

Force Measurement

The purpose of this section is to outline in detail how the normal and lateral forces exerted on the pin shaft will be measured. The basic idea is to measure the strain at different locations using strain gages. Strain gages are resistors that are attached to a specimen, and when the specimen is strained, the strain gages are strained and change their resistances. By knowing the location of the strain gages and the material and geometrical properties of the shaft itself, the loads applied to the shaft can be determined.

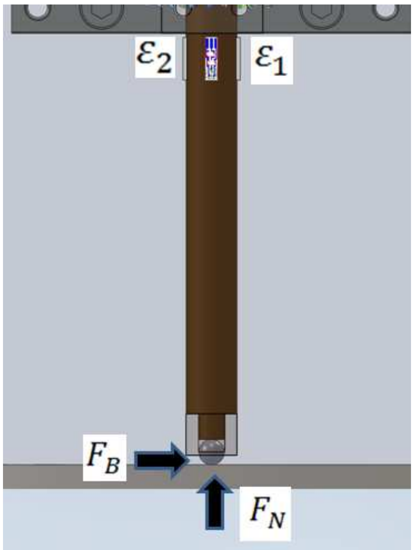
Geometrical and Material Selection

The geometry and material selection of the shaft are described in detail in the previous Pin Gantry System section starting on Pg. 40.

Strain Gage Configuration

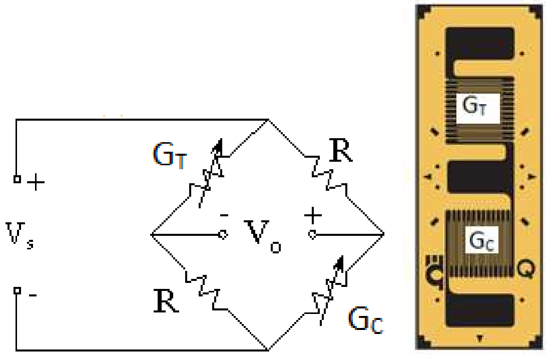
To measure the normal force and the frictional force, six strain gauges will be placed on the shaft. These strain gauges will be placed in pairs at three locations on the outside of the shaft. These locations will be 90° apart from each other and as far up the shaft as possible (just below the threads for the screw-drive) so that they will experience maximum strain, thus increasing the precision of our measurement. The set-up just described is pictured below in Figure 27.

Figure 27: Strain Gage locations



These strain gage pairs will each be arranged in what is known as a half Wheatstone bridge, shown in Figure 28. In this figure V_S is the excitation voltage, V_o is the measured voltage, R is a resistor, and G_C and G_T denote gages in compression and tension. The strain gages that are 180° from each other will be used to measure both the normal force and the component of the frictional force along the axis of which these gages are aligned (this is explained later).The remaining strain gage will be used to measure the remaining component of the frictional force. These bridges amplify the output voltage by two times the output voltage that would be seen if only one gage was used, thus increasing the accuracy of the measurements. The use of multiple gages also tends to eliminate noise, as we will measure the mean between the gages. Finally, by putting resistors into the half bridge that change resistance as a function of temperature in the same way the strain gages do, the temperature effects will be accounted for automatically. Each strain gage pair in the half bridge will be in a tee rosette configuration (also shown in figure 21), in which strain gages are placed perpendicular to one another so that one is strained axially and the other is strained in the opposite sign in the Poisson direction.

Figure 28: Half Bridge Circuit [22] and Tee Rosette Configuration [24]



Equations 17, 18, 19, 20, 21 and 22 show how the strain readings of the two half bridges that are 180˚ from each other can be used to determine the normal force and lateral force exerted on the shaft. The forces and strain gages referred to in these equations are depicted in Figure 27. Equations 17 and 18 show that the strain measured is the sum of the strain from the normal force compression and the bending strain from the lateral force. The strain from the normal force is equal on all strain gages in both sign and magnitude. The strain from the lateral force, however, is equal in magnitude and opposite in sign because this force will put one side of the shaft into tension and the other into compression. These differences are shown in equations 17 and 18.

$$\varepsilon_1 = \varepsilon_N + \varepsilon_B \qquad \text{(Eq. 17)}$$

$$\varepsilon_2 = \varepsilon_N - \varepsilon_B \qquad \text{(Eq. 18)}$$

By adding and subtracting these equations from each other, the strains due to normal force (Equation 19) and bending (Equation 20) can be determined.

$$\frac{\varepsilon_1+\varepsilon_2}{2} = \varepsilon_N \qquad \text{(Eq. 19)}$$

$$\frac{\varepsilon_1-\varepsilon_2}{2} = \varepsilon_B \qquad \text{(Eq. 20)}$$

These strains can then be used to determine the normal force (Equation 21) and the lateral force along the line of the two strain gages being read (Equation 22). Because the gages can be used to determine the normal force, this design will not need the normal force load cell that was present in the alpha design.

$$F_N = \frac{E\varepsilon_N\pi D^2}{4} \qquad \text{(Eq. 21)}$$

$$F_B = \frac{E\varepsilon_B\pi R^3}{4L} \qquad \text{(Eq. 22)}$$

A third strain gage (pictured on the front of the shaft in Figure 27) will be used to determine the other component of the frictional force by subtracting the strain due to the normal force (determined using the other two strain gages) and correlating the strain due to bending to the friction force using Equation 17. Once the lateral forces along two axes are determined, the magnitude of the lateral force can be determined using Equation 23.

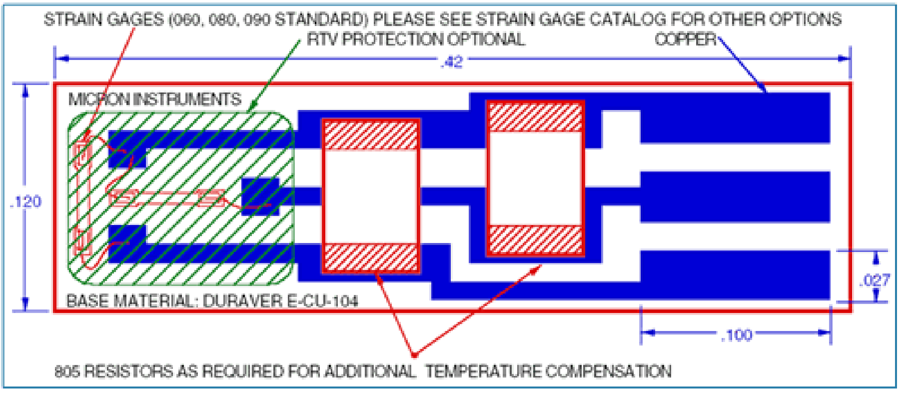
$$|F_F| = \sqrt{F_{Bx}^2 + F_{By}^2} \qquad \text{(Eq. 23)}$$

Using the equations above, and plugging in the largest and smallest forces that will be seen according to the specifications, the minimum (4.8×10^{-9}) and maximum (9.9×10^{-4}) strain expected to been seen can be determined. The strain gages that are chosen must be able to work over these strains, over the specified temperature and humidity, and be able to deliver measurements that are accurate to 1% of the force being measured.

Strain Gage Selection

Typical strain gages are only accurate to 1 micro-strain. This strain resolution will not meet the specifications. Micron Instruments makes semi-conductor strain gages that can, "Perform like a foil gage except that the resistive change is 30 to 55 times greater."[19] When this was discussed with their application engineer, he stated that their backed semiconductor strain gage half-bridge can easily read 1/10 micro-strain, and with accurate calibration it is possible to read 1/100 micro-strain [21]. This gage (Figure 28) is already configured in a half-bridge with temperature compensation. The two gages are placed perpendicular to each other and one gage would read axial strain while the other gage would read strain in the Poisson direction. Figure 28 also shows a silicone covering over the strain gages in green. This covering will help to eliminate damage from contact to the strain gages. It will also stop condensation and oils due to handling from changing the resistance of the strain gages. This gage will be used to measure the strain at the three locations need.

Figure 28: Micron Instrument SSGH Half-Bridge Gage [19]



If these gages only read 1/10 micro-strain accurately, will read up to .0046 N of lateral force accurately. If the gages read 1/100 micro-strain accurately, they will read up to .00046 N of normal force accurately. This would meet the specified low range measurement of .001N of lateral force. In the conversation with their application engineer he also said that the calibration needed to read 1/100 micro-strain accurately is difficult and reading such a small strain may be impossible because at that level the wind from the shaft moving may cause the gages to show strain. He also stated that these gages are not recommended for long term readings as they will see creep over time [21]. As a result, Micron Instrument recommended that we use a semi-conductor gage that does not have a backing and create our own Wheatstone bridges, but warned that this would be difficult to get right on the first try. Micron Instruments quoted what it would take to have them perform all the bonding and wiring needed to create a system that would work, but because the lead time required to get such a system made would cause our device to be incomplete for the Design Expo, we decided to accept that the strain gages selected will show changes due to creep. These issues have been discussed with Gordon Krauss, the team sponsor, and he has accepted that the least amount of lateral force read may be .046 N and that the creep may result in more error than anticipated. Appendix K discusses the system recommend by Micron Instruments further.

Analog to Digital Conversion

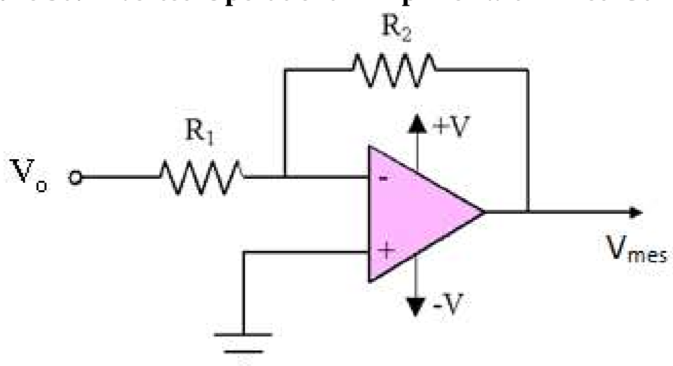
The next step in this process is to insure that the output voltage measurement can be read and stored by the USB-6009 device. This will require that the device can correctly convert the analog voltage signal to a digital signal. The range of the voltage signals expected from the half bridge can be determined using Equation 24. In this equation the gage factor (*GF*, the change of a gage's resistance per unit strain) is determined by equation 25 for the half bridge. This equation compensates for the fact that the gage in the Poisson direction will only see part of the strain that the gage in the axial direction will see.

$$V_o = \frac{V_s GF \varepsilon}{2} \qquad \text{(Eq. 24)}$$

$$GF = \frac{GF_o(1+\nu)}{2} \qquad \text{(Eq. 25)}$$

From the specifications given by Micron Instruments for the strain gage, the gage factor was determined to be 100.5. If we use the recommend excitation voltage of 5V and a minimum and maximum strain of 1/10 of micro-strain and 1.22 X 10^{-3} respectively, we can determine that the range of the output voltage will be 2.5 X 10^{-5} to .25 V. This low range voltage is well below the USB-6009's ability to convert accurately to a digital signal. For this reason, we will need to incorporate an inverting amplifier with a constant gain to amplify the signal so that the low end voltage will be high enough that it will be read by the USB-6009 (> 37.5 mV). The schematic of this circuit is shown in Figure 29. The voltage that will be read after gain is shown in Equation 26.

Figure 30: Inverted Operational Amplifier with Fixed Gain [23]



$$V_{mes} = -\frac{R_2}{R_1} V_0 \qquad \text{(Eq. 26)}$$

It is important to note that the USB-6009 will only read up to 20V. Because of this restriction, the team will also have to create a way to change the gain of the amplifier. To create a low cost system, the supply voltage will be 5V, the same as the excitation voltage for the gages. By changing out the resistors, different gains can be achieved. By having many different resistors whose gain is well understood, the voltage measured can always be near 5 volts. By always getting close to this voltage, the measurement can be more accurate.

Manufacturing

Once the shaft is made, the strain gages will be bonded onto the shaft using a high temperature bond that will ensure that the strain gages will work over the specified temperature range. The procedure that will be used is outlined by documents provided by Vishay Micro-Measurements and will be done under the supervision of Todd Webber, who has experience mounting strain gages. The procedure that will be used can be found in Appendix L.

We will also make the circuit required to accurately read the strain the shaft is experiencing. This will involve mounting a 15 pin female connector to a mini board that will be used to connect to the strain gages on the shaft through a male connector. These connectors will allow for the shaft to be removed from the tribometer without having to remove the entire circuit or de-solder wires. The circuits used to amplify the voltages from the strain gages and the power supply that will supply 5V to the operational amplifiers and Wheatstone bridges will be soldered onto the mini board that the female connector will mount to. The power supply will be built using the steps outlined by the manufacture found online at: http://www.ladyada.net/make/bbpsup/solder.html. This instruction shows where to place every part of the power supply, how to apply solder, and how to test and use the device.

The wires that will be attached to the strain gages will be soldered using solder that will not melt at the 150°C that the shaft may experience in operation. These wires will also be stressed relieved with tie wraps that will not melt at this temperature.

Both of the processes required for completing this system involve the risk of burning skin. This could be done with a solder iron needed to apply the solder to complete circuits or on the shaft when the bond is cured in an oven. During both of these activities special care will be taken to insure that injury is avoided. When soldering, we will be sure to always be aware of the solder iron.

Cost Analysis

A breakdown of the cost for the parts needed to supply the voltage and amplify the signals from the bridges as well as the strain gages that will be attached to the shaft can be found in Table 5. This cost includes the circuit board that these circuits will be made on, the operational amplifiers, and the kit needed to apply the excitation voltage. To allow the removal of the shaft, all wires from the strain gages will be routed through a 15-position sub connector. This will be able to plug into a female sub connector that will be located on the circuit board. It is also important to note that because we might accidentally break a strain gage while bonding it to the shaft, two extras will be ordered. The total cost for this system is $267.37. This cost does not include resistors used in the inverted amplifier system as there are some spare resistors in the X-50 lab that we will use. This cost also does not include the material used to prepare the surface of the shaft and the bonding material for the strain gages, as Todd Webber will allow us to use spare material from the ME 395/495 labs.

Table 5: Cost of Force Gage System

Part	Supplier	Part #	Qnty	Cost/part	Total
Dual Mini Board with 213 Holes	Radio Shack	276-148	1	1.99	1.99
LM741CN Operational Amplifier (8-Pin Dip)	Radio Shack	276-007	3	0.99	2.97
15-Position HD Female Solder D-Sub Connector	Radio Shack	276-1502	1	1.99	1.99
15-Position HD Male Solder D-Sub Connector	Radio Shack	276-1501	1	1.99	1.99
9- and 15-Position Shielded Metal D-Sub Connector Hood	Radio Shack	276-1508	1	2.99	2.99
12v 1A 2.1mm DC Adapter	Fundamental logic web store	N/A	1	7.99	7.99
Adjustable breadboard power supply - v1.0	Adafruit Industries	N/A	1	15	15
Backed Half Bridge Strain Gage	Micron Instruments	SS-080-050-1000PB-SSGH	5	46.49	232.45
				Total	267.37

Calibration and Verification of Force Measurement System

In order for the force measurement system to work correctly it must first be calibrated. This will be done by applying normal and lateral loads of known magnitudes to find an equation that relates the voltage read to the forces applied. Normal forces will be applied by hanging weights of known weight from a string connected to the shaft. Lateral forces will be applied in a similar manner; the strings will be attached to a shaft and run through a pulley that will be placed so that the tension of the string is on the shaft in the lateral direction. By doing this for different loads, and knowing the gains of the amplifier, which can be measured by applying a known voltage and reading the output, the force measurement system can be calibrated over the expected load range.

The creep associated with the strain gages will also be calibrated out. This will be done by applying a normal force and observing how the reading changes over time. By taking measurement s periodically at different normal forces, the creep of the strain gages can be taken into account.

This process would allow for the testing of the force measurement system even if the rest of the tribometer fails or is not built. This test could be done by placing the shaft in a vice and applying forces as described above.

Safety will be a concern when performing these calibration techniques. Special care will be taken to avoid dropping weights onto feet and pinching fingers in the pulley systems. To avoid the possibility of being electrocuted, the circuit board will be covered and the plug will be accessible so that it can easily be pulled if there is a problem.

Environmental Control

(Note: this section is not complete due to personal issues of the engineer working on this system.Because of these issues, this system was not worked on much after the initial alpha design. When this engineer was forced to drop the course because of these issues, the team decided, with approval from the sponsor, to not include an environmental system in the team’s prototype (see Appendix P for the approval of this). This section will outline what work was done before this group member dropped the course.)

Wear testing requires strict environmental control including temperature, humidity, and lubricant. A test chamber should surround the rotational system, and a linear slot should be cut out to allow the pin to reciprocate back and forth. An accordion style insulating device should follow the reciprocating pin, sealing the chamber from outside conditions. This device should be mounted to poles that connect to the linear slide, so that the shaft will not have an outside force acting on it. This device should also direct steam away from the strain gages and motors. The outside of the chamber will be surrounded by insulation to make it easier to control the temperature and humidity inside the chamber.

Temperature control can be done using a thermoelectric control (TEC) to cool the system to 0°C. This system runs current through a sandwich of two dissimilar metals which causes a temperature difference between the two different sides. To heat up to the required 100°C, a resistive heater should be used. A heat sink can be added to the TEC to increase the efficiency of the cooling system and a fan should be added in the environmental box to eliminate gradients. To close the control loop, a thermocouple can be used to read the temperature in the chamber. Based on the temperature, the voltage to the TEC/heater and fan can be varied to control the temperature.

Humidity control can be achieved by buying both an off-the-shelf humidifier to generate humidity, and desiccant material, such as silica gel, to absorb humidity. The humidifier works by atomizing water into the air and blowing it into the controlled environment. To absorb water out of the air, silica gel packets will be put into a chamber, and air will be circulated over them with a fan. By varying the fan speeds of both the humidifier and de-humidifier chambers, the tribometer should be able to control the humidity. The feedback for the humidity control can come from a capacitance sensor which measures the capacitance of the air. As the air becomes more humid the air has less capacitance.

It should be noted that normal humidifiers only work up to 50°C. For this reason we looked into various ways to boil water into the air inside the environmental control box. This may be difficult, as at high temperatures it will take a large amount of water to saturate the air and at low temperatures, the water may condense on the disk and shaft.

A lubricant bath tub could be added onto the sample clamp disk. The lubricant could only be tested at low speeds because the disk is spinning and would simply spin lubricant off creating a giant mess inside the test chamber.

This system can be tested to confirm it is working using a separate humidity and temperature sensor. Testing should be done to confirm that the system works at a wide range of temperatures and humidities. The sensors will also be placed in several parts of the box to confirm that gradients are small.

To ensure safety, this system should first be assembled and tested away from the tribometer. This will allow future teams to first work out the kinks in the environmental system itself. It can then be attached to the tribometer and tested. This should be done so that when the environmental system is assembled with the tribometer the team can concentrate on issues that may occur to the rest of the tribometer (ex: condensation build-up on electrical circuits or melting of parts that are not expected to see high temperatures.)

Design Analysis Assignment

A design analysis assignment was performed examining the material and manufacturing process selection, environmental sustainability, and safety aspects for our device. The complete assignment can be found in Appendix C. This section gives a general summary of what was learned from this assignment.

The material and manufacturing process selection was a key component of our final design, especially for the pin shaft. We originally assumed that the shaft would be made out of stainless steel due to its high strength and corrosion resistant attributes, but when it was discovered that the readily available stainless steels were very difficult to machine, we began more thorough research into the material selection process. By using Ashby’s Material Selection Basics [16], we were able to develop a set of material indices which, when analyzed with the CES software[27], gave us a viable collection of materials that could be used for the pin shaft. We ended up choosing bronze alloy 544 due to its availability and its ability to be machined easily. Without going through the material selection process, we may not have discovered this option and our design would have suffered.

Designing for environmental sustainability taught us how much strain our device really puts on the environment. The analysis performed on the aluminum sides was very eye-opening, as none of us realized how many resources were needed to make the quantity of aluminum used in our design. While we had already purchased the aluminum, and therefore went ahead and used it in our device, this section of the assignment made us all realize how much impact we, as engineers, have on the environment. A simple choice of materials for a relatively small device, such as ours, can have a significant impact upon everything around us. This is a lesson that we will all take with us as we begin our professional engineering careers.

Designing for safety was the final component of the assignment. While safety has always been highly stressed to us throughout our undergraduate careers, the use of the DesignSafe software was a new addition to our general safety knowledge. By performing this assignment, we were able to plan exactly how we would manufacture and assemble our device in a safe manner, which helped us organize our fabrication plan into a more robust strategy that was not only safer, but more time-efficient. More detail regarding DesignSafe is given in Appendix U.

Integration

The integration of each of the four subsystems into a final assembled device has been planned and verified using an assembled CAD model with frequent communication between team members. The strain gauges of the Force Measurement system will be attached to the bronze shaft of the Pin Gantry system. The entire Pin Gantry System will be mounted to the linear motion component of the Motion Control system. The Environmental Control system has been designed to allow the shaft access to the controlled environment box and to fit around the disk of the Motion Control system.

The programming for each respective controlled or measured variable will be a more collaborative effort between team members to avoid issues between separate commands and to ensure that control and measurements performed by the device will be made by a single central system. Inputs from every sensor and encoder will be routed through the central NI USB 6009 DAQ.

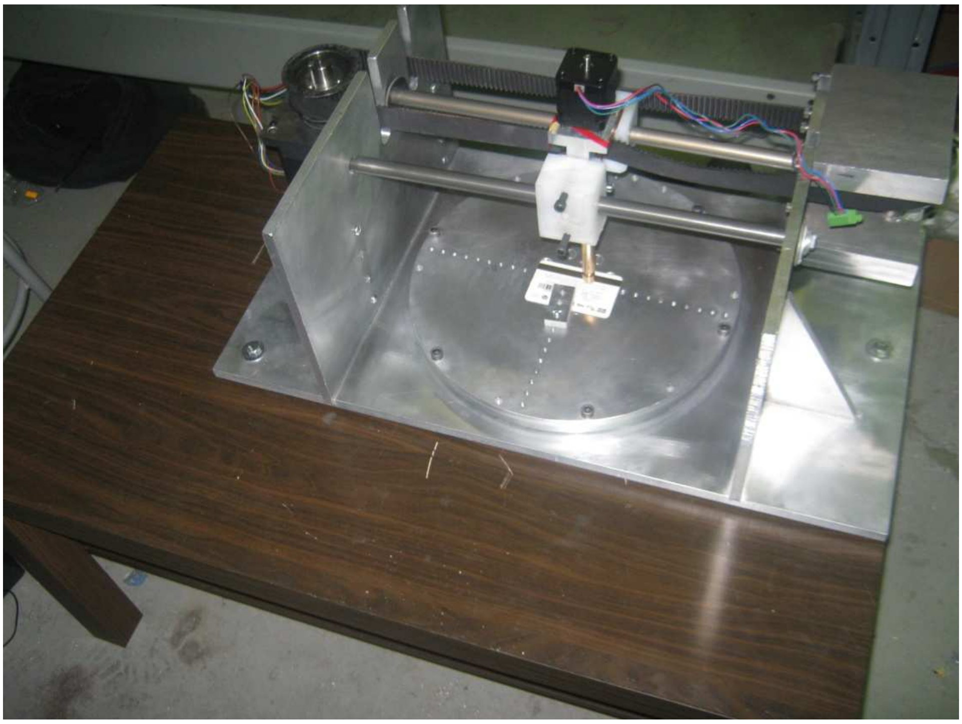
Total Cost Estimation

The total cost estimation for the assembled tribometer includes the costs for each subsystem. Because the entire environmental control system was removed from the design, the cost decreased from an originally estimated $3,100 to $2,500. A complete Bill of Materials detailing the cost item by item can be found in Appendix Q.

Final Design

Pictured below in Figure 31, is the final prototype as it was presented at the design expo on December 11, 2009. The prototype that was presented was incomplete, and was not tested to prove that it could meet the specifications set previously this semester. The linear motion system suffered from late shipments and a lack of time, while the rotational motion system suffered from poor manufacturing. The entire system lacked the proper hardware and software from an electronics standpoint to move at the desired speeds with proper feedback.

Figure 31: Final Prototype

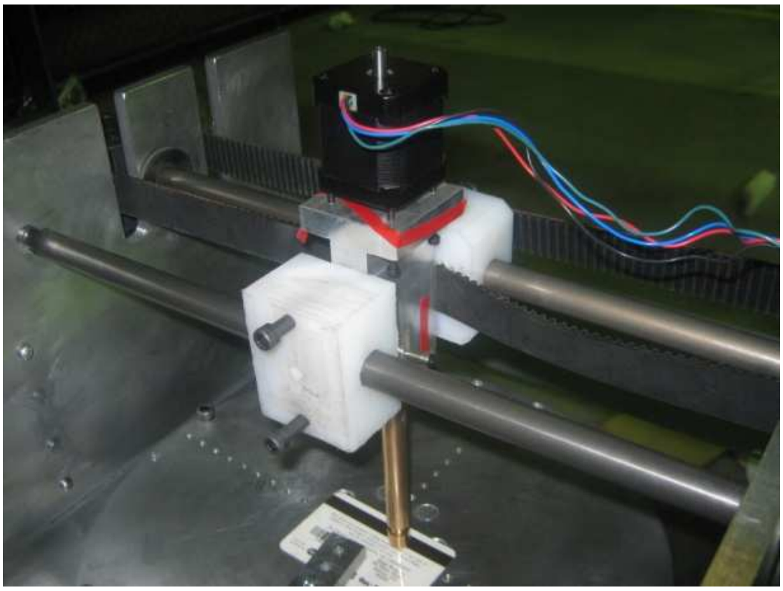


Motion Control

The rotational system was completed mechanically on December 4th, but there are several problems with the final version as built. The tolerances of the entire system were driven by a bearing that had loose tolerances on it, which resulted in a system that was slightly out of round while spinning. The bearing purchased from McMaster Carr was inexpensive, but was not of high enough quality to allow us to achieve acceptable rotational motion. This caused the gear which was mounted concentrically to the hole-pattern to be out of concentricity with the ten inch diameter bearing. Multiple parts out of alignment causes unequal forces on the motor pinion gear, and although it turns, at high speeds this will probably be a problem. The sample mount was forced to be made on a manually operated mill and made round with a band saw and grinder. The sample mount piece is out of round, and would most likely cause vibrations at higher speeds.

The linear system was not completed to what was specified in the plans. The major underlying problem was that the linear bearing, which was the backbone of the project, was not received until the day of the expo. This was due to Igus, the company donating it, not shipping until one week after they said they shipped it. Our team assumed it would be delivered on time, when it was not. We were forced to manufacture a linear bearing system ourselves on short notice, and it is not nearly of the same quality as the Igus system. This interim linear bearing system can be seen in place in Figure 32. This system was made in one day, and does not meet the tolerance or load requirements for the system. The Delrin slider and aluminum rail do not slide very well, and were not able to be aligned correctly, so using this system as a permanent solution is not. The plan is to install the real linear bearing system after this report is turned in.

Figure 32: Interim Linear Bearing System



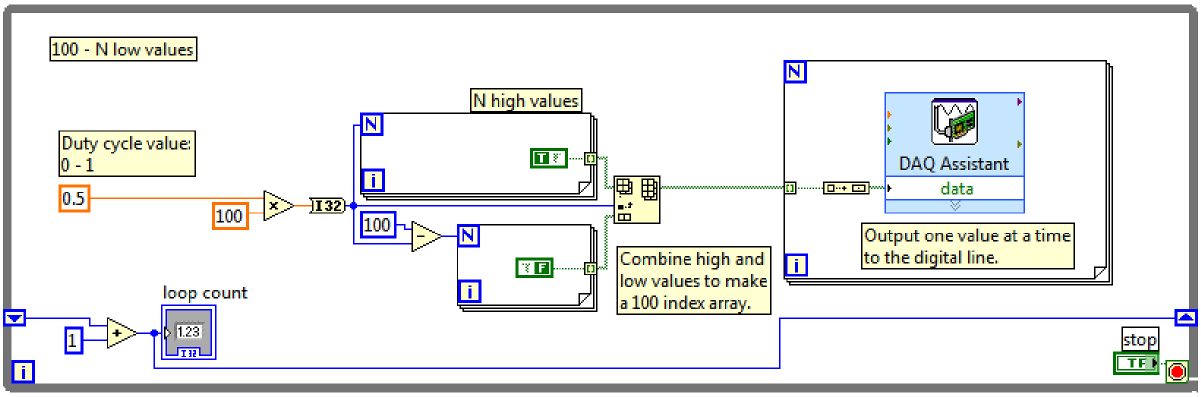
Despite those major problems the motion system went together pretty smoothly. It was extremely hard to manufacture because the pieces required a high tolerance, and had to be bolted to the table of mills and aligned properly. The other difficult part was that the pieces were larger than the travel in the mill so they had to be flipped, and re-aligned. Mistakes occurred from this process, and it led to many incorrectly drilled holes which detract from the appearance of the device, but do not affect its performance. Now that all parts have been manufactured and assembled, most of the systems are aligned.

The final prototype does not include any feedback mechanisms which can verify test parameters on the fly or keep the test system in a safe testing area. The potentiometer, which would be installed on the motor shaft, could not be mounted because we could not machine the motor shaft. This is addressed later on in the recommendations section (Pg. 60). The light sensor that detects rotations on the bottom rotational bearing was not mounted both because there was not enough time, and we did not have a DAQ system that could handle the amount of digital data coming in.

Electronics and Programming

The USB 6009 from National Instruments was initially to be used to control the entire system, but after borrowing a USB 6008 through a connection with a team member, we discovered that it was going to be difficult to control the system with the device. The smart move was to use everything on a trial basis to make sure it was in fact what we needed, and what we found was that the original plan was not going to work. The first problem dealt with acquiring all of the software from the web and finding the most up-to-date and current compatible software that would work with the USB DAQ. After hours of frustrating downloading and uploading firmware, the system could finally send and receive measurements. To verify that it would take measurements, we took a graph and saved data from a simple potentiometer which would model an analog input similar to what the stain gauges we were trying to observe. We received the motors later than we would have liked, and they were hooked up to the DAQ to for testing.   Figure 33 shows a screen shot of the test program which sends out signals based on speed of the loop.

Figure 33: Basic code to send pulses with 50% duty cycle as fast as the software will loop



The virtual instrument was programmed using LabView 2009 from National Instrumentswith help from their fourms. We utilized the DAQ assistant which comes with the drivers for the USB DAQ card. The problem is that this piece of software takes up a significant portion of computer bandwidth, which makes the loop run excessively slow. We did not attempt to go back and re-program to try to improve the test, as it was apparent that the USB DAQ would be not capable of achieving the speeds we need to send pulse to the motor.

We were able to run the small normal force stepper motor, but we saw speeds of around one cycle per second. This equates to only .15 revolutions per minute using the smallest micro stepping we could, which is a far cry from the goal of 600 revolutions per minute. The same code was applied to test if the larger motion-control motors could spin. They would not spin due to the fact that the DAQ card could not push 5V through to the NEMA 42 stepper motor controllers. Measuring with a volt meter showed only a voltage of 1.4V going across the leads when it was attached, but when it was not attached to the motor controller it would output 5V. We believe the DAQ simply did not have enough power to overcome a suspected internal resistance in the controller.

The problems behind the speed issue were tackled next. It turns out that the initial assumptions that the USB DAQ would be fast enough to generate a signal was completely wrong. The DAQ cannot generate signals based on hardware clocks due to the fact that it doesn't have any. Instead, it relies on clocks from the host computer transported through USB. This means that the signal we generate is "software timed", indicating it updates based on the speed of the loop running in the software. This would be fine if we were only trying to generate a few number of pulses per second, but we are try to generate upwards of 6000 pulse per second. We tried to use a controller that looped faster and utilized interrupts (hardware timers), but due to time constraints and a lack of programming and electronics understanding we were unable to successfully get it to work.

Pin Gantry

All parts of the pin gantry system were successfully manufactured and assembled in accordance with our design. However, due to issues with the force measurement system and programming difficulties, a fully validated, working system was not achieved.

Unfortunately, the desired stepper motor used in the design of the system was out of stock for twelve weeks. Due to the late notification of this issue, a less powerful stepper motor had to be used as a replacement. The stepper motor used was a NEMA 17 with a holding torque of 62 oz-in purchased from Keling while our original design called for a motor with a holding torque of 100 oz-in. To achieve a normal force of 200 N, the motor needed to be able to produce 76 oz-in of torque. Therefore our prototype will not be able to reach the maximum normal load specification of 200N. The replacement motor will be able to transmit a normal force of 172 N.

Two Smalley wave springs were successfully intertwined into one, effectively doubling the wire thickness and thus resulting in higher spring constant. This intertwined spring should theoretically be able to handle loads greater than 200 N, however, we were not able to perform a physical validation test. Our validation plan for the pin gantry system was dependent upon the successful integration of the force measurement system into the device. Due to multiple issues with the strain gages, the force measurement was not in acceptable working condition. Details regarding the force measurement system of the prototype are given in the next section.

We were able to successfully manufacture and assemble the aluminum chassis, bronze shaft, ball holder, and threaded rod. We were also able to rotate the stepper motor and attach it to the both the chassis and threaded rod. However, the threaded rod was not attached as snugly to the motor shaft as we would have liked. We placed set screws through the threaded rod aligned with the flat of the motor shaft in order to allow the threaded rod to move vertically as the motor rotated. There was a small amount of slop between the set screws and the motor shaft which meant that the motor could step one pulse, but the threaded rod would not move with it. It took multiple pulses for the threaded rod to move with the motor. This meant that our method of counting the number of pulses the stepper motor moved, and then calculating the respective vertical distance that the rotation yielded was not accurate.

Due to lack of time the electronic resistance measurement system was not manufactured with the device. The team decided that having working motion and force measurement systems were higher priorities than having an electronic measurement system. The design of the system can be found in Appendix T.

Force Measurement

Due to several issues that occurred during the last few weeks of this project, the force sensing system did not meet the specifications. This section of the report will explain what happened and the status of this system.

The first issue occurred when Micron Instruments did not ship the strain gages as quickly as we expected them to. When we had first discussed ordering the SSGH gages, Micron Instruments said that the gages were in stock and would ship the day the order was placed or the next day. We submitted a purchase order form into the Department of Mechanical Engineering Administration Office on 19 November 2009. When this was done, we were told that they would place our order on that day or the next day. Micron did not ship our order until 1 December 2009. The strain gages arrived on 3 December 2009, one week before the prototype was due.

On the day the gages arrived, they were given directly to Todd Wilber, the technician who we had asked to bond the gages onto the shaft for us, with the shaft they were to be placed on. We gave him verbal instructions on how to place the gages onto the shaft. He was told to place the gages so that the wires would come down the shaft towards the threaded end. We also gave him the information provided by Micron Instruments about how to unpack the gages, found in Appendix N, information about the gages, found in Appendix O, and how to bond the gages, found in Appendix L. He told us that a lot of other groups had asked him to bond gages to their projects and that he would do ours as soon as possible.

He contacted us on 7 December 2009 asking us to answer some questions he had. When we met with him, he showed us the shaft with the gages mounted on them. He had mounted them upside down and soldered wires directly to two of the gages instead of on the solder pads. We then explained to him what we wanted and that the way he had mounted and wired the gages was incorrect. He said he would re-wire the gages properly. He did this and gave us the shaft with mounted and wired gages on 8 December 2009.

Upon receiving the shaft, the team then noticed there was another issue. The gages that Micron Instruments had shipped did not come with resistors placed on them as the team expected. These resistors are needed to complete the half bridge. A call was made to Micron Instruments on 9 December 2009 asking if the gages were supposed to come with these resistors already placed on the backing as indicated on the technical drawings from their website. Micron Instruments informed us that these resistors were to be selected and placed by the customers once the gage was bonded to the shaft, so that the bridge could be accurately balanced. This was not specified anywhere on Micron Instruments website. The technician said that this was supposed to be implicitly obvious.

Because Todd Wilber had already put a protective coat over the strain gage and the team did not have any resistors flat chip resistors, the team decided to try and place a round resistor in series with the axial strain gage and read the voltage drop across the strain gage. When we asked Micron Instruments if they thought this was a good idea to do on short notice, they agreed that it was our best option. This seemed to only work on one strain gage at high loads. We believe this is for two reasons. First, we think that some of the gages were damaged when they were improperly wired. Secondly, we think that because we were using a voltage source and not a current source, this reading was not as accurate as it needed to be. After working on this system for an entire day and not making much progress, the team decided to focus on getting the rest of the system as completed as possible, instead of investing time in what the team thought may have been broken gages

The team did complete the circuit board and power supply as previously described. The voltage supply works as advertised and the operational amplifiers were wired to test. The team did remove the power to the amplifiers when they attempted the fix described in the previous paragraph. This was because the voltage expected to be seen was large, and the team expected that the range of voltage after amplification would be larger than the cut off voltage of the amplifier.

Fabrication Plan

A detailed, step-by-step fabrication plan for assembling the device can be found in Appendix S. Many of the required parts were manufactured by our team. The part drawings and manufacturing plans can be found in Appendices H and I, respectively. The rationale for the design of the parts, as well as part descriptions, can be found in the Engineering Analysis section beginning on Pg. 26.

Validation Results

Due to the circumstances previously discussed, we either did not have enough time or were unable to validate the prototype for many of the components. For this reason, we recommend that in the near future, the prototype be validated in the ways described in the engineering analysis section. This validation includes the testing of the motion systems, calibration and validation of the force measurement system once it is completed and the validation of the wear measurement. The tribometer must also be put through tests to make sure it meets the specifications. These tests would include making sure it can apply the maximum and minimum forces and that it can run tests as long as the specified one-million cycles. Once the testing is complete, the results will need to be analyzed so that the weak points in the design can be identified and fixed.

Discussion

This section contains a critique and discussion of our design. It includes what we would have done differently and what aspects of our design we would modify to improve the function of our prototype.

Nearly all of the issues with our device are directly caused by our team not having enough time to fully implement all of the aspects we planned on completing. Throughout the semester, we became more aware of how generating a detailed plan benefits the project. Unfortunately, we realized this too late and were still waiting on parts at the time of the design expo. If we had fully explored exactly what items we needed earlier, and proceeded to order them, our manufacturing, assembly, and validation tasks would all have gone much more smoothly.

The ordering of materials was also affected by incidences of poor communication by all parties involved with the project. The problems started when the mechanical engineering department did not inform us that they had changed where our packages would be shipped. Their emails informing us when our items arrived were also never received by any of our team members. This resulted in our team spending several days attempting to track down packages that were already sitting in the building, but not in the room we specified. This pushed back the manufacturing plan and put more pressure on an already tight timeline. We also failed to verify that some other packages had been shipped when they were supposed to have been, which significantly delayed our assembly process. Our group learned how important clear communication is for engineering projects and that assuming items have shipped is not a good practice.

Tolerances and machining also caused some bumps along the road towards a final prototype. The motion control system had several parts that were difficult to make because it required the removal of the vice and special alignment by clamping the part to the table. The group was already limited by the fact that it was down to three people to make these parts, and also limited by the machine time of everyone else in the shop trying to make parts. Even with resources utilized at another shop, the rotational parts did not align together as planned and the linear idler tensioning system was messed up by another group member. Some of the pieces that we thought would be the simplest turned out to be the most difficult to machine. The idler pulley is a good example. It was supposed to be simply pocketed out, with bearings pressed in afterwards. The idler turned into a three day ordeal that involved a basic CNC code, and a mill, which then created two unaligned holes. This was caused by the part not being perfectly aligned and round, and the CNC mill not being leveled. The bearings were also squeezed too hard to allow them to turn. At the end of the day, we received a piece of bronze, and pressed it into a fully bored out idler pulley. This is shown in Appendix P under a design change.

Because of the high quantity of parts that we were required to manufacture ourselves, we did not have the luxury to ensure that each part was perfectly made. Due to many group members' unfamiliarity with machining precision parts, and the low availability of machines in the shop, we were forced to use machined parts that did not always exactly match what was called for in our design. We feel that if we had been able to manufacture all of the parts to the tolerances required by our design, our prototype would have met most of the specifications we were striving to achieve.

According to our project plan, electronics and programming was to be performed after manufacturing and mechanical assembly were completed. Because both of these tasks took longer than planned, the programming and electronics tasks did not receive the necessary amount of time to be achieved. The team already had a lack of programming skills, and the electronics programming needed for this was not simple. The original plan to use a USB system from National Instruments was thrown out the window late in the assembly process, and another controller was brought in. Further thought would probably lead to a hybrid system between several different electronics systems of both low and high levels, but we simply ran out of time to tackle this process.

Recommendations

Motion Control

The design of the motion system is still regarded by our team as a solid design that was just too difficult to build given the problems that we ran into with logistics, losing a group member, and machining time and capability constraints. We believe the downfall of the design was a time issue rather than a failure of the design itself. Had the design been executed properly, we feel that it would have accomplished all of its tasks. Our recommendation would be to keep the overall ideas and fundamentals of the motion control system as well as the method for producing the motion, while improving individual components and tolerance levels of parts to create a better end product.

The motion system was not fully tested due to problems discussed in the Electronics and Programming section, which follows directly after this one, but we would recommend continuing this project to help further explore low-cost multi-test tribometry. We have several recommendations for the motion system that should be investigated to improve on this current design.

After running into several problems with the low-cost bearing purchased for this project, it would be advantageous for future teams to research better rotational bearings. Groups could even find bearings that have gears integrated into the sides of the bearing. A large bearing should probably still be used, but teams could also look into using a combination of small roller bearings on a round track. The bearings would also likely be better suited for the environmental system that should be integrated in the future.

Although the original plans called for pocketed out pieces and lightening holes, we simply ran out of time to machine all of these features. Making the system lighter will help it accelerate up to the specified speeds easier, and will make the entire device more portable. To make the "diet" easier and to increase the accuracy on the making of parts it would be a good idea to find a CNC shop that could create the major parts of the motion system.

One question that was brought up at the design expo was about belt slip. The design utilizes a belt tensioning device on the side opposite of the motor, and that should prevent belt slip altogether, but there is a way to detect it as well. If we put the potentiometer on a shaft that was integrated with the idler pulley used to keep the system tensioned, then we would be observing what the motors moves instead of the motor itself moving. This would require revamping the current idler/tensioning system, but the system could then detect if there was belt slip.

Electronics and Programming

We would advise obtaining the aid of someone who is well-versed with electronics and programming LabView or placing a student with advanced knowledge of these fields on the team to work specifically on developing the programming and electronics of this system. A significant obstacle in this project for our team was the lack of help and resources from an electronics and programming standpoint.

With a dedicated knowledge base of electronics and programming, the team and project would benefit the selection of correct equipment early on and would suffer less from last minute problems that occurred because of a lack of knowledge. We would recommend looking into different controllers/DAQ systems that can handle more raw digital data at a higher rate using interrupts or an FPGA. These devices use hardware instead of software and that would be the recommended path for future systems.

Safety should also be a bigger aspect in the electronics and programming. The USB DAQ did not have any safety system in place in case of error nous programming. On many systems there is a built in "watch dog timer" which ensures that all loops, threads, and programs do not get hung up and cause problems that would freeze the state of the hardware. There should also be emergency cut off switches integrated into the electronics that would cut the power supplies to the motor controllers in case of an emergency.

Integrating a set of control states, such as a disable control, operator control, and autonomous control states, is also recommended. The disable state would allow power to be on and sensor readings to take place, without allowing motion to occur. In operator control, the operator could jog the device around utilizing safety sensors, while still reading other sensors' data. One could use operator mode for tasks such as calibrating. Autonomous mode could be utilized to run tests that are programmed through the user interface on the computer connected to the tribometer.

Although we did not have much time to test, in what validation we did perform, we noticed that the stepper motor became extremely hot. The motor operated continuously for around 5 minutes, so we are not sure if steppers can handle continuous motion similar to what is observed in the current design. We would recommend analyzing this problem, and researching if steppers are used in continuous situations.

Pin Gantry

In order for our prototype to successfully meet the requirements and specifications, some changes must be made to the current assembled device. This section details what can be done to produce a fully functioning pin gantry system.

There are a few options for achieving the normal load application of 200 N. The use of a more powerful stepper motor is the simplest, and was what our design originally called for. However, the current stepper motor can achieve the load of 200 N by adding a gear train system between the motor shaft and threaded rod. This will require purchasing the appropriate gears and then machining and attaching them to the motor shaft and threaded rod. While using a gear train with the current stepper motor will be cheaper than purchasing a more powerful motor, it will also add unknown complexities to the linear reciprocating test. If the entire pin gantry system is moving back and forth at high speeds, there is a possibility of the gear train system getting thrown out of alignment and becoming ineffective. Therefore, we recommend the purchase of the more powerful NEMA 17 motor from Anaheim Automation along with the appropriate driver and power supply. These parts are given below.

Recommended Stepper Motor Items

Normal Force Motor	NEMA 17 Bipolar Stepper Motor	Anaheim Automation	17Y402D-LW4	$43.80
Microstepping Driver	Microstepping Driver 1600 ppr	Anaheim Automation	MBC25081	$80.00
Power Supply	Stepper Motor Power Supply	Anaheim Automation	PSAM24V2.7A	$100.00

To ensure that there is no slop between the threaded rod and the stepper motor shaft, we recommend machining a flat into the threaded rod rather than using set screws. The greater surface area of a flat compared to set screws will result in a tighter fit between the two parts and will allow little to no rotation of the shaft independent of the threaded rod.

Force Measurement

The team recommends that the previously alluded to system developed with the help of Micron Instruments and described in Appendix K be implemented in order to get a force measurement system that can meet the specifications. This system is expensive, but as seen by the results of our prototype, it is required to get a system that works.

There are also two gages left in the packaging that will be submitted to the sponsor with the prototype. These gages could be used to try a second time to get the current system to work. The flat resistors required would need to be purchased to make this attempt. These strain gages may also be returned to Micron Instruments when a new shaft is sent to Micron to try the design described in Appendix K.

Conclusion

After performing initial research, identifying specifications, conducting design analysis, and manufacturing and testing our device, we have produced a Final Design prototype. The final cost of our prototype, which we presented at the Design Expo on December 10, 2009, was $2,500. Despite our team's best efforts, we were not able to achieve many of the specifications originally set forth by our sponsor. Table 6 shows which specifications would be met if a working computer controller was acquired and validation had occurred. It also shows which specifications we know that we cannot meet based on our design, due to physical limitations of our device. Although our team is disappointed that we were not able to deliver a fully-functioning prototype that met all of cthe specifications, we each feel that this prototype is a respectable starting point for a system that costs only 4% of similar products on today's market.

Does not meet specifications
Design must be validated to ensure specifications
Environmental control not included

Table 6: Design Specifications

Specifications	Parameters	Page
Linear Testing Speed m/s (ft/s)	.01*-1* (.03-3.28)	
Rotational Testing Speed rad/s (RPM)	.3-20π (2.9-600)	56
Measurement Resolution of distance & force (%)	1	
Control Precision mm (in.)	1 (.04)	
Test Specimen Size Diameter mm (in.)	25.4-254 (1–10)	
Test Specimen diameter resolution mm (in.)	.025 (.001)	
Volume cm (in.)	60.96x60.96x91.44 (24x24x36)	
Normal Load Range N (lb)	.1-200 (.02-45)	57
Friction Coefficient Measurement	.01-1	
Stroke Length cm (in.)	0-25.4 (0-10)	
Data Acquisition Frequency (kHz)	20	
Operating Temperature Range °C (°F)	0-150 (32-302)	51
Temperature Resolution °C (°F)	5 (5)	51
Relative Humidity Range (% water)	0-100	51
Relative Humidity Resolution(%)	3	51
Electrical Contact Resistance Range (Ohms)	0-1000	57
Electrical Resistance Resolution(ohms)	1	57
Ball Sizes mm (in.)	1.59, 3.18, 6.35 (1/16 , 1/8, 1/4)	
Wear range mm (in.)	0-3.18 (0-1/8)	
Wear resolution in (µm)	9.8 x 10^{-5} (2.5)	
Total Cycles	0-1M	
Distance Measurement in (mm)	±.04 (±1)	
Perpendicularity of the Disk to the pin (degrees)	± 1	

Acknowledgements

Throughout the semester, our team received valuable input and guidance from multiple individuals, groups, and companies. Their ideas and tips helped us in every facet of our project from the initial design all the way to the manufacturing and assembly of our device. Our team would like to acknowledge the following people and thank them for their help.

- Gordon Krauss
- Dan Johnson
- Arup Gangopadyay
- Sean Hickman
- John Baker
- Bob Coury
- Marvin Cressy
- Todd Wilber
- Smalley Steel Ring Co.
- igus, Inc.

References

1. Carnes, K. (2005). The Ten Greatest Events in Tribology History [Electronic Version]. *Tribology and Lubrication Technology, June*

2. Davidson Physics. (2005) *Brief History of Friction.* Retrieved from http://webphysics.davidson.edu/faculty/dmb/PY430/friction/history.html

3. Buyanovskii, A. (1994). Tribological Test Methods and Apparatus. *Chemistry and Technology of Fuels and Oils, 30* (3), 133-147.

4. Krauss, Gordon. Personal INTERVIEW. 17 September 2009.

5. Gilett, C.E. US1534014 http://www.freepatentsonline.com/1534014.html

6. Bao, Frank W. 4051713 http://www.freepatentsonline.com/4051713.html

7. Nanovea (2009). Tribometer [Brochure].

8. CETR (2009). Comprehensive Materials Testing for Mechanical and Triobological Properties [Brochure].

9. ASTM. "ASTM G 99-05." Web. http://specs4.ihserc.com/DocViewFrame.aspx?sess=217736362&prod=SPECS4&docid=LBHYHBAAAAAAAAAA

10. ASTM. "ASTM G 99-05." Web. ASTM. "ASTM G 133-05." Web. http://specs4.ihserc.com/DocViewFrame.aspx?sess=217736362&prod=SPECS4&docid=LBHYHBAAAAAAAAAA

11. Herguth Laboratories, Inc. *Description Of Lubricating Oil Test With Pin-On-Disk Tribometer* [PDF document]. Retrieved from http://www.herguth.com/

12. Mark, M. A. (2004). *SRV Friction and Wear Testing of Automotive Components* [PowerPoint slides]. Retrieved from http://www.tribologytesting.com/

13. *Measuring Strain with a Strain Gauge*. National Instruments. 2003.

14. Gangopadyay, Arup. Personal INTERVIEW. 25 September 2009.

15. Baker, John. Personal INTERVIEW. 6 October 2009.

16. Ashby, Michael. Materials Selection in Mechanical Design 3rd ed. Butterworth-Heinemann(2005), pp. 65-84.

17. Hickman, Sean. Personal INTERVIEW. 9 October 2009.

18. “Patent Storm: U.S. Patents.” Web. http://www.patentstorm.us/

19. Johnson, Dan. “Materials Selection and Assignment.” Lecture. 13 October 2009.

20. Micron Instruments. "Backed Semiconductor Strain Gage Half Bridge – Temperature Compensated." Web. http://www.microninstruments.com/store/ssgh.aspx

21. Chelner, Herb. Personal INTERVIEW. 6 November 2009.

22. Tibhuvan University. "Stress, Strain, and Strain Gages." Web. http://mech.ioe.edu.np/stress.htm

23. Wright State University. "Operational Amplifier." Web. http://www.cs.wright.edu/~phe/EGR199/Lab_2/

24. Omegadyne. "Strain Gages Transducer Quality Dual Element 90° Biaxial Tee Rosette." Web. http://www.omegadyne.com/ppt/prod.html?ref=SGT_2-ELEMENT90

25. Free Circuit Diagrams. “Linear Scale Analog Ohm Meter.” Web. http://freecircuitdiagram.com/2008/08/21/using-current-source-to-make-linear-scale-analog-ohm-meter/

26. McMaster Carr. “McMaster Carr – Bronze.” Web. http://www.mcmaster.com/#bronze/=4x6hw2

27. CES EduPack. Software. Granta Design Limited, 2009.

Appendix A: Bios

Brian Hopton: I am a senior hailing from Novi, MI and I never thought I wanted to be an engineer because that is what my dad did. After joining the FIRST robotics team (503-Frog Force) at my high school, I quickly found out that engineers are the ones who actually solve all of the world's problems. After winning the national competition junior year, and placing 4th at nationals senior year I applied to mechanical engineering at the University of Michigan. I'm actually a fan of aerospace engineering, but after learning that aerospace companies hire more mechanical engineers than aerospace I quickly changed my priorities. Mechanical engineers actually do more hands on "dirty work" than many of the other disciplines working on computers or with chemicals and molecules. In my free time I mentor high school students on a FIRST robotics team (830-Rat Pack), and I also am the president of the SAE Aero Competition team (M-Fly). Playing paintball and guitar are also some hobbies of mine. After getting my undergraduate degree here, I will either pursue a masters in mechanical engineering or start a career in the aerospace industry.

Brian Kirby: I'm from Pinckney, Michigan. Pinckney is a small town that is located 30 minutes northwest of Ann Arbor. I have been interested in engineering since my freshman year of high school, when I took a class on robotics and automation. I liked that class because I got hands on experience designing and building automated fixtures. I ended up taking that class all four years of my high school career. In my junior year, I built a workcell that built baseball bat key chains from scratch. This workcell won the National Robotic Challenge Gold award for automated workcells.

The last two years, I've been interning at the Laboratory for Atmospheric and Space Physics (LASP) at the University of Colorado – Boulder. LASP has allowed me to work on three satellite instruments over these two summers as a calibration and test engineer. I enjoyed working there because they allow me to do many different types of tasks – designing, running experiments, analyzing data, machining ect. I also like the idea of my working being shot into space. I enjoy working on space technology so much, that I am currently enrolled in the SGUS program for Space Engineering here at the University of Michigan.

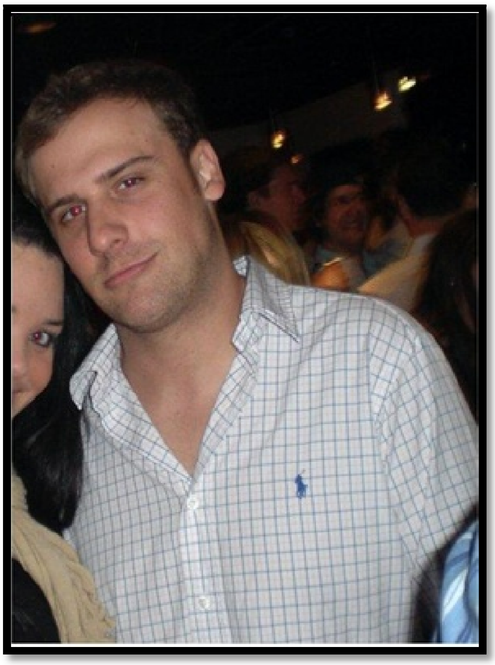
My plan is to graduate next year with my masters in Space Engineering and to work on satellites. I also wish at some point in my career to work on something like the Mars rovers and possibly manned missions.

Tristan Kreutzberg: Born and raised in Houston, TX, I grew up five minutes away from Johnson Space Center. Both of my parents were involved with NASA throughout their professional careers, so I was always aware of what was going on with the U.S. manned space program. Surprisingly, I had no interest in being an astronaut as a young kid – I was fascinated by dinosaurs and was going to be a paleontologist. However, as I grew up I became less interested in velociraptors and more interested in how things work. By the time I graduated high school I knew I wanted to be an engineer, but I wasn't sure exactly what profession I would pursue. I decided on mechanical because I figured it gave me the most options in terms of careers.

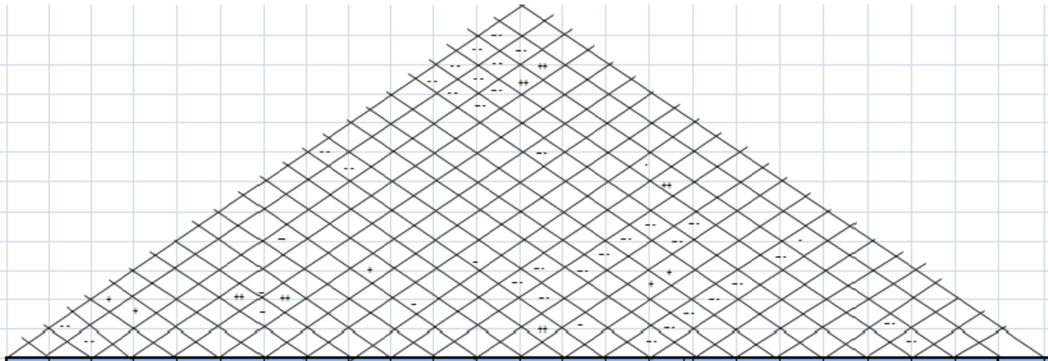
Now I'm in my fourth and final year as an undergrad and am looking forward to working as part of the manned space program, just like my parents. I've interned for Lockheed Martin back home in Houston for the past three summers, which has been a great experience. It confirmed for me that a career involving space is indeed the place for me.

I also enjoy running, playing guitar, and recording music on my laptop. I've also been known to play Ultimate Frisbee from time to time.

Justin Hresko: I hail from a small town in Michigan called Flushing. It's a pretty nice place however it is only 10 minutes outside the infamous city of Flint. So as you can guess, a large part of my family and friends work (or I should say worked) for General Motors. My dad and uncles were all engineers so I guess I just picked up where they left off. I am in my fifth year of undergraduate school here at Michigan and am glad to be finishing up. I am still unsure of what the future lies ahead of me. One option is to work for General Motors; however I don't know if in would like to end up out of state or for that matter overseas. Another option is a family owned debt collection agency. Not quite the company that breaks down doors, but they do collections for hospitals (including U of M), Kohl's Department Store, and law firms. In my spare time I enjoy playing and watching sports. I grew up with a baseball diamond in my back yard but I also played hockey, football, and tennis. I also enjoy video games, anything to do with Erin Andrews, and long walks (seriously). I also enjoy fishing and hunting. I have taken a couple hunting trips to Alaska and the northern part of Canada which were really intense and enjoyable.



Appendix B: QFD Chart



Relationships

++ Strong Positive
\+ Medium Positive
\- Medium Negative
-- Strong Negative

Design Criteria/Customer Needs	Weight	Linear Testing Speed (+)	Rotational Testing Speed (+)	Distance Measurement Resolution (-)	Volume (-)	Maximum Normal Force (+)	Test Specimen Size (+)	Safety Button and Saftey Shield	Weight (-)	Max Friction Force (+)	Data Acquisition Frequency (+)	Maximum Operating Temperature (+)	Maximum Test Radius (+)	Electrical Contact Resistace (+)	Maximum Stroke Length (+)	Cost (-)	Temperature Measurement Resolution (-)	Force Measurement Resolution (-)	Pin Wear Measurement Resolution (-)	Disk Wear Measurement Resolution (-)	Minimize Vibrations (-)	Amplitude of Oscilation (+)	Repeatability of Path (+)	Stopping Distance Resolution (-)	**Team 21**	**Nanovea**	**CETR**
Combining Testing Methods	10	3	3								1					3									5	1	1
Affordable	9	3	3	9	9	3			3		3			1		9	9	9	9	9	3	3	3	3	5	2	2
Reliability	7	1	1								3	1	1	1	1	9		3			9	3	3	3			
Meets ASTM standards	4	9	9			3	9			1			3		3	1	1	1	1	1					4	5	5
Accurate and Precise Data Collection	7	3	3	9							3			3		9	9	9	9	9	9	9	3	9	3	4	5
Operate Safely	8	3	3					9								3									4	5	5
Fits on a Small Table	8				9		3		3				1		1	3									5	1	2
Measure Electrical Resistance	7			3							3			9		3									4	3	5
Record and display data during test	8	3	3	3							9			3		3	3	9				9			4	4	4
Wear Depth Measurement	7			9												9			9	9					3	4	1
Force Measurement	8	1	1	9		9				9	1					9		9							4	5	5
Temperature control	4											9				3	9								3	5	4
Distance Measurement	7	3	3	9							1		1		3	9								9	3	4	4
Ambient Condition Measurement	5	1	1	9												3									3	5	5
Constant Speed during Testing	8	3	3	3							3					3									4	4	4
Repeatability	7	1	1	3												9					3		9		3	4	4
	Total	234	234	477	153	111	60	72	51	76	211	43	34	124	48	646	208	313	211	211	174	183	132	174			
	Scaled	0.491	0.491	1	0.321	0.233	0.126	0.151	0.107	0.159	0.442	0.09	0.071	0.26	0.101	1.354	0.436	0.656	0.442	0.442	0.365	0.384	0.277	0.365			
	Relative Weight	12%	12%	25%	8%	6%	3%	4%	3%	4%	11%	2%	2%	6%	2%	34%	11%	16%	11%	11%	9%	9%	7%	9%			
	Rank	4	4	2	13	16	19	18	20	17	6	22	23	15	21	1	9	3	6	6	11	10	14	11			
	Unit of Measuremen	m/s	rpm	%	ft^3	N	in^2		lbs	N	kHz	C	in	Ohms	in.	$	%	%	%	%	Hz	mm	%	mm			
Benchmarking	Team 21	1	600	1	12	200	400	Y	40	10	20	100	10	1000	10	400	1	1	1	1			1	< 1			
	Nanovea	0.1	500	N/A		60		Y	66		8	200			0.98												
	CETR	30	7,000	0.1		1,200		Y	N/A		20	1,000		1*10^6	2.95												

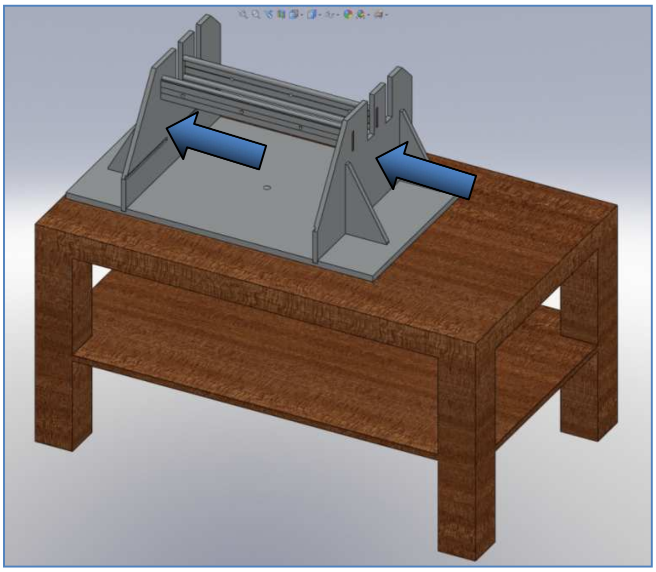
Appendix C: Design Analysis Assignment

Material and Manufacturing Process Selection

Chassis Sides Material Selection

This section will discuss how the material was selected for the sides of the chassis. The sides are pictured in the figure below and denoted by arrows.

Figure C1: A picture indicating the sides of the chassis



These sides are used to support the linear slide that the pin-gantry system is mounted to. These sides also support the motor, belt drive, and idler that are used to create the required linear motion. The sides must be able to withstand the cyclic forces that will be applied because of the force application performed by the pin-gantry system. The sides must also be as light weight as possible so that the device is portable. These pieces also have many holes and slots that require the selected material to be machinable to high tolerances. These parts must also have a high stiffness and low cost. Finally, the sides must be able to withstand temperatures up to 150°C and high humidity.

The first step that was taken to find the optimal material was to create a material indice. This is a process by which we create a parameter that consists of material properties that we want to maximize. In this case, we can start by using a parameter that is used to find the ratio of stiffness of a material to the density and cost of a material. The indice used is shown in the equation below where *E* is the Young's modulus, C_m is the cost of the material, and ρ is the density of the material.

— (Eq. C1)

By plotting this indice's denominator versus its numerator for several materials, we can determine how different materials compare to each other based on this parameter. The following figure shows a logarithmic graph of wide range of materials plotted against each other. The materials in the upper left corner of the graph are rated the highest on this parameter, while the materials in the lower right corner are the materials that rate the lowest.

Figure C2: Graph Comparing Materials Based on Material Indice [27]

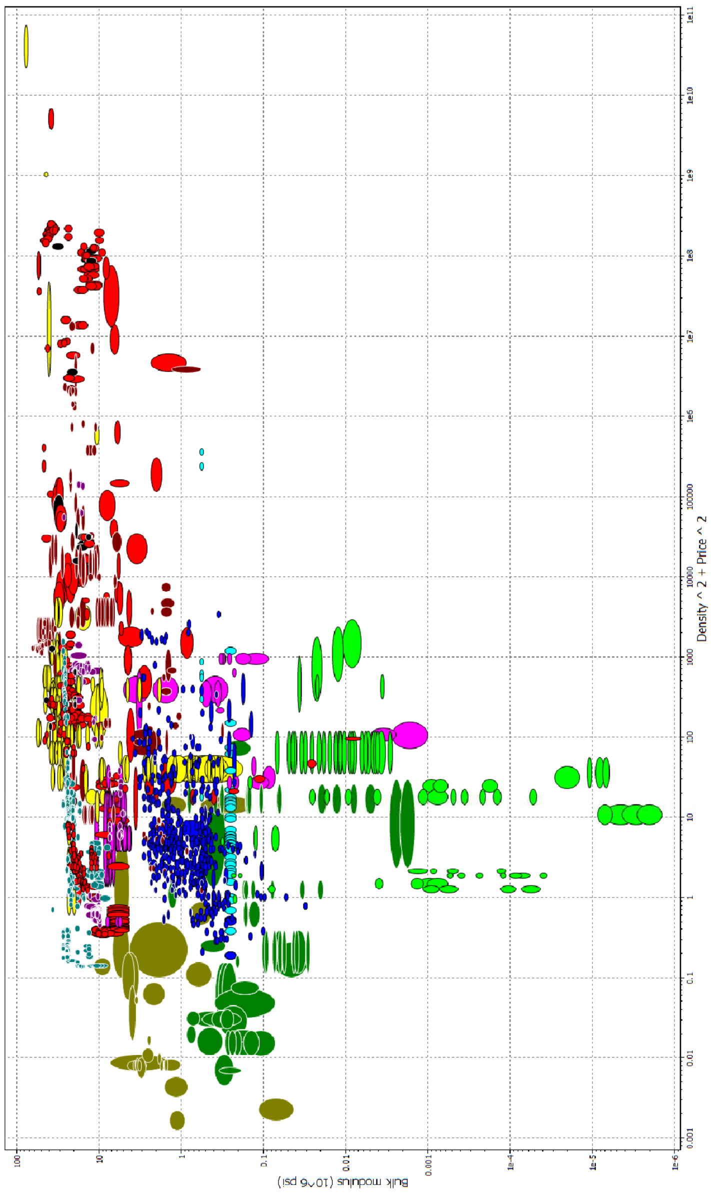
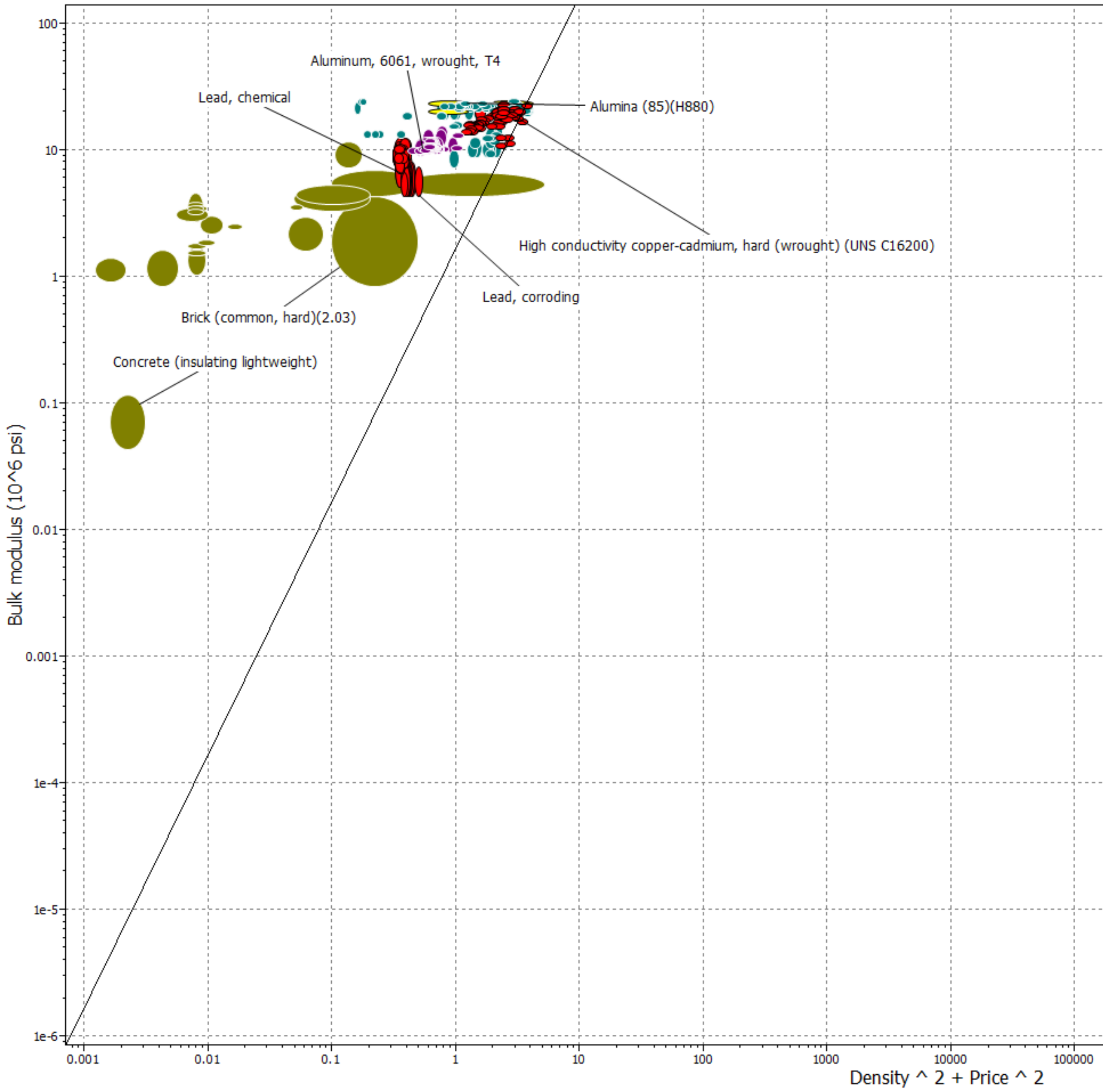


Figure C3: Comparison of Materials that can Withstand the Expected Conditions Based on Material Index [27]



From this process we can select five materials that have met the requirements that we have specified so far. These materials are Aluminum 6061, Copper-Cadmium, Alumina 85, Lead, and Concrete. We can then further analyze these materials based on some of the material properties already discussed as well as others. Significant properties are listed for these materials in Table C1.

Table C1: Comparison of Selected Materials [27]

	Materials				
Properties	**Concrete**	**Cu-Cd**	**Alum 6061**	**Lead**	**Alumina 85**
Yield Strength (KSI)	.0145	49.3	28	.87	24.8
Young's Modulus (MPSI)	.087	48.3	9.86	1.89	35.4
Fatigue Strength at 10^7 Cycles (KSI in$^{1/2}$)	.0218	32.6	30	.725	21.1
Price ($/lb)	.0188	1.45	.713	.438	.752
Density (lb/in^3)	.0325	.323	.0965	.409	.125
Notes	Low Wear Resistance	Not resistant to Acids	Used for Heavy Duty Structures	Poisonous	Only Machined by Water Jets

Using the information in Table C1, we decided that aluminum 6061-T4 is the best material to make the sides out of. This material maximizes stiffness to cost and density. It also has high fatigue strength and Young's modulus, while being able to withstand the expected conditions. Concrete was eliminated because of its low fatigue strength and its susceptibility to wear. Copper-cadmium was eliminated due to its not being able to withstand acids which may be used as lubricants and its high price. Lead was eliminated because it is poisonous to humans. Alumina was eliminated because it cannot be machined with mills as would be required for us to machine.

Chassis Sides Mass Production

Due to the small demand for tribometers, we expect that only 100 of these devices would be sold if the device was made commercially available. This means that only 200 sides would be made in mass production. Due to a low amount of parts being made and the requirement for high tolerances, mass production of the sides would most likely be done on a CNC Mill.

According to CES[27], aluminum 6061 can be easily machined on a mill. The process by which this would be done can be found in Appendix I. All required machining could be performed on this CNC Mill and would only require that the part be mounted in the mill twice. Once a program is written to do the machining, this part will require limited attention by an employee. For this reason, and the fact that most companies already own a CNC Mill, we believe this is the best option for mass producing this part.

Pin Shaft Material Selection

This section will discuss, in detail, how the material was selected for the pin shaft of the pin gantry system. The shaft is pictrued in the figure below.

Figure C4: Pin Shaft

The pin shaft of the tribometer performs multiple functions and is one of the most important parts of the design. The primary purpose of the shaft is to transmit a constant normal force between the ball at the tip of the shaft and the disk. It must also withstand the forces seen due to friction without deflecting more than 1° from vertical. The highest normal load that the shaft must be able to achieve is 200N, and the highest coefficient of friction applied is specified to be 1.0. Therefore, the shaft must be sufficiently strong to withstand these maximum specified loads. However, the pin shaft is also used to measure the forces by using strain gauges attached directly to the surface. Because strain gauges measure the forces by displacements on the surface, the shaft must be flexible enough to allow the measurement of very low normal and frictional loads. The tip of the shaft will also be exposed to the extreme environmental conditions of the environmental control box, so it must also be able to withstand temperatures from 0° - 150° C and 100% RH.

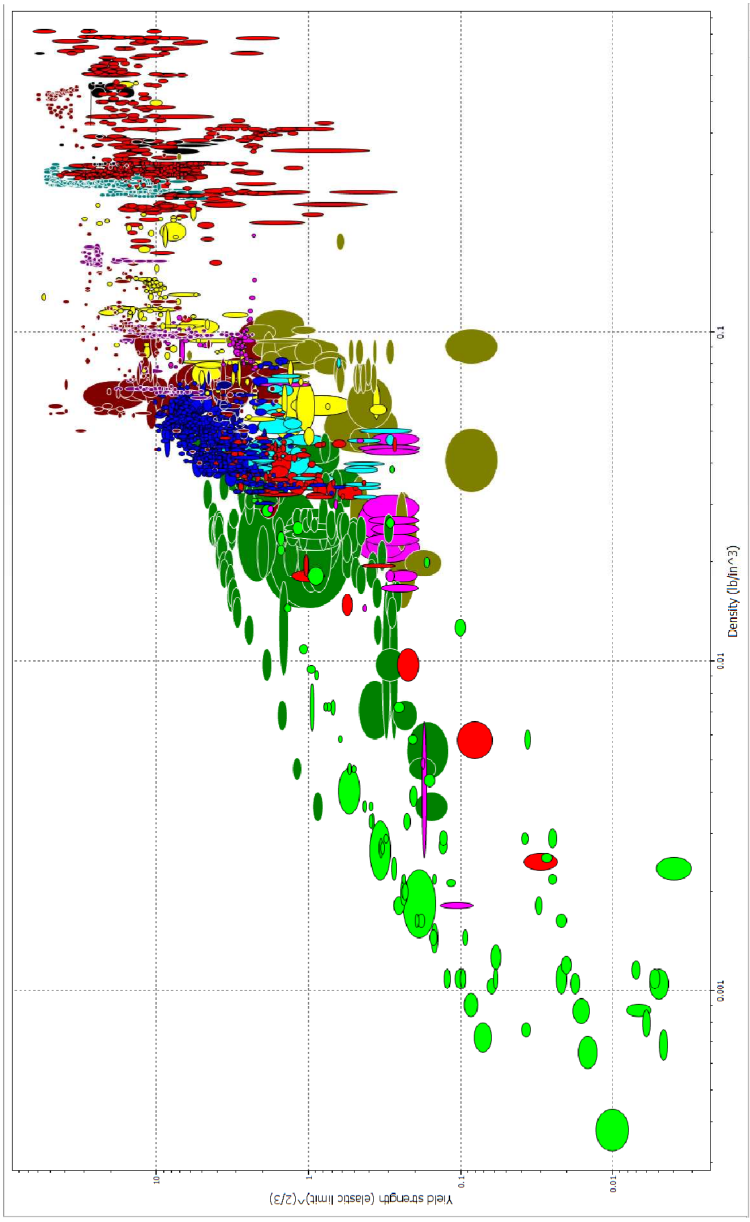
Appropriate material indices were derived by first determining which objectives and constraints were driving the design of the pin shaft. The shaft is subjected to both compressive and bending forces, however, the stresses due to bending are much greater than those seen from compression, so the pin was assumed to function as a beam. It was also assumed to have a circular cross-section because circular shafts can be evaluated quite easily and are commonly manufactured and sold in nearly all materials.

The objective was to minimize the weight of the beam because a light-weight pin gantry system will be less difficult to move in a linear reciprocating test. Two constraints were looked at – strength prescribed and stiffness prescribed. The pin beam must never plastically deform as this would ruin the alignment, force application, and force measurement aspects of the design. Therefore, a material index for a beam with minimum weight and strength prescribed, as given in Ashby's Materials Basics [16] is

$$M = \frac{\sigma_y^{\frac{2}{3}}}{\rho} \qquad \text{(Eq. C2)}$$

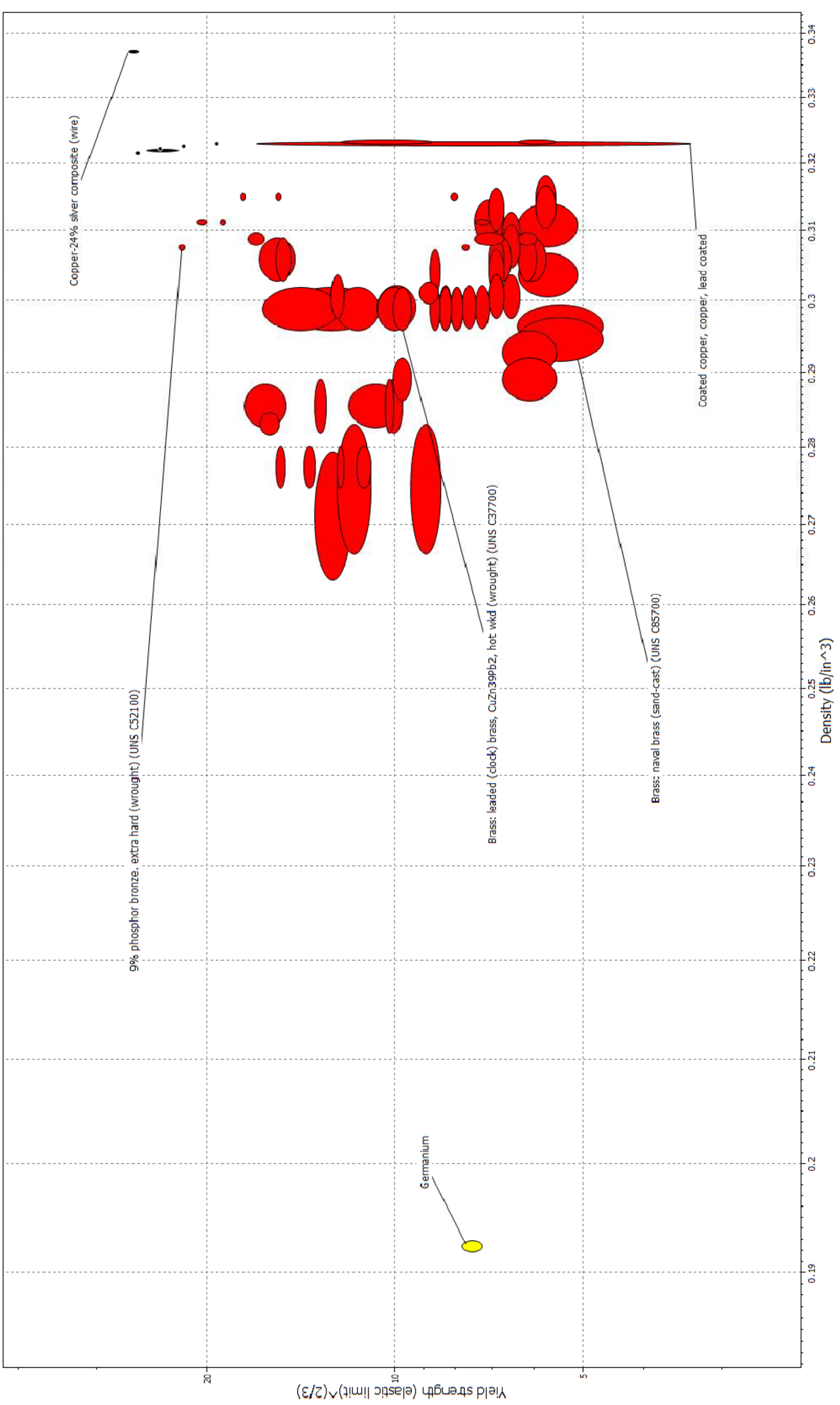
Where σ_y is the yield stress and ρ is the material density. This index should be maximized to give the strongest, lightest beam possible. By plotting the numerator vs the denominator with CES[27], we are able to observe how different materials compare to each other with respect to strength and density. A plot of the graph is shown below.

Figure C5: Graph Comparing Material Index [27]



We can then begin eliminating materials that do not meet other required criteria such as temperature, corrosion resistance, and stiffness. A graph showing a reduced sample of suitable materials is shown below.

Figure C6: Comparison of Suitable Materials [27]



After identifying which materials would be strong enough for the pin shaft, we then looked at the stiffness requirement. We did not want a beam that was too stiff and would not deflect enough, but it had to be stiff enough to achieve the specification of not deflecting more than 1° from vertical. By taking a safety factor of 2 into account, we determined the minimum value of Young's Modulus, *E*, that would allow the pin to deflect no more than 0.5° from vertical at the highest normal and frictional loads. The materials that had been previously identified for their high strength to weight ratios were then examined for their stiffness properties. A strong material with a Young's Modulus that was as low as possible without deflecting too much was desired. The top five materials identified using the CES software[27] are displayed below.

Table C2: Comparison of Top Five Materials

	Materials				
Properties	**Bronze Alloy 544**	**Manganese Bronze, C86500**	**Aluminum Bronze E, forged**	**Naval Brass**	**Silicon Bronze CuSi3Mn1**
Yield Strength (KSI)	60	24.9	36.3	10.2	65.3
Young's Modulus (MPSI)	15	14.9	16.7	13.8	14.8
Fatigue Strength at 10^7 Cycles (KSI $in^{1/2}$)	34.5	21.8	35.1	18.9	33.4
Price ($/lb)	1.62	1.04	1.57	1.15	1.43
Density (lb/in^3)	.321	.299	.275	.292	.308
Notes	Easily machinable.	Parts in contact w/ salt and fresh water	Used in marine shafts	Sand cast	Hard (wrought)

The final choice for the pin shaft was Bronze Alloy 544. It was chosen because it was strong enough to withstand the highest loads seen in our application, while also deflecting enough to allow the measurement of very low loads. It is also corrosion resistant and able to function in the temperature ranges that it will be exposed to in our device. What sets it apart from the other materials in the table is that it is readily available, and highly machinable. It is a little more expensive than the other materials, but less than 0.5 lbs of the material is required for the shaft in our design.

Pin Shaft Mass Production

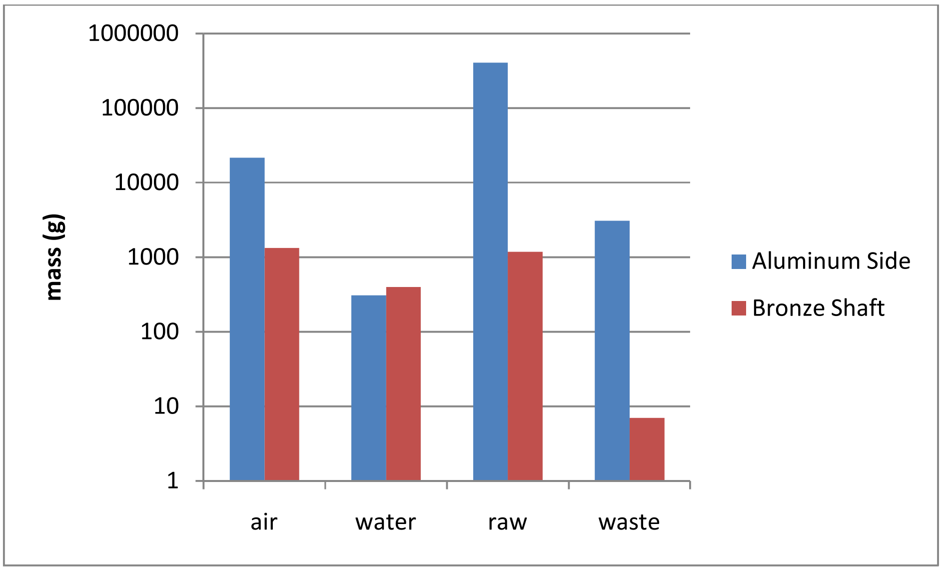
Due to the small demand for tribometers, we expect that only 100 of these devices would be sold if the device was made commercially available. This means that only 100 pin shafts would be made in mass production. Due to the low amount of parts being made and the requirement for high tolerances, mass production of the shafts would most likely be done on a lathe and CNC Mill.

According to both the CES[27] software and the supplier's website [26], bronze 544 can be easily machined. The process by which this would be done can be found in Appendix I. The machining to be performed on the lathe would be performed first, and then the part would be finished on a CNC Mill, in a rotational chuck. Once a program is written to do the machining, this part will require limited attention by an employee. For this reason, and the fact that most companies already own lathes and CNC Mills, we believe this is the best option for mass producing this part.

Environmental Performance

To analyze the environmental impact of the selected materials we used a program called SimaPro. This program allows you to select materials, put in the weight of those materials, and it gives you the ways that those materials impact the environment and how much of an impact they have.

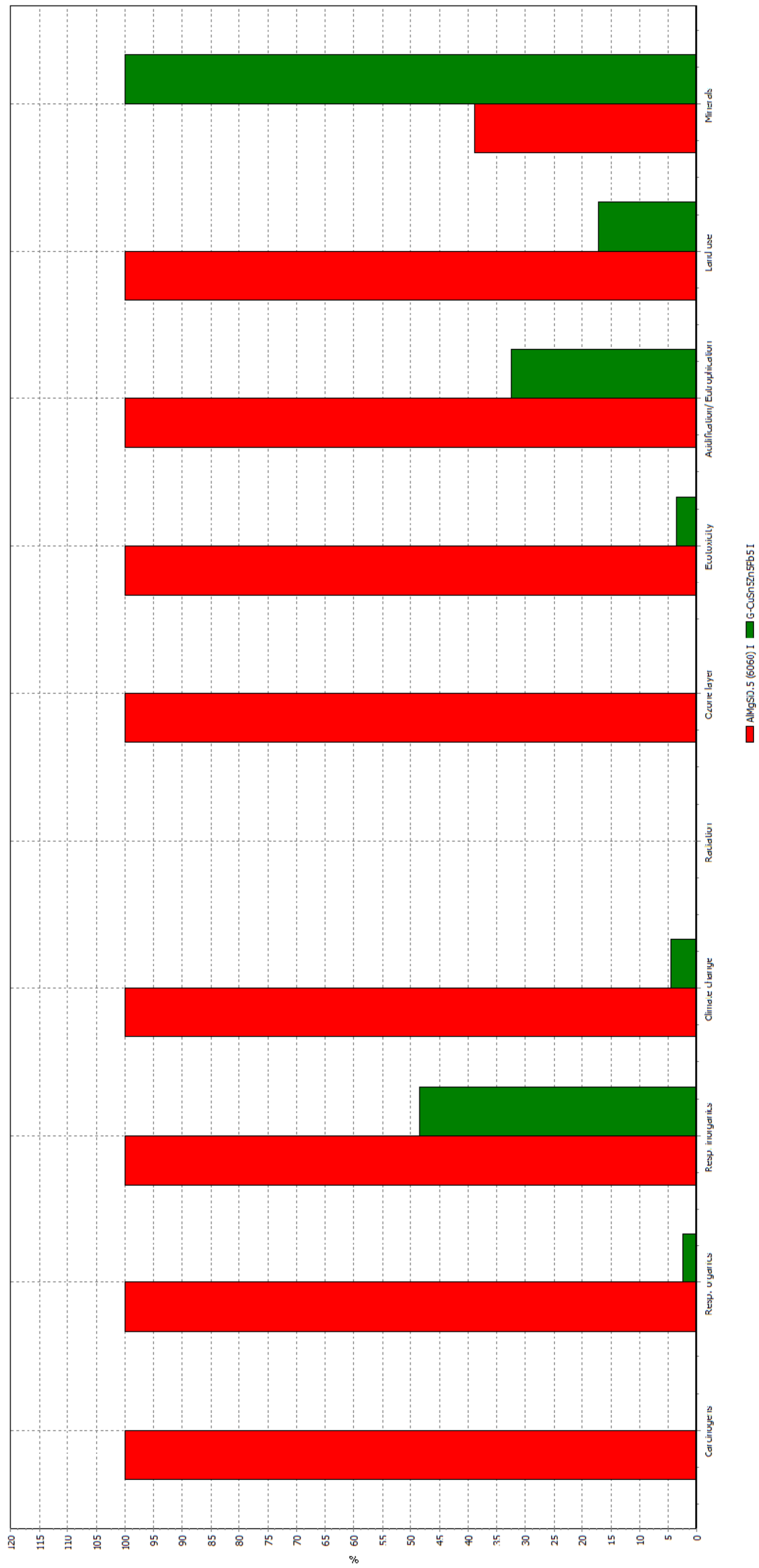
The first thing we did was to see how much air emissions, water emissions, raw materials, and solid waste would be created by each part. A graph logarithmic graph showing this can be found below. This graph indicates that the aluminum sides create a lot of damage to the environment, most significantly in raw materials. Most of the mass in the raw materials section is the water that is used in making the materials.



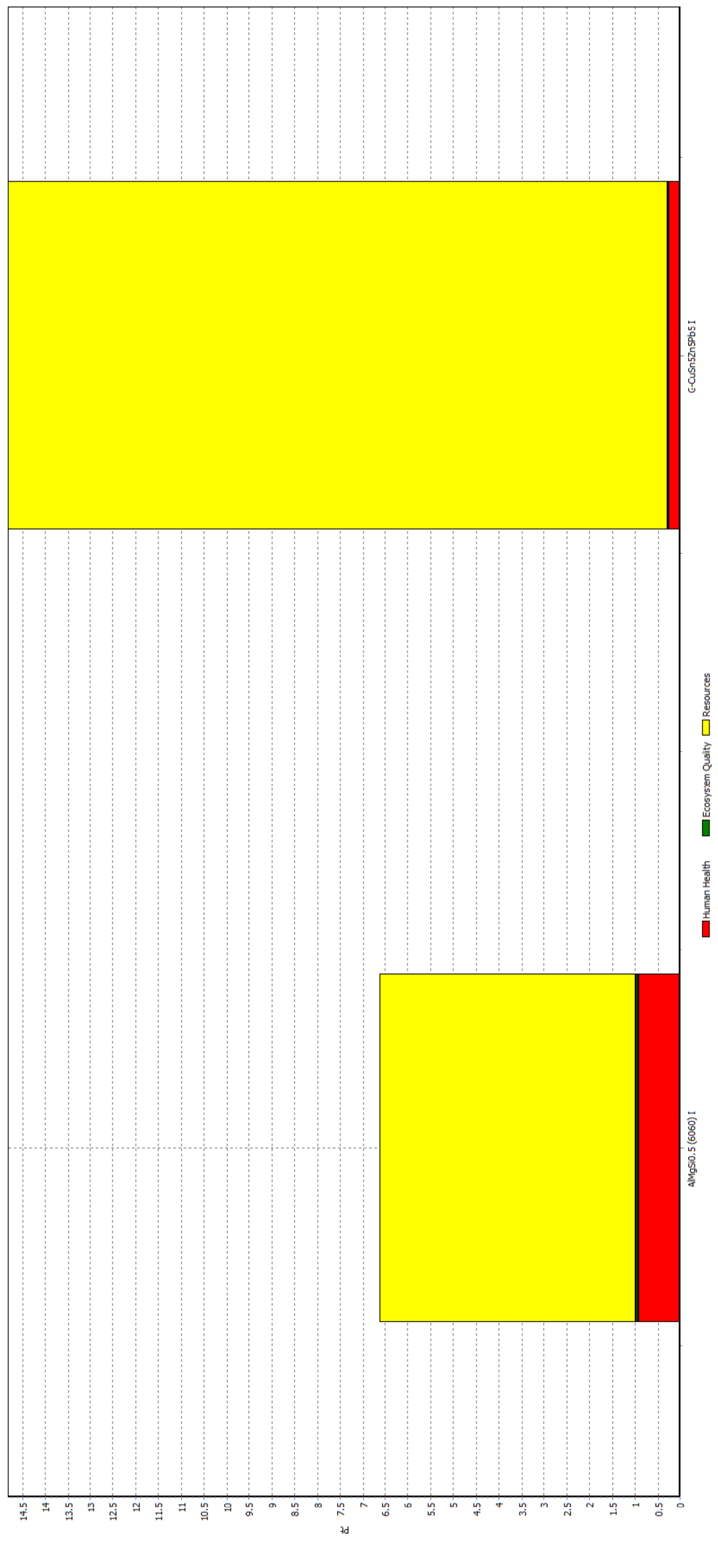
We then graphed E199 impact categories. This graph shows how the two materials compare in these categories. The material with the most impact is shown as 100% and the other material is shown in percentage of the one with the most impact. This graph shows that again the aluminum sides will make more of an environmental impact than the bronze shaft. This is true in every category except for minerals. This graph is shown on the next page.

We also create a graph that shows the normalized "meta-categories." These categories include the impact on human health, the ecosystem, and resources. The categories are plotted versus the percentage of damaged caused by an average European in one year. This graph, on Pg. 81 shows that the resources used to make both parts are more significant than the damage caused to human health or ecosystem quality and even this impact is significantly less than 1%.

Lastly, we made a chart showing how each material scores in total E199 points. This chart is shown on Page 82 and indicates that the bronze shaft has more of an environmental impact due to its use of resources.


Carcinogens
Resp. organics
Resp. inorganics
Climate change
Radiation
Ozone layer
Ecotoxicity
Acidification/ Eutrophication
Land use
Minerals
%
AlMgSi0.5 (6060) I
G-CuSn5ZnPb5 I

Human Health
Ecosystem Quality
Resources
AlMgSi0.5 (6060) I
G-CuSn5ZnPb5 I
0.072
0.07
0.068
0.066
0.064
0.062
0.06
0.058
0.056
0.054
0.052
0.05
0.048
0.046
0.044
0.042
0.04
0.038
0.036
0.034
0.032
0.03
0.028
0.026
0.024
0.022
0.02
0.018
0.016
0.014
0.012
0.01
0.008
0.006
0.004
0.002
0


Pt
AlMgSi0.5 (6060) I
G-CuSn5Zn5Pb5 I
Human Health
Ecosystem Quality
Resources

After considering all the things, we can still consider what will happen to the device over its life cycle. For both of these products, a long life time is expected, with the shaft having a lesser life time. This is because it will see large cyclic loading and eventually break. When this happens, the shaft will need to be replaced, but the old shaft can be melted down and recycled. The same can be done with the aluminum sides when the tribometer will no longer be used.

Because the shaft will most likely need to be replaced several times over the lifetime of a tribometer, we have determined it will have more of an environmental impact; however, overall, neither material will make a significant impact to the environment because very few tribometers are expected to be built.

Appendix D: Functional Decomposition

1. Hold test specimens
    1.1. Hold ball
        1.1.1. Different sizes
            1/16-1/4 inch
        1.1.2. Different finish
        1.1.3. With or without spin
        1.1.4. Safety
            1.1.4.1. Does not fly out
                1.1.4.1.1. Debris does not leave immediate test area
            1.1.4.2. Does not damage ball
        1.1.5. Easy to change out
        1.1.6. Perpendicular (±1degree)
        1.1.7. Dampen Vibrations
    1.2. Hold disk
        1.2.1. Different sizes
        1.2.2. Different types
        1.2.3. Safely
            1.2.3.1. Does not fly out
                1.2.3.1.1. Debris does not leave immediate disk area
            1.2.3.2. Does not damage disk
        1.2.4. Easy to change out
        1.2.5. Keep level (±1 degree)
        1.2.6. Dampen Vibrations
2. Perform rotational movement.
    2.1. Spin plate/disk
        2.1.1. Set speed
            2.1.1.1. Capable of achieving 1 m/s.
            2.1.1.2. Maintain same speed
                2.1.1.2.1. Measure speed
    2.2. Variable radius.
        2.2.1. Set radius
        2.2.2. Measure radius
    2.3. Maintain same path.
        2.3.1. Check relative location
3. Perform linear reciprocating movement.
    3.1. Achieve top speed through middle 30% of stroke length.
    3.2. Capable of achieving 1 m/s.
    3.3. Variable stroke length
        3.3.1. Set stroke length
        3.3.2. Measure stroke length
    3.4. Stop within 1 mm each time.
    3.5. Maintain same path
4. Perform custom combined movement.

- 4.1. Linear and rotational movement.
  - 4.1.1. Program set path
  - 4.1.2. Follow set path
    - 4.1.2.1. Control position
      - 4.1.2.1.1. Measure position
      - 4.1.2.1.2. Respond with correct movement
      - 4.1.2.1.3. Control speed
        - 4.1.2.1.3.1. Measure speed
        - 4.1.2.1.3.2. Supply force required to maintain constant speed
- 4.2. Cross path testing.

5. Apply constant normal force
   - 5.1.1. Variable normal force
     - 5.1.1.1. Set normal force
     - 5.1.1.2. Constant within 1%.
6. Measure friction force.
   - 6.1. Data acquisition up to 20 kHz.
   - 6.2. Instantaneous results.
   - 6.3. Store results
   - 6.4. Measure normal force
   - 6.5. Measure lateral force
     - 6.5.1. Measure force in both directions
7. Set and maintain ambient conditions.
   - 7.1. Maintain specified temperature up 150°C within +/- 5°C.
     - 7.1.1. Set temperature
     - 7.1.2. Measure temperature
     - 7.1.3. Record and Save temperature
     - 7.1.4. Change temperature
       - 7.1.4.1. Add heat
       - 7.1.4.2. Dissipate heat
   - 7.2. Maintain specified humidity
     - 7.2.1. Set humidity
     - 7.2.2. Measure humidity
     - 7.2.3. Change humidity
     - 7.2.4. Record and Save humidity reading
8. Ability to use lubricant.
   - 8.1. Lubricant covers disk.
     - 8.1.1. Stops liquid from splashing away from disk
       - 8.1.1.1. Stops lubricant from leaking onto motors/wires ect.
   - 8.2. Allows lubricant to be easily removed from test
   - 8.3. Ability to circulate lubricant
9. Measure electrical resistance.
   - 9.1. Instantaneous readings between pin and disk.
     - 9.1.1. Up to 1000Ω ± 1%
     - 9.1.2. Record electrical resistance

10. Measure wear.
    10.1. Pin wear during test.
        10.1.1. Record pin wear
    10.2. Disk wear during test
        10.2.1. Record disk wear
11. Stop test at set wear path distance
    11.1. Set wanted distance
    11.2. Measure distance of test
        11.2.1. Measure speed vs time
    11.3. Stop movement
    11.4. Record and Save distance

Appendix E.1: Component Concepts

Force Measurement

To measure the lateral forces caused by friction and the normal force of the pin against the disk.

Wire Attached to Strain Gauge

A wire is attached to the pin at one end and a strain gauge at the other. When the pin is displaced laterally due to friction, the wire is pulled taught and the friction force is recorded by the strain gauge. Only works in one direction.

Laser

A laser inside the pin shaft is oriented vertically upwards. As the angle of the pin changes due to the frictional force acting on the ball, the laser beam will also change. A sensor in the arm connected to the pin shaft will measure the distance the laser travels which can then be used to calculate the frictional force since the material properties and lengths of the pin will be known.

Strain Gauge Rosette

A strain gauge rosette will be placed on the pin shaft. This will allow the measurement of the lateral forces in the X and Y planes.

6-Axis Load Cell

A 6-axis load cell would be placed under the disk or in the pin shaft. These load cells can be bought online and would be very simple to install. However, they are also quite expensive (thousands of dollars).

Variable Path Motion

The tribometer must be able to perform rotational and linear reciprocating tests. It would also be highly beneficial if it can perform a user-designed combination test that involves crossing paths.

Pin - All Motion

The disk will remain completely stationary, and all movement will be performed by the pin. This design is similar to the laser cutter used in the shop.

Disk – All Motion

The pin will remain completely stationary, and all movement will be performed by the disk. It will be able to rotate as well as move linearly back and forth.

Pin – Reciprocating, Disk – Rotating

The disk will be able to rotate and the pin will be able to move linearly back and forth. Either one component can be in motion at a time, or both can be utilized simultaneously to achieve complex paths.

Magnets with Rotating Disk

This design is similar to the Disk – All Motion design. A rotating disk will rest on a magnetic platform which can reciprocate linearly due to changing magnetic fields induced by changing the current direction in wires around the outside of the test area.

Wear Measurement

Measuring the wear of the pin and disk in real time would give researchers valuable information and insight into how materials wear out.

LVDT

A Linear Variable Displacement Transformer would be mounted to the pin shaft. As the pin moved vertically downwards due to wear of the ball and disk, the LVDT would measure the distance travelled. These devices are highly accurate and can be purchased already manufactured.

Sensor Layer Illumination

The ball will be embedded with a certain element such as Chromium. As the ball wears down, more chromium is exposed. The amount of wear is measured by shining a known wavelength of light on the worn material and calculating the amount of chromium which is present. This method is complicated and requires expensive equipment.

White Light Interferometer

A very high quality 3D camera can observe the wear path in the disk at various intervals. This device is highly accurate but is also very expensive. It would also require stopping the test periodically to take a measurement.

Radioactive Tracer

This is similar to sensor layer illumination, but uses radioactive tracers in the ball instead of chromium. By measuring the amount of radioactive material present on the disk, one can calculate the amount of wear that has taken place.

Temperature Change

Wear rate can be estimated from the surface temperature at the pin-disk interface. This method is not very accurate.

Worm Drive Encoder

This method requires the use of the screw with worm drive to apply the normal force. As ball wears down, the normal force will decrease. Therefore, the worm drive rotates which screws the pin downwards into the disk and maintains the desired normal force. An encoder on the worm drive will measure how far it has rotated, thus measuring how far down the pin has moved. This is the total wear measurement.

Holding Disk

The disk should be held tightly during testing to prevent slipping or vibrations. The sample also needs to be centered with respect to the motor if the disk is to be rotated. Being able to accommodate various shapes and sizes of samples would be beneficial. Holding the disk satisfactorily is an important safety consideration.

Chuck

A large chuck can be used to hold samples of various shapes and sizes. The chuck will ensure the sample is centered. Our team will have to manufacture the chuck to fit in our device and perform the tasks we require of it.

Clamp
Clamps can be used to hold the sample onto a large, flat base. It can accommodate various shapes and sizes of samples.

Screw
A base with holes drilled at varying intervals will hold the sample. Samples of varying sizes and shapes will be placed on this base and centered using a dowel screw. This requires an existing hole in the center of the sample. Stops will be screwed into the appropriate holes along the base depending on the size of the sample. Set screws will then be inserted into the stops and will hold the sample in place by pressing against the sides of the sample.

Holding Pin
The tribometer must be able to hold balls of different sizes (1/4, 1/8, 1/16 in.). The balls should not rotate when the pin and disk rub against each other. Several concepts were generated that could solve the problem. These concepts are outlined in this section.

Screw on with Washer
In this concept the pin has a place where the ball can fit and a threaded outside shaft. The ball is placed in a washer that holds it at the diameter of the ball. The washer and ball are then placed in a cup-shaped object that holds the ball and washer. This cup has threads that allow it to be screwed onto the pin. This cup can be screwed on tight enough so that the ball cannot move.

Screw of Different Sizes
This concept is very similar to the screw on with washer technique. The difference in this concept is that it does not utilize a washer. This concept would have three different cups that accommodate the different size balls. This will allow more of the ball to be worn away before the ball has to be changed out.

Set Screws
In this concept, the ball is held in place using set screws. These screws would push the ball up and into the pin. They would allow for the different size balls. To accommodate smaller balls, the screws would just have to thread in farther.

Magnets
This concept holds metal balls in using electromagnets. If this concept was used, the ball would not need to be held from multiple sides by physical restraint. This would allow the ball to be exposed, and wear tests could, theoretically, run until the ball was completely worn away.

Control Systems
The team researched and determined several concepts to control the movement of the disk and pin. This section outlines the concepts the team explored.

National Instruments C-Rio / C-Daq
This is a real time controller that allows you to automatically publish a front-panel interface in LabVIEW. This system can also do real time data logging and analysis of the control. This is great because one of the requirements of the tribometer is to record the velocity of the pin relative

to the disk during the test. A user could easily take the stored data and report what the velocity was at any point during a test.

National Instruments – USB

National Instruments (NI) sells devices such as the USB-6008 which it markets as a low cost (≈$300) way to control motors. Using this system, the team can control two motors and monitor up to 8 input signals. Data can be logged similarly to the C-Rio systems. For more inputs and outputs, higher accuracy, and more power NI recommends using the NI USB-6210 and NI USB-6211.

National Instruments – PCI Card

Using a PCI card such as the NI PCI-6221 will work similar to the USB devises discussed above; however, the PCI cards are known for sampling at high rates with incredible accuracy. These devices also cost twice as much as the USB devices.

Audrino

Audrino is a type of single board microcontroller. These devices are programmed using C or C++ code. They can sample at high rates, while controlling up to 14 digital I/O pins and 6 analog inputs. While these devices have low prices, they do not give you an interface for the user to change the motion and monitor what is happening directly. Also, data logging with these devices is notoriously difficult.

Temperature Control

Thermoelectric with Heat Sink

This method consists of utilizing thermoelectric Peltier modules in control the temperature. Heating and cooling can both be accomplished using a single Peltier device by reversing the direction of the current. The Peltier modules will be attached to a heat sink in order to further improve the thermal efficiency. This method is inexpensive and convenient because it takes care of both heating and cooling.

Hot Plate

This method was drawn up as a way to heat the test specimen and not the atmosphere of the testing area. Due to the fact that an atmospheric temperature between 0-100 Celsius is desired this method is not going to be used.

Heat Pump

Inducing heat by connecting the test area to a heat pump was also analyzed. Although this method was very efficient and reliable it proved to be much too expensive for our budget. Also considering that controlling temperature is a secondary requirement, we could not find any justification for using this method.

Humidity Control

Humidifier / Dehumidifier

This method basically consists of attaching a premade humidifier or humidifier to control the relative humidity within the system. Although this method would be reliable and efficient its costs would be too high for our budget.

Silica Gel Desiccant

This method would consist of blowing air over a desiccant such as silica gel in order to provide dry air into the system. Silica gel absorbs moisture and decreases relative humidity. This process would be very inexpensive and efficient at the same time.

Humidity Inducing Salt Solution

Saturated salt solutions in water release a relative humidity within its environment depending on its saturation. This method although proven to be reliable would take too long to induce a desired humidity level. The desired time to reach a certain RH would be in minutes and not hours or days, as it would take for the saturated salt solution.

Appendix E.2: Component Pugh Charts

Application of Normal Force

Selection Criteria	Weight	Concept A Dead weights		Concept B Cable		Concept C Hydraulic		Concept D Pneumatic		Concept E Spring		Concept F Screw w/ Worm Drive	
		Rating	Weighted	Rating	Weighted	Rating	Weighted	Rating	Weighted	Rating	Weighted	Rating	Weighted
Complexity	9	9	81	6	54	1	9	3	27	6	54	6	54
Cost	9	9	81	6	54	1	9	3	27	9	81	6	54
Durability	1	9	9	6	6	6	6	6	6	3	3	9	9
Reliability	3	9	27	6	18	3	9	3	9	3	9	6	18
Weight/Size	3	1	3	9	27	1	3	3	9	9	27	9	27
Manufacturability	9	9	81	6	54	1	9	9	81	6	54	6	54
Accuracy/Precision	3	6	18	6	18	6	18	6	18	3	9	6	18
Safety	3	3	9	9	27	6	18	9	27	6	18	9	27
Practical	3	3	9	3	9	1	3	1	3	6	18	6	18
Variable range	9	3	27	9	81	9	81	9	81	6	54	9	81
Easy to use	1	3	3	6	6	6	6	6	6	6	6	6	6
Total Score			348		354		171		294		333		366
Rank			3		2		6		5		4		1

Force Measurement

Selection Criteria	Weight	Concept A 6-axis load cell		Concept B strain gauge/s		Concept C laser		Concept D current induced		Concept E multiple load cell:	
		Rating	Weighted	Rating	Weighted	Rating	Weighted	Rating	Weighted	Rating	Weighte
Complexity	9	9	81	6	54	1	9	1	9	6	54
Cost	9	-3	-27	9	81	6	54	6	54	6	54
Durability	3	9	27	6	18	6	18	3	9	6	18
Reliability	3	9	27	6	18	3	9	3	9	6	18
Weight/Size	1	3	3	9	9	1	1	3	3	3	3
Manufactorability	9	9	81	3	27	1	9	1	9	3	27
Accuracy/Precision	6	9	54	6	36	1	6	1	6	6	36
Safety	3	9	27	9	27	3	9	3	9	9	27
Practical	3	1	3	9	27	3	9	1	3	6	18
Variable range	3	3	9	3	9	6	18	6	18	3	9
Easy to use	1	9	9	6	6	1	1	3	3	6	6
cal	1	9	9	6	6	3	3	3	3	9	9
			0		0		0		0		0
Total Score			303		318		146		135		279
Rank			2		1		4		5		3

Variable Path Motion

Selection Criteria	Weight	Concept A Pin-All Motion		Concept B Pin Reciprocating		Concept C Pin Spinning		Concept D Disk-All Motion	
		Rating	Weighted	Rating	Weighted	Rating	Weighted	Rating	Weighted
Complexity	9	3	27	6	54	3	27	3	27
Cost	9	6	54	6	54	6	54	3	27
Durability	3	3	9	6	18	3	9	3	9
Reliability	3	6	18	6	18	1	3	3	9
Speed	1	1	1	1	1	1	1	1	1
Manufacturability	9	3	27	3	27	3	27	3	27
Accuracy/Precision	9	3	27	6	54	3	27	2	18
Control of Motion/Speed	9	3	27	6	54	1	9	3	27
Control of Environment	1	1	1	1	1	1	1	1	1
Safety	3	1	3	1	3	1	3	1	3
Practical	3	4	12	3	9	3	9	2	6
Versitility	9	9	81	6	54	3	27	3	27
Easy to use	1	6	6	6	6	3	3	3	3
Weight/Size	1	6	6	6	6	6	6	1	1
Meets Standards	3	3	9	6	18	3	9	2	6
Total Score		308		377		215		192	
Rank		2		1		3		4	

Wear Measurement

Selection Criteria	Weight	Concept A LVDT - linear vertical displacement transducer w/ reference		Concept B Sensor Layer illumination		Concept C White Light Interferometer		Concept D Radioactive Tracer		Concept E Temp. Change		Concept F Worm Drive with Encoder	
		Rating	Weighted	Rating	Weighted	Rating	Weighted	Rating	Weighted	Rating	Weighted	Rating	Weighted
Complexity	9	6	54	1	9	3	27	1	9	3	27	6	54
Cost	9	3	27	1	9	-9	-81	1	9	3	27	6	54
Durability	3	6	18	1	3	6	18	1	3	3	9	6	18
Reliability	3	1	3	9	27	9	27	9	27	1	3	3	9
Speed	1	3	3	6	6	3	3	9	9	3	3	3	3
Manufacturability	9	9	81	1	9	1	9	1	9	3	27	3	27
Accuracy/Precision	9	6	54	9	81	9	81	9	81	1	9	3	27
Safety	3	9	27	3	9	6	18	1	3	9	27	9	27
Practical	3	9	27	1	3	1	3	1	3	3	9	9	27
Versatility	9	1	9	1	9	3	27	1	9	3	27	3	27
Easy to use	1	9	9	3	3	3	3	-9	-9	3	3	9	9
Weight/Size	1	9	9	6	6	3	3	6	6	3	3	9	9
Total Score		240		144		138		159		174		291	
Rank		2		5		6		4		3		1	

Holding Disk

Selection Criteria	Weight	Concept A Chuck		Concept B Clamp		Concept C Screw	
		Rating	Weighted	Rating	Weighted	Rating	Weighted
Complexity	9	3	27	3	27	6	54
Cost	9	1	9	3	27	9	81
Durability	3	6	18	3	9	3	9
Repeatablity	9	9	81	3	27	3	27
Accuracy	9	6	54	3	27	3	27
Reliability	3	6	18	3	9	3	9
Speed	1	9	9	6	6	1	1
Manufacturability	9	1	9	3	27	9	81
Safety	3	6	18	3	9	3	9
Practical	3	6	18	3	9	3	9
Versitility	9	6	54	9	81	3	27
Easy to use	1	9	9	3	3	1	1
Weight/Size	1	1	1	3	3	6	6
Total Score		325		264		341	
Rank		2		3		1	

Holding Pin

Selection Criteria	Weight	Concept A Screw-washer		Concept B Screw Sizes		Concept C Set screws		Concept D Magnets	
		Rating	Weighted	Rating	Weighted	Rating	Weighted	Rating	Weighted
Complexity	9	9	81	9	81	6	54	1	9
Cost	9	9	81	3	27	9	81	3	27
Durability	6	6	36	9	54	9	54	6	36
Reliability/Repeatability	9	6	54	9	81	3	27	3	27
Weight/Size	1	9	9	9	9	9	9	6	6
Manufactorability	9	9	81	1	9	6	54	1	9
Safety	3	6	18	9	27	6	18	1	3
Practical	6	9	54	6	36	9	54	1	6
Variable sizes	3	9	27	6	18	9	27	9	27
Easy to use	3	9	27	9	27	6	18	3	9
Proper Hold	9	6	54	9	81	6	54	3	27
Total Score		522		450		450		186	
Rank		1		2		3		4	

Control Systems

Selection Criteria	Weight	Concept A C-Rio/C-DAQ		Concept B NI-PCI card		Concept C NI-USB		Concept D Audrino	
		Rating	Weighted	Rating	Weighted	Rating	Weighted	Rating	Weighted
Complexity	6	3	18	3	18	3	18	1	6
Cost	9	1	9	3	27	6	54	6	54
Durability	3	6	18	6	18	6	18	3	9
Reliability	3	6	18	6	18	6	18	3	9
Processing Power	9	6	54	3	27	1	9	3	27
Accuracy/Precision BITS	9	6	54	3	27	3	27	3	27
Practical	3	6	18	6	18	6	18	3	9
Versitility	3	6	18	3	9	3	9	1	3
Easy to use	9	3	27	3	27	3	27	1	9
Knowledge Base	9	6	54	6	54	6	54	3	27
Total Score		288		243		252		180	
Rank		1		3		2		4	

Motors

Selection Criteria	Weight	Concept A DC w/ Encoders		Concept B Steppers	
		Rating	Weighted	Rating	Weighted
Complexity	9	3	27	3	27
Cost	9	3	27	3	27
Durability	3	3	9	3	9
Reliability	3	6	18	9	27
Speed	6	3	18	3	18
Accuracy/Precision	9	6	54	6	54
Control of Motion/Speed	9	3	27	6	54
Practical	3	6	18	9	27
Versitility	9	6	54	6	54
Easy to use	3	3	9	6	18
Total Score		261		315	
Rank		2		1	

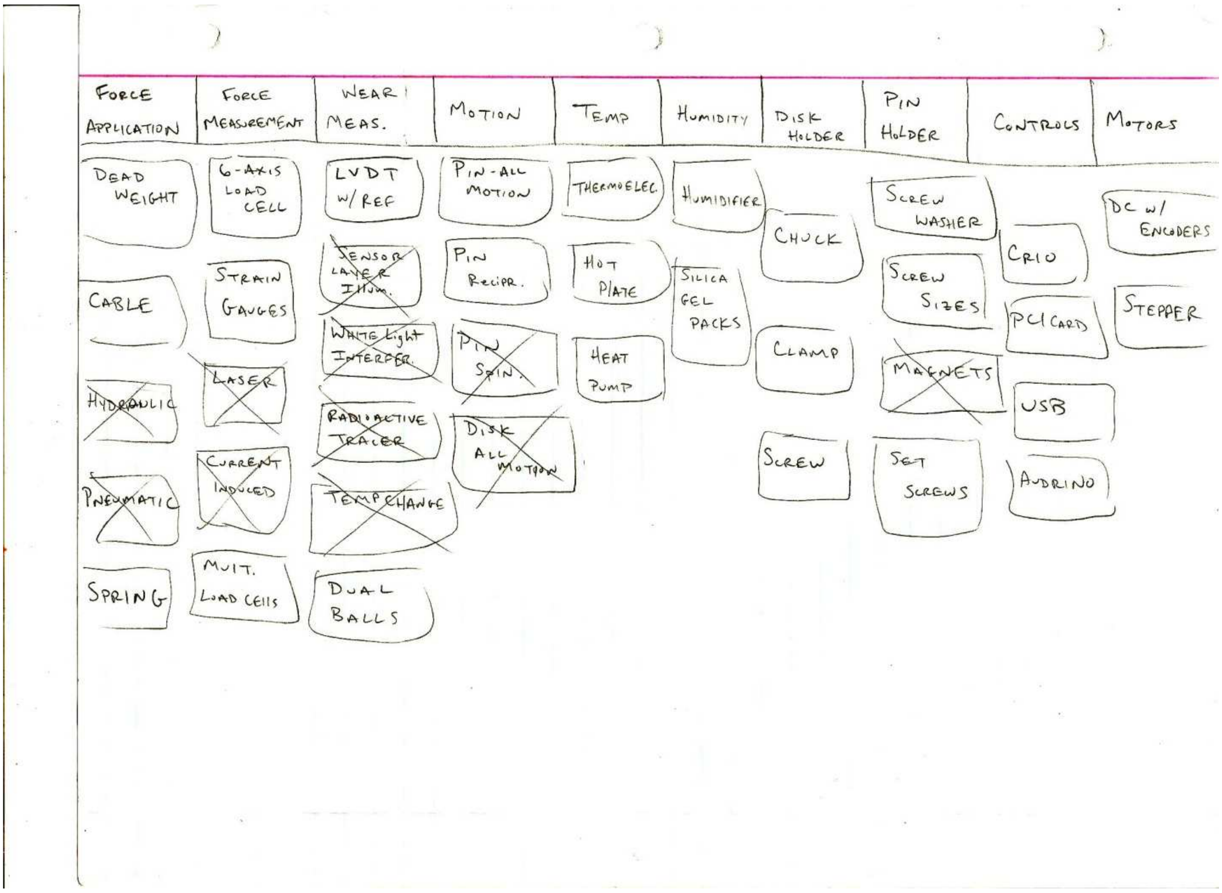
Temperature Control

Selection Criteria	Weight	Concept A Thermoelectric w/ heat sink		Concept B Hot Plate		Concept C Heat Pump	
		Rating	Weighted	Rating	Weighted	Rating	Weighted
Complexity	9	3	27	6	54	3	27
Cost	9	6	54	3	27	1	9
Durability	3	6	18	6	18	6	18
Reliability	3	6	18	3	9	6	18
Speed	1	3	3	3	3	6	6
Precision	9	6	54	6	54	9	81
Temperature Range	9	6	54	3	27	6	54
Safety	3	6	18	3	9	6	18
Practical	3	6	18	1	3	3	9
Easy to use	1	3	3	3	3	6	6
Weight/Size	3	6	18	3	9	1	3
							0
Total Score		285		216		249	
Rank		1		3		2	

Humidity Control

Selection Criteria	Weight	Concept A Humidifier		Concept B Dehumidifier		Concept C Silica Gel Packs		Concept D Humidity Inducing Salt Solution	
		Rating	Weighted	Rating	Weighted	Rating	Weighted	Rating	Weighted
Complexity	9	3	27	3	27	6	54	3	27
Cost	9	6	54	6	54	9	81	9	81
Durability	3	9	27	9	27	3	9	3	9
Reliability	3	6	18	6	18	3	9	3	9
Rate of Humidity Flux	1	3	3	3	3	3	3	3	3
Accuracy/Precision	9	6	54	6	54	3	27	3	27
Safety	3	9	27	9	27	6	18	6	18
Practical	3	6	18	6	18	9	27	1	3
Versitility	3	3	9	3	9	1	3	3	9
Easy to use	1	6	6	6	6	6	6	6	6
Weight/Size	3	3	9	3	9	3	9	3	9
Total Score		252		252		246		201	
Rank		1		1		3		4	

Appendix F: Component Flow Chart



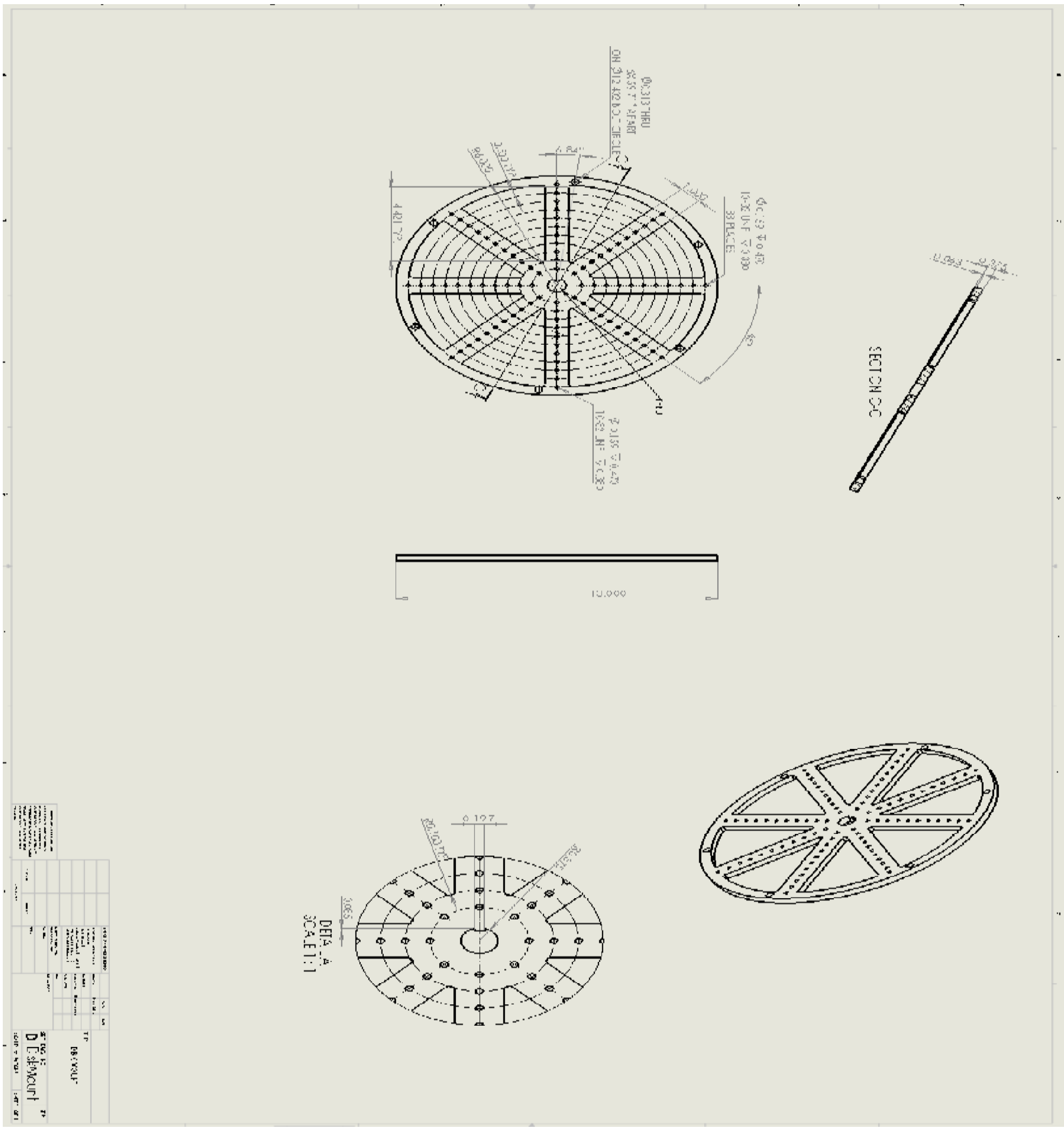
Appendix G: Assembled Pugh Chart

Selection Criteria	Weight	Concept A: Dead weight, multiload, LVDT, All Pin, TEC, gel, chuck, screw washer, pci,stepper		Concept B: cable, strain gauge, lvdt, pi-all, tec, gel pack, screw, screw washer, pci, dc motors		Concept C: dead weight multi load, lvdt, pin recip, tec, humid, screw, screw washer, pci stepper		Concept D: Screw/worm drive, strain gauges and load cell, pin recip, TEC, humid /gel pack, screws, screw sizes, USB, DC motors		Concept E: Super Awesome: pneumatic, 6-axis, 3D-visualizer, TEC, Humid/Dehumid,screw sizes, crio, stepper, Pin_ric		Concept F: Super Cheap: Dead Weight, Strain Gauge, No Wear, Pin-All, No Temp, No Humid, Screw Washers, USB, DC	
		Rating	Weighted	Rating	Weighted	Rating	Weighted	Rating	Weighted	Rating	Weighted	Ranking	Weighted
Complexity	9	6	54	6	54	6	54	6	54	3	27	9	81
Cost	9	6	54	6	54	6	54	6	54	1	9	9	81
Durability	3	6	18	3	9	9	27	9	27	6	18	1	3
Repeatability	6	3	18	6	36	3	18	6	36	9	54	0	0
Weight/Size	1	1	1	6	6	1	1	9	9	6	6	1	1
Manufacturability	9	6	54	3	27	6	54	3	27	3	27	9	81
Accuracy/Precision	6	3	18	6	36	6	36	9	54	9	54	1	6
Safety	6	3	18	3	18	6	36	6	36	9	54	9	54
Practical	6	6	36	6	36	3	18	6	36	1	6	1	6
Easy to use	3	6	18	6	18	6	18	9	27	6	18	9	27
Versitility	6	6	36	6	36	3	18	3	18	9	54	0	0
Total Score			325		330		334		378		327		340
Rank			6		4		3		1		5		2

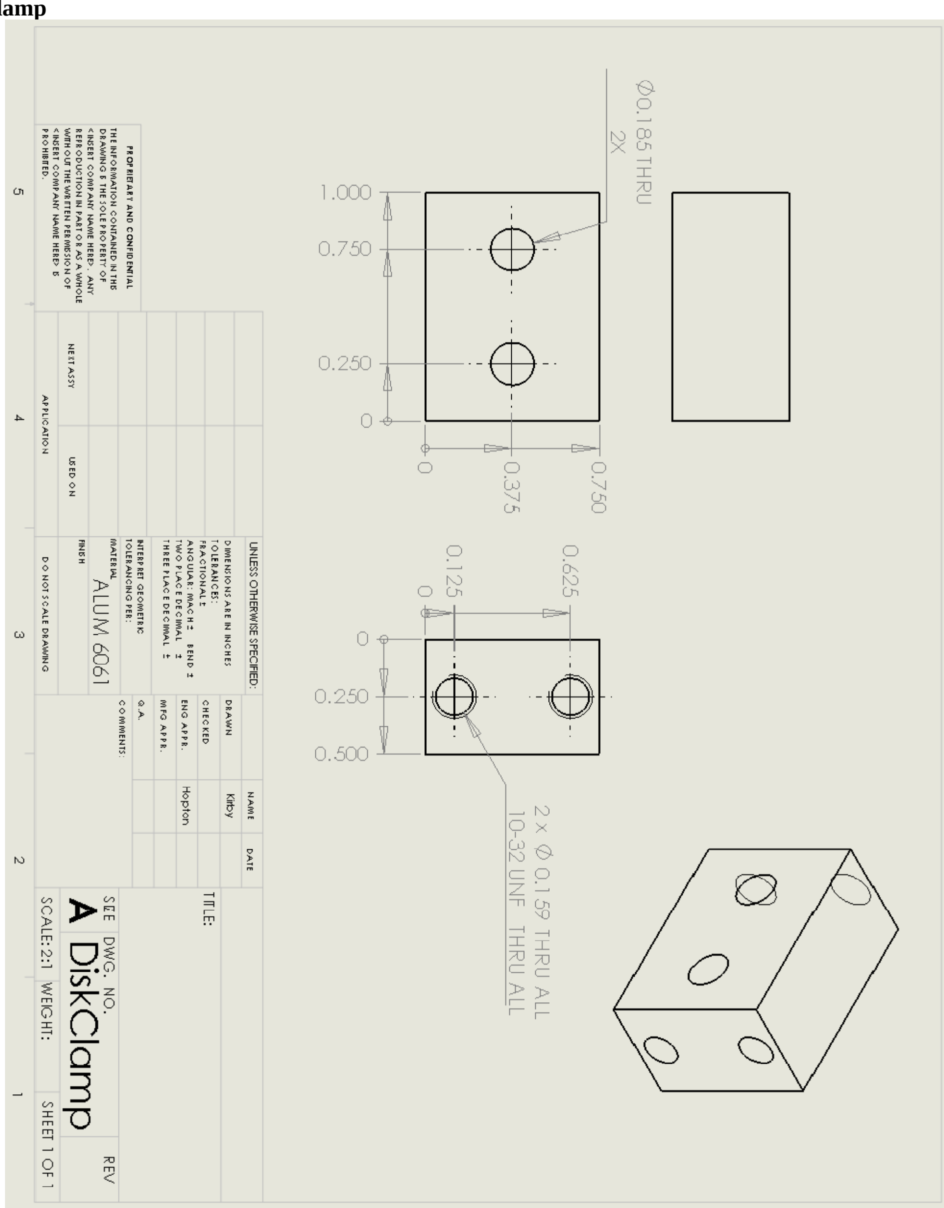
Appendix H: Part Drawings

This appendix contains the part drawings for each custom piece that we are machining ourselves.

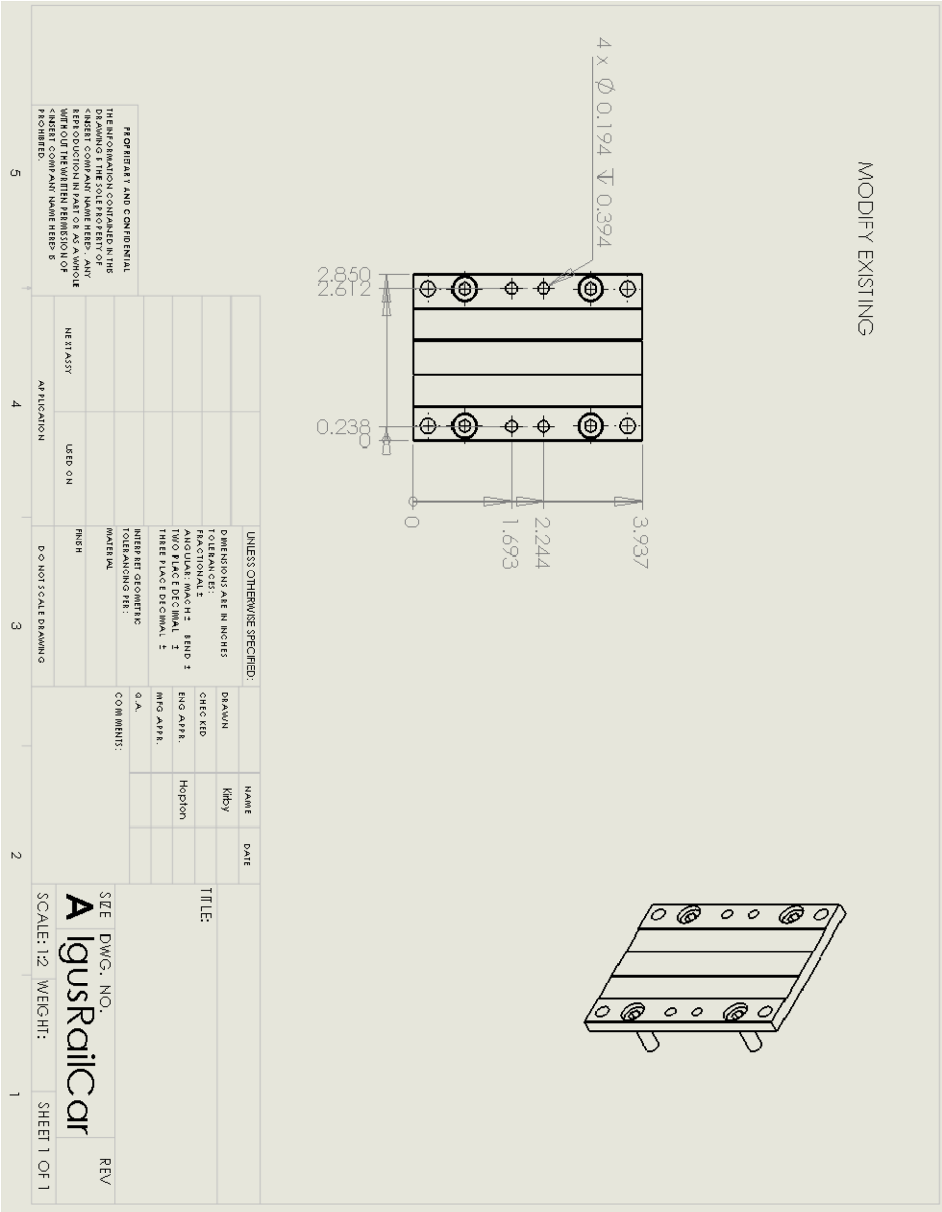
H.1: Disk Mount



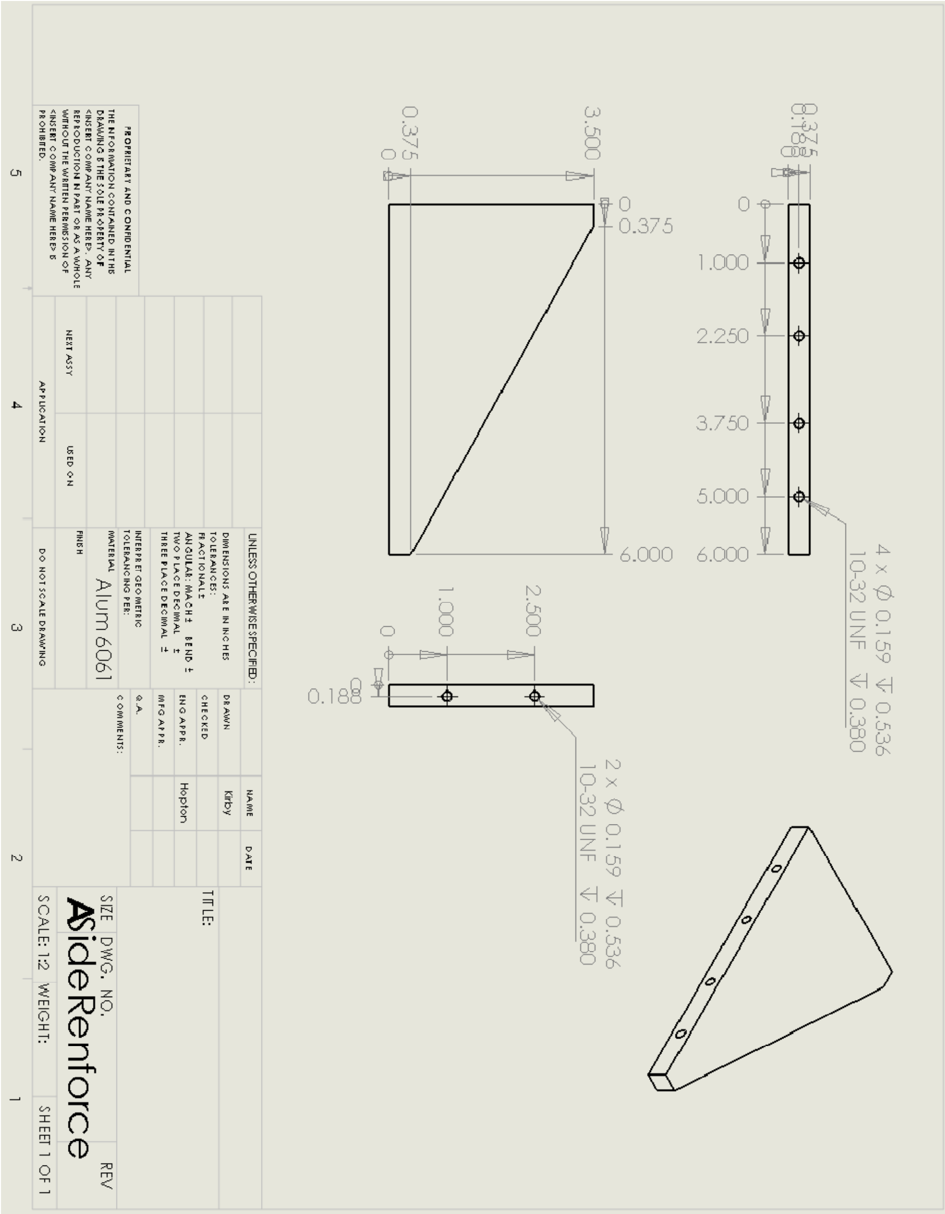
H.2: Disk Clamp



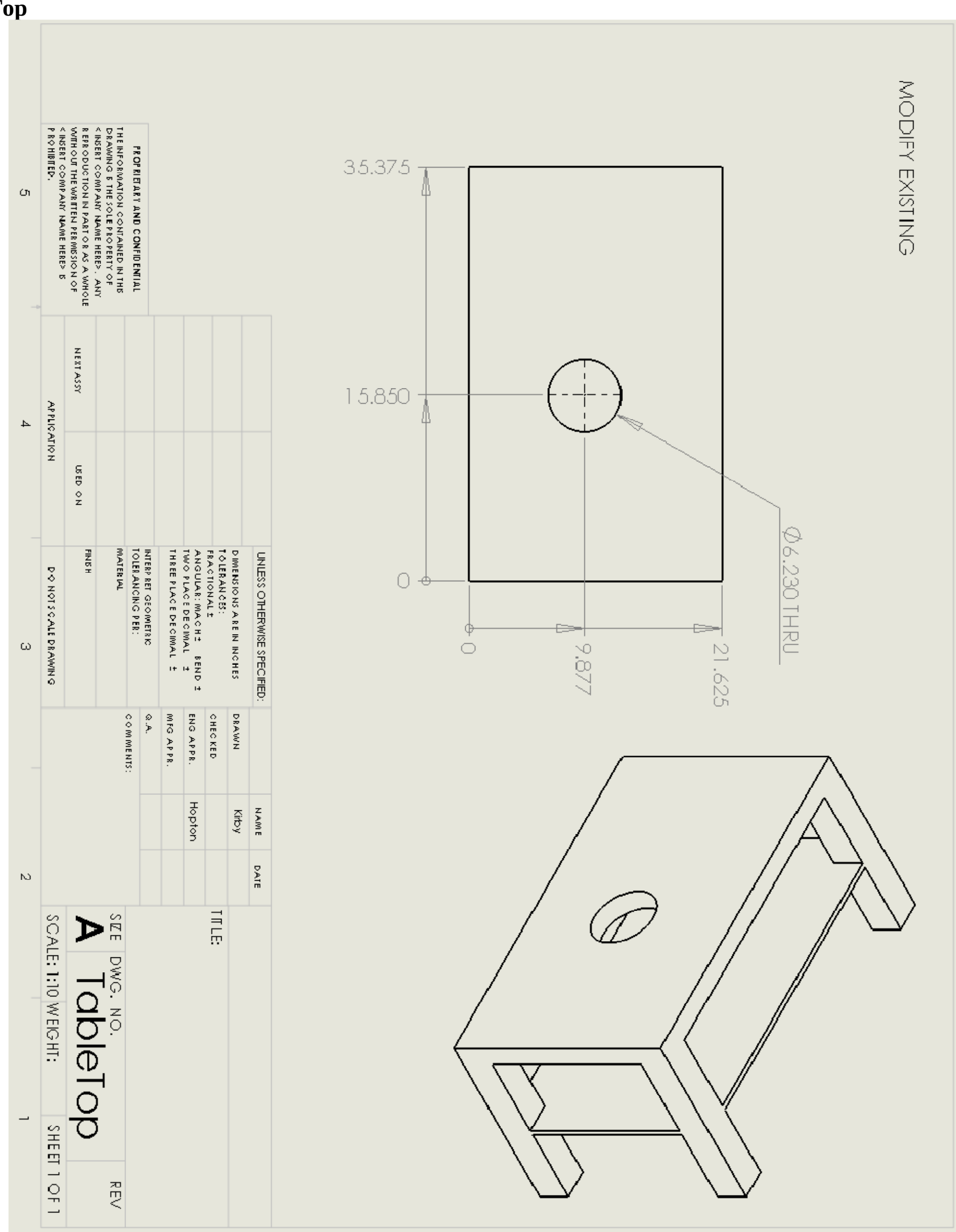
H.3: Igus Rail Car



H.4: Side Reinforcement

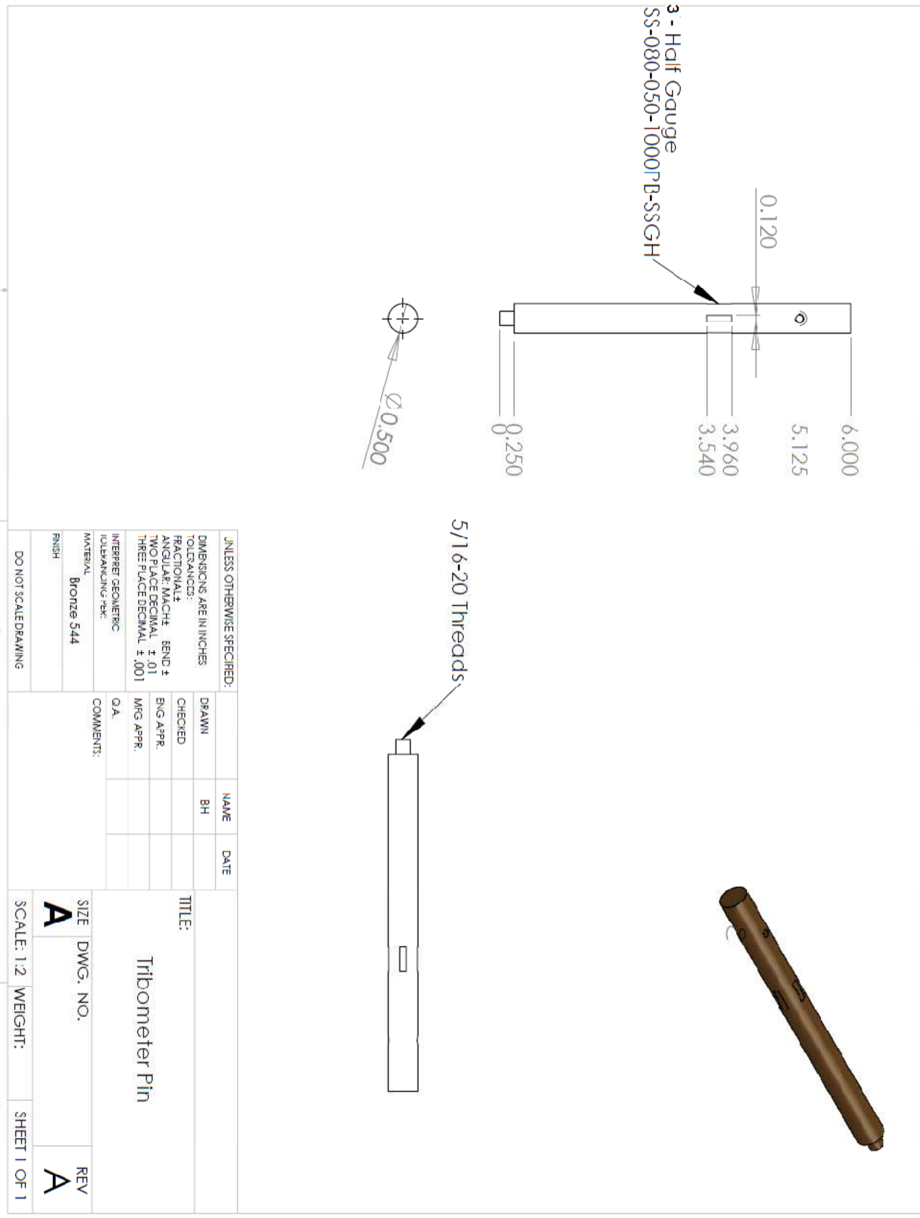


H.5: Table Top

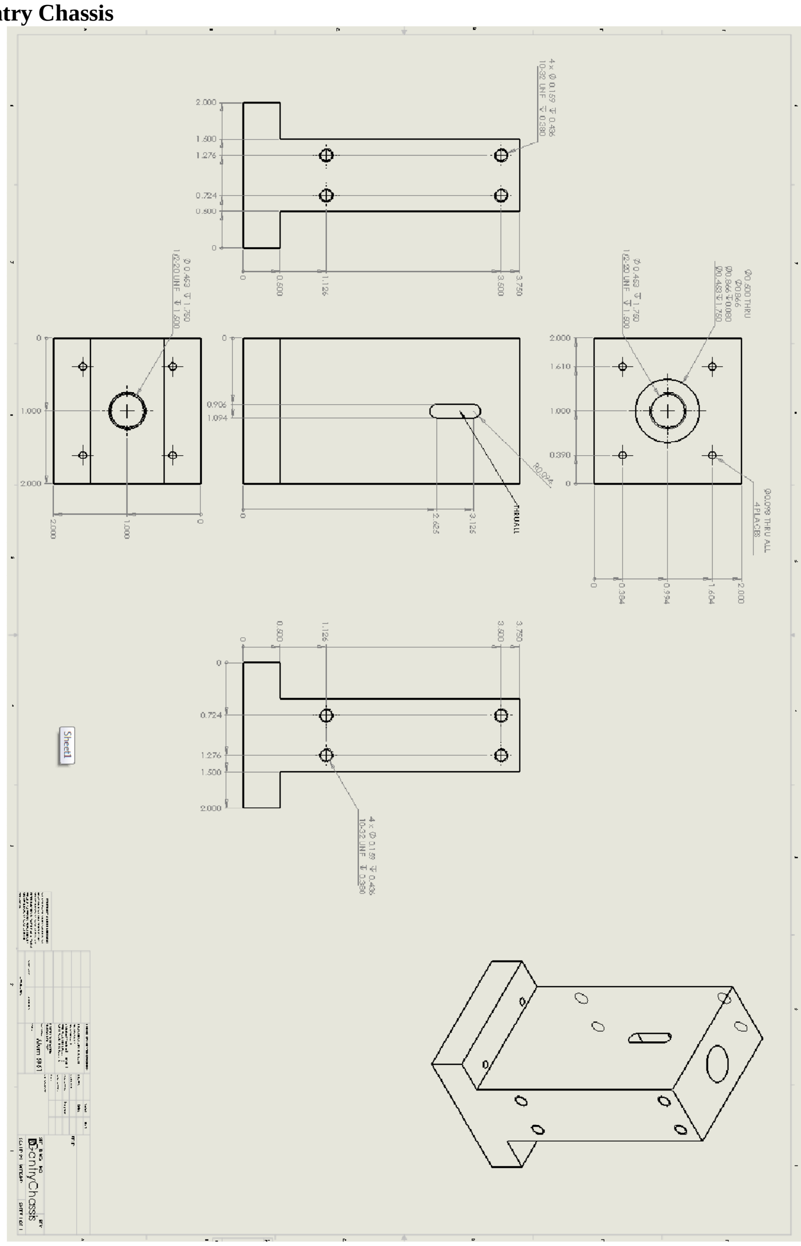


MODIFY EXISTING
35.375
15.850
0
Ø6.230 THRU
0
9.877
21.625
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
DRAWN
Kirby
CHECKED
ENG APPR.
Hopton
MFG APPR.
Q.A.
COMMENTS:
NAME
DATE
TITLE:
SIZE
A
DWG. NO.
TableTop
REV
SCALE: 1:10 WEIGHT:
SHEET 1 OF 1
NEXT ASSY
USED ON
APPLICATION
DO NOT SCALE DRAWING
MATERIAL
FINISH
PROPRIETARY AND CONFIDENTIAL

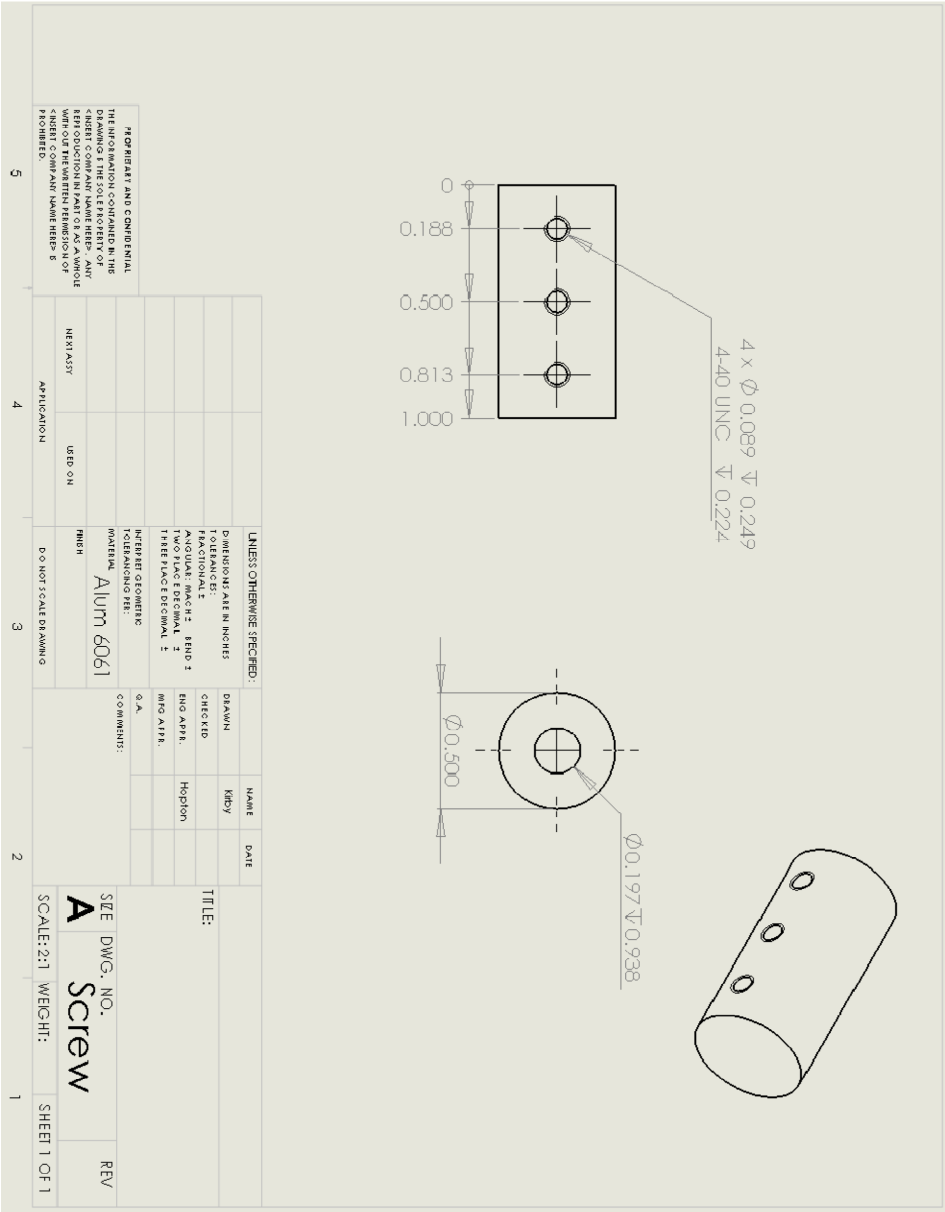
H.6: Bronze Pin



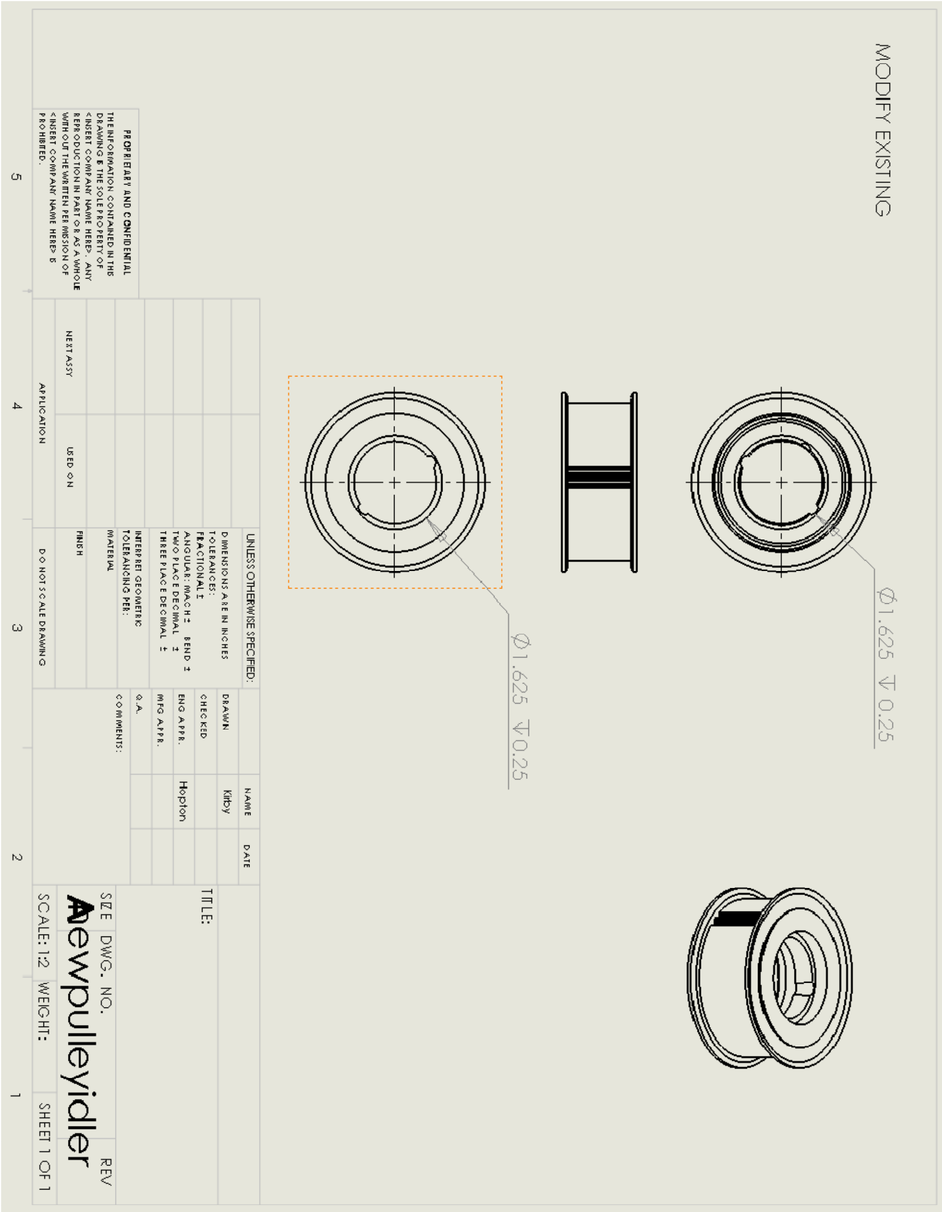
H.7: Aluminum Gantry Chassis



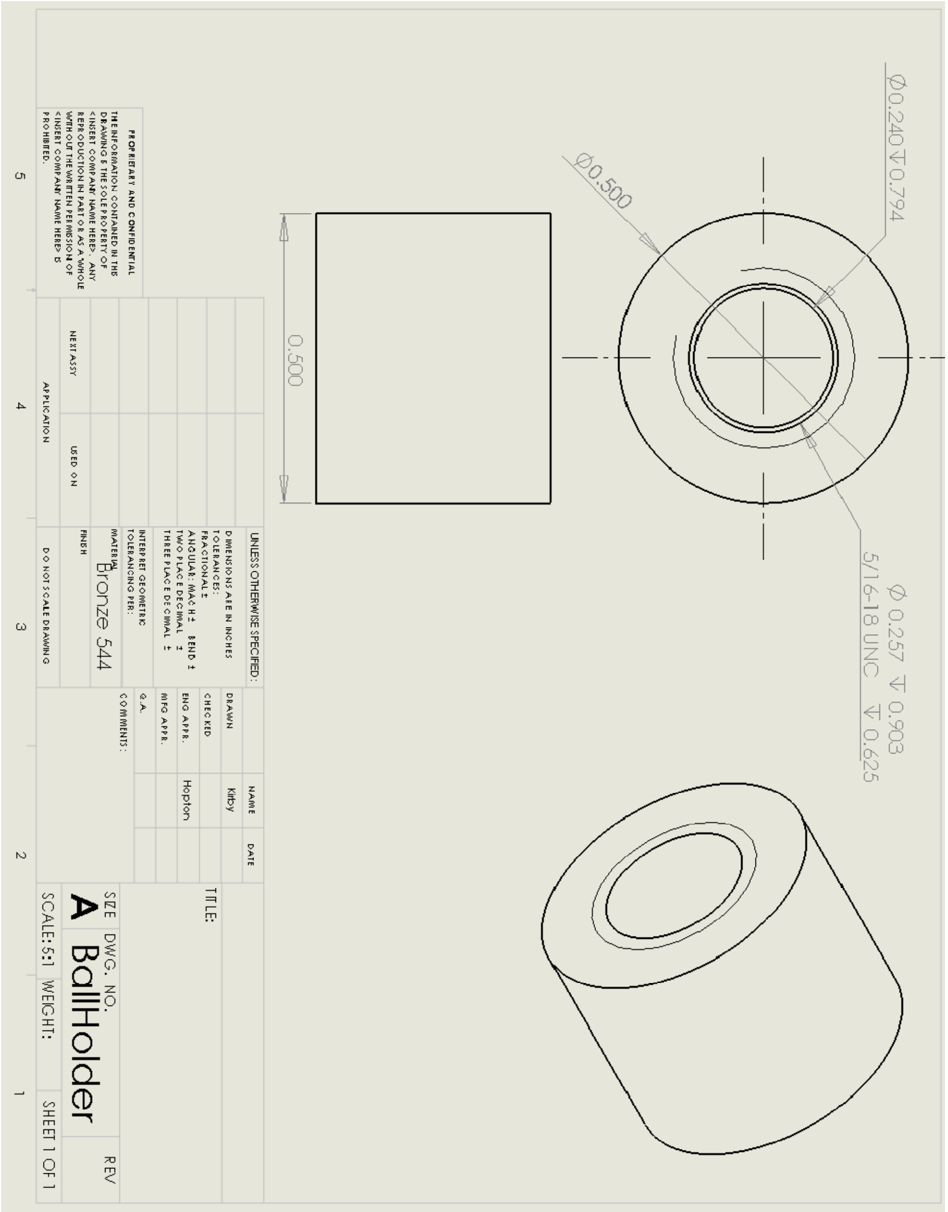
H.8: Threaded Screw Rod



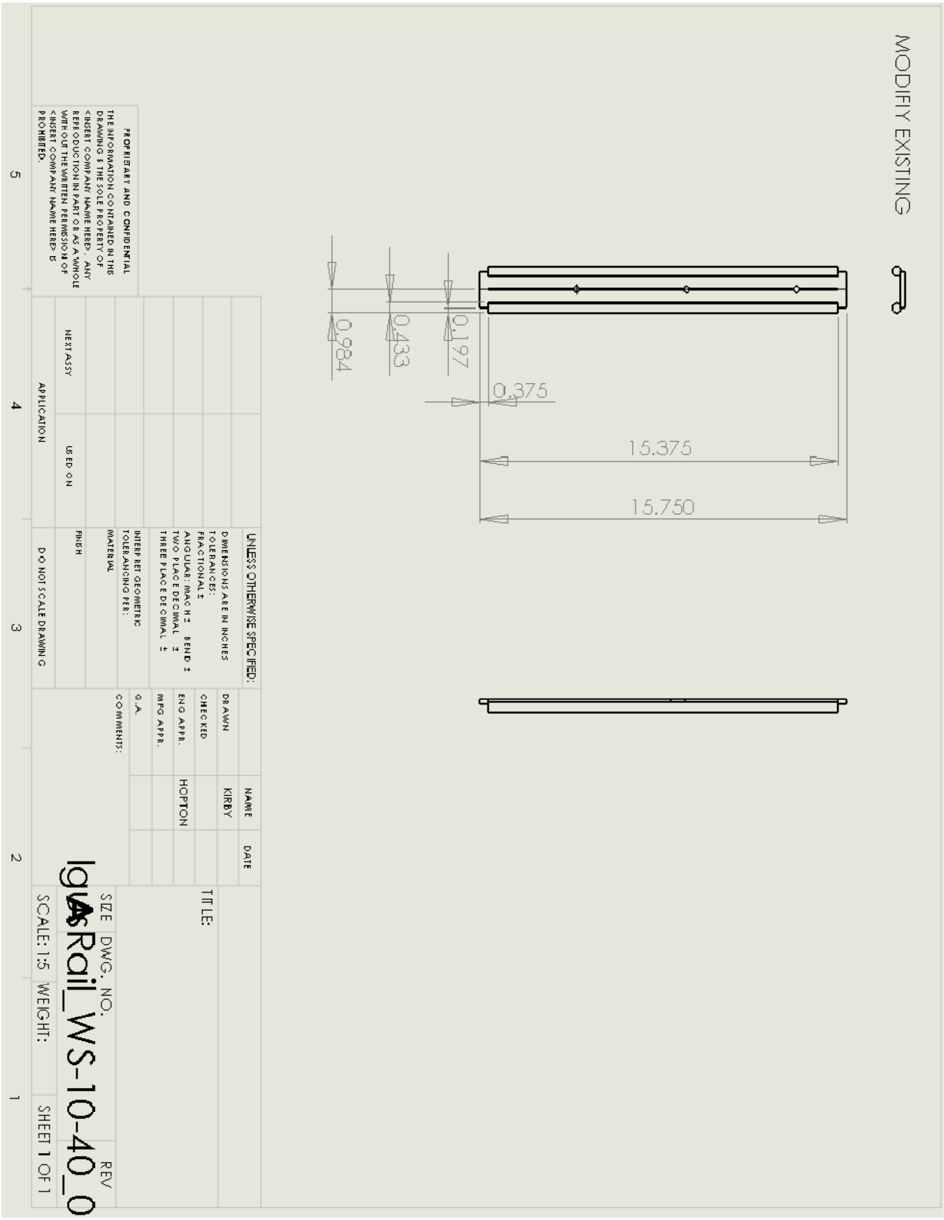
H.9: Pulley Idler



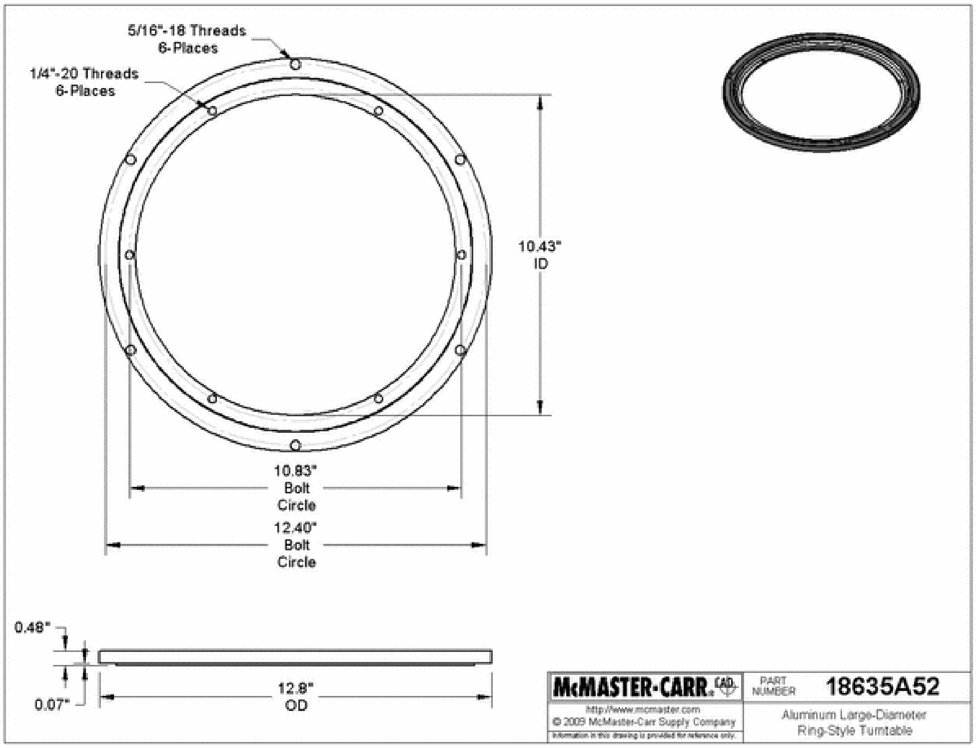
H.10: Ball Holder



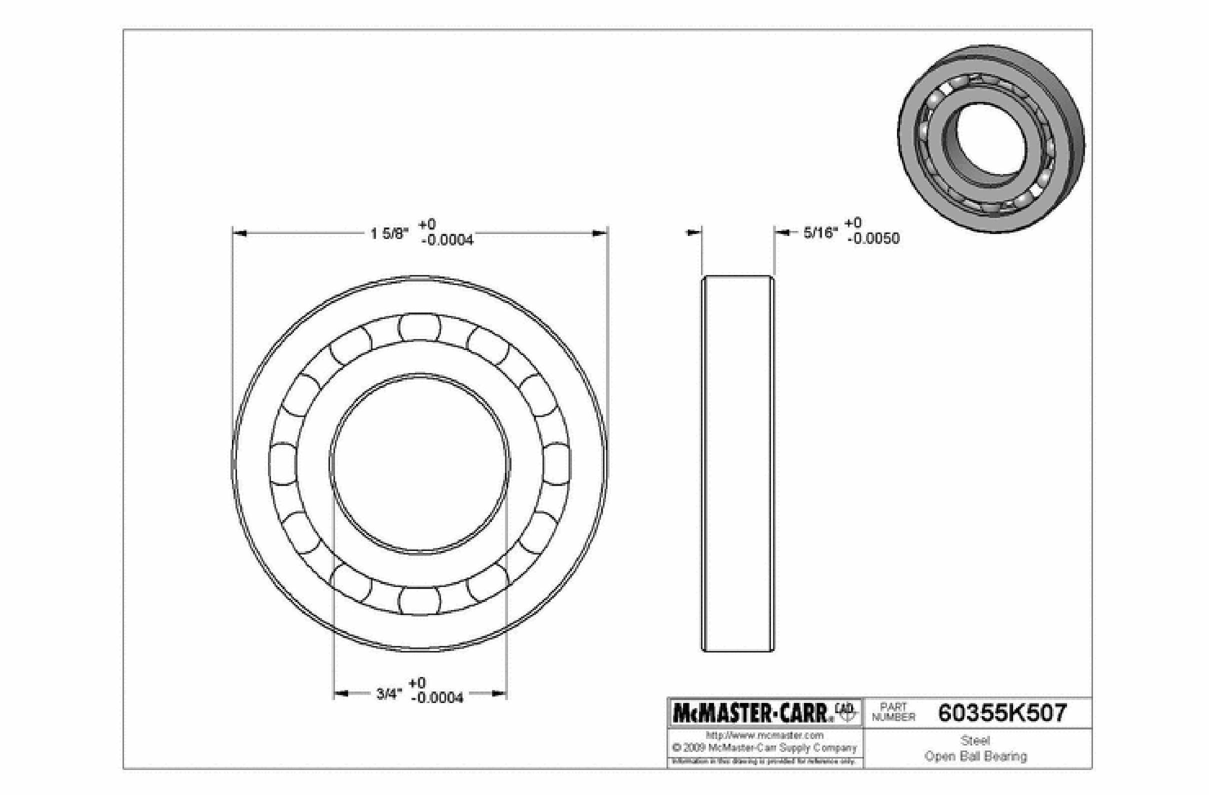
H.11: Igus Rail



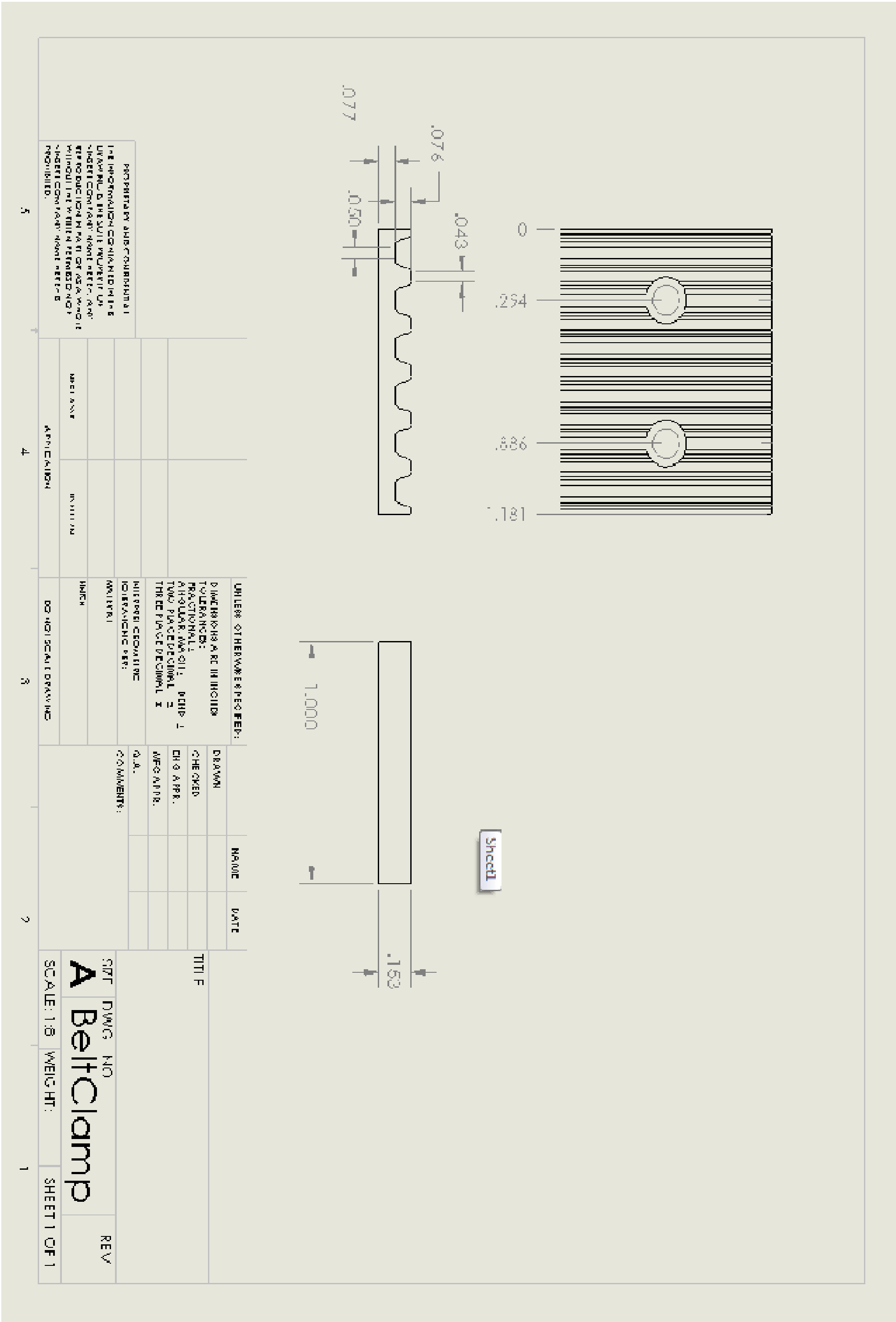
H.12: Rotational Motion Bearing



H.13: Open Ball Bearing



H.14: Belt Clamp



H.15: Bottom Plate

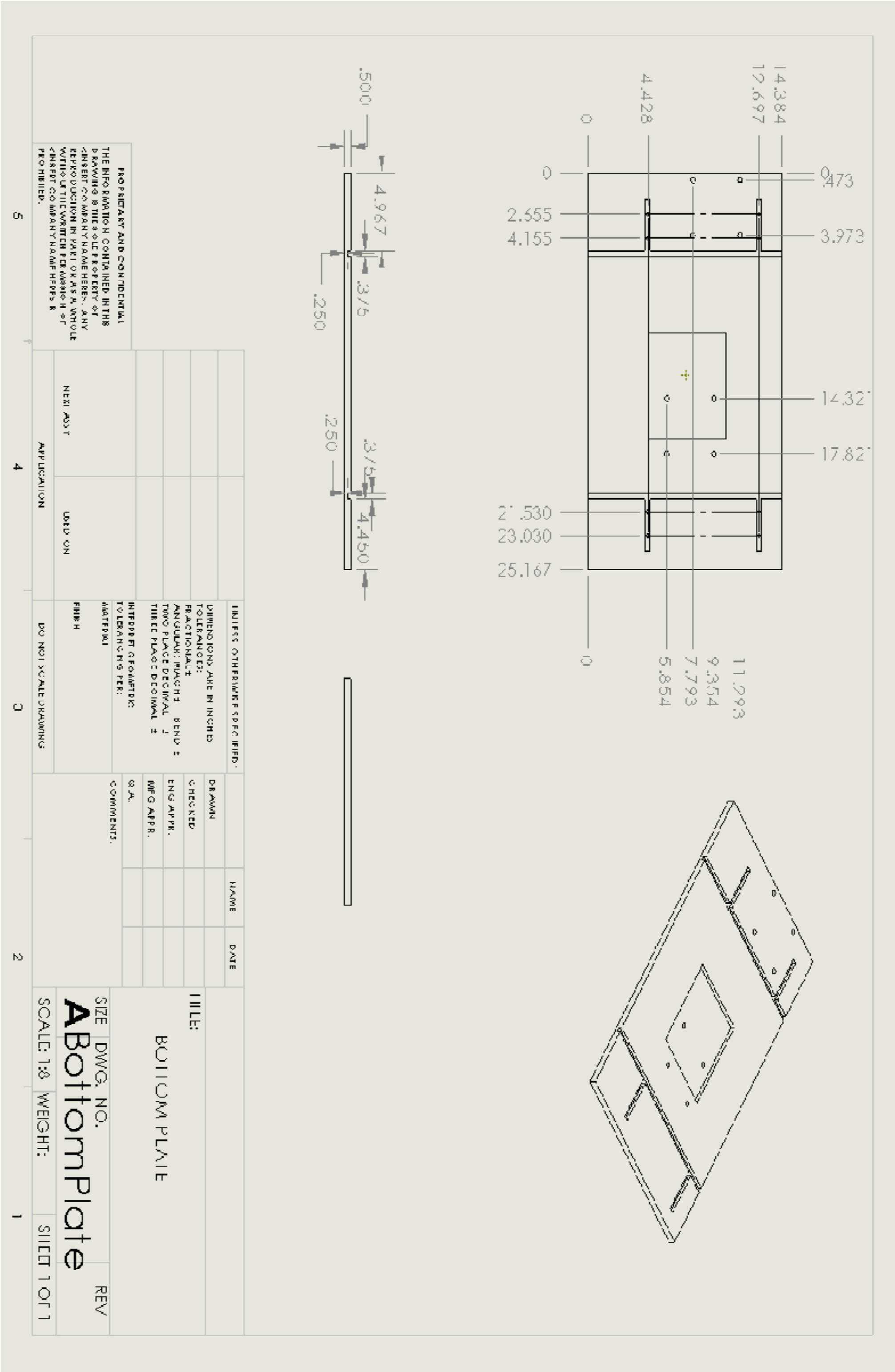


Figure H.16: Side

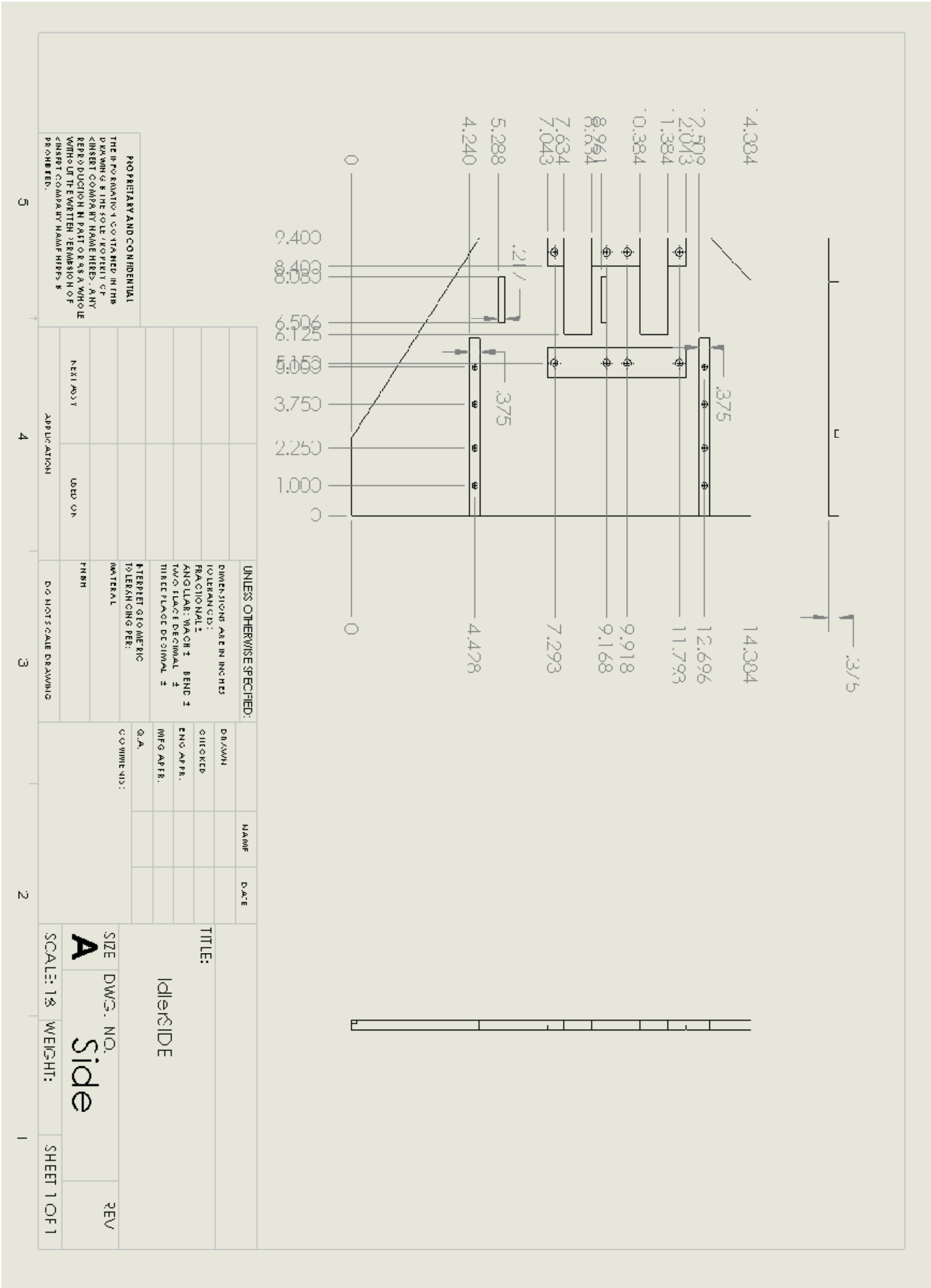
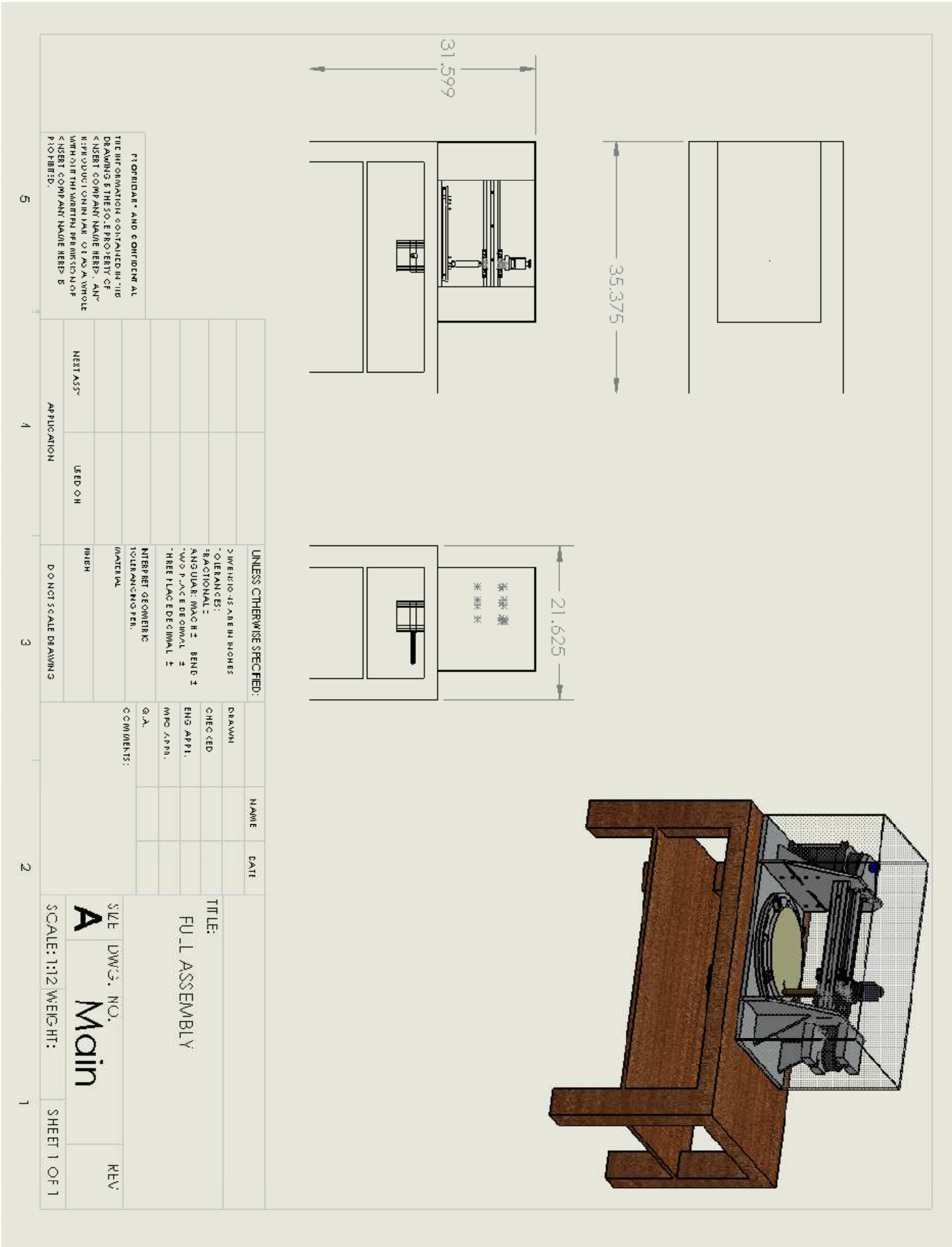


Figure H.17: Main Assembly



Appendix I: Step-by-Step Manufacturing Plans

This appendix contains the tools and machine feeds and speeds that will be used to manufacture the custom parts we are machining ourselves.

I.1: Bronze Shaft

Step	Operation	Machine	Cutting Tool	Cutting Speed	Notes
1	Cut to Length	Band Saw	Band Saw	50 fpm	
2	Make End Smooth	Lathe	¼ End Mill	350 fpm	
3	Drill Shoulder Bolt Holes	Mill	3/16 Drill	175 fpm	Use lubricant
4	Tap Holes	Tap			Use lubricant
5	Turn Bottom End	Lathe	1/8 End Mill	350 fpm	

I.2: Ball Holder

Step	Operation	Machine	Cutting Tool	Cutting Speed	Notes
1	Cut to Length	Band Saw	Band Saw	50 fpm	
2	Mill Flat	Mill	.25 Face Mill	250 fpm	
3	Drill Hole	Mill	.24 Dia Drill	175 fpm	Use lubricant
4	Tap Hole	Tap			Use lubricant

I.3: Aluminum Chassis

Step	Operation	Machine	Cutting Tool	Cutting Speed	Notes
1	Mill Block	Mill	.25 Face Mill	850 fpm	
2	Mill Middle	Mill	.25 Face Mill	850 fpm	
3	Drill Center Hole	Mill	.453 Dia. Drill	400 fpm	Use lubricant
4	Drill Center Hole	Mill	.5 Dia Drill	400 fpm	Use lubricant
5	Thread Center Hole	Tap			Use lubricant
6	Drill Front Screw Holes	Mill	.159 Dia Drill	400 fpm	Use lubricant
7	Drill Back Screw Holes	Mill	.159 Dia Drill	400 fpm	Use lubricant
8	Drill Bottom Screw Holes	Mill	.098 Dia Drill	400 fpm	Use lubricant
9	Mill Slot	Mill	.188 Ball Mill	250 fpm	Use lubricant
10	Tap Holes				Use lubricant

I.4: Steel Threaded Rod

Step	Operation	Machine	Cutting Tool	Cutting Speed	Notes
1	Center Bore	Mill	.197 Dia Drill	40 fpm	Use lubricant
2	Drill Holes	Mill	.089 Dia Drill	40 fpm	Use lubricant
3	Tap Holes	Tap			Use lubricant

I.5: Bottom Plate-6061 AL .5”plate

Step	Operation	Machine	Tool	Speed	Notes
1	Rough Cut	Band Saw	Band Saw	Fast	
2	Clamp down	Mill			To table
3	Create flat	Mill	.5” End Mill	800 rpm	
4	Zero	Mill	Edge finder		
5	Mill slot	Mill	3/8 End Mill	350 fpm	
6	Gear Clearence	Mill	.5” end mill	350 fpm	
7	Drill Hole Pattern	Mill	#10 drill	1000rpm	
8	Motor Mount	Mill	9mm drill	1000 rpm	
9	Counter sink	Hand drill	Countersink	high	

I.6: Idler Side-6061 AL 3/8” plate

Step	Operation	Machine	Tool	Speed	Notes
1	Rough Cut	Band Saw	Band Saw	Fast	
2	Clamp down	Mill			To table
3	Create flat	Mill	.5” End Mill	800 rpm	
4	Zero	Mill	Edge finder		
5	Mill slot	Mill	3/8 End Mill	350 fpm	
6	Gear Clearence	Mill	.5” end mill	350 fpm	
7	Drill Hole Pattern	Mill	#10 drill	1000rpm	
8	Motor Mount	Mill	9mm drill	1000 rpm	
9	Counter sink	Hand drill	Countersink	high	

I.7: Drive Side- 6061 AL 3/8” plate

Step	Operation	Machine	Tool	Speed	Notes
1	Rough Cut	Band Saw	Band Saw	Fast	
2	Clamp down	Mill			To table
3	Create flat	Mill	.5” End Mill	800 rpm	
4	Zero	Mill	Edge finder		
5	Mill slot	Mill	3/8 End Mill	350 fpm	
6	Gear Clearance	Mill	.5” end mill	350 fpm	
7	Drill Hole Pattern	Mill	#10 drill	1000rpm	
8	Motor Mount	Mill	9mm drill	1000 rpm	
9	Counter sink	Hand drill	Countersink	high	

I.8: IdlerMount-6061 AL 1" plate

Step	Operation	Machine	Tool	Speed	Notes
1	Rough Cut	Band Saw	Band Saw	Fast	
2	Clamp down	Mill	Vice	-	
3	Create flat	Mill	.5" End Mill	800 rpm	
4	Zero	Mill	Edge finder	500rpm	

I.9: Linear Bearing Gantry Side- 6061 Aluminium

Step	Operation	Machine	Tool	Speed	Notes
1	Clamp	Mill	vice	-	
2	Zero	Mill	Edge finder	500 RPM	
3	Center Drill	Mill	Center Drill	800 RPM	
4	Drill	Mill	#10 Drill	800 RPM	

I.10: Linear Bearing Slide Material

Step	Operation	Machine	Tool	Speed	Notes
1	Clamp	Mill	vice	-	
2	Mill flat	Mill	3/8 end mill	800 rpm	
3	zero	Mill	Edge finder	800 rpm	
4	Mill ends	Mill	3/8" end mill	800 rpm	

I.11: IdlerMount

Step	Operation	Machine	Tool	Speed	Notes
1	Clamp	Mill	vice	-	
2	Mill flat	Mill	.5" end mill	700 rpm	
3	zero	Mill	Edge finder	700 rpm	
4	Mill ends	Mill	.5"" end mill	700 rpm	
5	Mill slot	Mill	.5" end mill	700rpm	
6	Flip			-	
7	zero	Mill	Edge finder	500rpm	
8	Center drill	Mill	Center drill	600rpm	
9	drill	Mill	.196	1000rpm	
10	tap	Mill	10-32 tap	-	

Appendix J: Location of Strain Gauges

This appendix provides multiple views of the strain gages to show exactly where and how they will be placed on the shaft.

Figure J.1

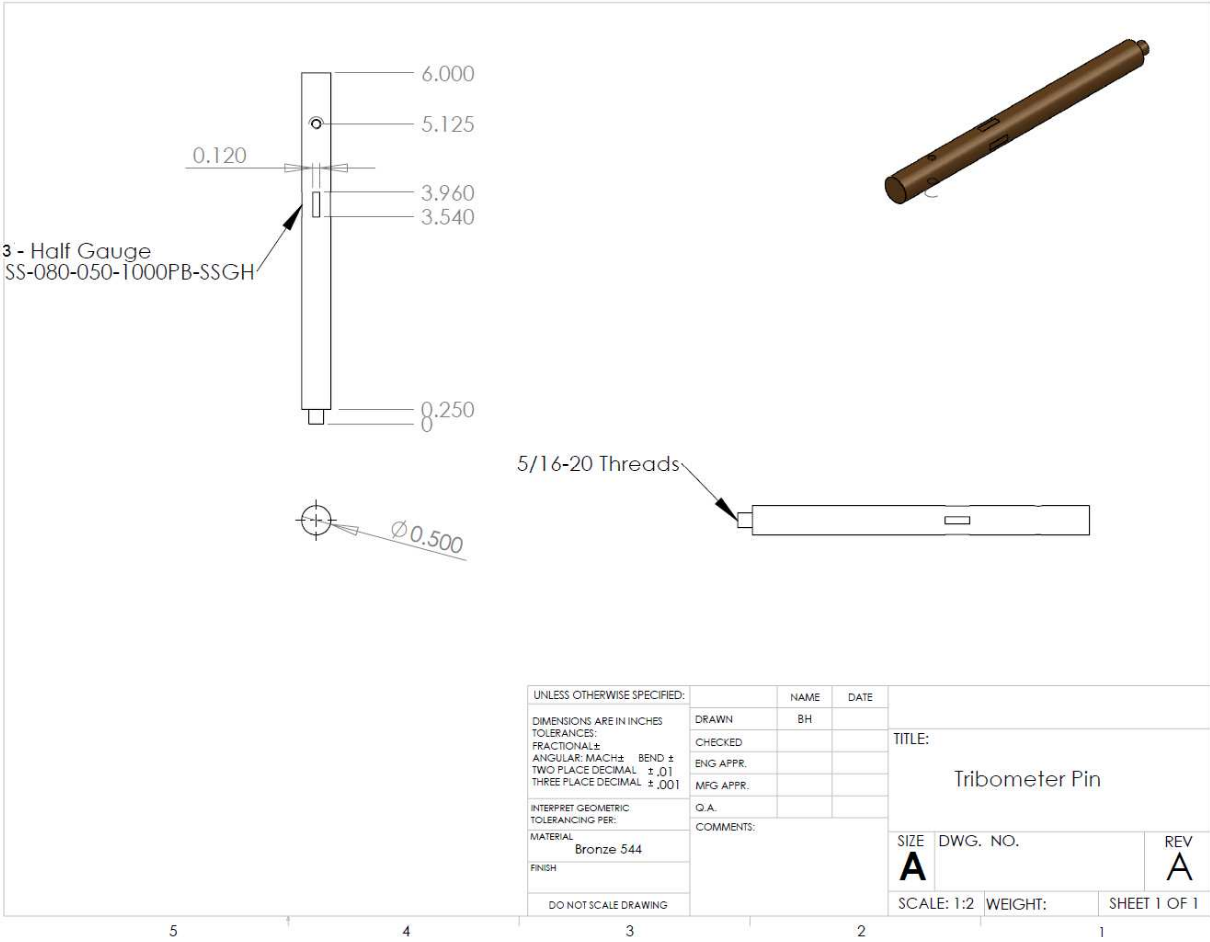

6.000
5.125
0.120
3.960
3.540
3 - Half Gauge
SS-080-050-1000PB-SSGH
0.250
0
Ø0.500
5/16-20 Threads
UNLESS OTHERWISE SPECIFIED:
NAME
DATE
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL±
ANGULAR: MACH± BEND ±
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.001
DRAWN
BH
CHECKED
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
INTERPRET GEOMETRIC
TOLERANCING PER:
MATERIAL
Bronze 544
FINISH
DO NOT SCALE DRAWING
TITLE:
Tribometer Pin
SIZE
A
DWG. NO.
REV
A
SCALE: 1:2
WEIGHT:
SHEET 1 OF 1
5
4
3
2
1

Figure J.2

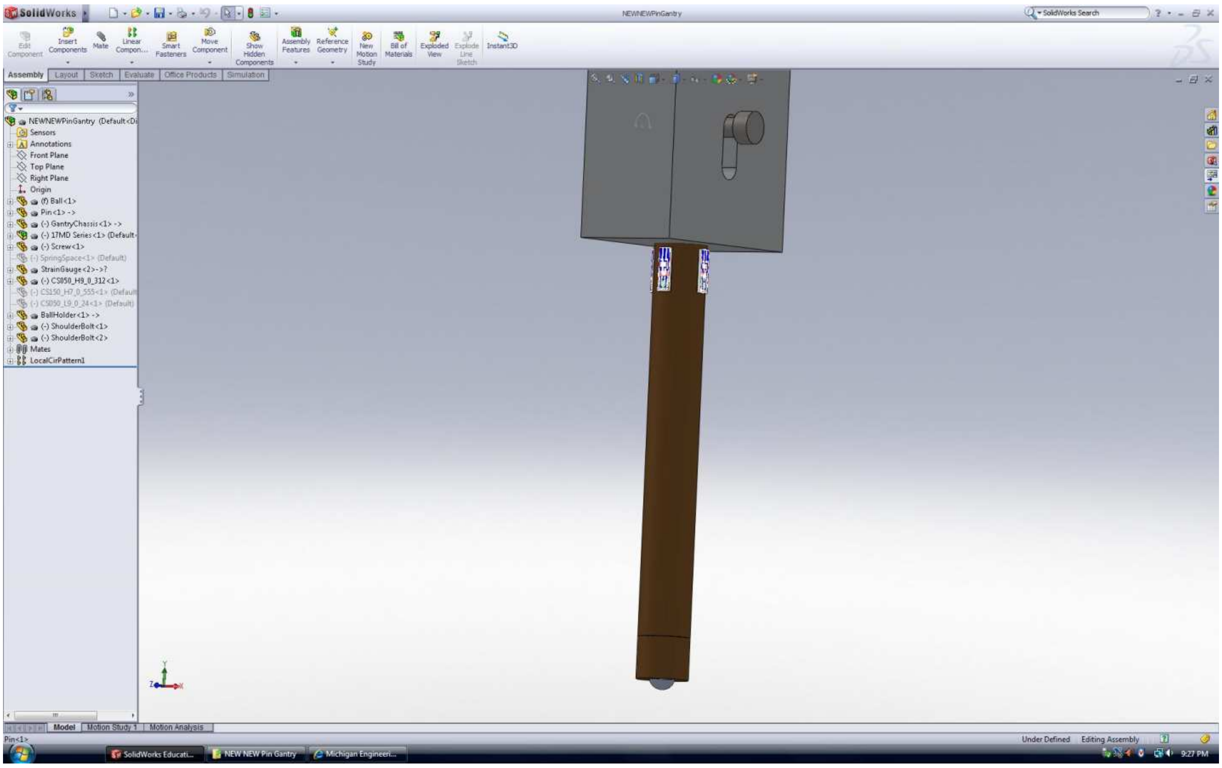
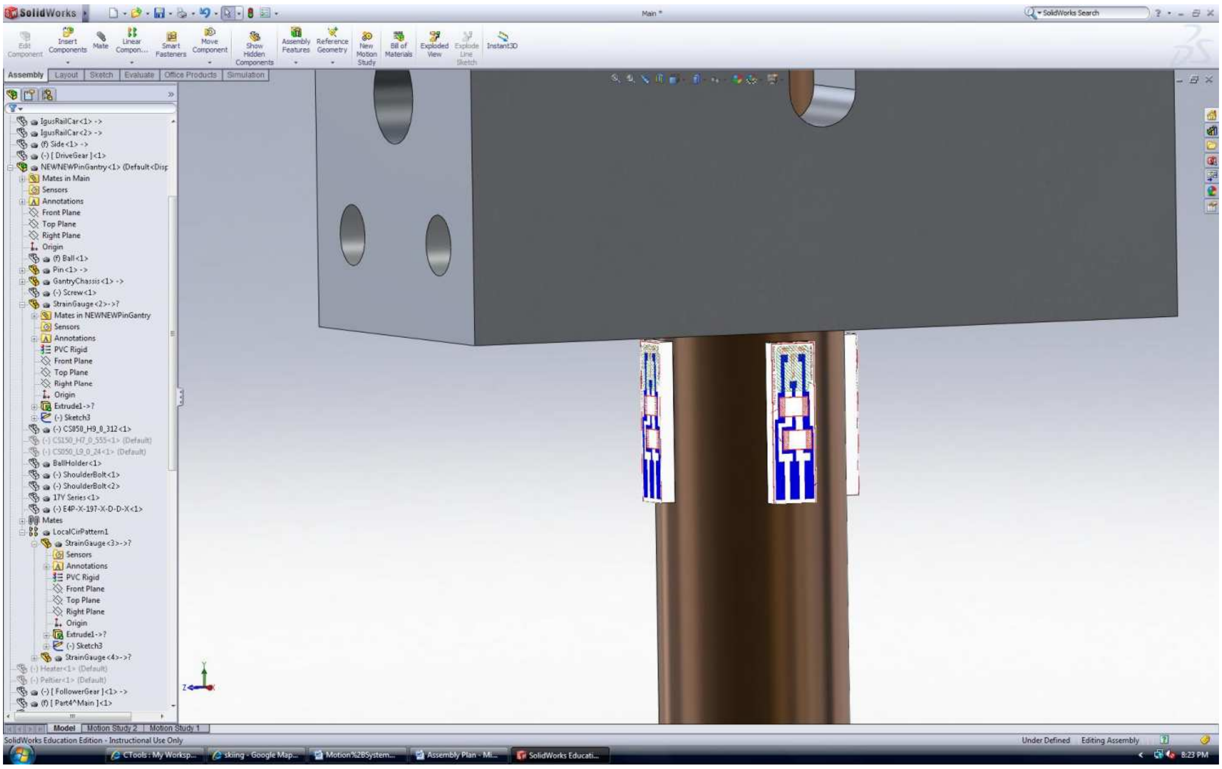


Figure J.3

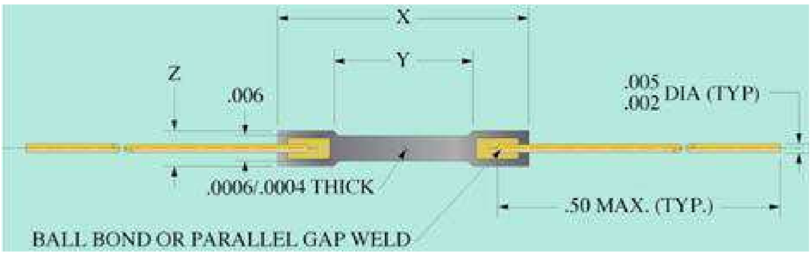


Appendix K: Micron Instruments Design for Force Measurement

This appendix describes the force measurement system that could be built by Micron Instruments to improve the accuracy of the tribometer.

This system will use semi conductor bar gages and will be arranged similarly to the system described in the report. These strain gages would be arranged in a tee rosette and wired in a half bridge to amplify the signal and to compensate for temperature. The difference from the designed mentioned in this report, other than the fact that these gages would not have a backing, comes from the way the third gage that is used to get the second component of friction force would work. Instead of having a tee rosette configuration, two gages would be placed in the axial direction. 180° from this location, two more gages would be placed in the axial direction. These gages would be wired into a full Wheatstone bridge in order to get better accuracy. We have selected a bar gage unit (part# SS-060-033-1000P) which will meet the requirements. This unit is pictured in Figure K.1, below. For the unit selected the dimension for X is .06", for Y is .033", and for Z is .008".

Figure K.1: Micron Instrument Bar Gage



The strain gages will need to be placed and bonded to the outside of the shaft and wired in the correct bridges by Micron Instruments. This should be done by Micron Instruments because of the accuracy required and the size of the gages that we have selected. It has been advised that the gages be bonded on , placed in the Wheatstone bridges and balanced by someone with experience, to achieve the accuracy required. It has been determined that the money needed to have Micron Instruments perform these tasks is well worth the price. A cost break down given to us by Micron can be found in on the next page.

This price also shows a silicone covering for the strain gages. This covering will help to eliminate damage from contact to the strain gages. It will also stop condensation and oils from handing from changing the resistance of the strain gages. Lastly, this covering will help to keep thermal gradients from changing the resistance of the strain gages from the lights. (Herb Chelner) This price also includes putting the wires on that we will use to read the output from the bridges and apply the excitation voltage.

If this strain gage system is used in an ensuing project, Herb Chelner at Micron Instruments should be contacted. He is the President at Micron Instruments and has been very helpful in suggesting this design. He is also the one who created the quote listed. His contact information is as follows: phone: 805-522-1468 email: hchelner@microninstruments.com . It should be noted that this process would require a detail drawing be made of the system and that when the shaft is delivered to Micron Instruments it will take an estimated three weeks for them to return a finished product.

Cost Analysis from Micron Instruments

	COSTS				
Materials					
Gages per rod		**rods**	1	2	3
SS-060-033-1000P		S2	2	4	6
		S4	1	2	3
gage cost per rod		40.14	80.28	160.56	192.84
		81.43	81.43	162.86	244.29
		32.14			
Gaging	126.2		126.2		
	50		100		
	92.87			185.74	
	45			180	
	75.01				225.03
	40				240
Bal. TC	11.91	3	35.73		
	6.75	6		40.5	
	5.75	9			51.75
Bridge Completion on rods					
	35.36	3	106.08		
	26.19	6		157.14	
	19.64	9			176.76
Silicone			30	45	55
Exit wires plus strap		8	24	40	48
Final balance output mv/temp					
	31	3	93		
	20	6		120	
	15	9			135
tot			676.72	1091.8	1368.67
Unit cost			676.72	545.9	456.2233

Appendix L: Strain Gage Bonding Procedure

VISHAY MICRO-MEASUREMENTS

Application Note B-129-8

Surface Preparation for Strain Gage Bonding

1.0 Introduction

Strain gages can be satisfactorily bonded to almost any solid material if the material surface is *properly* prepared. While a properly prepared surface can be achieved in more than one way, the specific procedures and techniques described here offer a number of advantages. To begin with, they constitute a carefully developed and thoroughly proven system; and, when the instructions are followed precisely (along with those for gage and adhesive handling), the consistent result will be strong stable bonds. The procedures are simple to learn, easy to perform, and readily reproducible.

Furthermore, the surface preparation materials used in these procedures are, unless otherwise noted, generally low in toxicity, and do not require special ventilation systems or other stringent safety measures. Of course, as with any materials containing solvents or producing vapors, adequate ventilation is necessary.

The importance of attention to detail, and precise adherence to instructions, cannot be overstressed in surface preparation for strain gage bonding. Less thorough, or even casual, approaches to surface preparation may sometimes yield satisfactory gage installations; but for *consistent* success in achieving high-quality bonds, the methods given here can be recommended without qualification. Fundamental to the Vishay Micro-Measurements system of surface preparation is an understanding of *cleanliness* and *contamination*. All open surfaces not thoroughly and freshly cleaned must be considered contaminated, and require cleaning immediately prior to gage bonding. Similarly, it is imperative that the materials used in the surface preparation be fresh, clean, and uncontaminated. It is worth noting that strain gages as received from Vishay Micro-Measurements are chemically clean, and specially treated on the underside to promote adhesion. Simply touching the gages with the fingers (which are always contaminated) can be detrimental to bond quality.

The Vishay Micro-Measurements system of surface preparation includes five basic operations. These are, in the usual order of execution:

1. *Solvent degreasing*
2. *Abrading (dry and wet)*
3. *Application of gage layout lines*
4. *Conditioning*
5. *Neutralizing*

These five operations are varied and modified for compatibility with different test material properties, and exceptions are introduced as appropriate for certain special materials and situations.

The surface preparation operations are described individually in *Section 2.0*, following a summary of the general principles applicable to the entire process. *Section 3.0* discusses special precautions and considerations which should be borne in mind when working with unusual materials and/or surface conditions.

The various surface preparation and installation accessories referred to throughout this Application Note are Vishay Micro-Measurements Accessories, listed in Strain Gage Accessories Data Book and available directly from Vishay Micro-Measurements.

As a convenience to the gage installer in quickly determining the specific surface preparation steps applicable to any particular test material, *Section 4.0* includes a chart listing approximately 75 common (and uncommon) materials and the corresponding surface preparation treatments.

2.0 Basic Surface Preparation Operations and Techniques

2.1 General Principles of Surface Preparation for Strain Gage Bonding

The purpose of surface preparation is to develop a chemically clean surface having a roughness appropriate to the gage installation requirements, a surface alkalinity corresponding to a pH of 7 or so, and visible gage layout lines for locating and orienting the strain gage. It is toward this purpose that the operations described here are directed.

As noted earlier, cleanliness is vital throughout the surface preparation process. It is also important to guard against recontamination of a once-cleaned surface. Following are several examples of surface recontamination to be avoided:

a. Touching the cleaned surface with the fingers.
b. Wiping back and forth with a gauze sponge, or reusing a once-used surface of the sponge (or of a cotton swab).
c. Dragging contaminants into the cleaned area from the uncleaned boundary of that area.
d. Allowing a cleaning solution to evaporate on the surface.
e. Allowing a cleaned surface to sit for more than a few minutes before gage installation, or allowing a partially prepared surface to sit between steps in the cleaning procedure.

APPLICATION NOTE

Surface Preparation for Strain Gage Bonding

Beyond the above, it is good practice to approach the surface preparation task with freshly washed hands, and to wash hands as needed during the procedure.

2.2 Solvent Degreasing

Degreasing is performed to remove oils, greases, organic contaminants, and soluable chemical residues. Degreasing should always be the first operation. This is to avoid having subsequent abrading operations drive surface contaminants into the surface material. Porous materials such as titanium, cast iron, and cast aluminum may require heating to drive off absorbed hydrocarbons or other liquids.

Degreasing can be accomplished using a hot vapor degreaser, an ultrasonically agitated liquid bath, aerosol type spray cans of CSM-2 Degreaser, or wiping with GC-6 Isopropyl Alcohol. One-way applicators, such as the aerosol type, of cleaning solvents are always preferable because dissolved contaminants cannot be carried back into the parent solvent. Whenever possible, the entire test piece should be degreased. In the case of large bulky objects which cannot be completely degreased, an area covering 4 to 6 in [100 to 150mm] on all sides of the gage area should be cleaned. This will minimize the chance of recontamination in subsequent operations, and will provide an area adequately large for applying protective coatings in the final stage of gage installation.

2.3 Surface Abrading

General — In preparation for gage installation, the surface is abraded to remove any loosely bonded adherents (scale, rust, paint, galvanized coatings, oxides, etc.), and to develop a surface texture suitable for bonding. The abrading operation can be performed in a variety of ways, depending upon the initial condition of the surface and the desired finish for gage installation. For rough or coarse surfaces, it may be necessary to start with a grinder, disc sander, or file. (*Note: Before performing any abrading operations, see Section 3.1 for safety precautions.*) Finish abrading is done with silicon-carbide paper of the appropriate grit, and recommended grit sizes for specific materials are given in *Section 4.0*.

If grit blasting is used instead of abrading, either clean alumina or silica (100 to 400 grit) is satisfactory. In any case, the air supply should be well filtered to remove oil and other contaminant vapors coming from the air compressor. The grit used in blasting should not be recycled or used again in surface preparation for bonding strain gages.

The optimum surface finish for gage bonding depends somewhat upon the nature and purpose of the installation. For general stress analysis applications, a relatively smooth surface (in the order of 100µin [2.5µm] rms) is suitable, and has the advantage over rougher surfaces that it can be cleaned more easily and thoroughly. Smoother surfaces, compatible with the thin "gluelines" required for minimum creep, are used for transducer installations. In contrast, when very high elongations must be measured, a rougher (and preferably cross-hatched) surface should be prepared. The recommended surface finishes for several classes of gage installations are summarized in Table I, below.

TABLE I

CLASS OF INSTALLATION	SURFACE FINISH, rms	
	µin	µm
General stress analysis	63 - 125	1.6 - 3-2
High elongation	>250	>6.4
	cross-hatched	
Transducers	16 - 63	0.4 - 1.6

Wet Abrading — Whenever *M-Prep* Conditioner A is compatible with the test material (see *Section 4.0*), the abrading should be done while keeping the surface wet with this solution, if physically practicable. Conditioner A is a mildly acidic solution which generally accelerates the cleaning process and, on some materials, acts as a gentle etchant. It is not recommended for use on magnesium, synthetic rubber, or wood.

2.4 Gage-Location Layout Lines

The normal method of accurately locating and orienting a strain gage on the test surface is to first mark the surface with a pair of crossed reference lines at the point where the strain measurement is to be made. The lines are made perpendicular to one another, with one line oriented in the direction of strain measurement. The gage is then installed so that the triangular index marks defining the longitudinal and transverse axes of the grid are aligned with the reference lines on the test surface.

The reference, or layout, lines should be made with a tool that *burnishes*, rather than scores or scribes, the surface. A scribed line may raise a burr or create a stress concentration. In either case, such a line can be detrimental to strain gage performance and to the fatigue life of the test part. On aluminum and most other nonferrous alloys, a 4H drafting pencil is a satisfactory and convenient burnishing tool. However, graphite pencils should never be used on high-temperature alloys, where the operating temperature might cause a carbon embrittlement problem. On these and other hard alloys, burnished alignment marks can be made with a ballpoint pen or a round-pointed brass rod. Layout lines are ordinarily applied following the abrading operation and before final cleaning. All residue from the burnishing operation should be removed by scrubbing with Conditioner A as described in the following section.

2.5 Surface Conditioning

After the layout lines are marked, Conditioner A should be applied repeatedly, and the surface scrubbed with cotton tipped applicators until a clean tip is no longer discolored by the scrubbing. During this process the surface should be kept constantly wet with Conditioner A until the cleaning is completed. *Cleaning solutions should never be allowed to*

For technical support, contact
micro-measurements@vishay.com
Document Number: 11129
Revision 14-Apr-09

Surface Preparation for Strain Gage Bonding

dry on the surface. When clean, the surface should be dried by wiping through the cleaned area with a single slow stroke of a gauze sponge. The stroke should begin inside the cleaned area to avoid dragging contaminants in from the boundary of the area. Then, with a fresh sponge, a single slow stroke is made in the opposite direction. The sponge should never be wiped back and forth, since this may redeposit the contaminants on the cleaned surface.

2.6 Neutralizing

The final step in surface preparation is to bring the surface condition back to an optimum alkalinity of 7.0 to 7.5pH, which is suitable for all Vishay Micro-Measurements strain gage adhesive systems. This should be done by liberally applying *M-Prep* Neutralizer 5A to the cleaned surface, and scrubbing the surface with a clean cotton-tipped applicator. The cleaned surface should be kept completely wet with Neutralizer 5A throughout this operation. When neutralized, the surface should be dried by wiping through the cleaned area with a *single* slow stroke of a clean gauze sponge. With a fresh sponge, a *single* stroke should then be made in the opposite direction, beginning with the cleaned area to avoid recontamination from the uncleaned boundary.

If the foregoing instructions are followed precisely, the surface is now properly prepared for gage bonding, and the gage or gages should be installed as soon as possible.

3.0 Special Precautions and Considerations

3.1 Safety Precautions

As in any technical activity, safety should always be a prime consideration in surface preparation for strain gage bonding. For example, when grinding, disc sanding, or filing, the operator should wear safety glasses and take such other safety precautions as specified by his organization or by the Occupational Safety and Health Administration (OSHA).

When dealing with toxic materials such as beryllium, lead, uranium, plutonium, etc., all procedures and safety measures should be approved by the safety officer of the establishment before commencing surface preparation.

3.2 Surfaces Requiring Special Treatment

Concrete — Concrete surfaces are usually uneven, rough, and porous. In order to develop a proper substrate for gage bonding, it is necessary to apply a leveling and sealing precoat of epoxy adhesive to the concrete. Before applying the precoat, the concrete surface must be prepared by a procedure which accounts for the porosity of this material.

Contamination from oils, greases, plant growth, and other soils should be removed by vigorous scrubbing with a stiff-bristled brush and a mild detergent solution. The surface is then rinsed with clean water. Surface irregularities can be removed by wire brushing, disc sanding, or grit blasting, after which all loose dust should be blown or brushed from the surface.

The next step is to apply Conditioner A generously to the surface in and around the gaging area, and scrub the area with a stiff-bristled brush. Contaminated Conditioner A should be blotted with gauze sponges, and then the surface should be rinsed thoroughly with clean water. Following the water rinse, the surface acidity must be reduced by scrubbing with Neutralizer 5A, blotting with gauze sponges, and rinsing with water. A final thorough rinse with distilled water is useful to remove the residual traces of water-soluble cleaning solutions. Before precoating, the cleaned surface must be thoroughly dried. Warming the surface gently with a propane torch or electric heat gun will hasten evaporation.

Vishay Micro-Measurements M-Bond AE-10 room-temperature-curing epoxy adhesive is an ideal material for precoating the concrete. For those cases in which the test temperature may exceed the specified maximum operating temperature of AE-10 (+200°F [+95°C]), it will be necessary to fill the surface with a higher temperature resin system such as M-Bond GA-61.

In applying the coating to the porous material, the adhesive should be worked into any voids, and leveled to form a smooth surface. When the adhesive is completely cured, it should be abraded until the base material begins to be exposed again. Following this, the epoxy surface is cleaned and prepared conventionally, according to the procedure specified in *Section 4.0* for bonding gages to epoxies.

Plated Surfaces — In general, plated surfaces are detrimental to strain gage stability, and it is preferable to remove the plating at the gage location, if this is permissible. Cadmium and nickel plating are particularly subject to creep, and even harder platings may creep because of the imperfect bond between the plating and the base metal. When it is not permissible to remove the plating, the surface should be prepared according to the procedure given in *Section 4.0* for the specific plating involved. Note that it may be necessary to adjust testing procedures to minimize the effects of creep.

Use of Solvents on Plastics — Plastics vary widely in their reactivity to solvents such as those employed in the surface preparation procedures described here. Before applying a solvent to any plastic, *Section 4.0*, which includes most common plastics, should be referred to for the recommended compatible solvent. For plastics not listed in *Section 4.0*, the manufacturer of the material should be consulted, or tests should be performed to verify nonreactivity between the solvent and the plastic.

Dimensional or Mechanical Changes Due to Surface Preparation — For most materials, strain measurement results are usually not significantly changed by the surface preparation procedures described in this Application Note. Even with appreciable material removal, effects on the static mechanical properties of the test part are generally negligible compared to other error sources in the experiment. It should be understood, however, that removal of a plated or hardened surface layer, or of a surface layer with significant residual stresses, may noticeably affect the fatigue life or the wear characteristics of the part when operated under dynamic service conditions.

Silicone Contamination —The properties of silicones which make them excellent lubricants and mold-release agents also make them the enemies of adhesion, and therefore potentially the most serious of contaminants to be encountered in the practice of bonding strain gages. The problem is compounded by the high natural affinity of the silicones for most materials, and by their tendency to migrate. Furthermore, since silicones are relatively inert chemically, and unaffected by most solvents, they are among the most difficult surface contaminants to remove.

The best practice is to keep the gage-bonding area free of silicones. This may not be as easy as it sounds, since the widely used silicones can be introduced from a variety of sources. For instance, many hand creams and cosmetics contain silicones, and these should not be used by persons involved in gage installation. Some of the machining lubricants also contain silicones, and such lubricants should be avoided when machining parts that are to have strain gages installed. Similarly, silicone-saturated cleaning tissues for eyeglasses should not be used in the gage-bonding area or by gage-installation personnel.

Regardless of efforts to avoid silicones, contamination may still occur. Light contamination can sometimes be removed by cleaning with Conditioner A, preferably heated to +200°F [+95°C]. More severe cases may require special cleaning solutions and procedures, recommendations for which should be obtained from the manufacturer of the silicone compound involved in the contamination.

4.0 Index of Test Materials and Surface Preparation Procedures

In this section, the specific step-by-step surface preparation procedures are given for approximately 75 different materials. For compactness, and convenient, quick access to the procedure for any particular material, the information is presented in chart form in Table II. The test materials are listed alphabetically, from ABS Plastics to Zirconium; and the complete procedure for each material is defined by one or more digits in each of the applicable operations columns of the table. Each digit identifies the required operation and specifies the step number for that operation in the complete procedure.

For example, assume that the necessity arises for bonding one or more strain gages to a brass test specimen. Reading down the *Specimen Material* column of Table II to Brass, and following that row across the table to the right, the first step in surface preparation consists of degreasing the specimen with CSM-2 Degreaser. The symbol (1) in the *Isopropyl Alcohol* column indicates that this is a suitable substitute degreasing operation. Continuing across the row, the second operation calls for abrading the specimen surface with 320-grit silicon-carbide paper. In the third operation the specimen is reabraded with 400-grit silicon-carbide paper, wet lapping with Conditioner A if feasible. The fourth and fifth operations consist of applying layout lines for locating the gages, and scrubbing the surface clean with Conditioner A. Cleaning with isopropyl alcohol is the final operation in the procedure.

In the *Remarks* column, it is recommended that the gages be installed within 20 minutes after completing the surface preparation, because the freshly bared brass surface tends to oxidize rapidly. In addition, in the *Grit Blast* column, the gage installer is specifically advised not to substitute grit blasting for other surface abrading methods, in order to avoid significantly altering the surface condition of this relatively soft material. Surface preparation procedures for other materials are defined similarly in the table, and, in many cases, accompanied by special warnings or recommendations in the *Remarks* column. When an operation not included in the first ten column headings is required, it is indexed in the *Other* column, with an arrow pointing to the *Remarks* column where the operation is specified.

Additional References

For additional information, refer to Instruction Bulletins listed below:

B-127, "Strain Gage Installations with M-Bond 200 Adhesive".

B-130, "Strain Gage Installations with M-Bond 43-B, 600, and 610 Adhesives".

B-137, "Strain Gage Applications with M-Bond AE-10, AE-15, and GA-2 Adhesive Systems".

Important Notice

The procedures, operations, and chemical agents recommended in this Application Note are, to the best knowledge of Vishay Micro-Measurements, reliable and fit for the purposes for which recommended. This information on surface preparation for strain gage bonding is presented in good faith as an aid to the strain gage installer; but no warranty, expressed or implied, is given, nor shall Vishay Micro-Measurements be liable for any injury, loss, or damage, direct or consequential, connected with the use of the information. Before applying the procedures to any material, the user is urged to carefully review the application with respect to human health and safety, and to environmental quality

Surface Preparation for Strain Gage Bonding

TABLE II
Index of Test Materials and Surface Preparation Procedures (Sheet 1 of 3)

See Section:	2.2		2.3				2.4	2.5	2.6		
SPECIMEN MATERIAL	CSM-2 DEGREASER	GC-6 ISOPROPYL ALCOHOL	GRIT-BLAST†	220-GRIT ABRASIVE PAPER	320-GRIT ABRASIVE PAPER	400-GRIT ABRASIVE PAPER††	GAGE LOCATION LAYOUT	CONDITIONER A (SCRUB)	NEUTRALIZER 5A	OTHER	REMARKS
ABS PLASTICS		1	No			2	3	4	5		ABS plastics may be affected by ketones, esters, aromatics, and chlorinated hydrocarbons.
ACRYLICS		1				2	3	4	5		Acrylics may be affected by ketones, esters, aromatics, and chlorinated hydrocarbons.
ALUMINUM, ALCLAD	1					2	3	4	5		Alclad coating must be removed prior to gage installation by abrading with 180-grit or 220-grit silicon-carbide paper. Test for completeness of Alclad removal: (a) swab area with 10% sodium hydroxide solution - area will darken within 60 sec if cladding is completely removed; (b) neutralize surface with Conditioner A; (c) flush area with distilled water. Proceed with surface preparation as specified at left.
ALUMINUM, ANODIZED	1					2	3	4	5		Black or colored anodizing must be removed prior to gage installation. Use nonchlorinated household cleaner to strip sealer. Clear anodized surface acceptable for elastic strain level only.
ALUMINUM, CASTINGS	1		(2)	(2)	2	3	4	5	6		Gages should be bonded within 30 min after final surface preparation.
ALUMINUM, WROUGHT	1		(2)		2	3	4	5	6		Gages should be bonded within 30 min after final surface preparation.
ANTIMONY		1	No			2	3	4	5		
ASPHALT		1	No				3		4	2	Often necessary to grind, disc sand, or file surface.
BERYLLIUM	1		No				3	4	5	2	Obtain safety department approval for surface removal. Abraded particles must be kept wet to prevent becoming airborne, and must be properly disposed of. Some individuals may develop allergic reaction. Gloves should be worn.
BERYLLIUM COPPER	1*	(1)*				3	4	5	6	2	Obtain safety department approval for surface removal. Abraded particles must be kept wet to prevent becoming airborne, and must be properly disposed of. Some individuals may develop allergic reaction.
BISMUTH		1	No			2	3	4	5		
BONE		(3)								1,2,3	Clean with ether under proper ventilation, then scrape surface, and redry with ether.
BORON-EPOXY COMPOSITES		1,5*	No			2	3	4	(5*)	(2)	Scrub with a slurry of pumice powder and Conditioner A.
BRASS	1	(1)	No		2	3	4	5**			Install gages within 20 min of final surface preparation.
BRICK		1,3*							4*	2	Wire-brush or disc-sand, and remove dust with dry paint brush. Fill and seal surface with epoxy adhesive, such as Micro-Measurements AE-10, and sand smooth after adhesive is cured.
BRONZE	1	(1)			2	3	4	5**			Install gages within 20 min of final surface preparation.
CADMIUM PLATE	1	(1)	No		2	3	4	5	6		Cadmium plating has a tendency to creep, and the plating should be removed if permissible.
CARBON (see GRAPHITE)											
CERAMICS		1*			2		3	4	5*		
CHROMIUM PLATE	1	(1)			2		3	4	5		Chromium plating should be removed at gage installation site if permissible.
CONCRETE										1	See Concrete Section 3.2 in text.
COPPER AND COPPER-BASED ALLOYS	1	(1)	No		2	3	4	5**	No		Install gages within 20 min after final surface preparation.

SPECIAL NOTES
* Heating specimen will help drive out oils, moisture, and solvent.
** Rinse with distilled water, and wipe dry.
† Use clean (filtered), dry air. Do not recycle alumina or silica grits.
†† Wet lap with Conditioner A when compatibility is indicated by "Conditioner A (Scrub)" column.
( ) Parentheses indicate alternate step(s).

APPLICATION NOTE

Instruction Bulletin B-130

Vishay Micro-Measurements

Strain Gage Installations with M-Bond 43-B, 600, and 610 Adhesive Systems

INTRODUCTION

Vishay Micro-Measurements M-Bond 43-B, 600, and 610 adhesives are high-performance epoxy resins, formulated specifically for bonding strain gages and special-purpose sensors. When properly cured, these adhesives are useful for temperatures ranging from -452° to +350°F [-269° to +175°C] with M-Bond 43-B, and to +700°F [+370°C] for short periods with M-Bond 600 and 610. In common with other organic materials, life is limited by oxidation and sublimation effects at elevated temperatures. M-Bond 43-B is particularly recommended for transducer applications up to +250°F [+120°C], and M-Bond 610 for transducers up to +450°F [+230°C].

For proper results, the procedures and techniques presented in this bulletin should be used with qualified Vishay Micro-Measurements installation accessory products (refer to Vishay Micro-Measurements Strain Gage Accessories Databook). Accessories used in this procedure are:

CSM Degreaser or GC-6 Isopropyl Alcohol	CSP-1 Cotton Applicators
Silicon-Carbide Paper	MJG-2 Mylar® Tape
M-Prep Conditioner A	TFE-1 Teflon® Film
M-Prep Neutralizer 5A	HSC-X Spring Clamp
GSP-1 Gauze Sponges	GT-14 Pressure Pads and Backup Plates

MIXING INSTRUCTIONS

Since M-Bond 43-B is a solvent-thinned, precatalyzed epoxy mixture, it is applied at room temperature directly as received. The M-Bond 600 and 610, on the other hand, are two-component systems. These must be mixed as follows:

1. Resin and curing agent bottles must be at room temperature before opening.

2. Using the disposable plastic funnel, empty contents of bottle labeled “Curing Agent” into bottle of resin labeled “Adhesive”. Discard funnel.

3. After tightening the brush cap (included separately), thoroughly mix contents of this “Adhesive” bottle by vigorously shaking it for 10 seconds.

4. Mark bottle with date mixed in space provided on the label.

Allow this freshly mixed adhesive to stand for at least one hour before using.

SURFACE PREPARATION

The extensive subject of surface preparation techniques is covered in Application Note B-129. Metal surface cleaning procedures usually involve solvent degreasing with either CSM Degreaser or GC-6 Isopropyl Alcohol, abrading, and cleaning with M-Prep Conditioner A, followed by application of M-Prep Neutralizer 5A. When practical, these preparation procedures should be applied to an area significantly larger than that occupied by the gage. Surfaces should be free from pits and irregularities. Porous surfaces may be precoated with a filled epoxy, such as M-Bond GA-61, which is then cured and abraded.

SHELF LIFE AND POT LIFE

At room temperature, M-Bond 600 has a useful storage life of approximately three months, while M-Bond 43-B and M-Bond 610 will last about nine months.

Once opened and mixed, M-Bond 600 and 610 have room-temperature pot lives of two weeks and six weeks, respectively. Since M-Bond 43-B is supplied already mixed, its pot life is about the same as its shelf life when kept in a tightly closed container.

These periods of adhesive usefulness can be increased by refrigeration at +30° to +40°F [0° to +5°C]. Check individual adhesive kit labels for details. Never open a refrigerated bottle until it has reached room temperature.

GAGE INSTALLATION

The basic steps for bonding gages using M-Bond 43-B, 600, and 610 adhesives are given on the following pages.

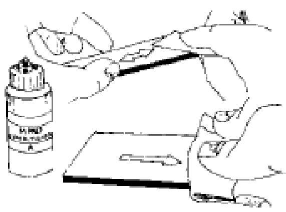
HANDLING PRECAUTIONS

Epoxy resins and hardeners may cause dermatitis or other allergic reactions, particularly in sensitive persons. The user is cautioned to: (1) avoid contact with either the resin or hardener; (2) avoid prolonged or repeated breathing of the vapors; and (3) use these materials only in well-ventilated areas. If skin contact occurs, thoroughly wash the contaminated area with soap and water immediately. In case of eye contact, flush immediately and secure medical attention.
Rubber gloves and aprons are recommended, and care should be taken not to contaminate working surfaces, tools, container handles, etc. Spills should be cleaned up immediately. For additional health and safety information, consult the Material Safety Data Sheet, which is available upon request.

Step 1

Thoroughly degrease the gaging area with solvent, such as CSM Degreaser or GC-6 Isopropyl Alcohol. The former is preferred, but there are some materials (e.g., titanium and many plastics) that react with CSM. In these cases, GC-6 Isopropyl Alcohol should be considered. All degreasing should be done with uncontaminated solvents-thus the use of "one-way" containers, such as aerosol cans, is highly advisable.

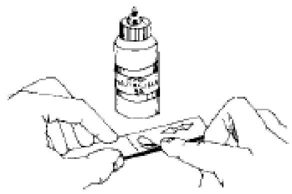
Step 2



Preliminary dry abrading with 220- or 320-grit silicon-carbide paper is generally required if there is any surface scale or oxide. Final abrading is done by using 320- or 400-grit silicon-carbide paper on surfaces thoroughly wetted with M-Prep Conditioner A; this is followed by wiping dry with a gauze sponge.

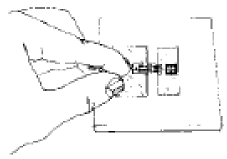
With a 4H pencil (on aluminum) or a ballpoint pen (on steel), burnish (do not scribe) whatever alignment marks are needed on the specimen. Repeatedly apply Conditioner A and scrub with cotton-tipped applicators until a clean tip is no longer discolored. Remove all residue and Conditioner by again slowly wiping through with a gauze sponge. Never allow any solution to dry on the surface because this invariably leaves a contaminating film and reduces chances of a good bond.

Step 3



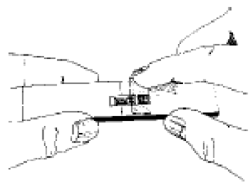
Now apply a liberal amount of M-Prep Neutralizer 5A and scrub with a cotton-tipped applicator. With a single, slow wiping motion of a gauze sponge, carefully dry this surface. Do not wipe back and forth because this may allow contaminants to be redeposited on the cleaned surface.

Step 4



Remove a gage from its mylar envelope with tweezers, making certain not to touch any exposed foil. Place the gage, bonding side down, onto a chemically clean glass plate or empty gage box. If a solder terminal is to be incorporated, position it next to the gage. While holding the gage in position with a mylar envelope, place a short length of MJG-2 mylar tape down over about half of the gage tabs and the entire terminals.

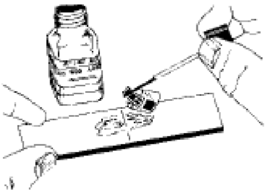
Step 5



Remove the gage/tape/terminal assembly by peeling tape at a shallow angle (about 30°) and transferring it onto the specimen. Make sure gage alignment marks coincide with specimen layout lines. If misalignment does occur, lift the end of the tape at a shallow angle until assembly is free. Realign and replace.
Use of a pair of tweezers often facilitates this handling.

Note: A "hot-tack" method of positioning can be used, which eliminates need for taping. This method is explained after Step 9.

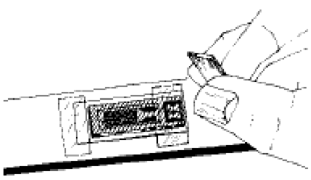
Step 6



Now, by lifting at a shallow angle, peel back one end of the taped assembly so as to raise both gage and terminal. By curling this mylar tape back upon itself, it will remain in position, ready to be accurately repositioned after application of adhesive.

Coat the gage backing, terminal, and specimen surface with a thin layer of adhesive. Also coat the foil side of open-faced gages. Do not allow the adhesive applicator to touch the tape mastic. Permit adhesive to air-dry, by solvent evaporation, for 5 to 30 minutes at +75°F [+24°C] and 50% relative humidity. Longer air-drying times are required at lower temperatures and/or higher humidities. ***Note: An additional drying step with 43-B is beneficial for large gages. Place the unclamped installation in an oven for 30 minutes at +175°F [+85°C] following the air-dry step above.***

Step 7

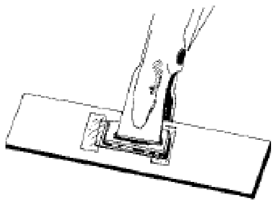


Return the gage/terminal assembly to its original position over the layout marks. Use only enough pressure to allow the assembly to be tacked down. Overlay the gage/ terminal area with a piece of thin Teflon sheet (TFE-1). If necessary, anchor the Teflon in position across one end with a piece of mylar tape.

Cut a 3/32-in [2.5-mm] thick silicone gum pad and a metal backup plate (GT-14) to a size slightly larger than the gage/terminal areas, and carefully center these. Larger pads may restrict proper spreading of adhesive, and entrap residual solvents during cure process.

Note: Steps 6, 7, and 8 must be completed within 30 minutes with M-Bond 600, 4 hours with M-Bond 610, and 24 hours with M-Bond 43-B.

Step 8



Either spring clamps or deadweight can be used to apply pressure during the curing cycle. For transducers, 40 to 50 psi [275 to 350 kN/m2] is recommended and 10 to 70 psi [70 to 480 kN/m2] for general work. Place the clamped gage/specimen into a cool oven and raise temperature to the desired during level at a rate of 5° to 20°F [3° to 11°C] per minute. Air bubbles trapped in the adhesive, uneven gluelines, and high adhesive film stresses often result from starting with a hot oven. Time-versus-temperature recommendations for curing each adhesive are given on the next page.

Step 9

Upon completion of the curing cycle, allow oven temperature to drop to at least 100°F [55°C] before removing the specimen. Remove clamping pieces and mylar tape. It is advisable to wash off the entire gage area with either RSK Rosin Solvent or toluene. This should remove all residual mastic and other contamination. Blot dry with a gauze sponge.

"Hot-Tack" Method of Gage Installation
This procedure eliminates all need for taping to prevent movement of the gage during mounting, and is especially suited to M-Bond 43-B and M-Bond 600.

1. After completing the preceding Steps 1, 2, and 3, remove a gage from its mylar envelope using clean tweezers.
2. Coat the bonding side of gage and gaging area of the specimen with adhesive, and set each aside to air-dry for at least 15 minutes. M-Bond 43-B may dry for up to 24 hours.

3. Using tweezers, position gage onto the specimen. A properly cleaned dental probe may help.

4. To anchor the gage, use a 15- to 25-watt soldering iron with a new conical tip. This is usually done by hot-tack-setting the adhesive at two spots (such as opposite gage-alignment marks) while temporarily holding the gage down with a mylar envelope. A little experimentation may be required to learn the correct iron temperature and hot tip contact time. These depend upon type of adhesive used and thermal conductivity of the base material.

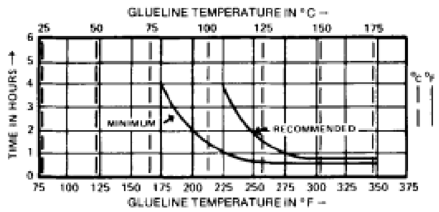
5. If the gage is open-faced, apply a thin coating of adhesive to its face and allow to dry for at least five minutes before overlaying with a Teflon sheet (as described in Step 7). Proceed with Steps 8 and 9.

RECOMMENDED CURE SCHEDULE

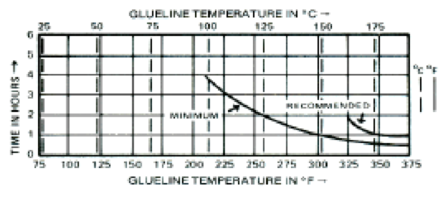
It should be noted that the following curves represent a range of time-versus-temperature; however, the upper limits of both time and temperature should be employed whenever possible, while keeping in mind the possible effect on the heat treat condition of the substrate material.

M-Bond 43-B: 2 hours at +375°F [+190°C].

M-Bond 600: Cure at temperature for time period specified by graph below.



M-Bond 610: Cure at temperature for time period specified by graph below.



POSTCURING

Postcures with the clamping fixture removed are usually required for stable transducer applications. Postcuring can be done following Step 9 above, or after wiring the transducer (subject to temperature limits of solder and wire insulation).

M-Bond 43-B: 2 hours at +400°F [+205°C].

M-Bond 600: 1 to 2 hours at 50°F [30°C] above maximum operating or curing temperature, whichever is greater.

M-Bond 610: 2 hours at 50° to 75°F [30° to 40°C] above maximum operating or curing temperature, whichever is greater.

FINAL INSTALLATION PROCEDURES

1. Refer to Strain Gage Accessories Databook to select an appropriate solder, and attach leadwires. Be sure to remove solder flux with Rosin Solvent. Gage tabs and terminals can be cleaned prior to soldering by light abrading with pumice to remove the adhesive film. This pumicing is not required with gages having integral leads (Options L and LE) or pre-attached solder dots. See Application Note TT-606, "*Soldering Techniques for Lead Attachment to Strain Gages with Solder Dots.*" General soldering instructions are discussed in Application Note TT-609, "*Strain Gage Soldering Techniques.*"

2. Select and apply protective coatings according to recommendations given in Strain Gage Accessories Databook.

ELONGATION CAPABILITIES

M-Bond 43-B:
1% at -452°F [-269°C]; 4% at +75°F [+24°C]; 2% at +300°F [+150°C].

M-Bond 600 & 610:
1% at -452°F [-269°C]; 3% from room temperature to 500°F [+260°C].

Mylar and Teflon are Registered Trademarks of DuPont.

Appendix M: Belt Drive Design

Industrial Belt Design - Drive Detail Report

Design Flex® Pro by the Gates Corporation

Designed For:

Provided By: brian hopton
university of michigan
gg brown
ann arbor, Michigan 48109
United States
bhop23@gmail.com
248767557 Phone

Application: **450 Team 21 Linear Belt Drive**

INPUT

Drive Information

Speed Ratio: 1.00
Input Load: 15 N-m
Service Factor: 1.6
Design Power: 24 N-m
Center Distance: 20 in +/-10%

	DriveR	DriveN
RPM:	375.0	375.0 +/-4%

Bushings Checked: Any
Belts Checked: Poly Chain Carbon, PowerGrip GT2, TruMotion, PowerGrip Timing, Poly Chain GT2, PowerGrip HTD, PowerGrip GT

SELECTED DRIVE

Belt Type: **PowerGrip GT2 - 5M**
Speed Ratio: **1.0**
dN RPM: **375.0**
Rated Load: **26.06 N-m, ODR: 1.09**
Belt Pull: **124 lb**
Center Distance: 21.26 in
Install/Take-Up Range: 20.41 in to 21.30 in
Noise: 46 dB @ 275 Hz

	Belt	DriveR	DriveN
Part No:	5MR-1300-15	P44-5MGT-15	P44-5MGT-15
Product No:	9390-7260	7709-1044	7709-1044
Top Width:	--	0.89 in	0.89 in
Weight:	0.18 lb	0.92 lb	0.92 lb
Rim/Belt Speed:	271 ft/min	266 ft/min	266 ft/min
RPM:	63.5	375.0	375.0
Bushing Part No:	--	1108	1108
Bore:	--	0.5 in - 1.125 in	0.5 in - 1.125 in
Pitch Diameter:	--	2.76 in	2.76 in

TENSION

	New Belt	Used Belt
Static Tension (Per rib/strand):	58 to 64 lb	41 to 47 lb
Static Belt Pull:	117 to 129 lb	82 to 94 lb
Rib/Strand Deflection Distance:	0.33 in	0.33 in
Rib/Strand Deflection Force:	4.3 to 4.7 lb	3.2 to 3.6 lb
Sonic Tension Meter:	260 to 286 N	182 to 208 N
Belt Frequency:	60 to 63 Hz	50 to 54 Hz

505C/507C Model STM Settings: Weight: 4.1 g/m,Width: 15 mm/#R, Span: 540 mm

When planning to re-install used belts, measure and record the tension before removing and re-install at the recorded tension.

NOTES

- This report: (1) only applies to Gates' products; (2) contains confidential information; (3) may only be disclosed to support the sale or maintenance of our products; and (4) is not a guarantee of performance.
- Buyer has sole responsibility for the selection and testing of products for any intended use which may not include flight-related aircraft applications.

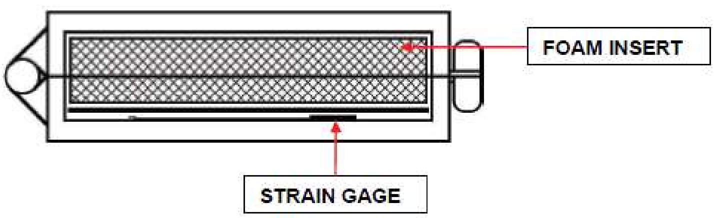
Appendix N: Instructions to Unpack the Strain Gages

RECOMMENDATIONS FOR REMOVING STRAIN GAGES FROM BOXES

The strain gage is secured in the strain gage box by two adhesive "dots".
A foam insert backed with a layer of paper may protect the gage.

1. Open the box.
2. If there is a foam insert **(See Fig. 1)** remove it using a pair of tweezers. Lift the foam insert vertically. Do not exert any pressure on the foam pad as this may damage the gage.

FIGURE 1.
Strain Gage Box



3. Do not attempt to remove the gage from the box until the wires attached to the adhesive dots have been cut.
4. Place the blade of an "X-ACTO" knife (or similar) on the wire as close as possible to the adhesive dot. Apply a **FIRM DOWNWARD PRESSURE** to sever the wire **(See Fig. 2).**

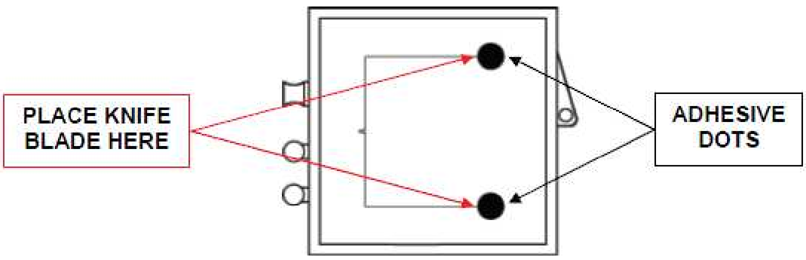


FIGURE 2.
Strain gage secured in strain gage box

Do not use a back and forth cutting motion to cut the wire as this may cause the wire to deform and/or detach from the strain gage.

Repeat for the second wire.

080230

RECOMMENDATIONS FOR REMOVING STRAIN GAGES FROM BOXES

4509 Runway Street • Simi Valley, California 93063 • Phone:(800) 638-3770 or (805) 522-4676 • Fax: (805) 522-4982

Appendix O: Strain Gage Information

MICRON INSTRUMENTS

BACKED STRAIN GAGE HALF BRIDGE (SSGH)

Temperature Compensated

BACKED SEMICONDUCTOR STRAIN GAGE HALF BRIDGE - TEMPERATURE COMPENSATED

STRAIN GAGE HALF BRIDGE

Micron Instruments offers a range of semiconductor backed half bridge strain gages. The gages are installed on a flexible insulated circuit with easy to solder pads which can be bent without hurting the gage and will perform like a foil gage except that the resistive change is 30 to 55 times greater.

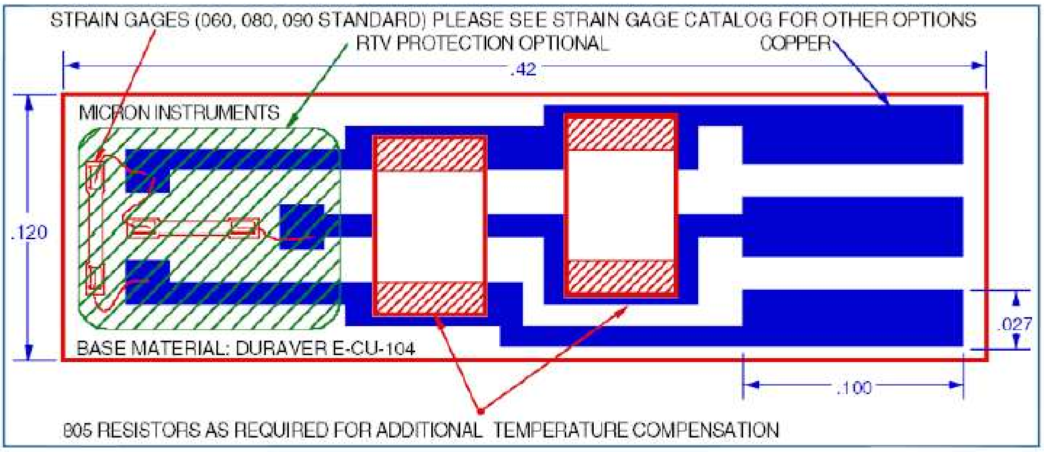
There are two gages on a backing, one longitudinal and one transverse. These gages are thermally matched to track each other before installing onto the backing. When used as one side of the bridge, they compensate each other thermally even when bonded since they both see the same thermal expansion. When both gages have a gage factor (GF) of 140 the transverse gage will normally see the Poisson's effect, which for most steels is 0.3. This reduces the transverse gage factor to 42. The average GF for the half bridge would be 91.

FEATURES

- Backed SSGH's are as easy to install as foil gages
- There are two gages on a backing, one longitudinal and one transverse.
- SSGH's are suggested for use in prototyping or proof of concept

SPECIFICATIONS

- Bar gages ranges available 120 ohm up to 1000 ohms
- Czochralski pulled boron doped silicon
- Base material Duraver E-CU-104
- Dimensions base .42"x.12"
- Copper solder pads



MICRON'S semiconductor strain gages are made from "P" doped bulk silicon. They have no P/N junction.
The silicon is etched to shape, eliminating the potential for molecular dislocation or cracks, thereby optimizing performance.

4509 Runway Street • Simi Valley, California 93063 • Phone:(800) 638-3770 or (805) 522-4676 • Fax: (805) 522-4982

sensors@microninstruments.com • www.microninstruments.com

STRAIN GAGE HALF BRIDGE

Backed Temperature Compensated Semiconductor Strain Gages Half Bridge (SSGH)

PART NUMBER	Width	Lead Attachment	Thickness	Resistance Ohms@ 78° F	Gage Factor	TCGF	TCR
SS-060-022-500PB-SSGH	.008	WL	.0004	540 ±50	150 ±10	-13%	17%
SS-060-033-500PB-SSGH	.008	WL	.0004	540 ±50	140 ±10	-13%	15%
SS-060-033-1000PB-SSGH	.008	WL	.0004	1050 ±75	155 ±10	-18%	24%
SS-080-050-120PB-SSGH	.008	WL	.0004	120 ±20	120 ±10	-9%	5%
SS-080-050-230PB-SSGH	.008	WL	.0004	230 ±30	120 ±10	-9%	5%
SS-080-050-345PB-SSGH	.008	WL	.0004	345 ±40	140 ±10	-13%	16%
SS-080-050-500PB-SSGH	.008	WL	.0004	540 ±50	140 ±10	-13%	16%
SS-080-050-1000PB-SSGH	.008	WL	.0004	1050±75	155 ±10	-18%	24%
SS-090-060-500PB-SSGH	.009	WL	.0004	540 ±50	140 ±10	-13%	16%
SS-090-060-1150PB-SSGH	.008	WL	.0004	1125 ±75	155 ±10	-18%	24%

Strain Gage Ordering Information

SS-080-050-500 P B SSGH

Model

Total Length

Active Length

Nominal Resistance at 78°F

Half Bridge

Back Gage

Dopant

Standard Gage Specifications

Material	Czochralski pulled boron doped silicon.
Leads	.002 dia. gold.
Contact Pad	Gold nickel fused, aluminum, or Hi-Temp.
Lead Attachment	Parallel gap welded with epoxy reinforcement or ball bonded.
Operating Strain	±2000 μ inch/inch (3000 μ inch/inch max.)
Linearity	Better than ±0.25% to 600 μ inch/inch
	Better than ±1.5% to 1500 μ inch/inch
Max. Operating Temp.	+278°F Bonded.

Standard Bridge Matching

Temperature °F	**0°**	**78°**	**278°**	
Standard Matching	**±0.6%**	**±0.4%**	**±0.4%**	**Percent of Base Resistance**

TERMS & CONDITIONS

Minimum Order:	$50.00
FOB:	Simi Valley, California
Terms:	Net 30 Days
	Credit Cards: Visa, Master Card, American Express
Effective Date:	March 2008

Prices Subject to Change without Notice

4509 Runway Street • Simi Valley, California 93063 • Phone: (800) 638-3770 or (805) 522-4676 • Fax: (805) 522-4982

sensors@microninstruments.com • www.microninstruments.com

Appendix P: Description of Engineering Changes

There were three significant engineering changes. The documentation for these engineering changes can be found below.

1)

What has changed: The motor that will apply the normal force has changed from a NEMA 17Y4 with 100 oz-in of holding torque to a NEMA 17Y3 with 62 oz-in of holding torque. Information about these motors can be found here: http://www.anaheimautomation.com/products/stepper/stepper-motors.php?tID=75&pt=t&cID=19

What part of the project does this impact: This change impacts the force application and the linear drive system. Because the holding torque will be lower, the tribometer will only be able to apply 172 N of force instead of the specified 200N. This change will also lessen the weight of this system by 0.3 lbs, allowing for slightly quicker acceleration in the linear motion system. The mounting for this system will remain the same, the change of the motors only changes the length of the motor.

Why was this change made: The motor originally picked out was out of stock and would take 3 months to be delivered. This would make it impossible for the team to meet the deadline

Who made this change: Tristan Kreutzberg 12/1/09

Authorization: Professor Gordon Krauss 12/7/09

2)

What has changed: The tribometer will no longer have any type of environmental system

What part of the project does this impact: The environmental system.

Why was this change made: Our team member, Justin Hresko, has decided to drop this course. He was responsible for this system and made little to no progress on its design. It would be impossible for the team to design, order the required materials, and build by this Thursday.

Below is the email received from Justin on Saturday the 5th of December telling us about dropping the course:

Hey guys,

To make it short and sweet i don't think i will be completing the course this semester. I've been trying to work on my system since 10 this morning and just dont have much to show for it. Even though it shouldn't be that difficult, my mind is just not in the right place to finish up. Even yesterday i felt i was just standing around and contributing little to the overall project and being more of a nuisance than any help. I apologize that you all had to put up with my shenanigans these past few weeks, and *wish you the best of luck.*

Sincerely,
Justin

Who made this change: Brian Kirby, Brian Hopton, Tristan Kreutzberg 12/5/09

Authorization: Professor Gordon Krauss 12/7/09

3)

What has changed: The idler pulley will utilize a bronze shaft press-fit into place as a connecting part between the idler pulley and idler shaft

What part of the project does this impact: The linear motion system.

Why was this change made: The idler shaft was too large to fit in the idler pulley. Originally, the pulley was to have a larger hole machined, but due to a machining error, the fit was not precise enough. A large bronze shaft was acquired and machined to the appropriate dimensions to give a correct fit between the pulley and shaft.

Who made this change: Brian Kirby, Brian Hopton, Tristan Kreutzberg 12/2/09

Authorization: Professor Gordon Krauss 12/3/09

Appendix Q: Bill of Materials

ME 450 Team 21 Build of Materials

Part #	Name	Description	Supplier	Supplier Part Number	Qty	Cost	Total Cost
M-1	Linear Drive Motor	NEMA 42 Stepper Motor,	Keling Technology, Inc	KL42H2150-42-8A	2	$279.00	$558.00
M-3	Linear Sensor	3 turn potientometer 10K	Digikey	SP533-10K-ND	1	$19.38	$19.38
M-4	Light Sensor	light sensor from robotics team used to measure rotational speed	FIRST 830	-	1	$0.00	$0.00
M-5	Linear Bearing	Igus Linear Bearing System	Igus	WW-10-40-10	2	$0.00	$0.00
M-6	Lazy Susan Bearing	12" diameter bearing for rotational test	McMaster Carr	18635A52	1	$31.59	$31.59
M-7	Disk Adapter	Adapts the lazy susan bearing and to the disk clamps, 3/8" AL Disk 12" Dia	ALRO	DROP	1	$40.00	$40.00
M-8	Disk Clamp	Adapter to go from the disk adapter to hold down the disk sample	Shop Supplies	-	8	$0.00	$0.00
M-9	Clamp Screws	10-32 Screws Used to clamp down the clamps	Shop Supplies	-	16	$0.00	$0.00
M-10	Disk Clamp Screws	10-32 Screws Used to hold the disk	Shop Supplies	-	16	$0.00	$0.00
M-11	Base Plate	1/2" aluminium plate 28"x24"	ALRO	DROP	1	$100.00	$100.00
M-12	Side Plates	3/8" aluminium used for side walls	ALRO	DROP	1	$50.00	$50.00
M-13	Motor Mount	mounts the pulley motor to the side plates	ALRO	DROP	1	$30.00	$30.00
M-14	Timing Pulley	Gates 5mm Pitch GT2 Timing Pulley	Motion Industries	P44-5MGT-25	2	$73.48	$146.96
M-15	Timing Belt	Power Grip GT2,	Motion Industries	5MR 1300 25	1	$58.50	$58.50
M-16	Idler Mount	a mount made for the ideler pulley to attach to the side plate	ALRO	DROP	1	$20.00	$20.00
M-17	Encoder Mount	holds the encoder stationary while the assembly spins.	ALRO	DROP	1	$5.00	$5.00
M-18	5mm Key Stock	5mmx5mm Key stock for the stepper motors	McMaster	92288A120	1	$3.49	$3.49
M-19	Table	The main table that the entire assembly will sit on	IKEA	LACK coffe-table	1	$20.00	$20.00
M-2	Stepper Motor Driver	KL-11078 Microstepping Driver	Keling Technology, Inc	KL-11078	2	$199.00	$398.00
M-20	Pinion Gear	24 teeth 1.5 PD 5/8th bore	McMaster Carr	5172T22	1	$18.23	$18.23
M-21	Follower Gear	80 teeth 1.5 PD 3/4 bore	McMaster Carr	5172T25	1	$64.16	$64.16
M-22	Idler Bearings	.75 bore 5/16 width ball bearings	McMaster Carr	60355K507	3	$6.22	$18.66
						TOTAL	$1,581.97

G-1	Normal Force Motor	NEMA 17 Bipolar Stepper Motor	Keling Technology	KL17H247-168-4B	1	$20.00	$20.00
G-2	Microstepping Driver	Microstepping Driver 1600 ppr	Keling Technology	KL-4020	1	$39.95	$39.95
G-3	Power Supply	Stepper Motor Power Supply	Keling Technology	KI-320-369.6A 110V/220V	1	$59.95	$59.95
G-4	Bronze Shaft	Hi-Strength Bearing-Grade Bronze (Alloy 544) .5" Dia, 1' length	Mcmaster Carr	8971K711	1	$16.82	$16.82
G-5	Wave Spring 1	Spring Loads 4.3 - 15 lbs	Smalley	CS050-H9	1	$0.00	$0.00
G-6	Wave Spring 2	Spring Loads 1.2 - 5 lbs	Smalley	CS050-L9	1	$0.00	$0.00
G-7	Wave Spring 3	Spring Loads 14.5 - 60 lbs	Smalley	custom	1	$0.00	$0.00
G-8	Spring	Spring Loads .183 - 1.26 lbs	Mcmaster Carr	1986K11	1	$10.15	$10.15
G-10	Ball Bearing 1	1/16" Steel Ball Bearing	Mcmaster Carr	96455K71	1	$4.34	$4.34
G-11	Ball Bearing 2	1/8" Steel Ball Bearing	Mcmaster Carr	96455K49	1	$1.98	$1.98
G-12	Ball Bearing 3	1/4" Steel Ball Bearing	Mcmaster Carr	96455K52	1	$3.11	$3.11
G-13	Encoder	E4P OEM Mini Optical Kit Encoder	US Digital	E4P-360-197-D-D-D-3	1	$32.25	$32.25
G-14	Encoder Cable	4-Pin Micro / Unterminated 4-Wire Discrete Cable	US Digital	CA-MIC4-W4-NC-2	1	$7.30	$7.30
G-15	shoulder bolts	3/16 Shoulder bolt	Mcmaster Carr	93996A718	2	$3.99	$7.98
G-16	Steel Threaded rod	Grade B-8 18-8 Stainless Steel 1/2"-20 Thread, 1' Length	Mcmaster Carr	91187A560	1	$8.62	$8.62
						TOTAL	$203.83

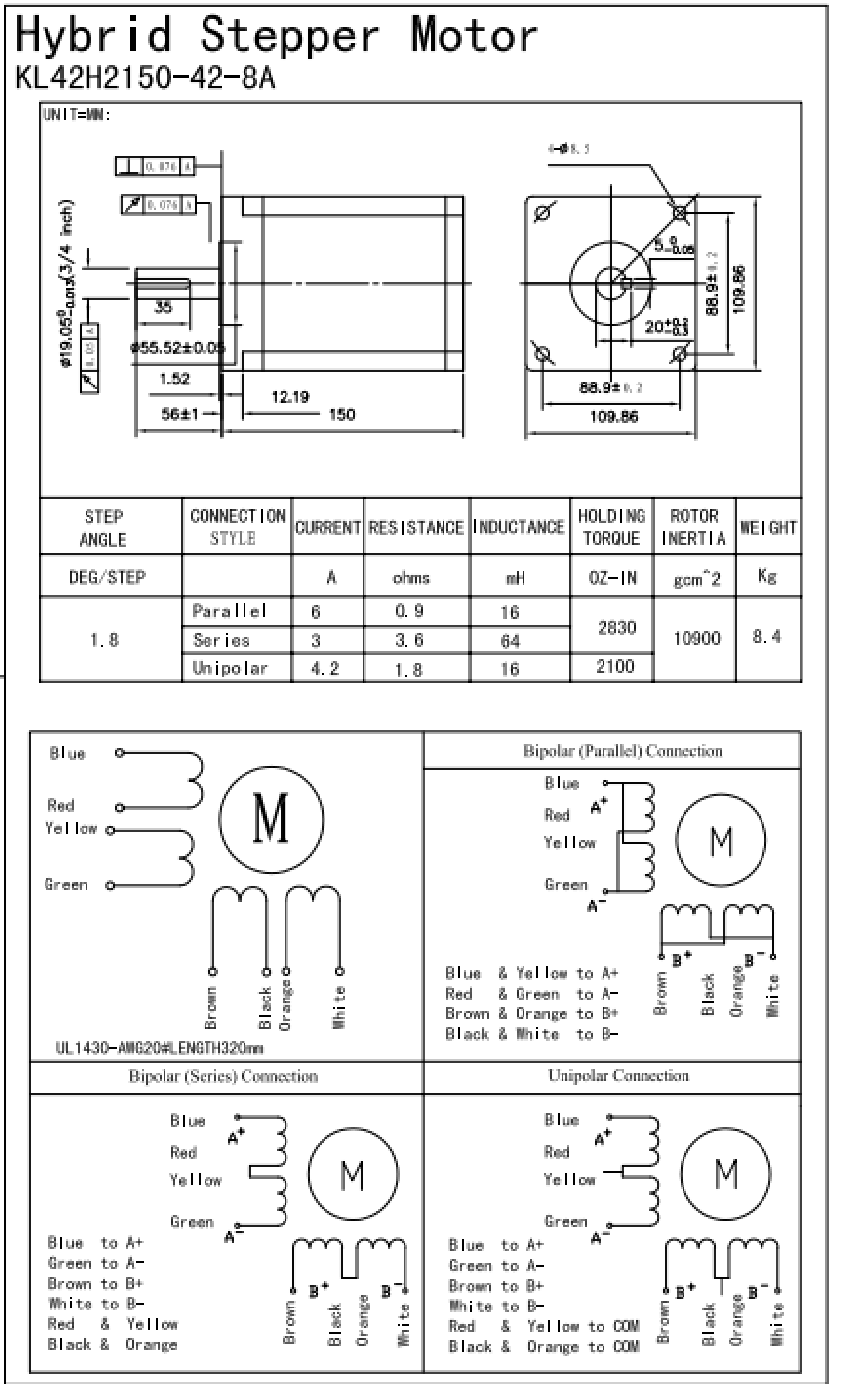
E-1	Strain Gauges	Piezoelectric gauges half bridge	Micron Instruments	SS-080-050-1000PB-SSGH	5	46.49	441
E-2	Mini-board	Dual Mini Board with 213 Holes	Radio Shack	276-148	1	$1.99	$1.99
E-3	Op-amp	LM741CN Operational Amplifier (8-Pin Dip)	Radio Shack	276-007	3	$0.99	$2.97
E-4		15-Position HD Female Solder D-Sub Connector	Radio Shack	276-1502	1	$1.99	$1.99
E-5		15-Position HD Male Solder D-Sub Connector	Radio Shack	276-1501	1	$1.99	$1.99
E-6		15-Position Shielded Metal D-Sub Connector Hood	Radio Shack	276-1508	1	$2.99	$2.99
	Dc adapter	9V 1A 2.1mm DC Adapter	adafruit industries		1	$7.99	$7.99
		Adjustable breadboard power supply - v1.0	adafruit industries		1	$15.00	$15.00
E-7	DAQ SYSTEM	USB DAQ	National Instruments	USB-6009	1	$250.00	$250.00
						TOTAL	$725.92

TOTAL COST	$2,511.72

Key	
M	Motion Systems
G	Pin Gantry
T	Temp. and Humidity System
E	Force Measurement

Appendix R: Motor Diagrams

Figure R1: Keling NEMA 42 Motor Diagram



STEP ANGLE	CONNECTION STYLE	CURRENT	RESISTANCE	INDUCTANCE	HOLDING TORQUE	ROTOR INERTIA	WEIGHT
DEG/STEP		A	ohms	mH	OZ-IN	gcm^2	Kg
1.8	Parallel	6	0.9	16	2830	10900	8.4
	Series	3	3.6	64			
	Unipolar	4.2	1.8	16	2100		

Figure R2: Keling NEMA 42 Driver and Power Supply

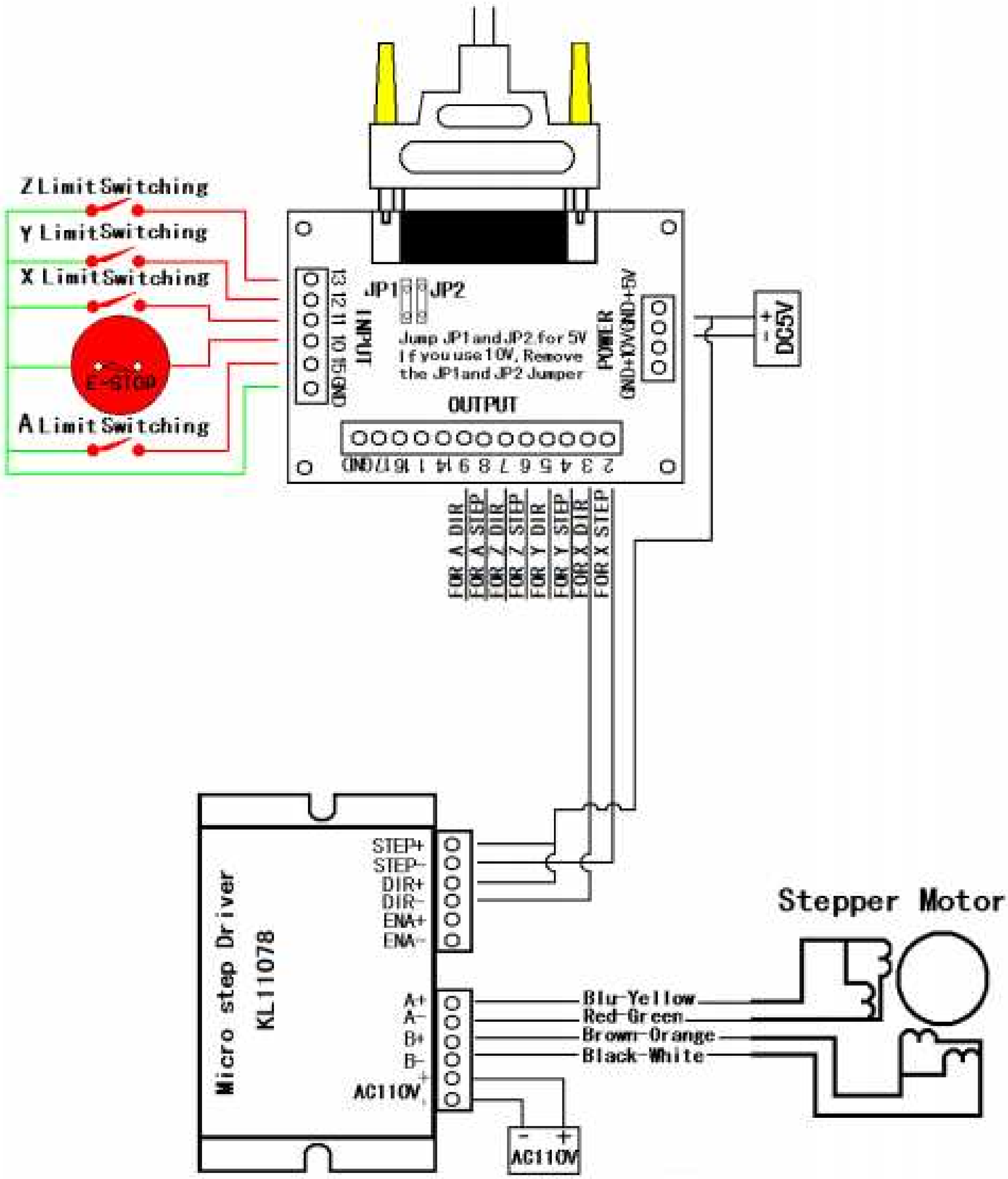
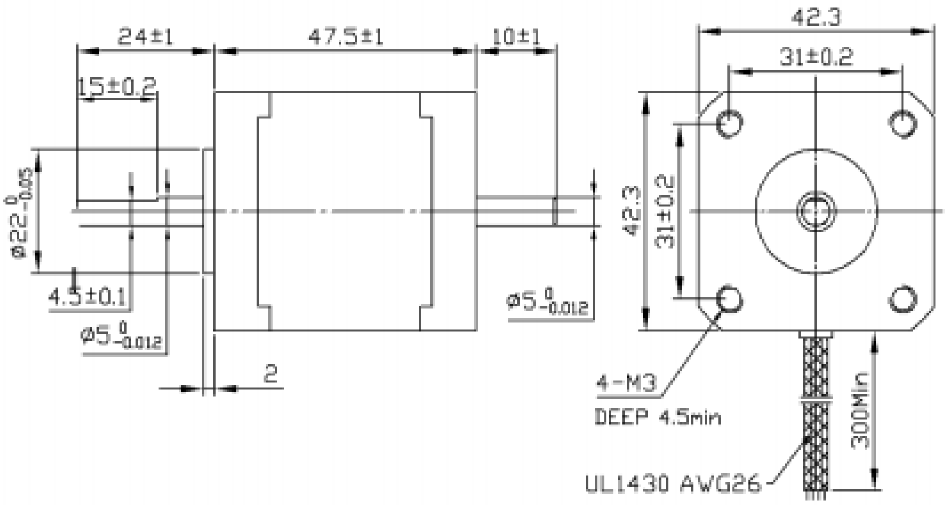


Figure R3: Keling NEMA 17 Motor Diagram

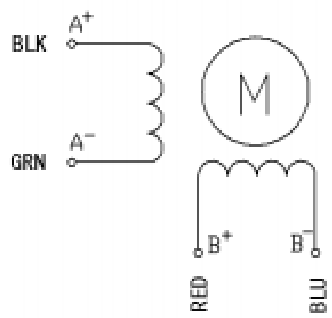
Hybrid Stepper Motor

KL17H247-168-4B

UNIT=MM

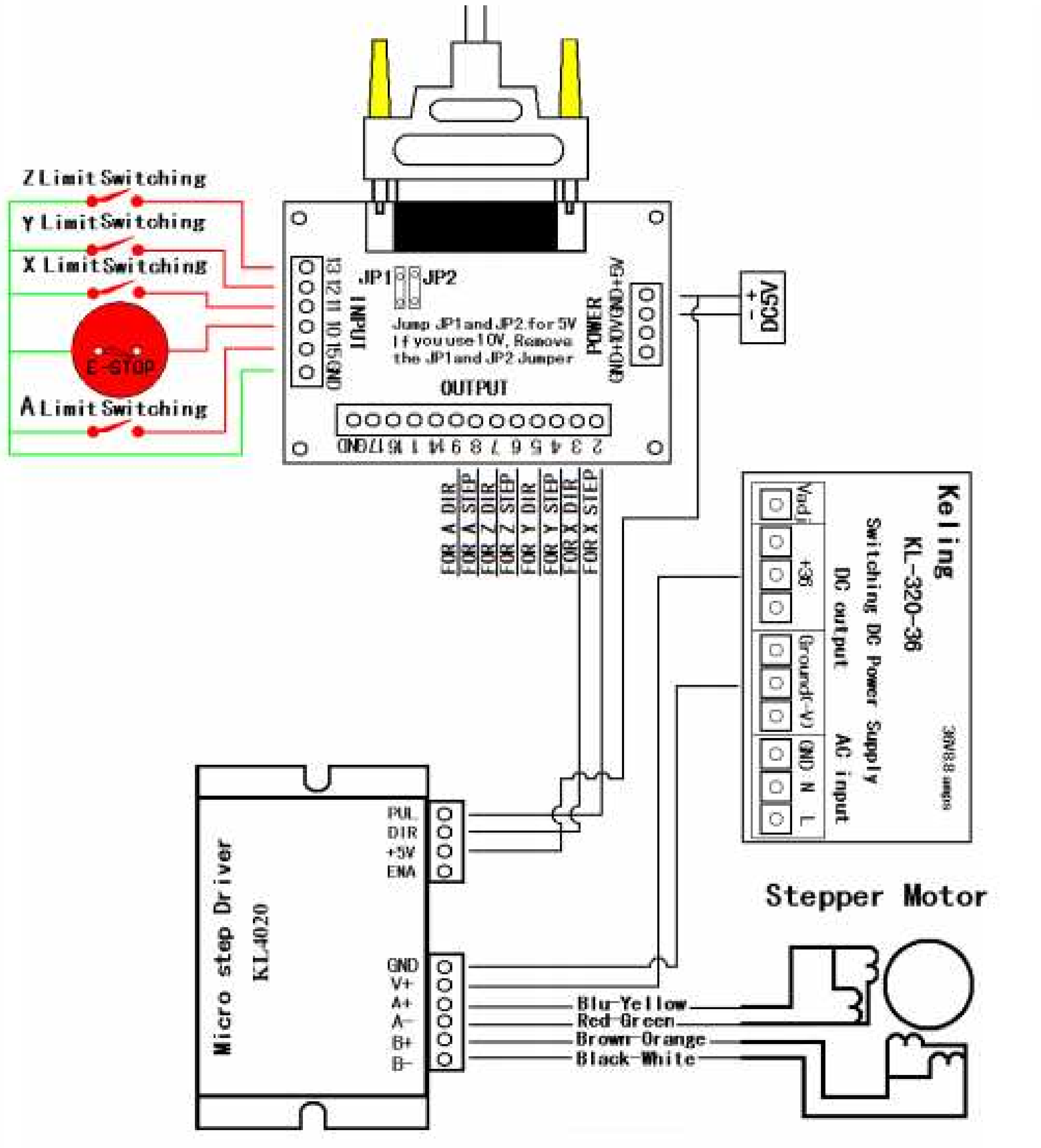


MODEL	PHASE	STEP PHASE	RATED VOLTAGE	CURRENT /PHASE	RESISTANCE /PHASE	INDUCTANCE /PHASE	HOLDING TORQUE	ROTOR INERTIA	WEIGHT
		DEG/STEP	V	A	ohms	mH	OZ-IN	gcm^2	Kg
	2	1.8	2.8	1.68	1.65	2.8	61.6	68	0.35

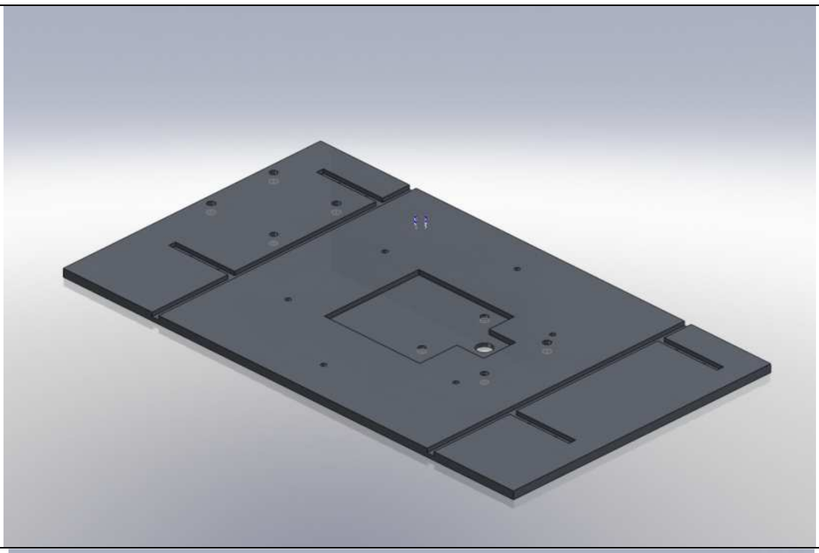
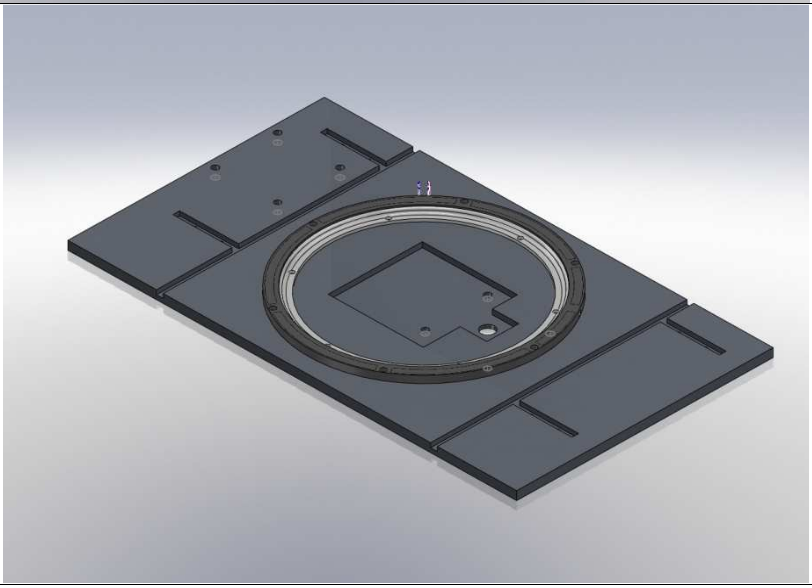


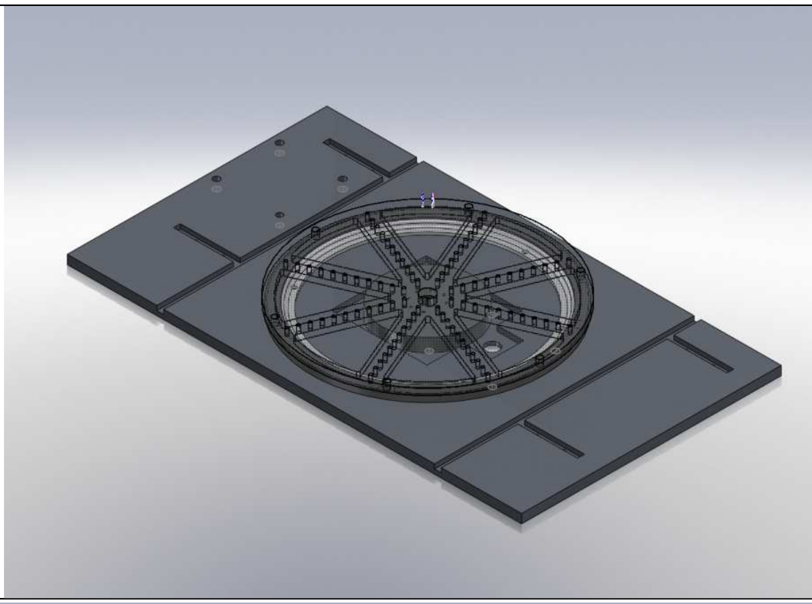
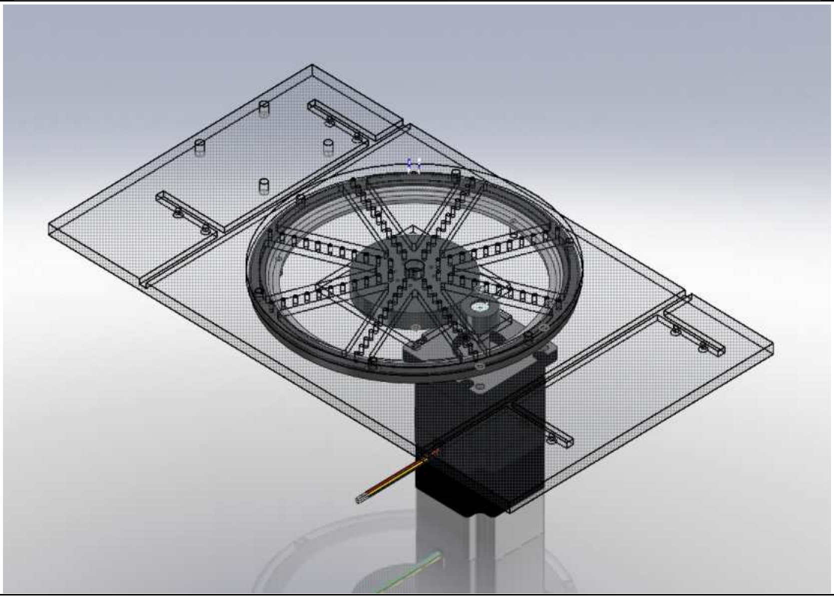
Black to A+
Green to A−
Red to B+
Blue to B−

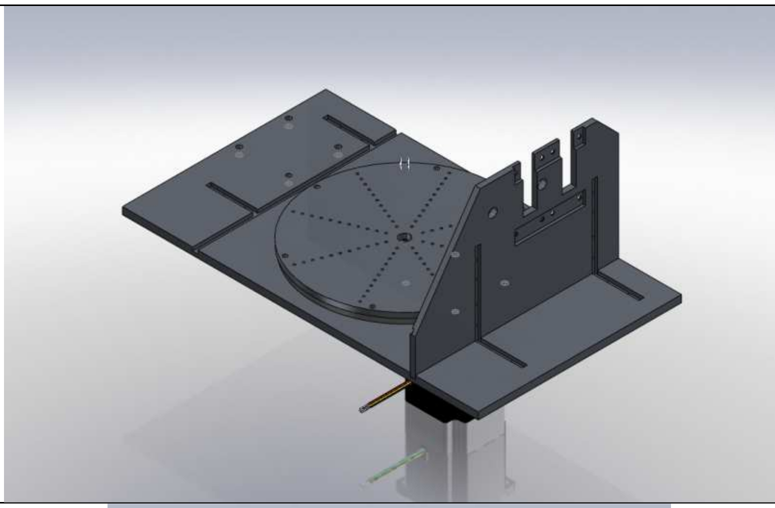
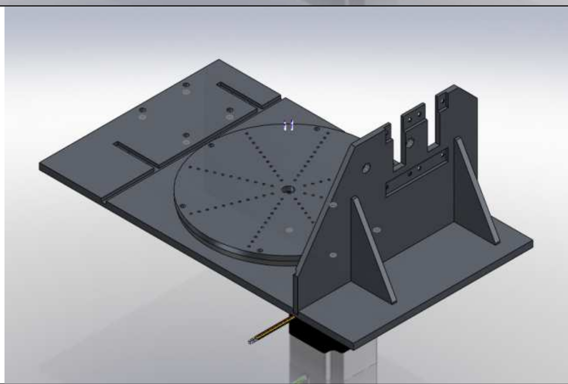
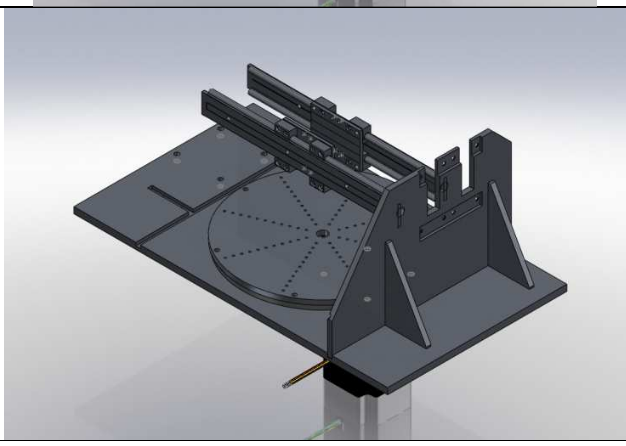
Figure R4: Keling NEMA 17 Driver and Power Supply

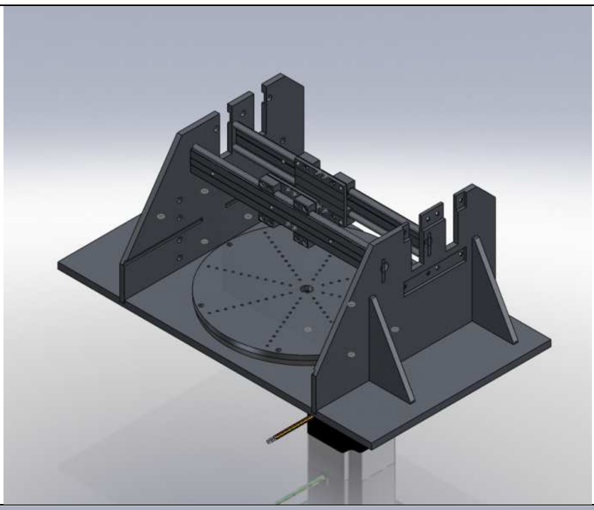
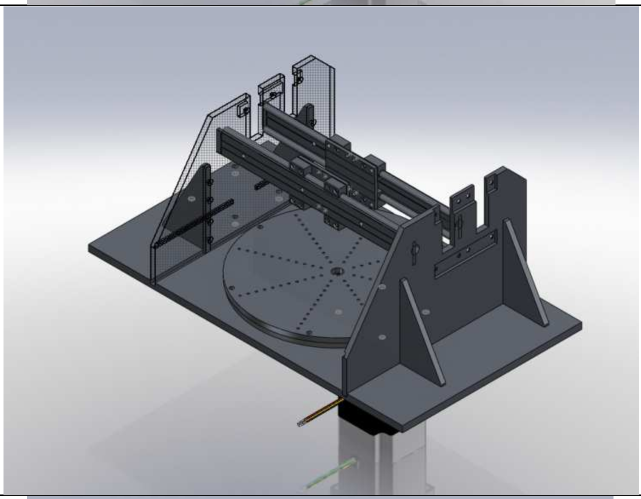
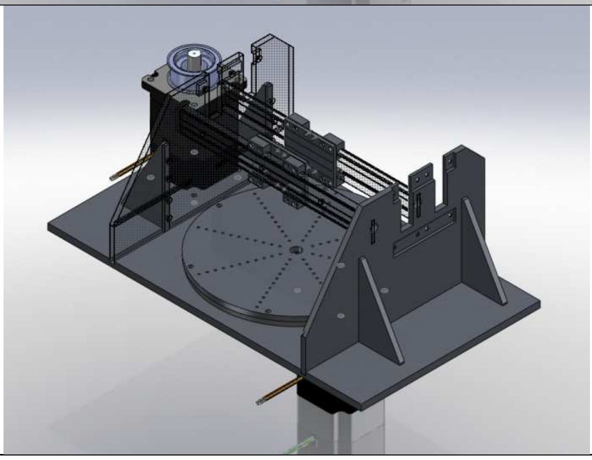


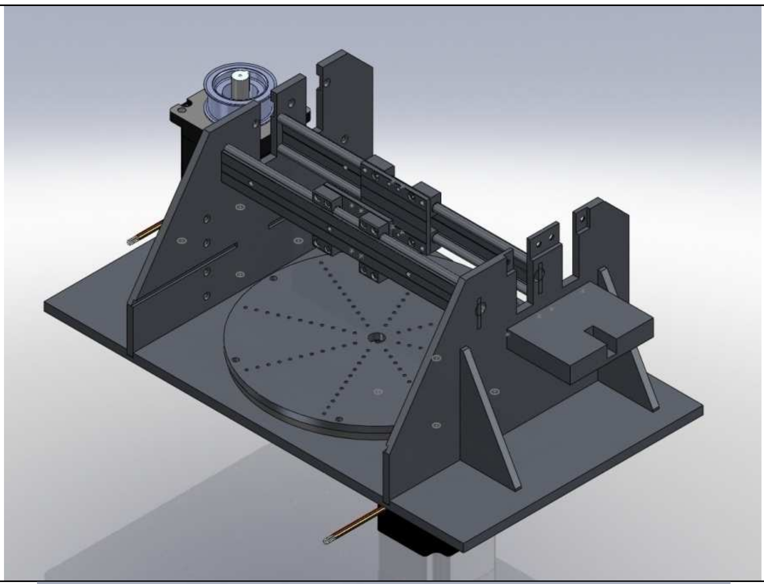
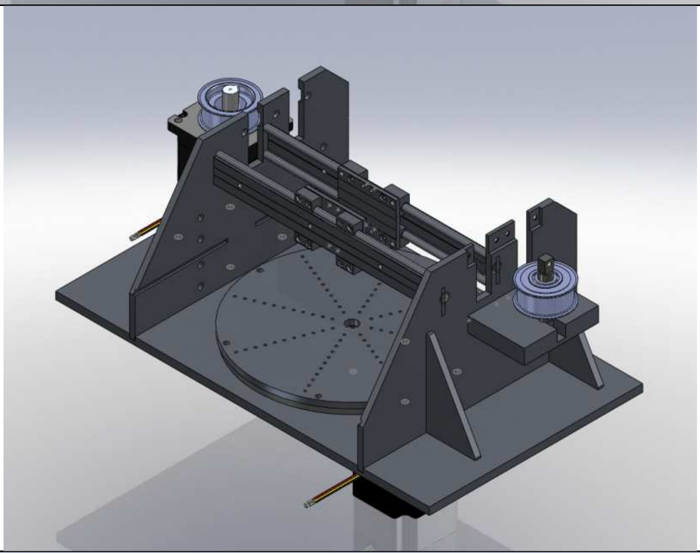
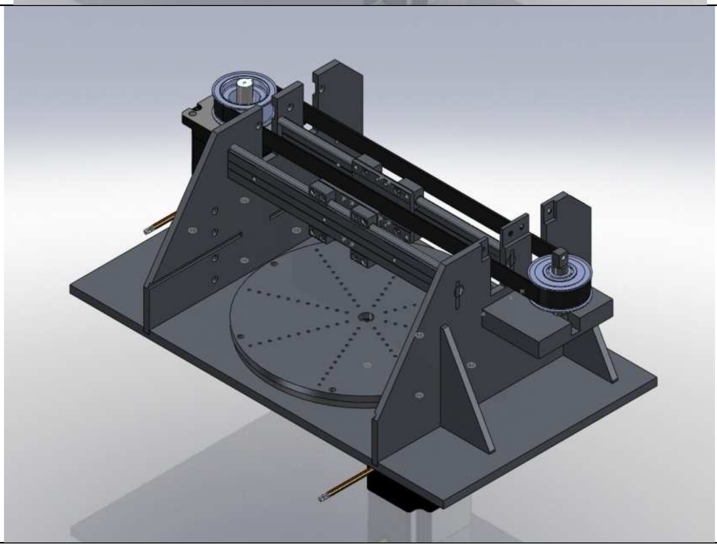
Appendix S: Step by Step Assembly Instructions

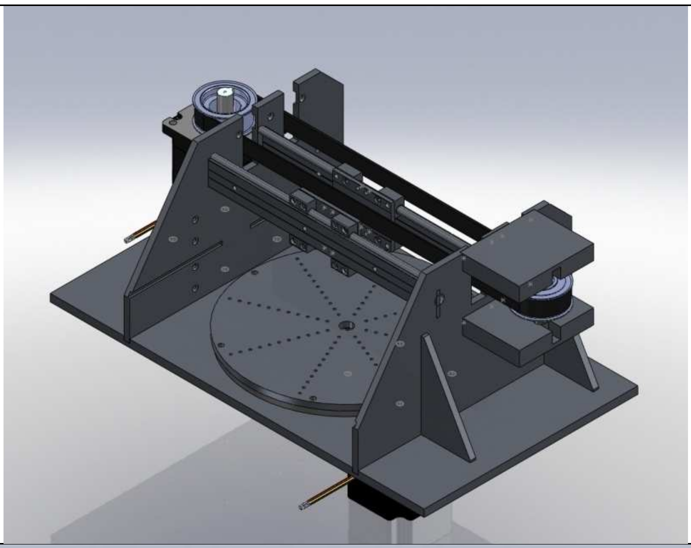
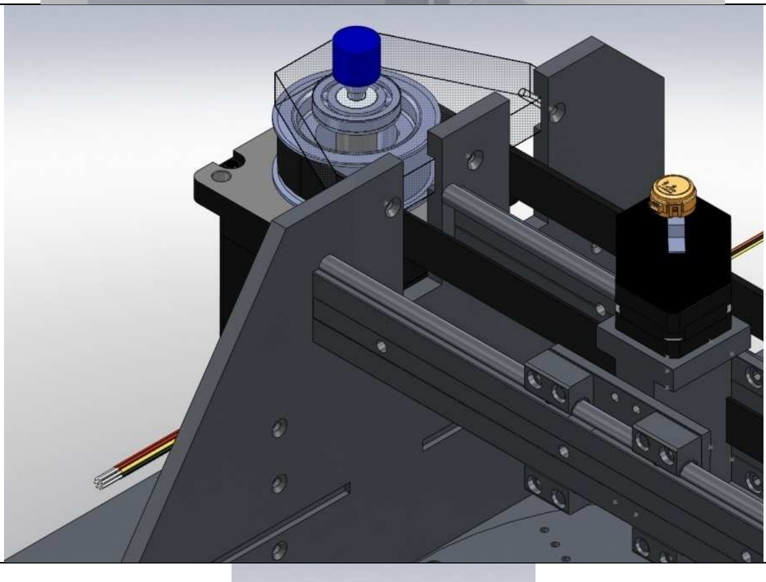
Step Number	Description	Picture
1	Start with the base with slots and holes machined for everything.	
2	Install the 10” diameter bearing with 6 ¾” long ¼”-20 machine screws. Apply loctite so they do not come loose.	

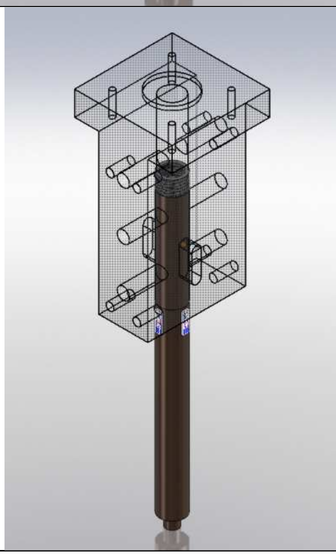
3	Screw the follower gear onto the sample mount with 8 1” long ¼”-20 machine screws. Then screw the sample mount onto the large diameter bearing with 6 ½” long 5/16-18 machine screws. Use loctite.	
4	Install the NEMA 42 motor with drive gear, below the base plate using 5/16” threaded rod. Use nuts to properly hang the motor level. Check level and alignment. Grease gears using lithium grease.	

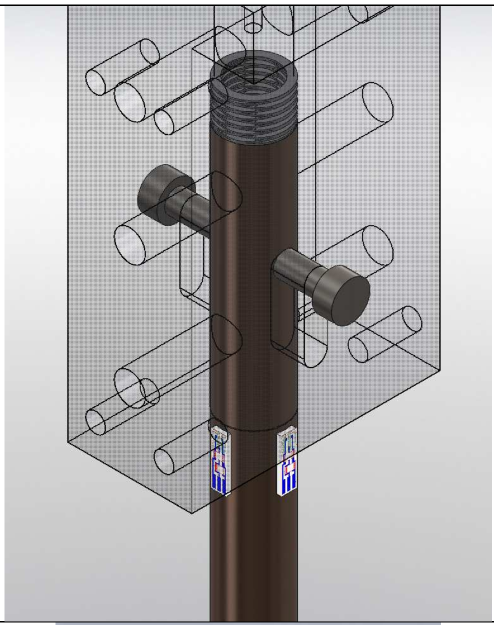
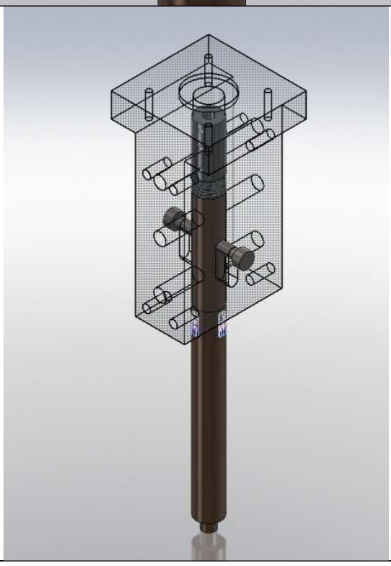
5	Slide right side into place	
6	Attach 2 side reinforcers with using 12 ½" long 10-32 screws.	
7	Slide the linear sliders onto the rails, and press the rails into the slots on the side.	

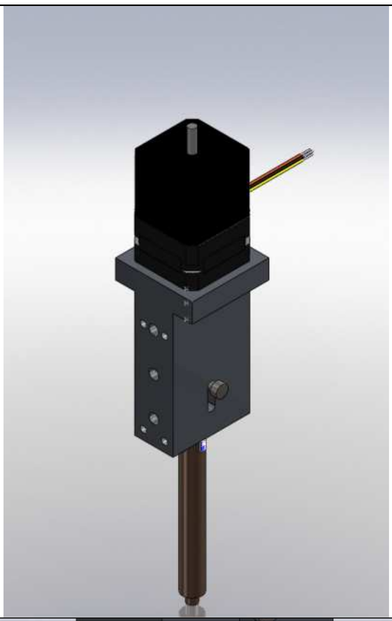
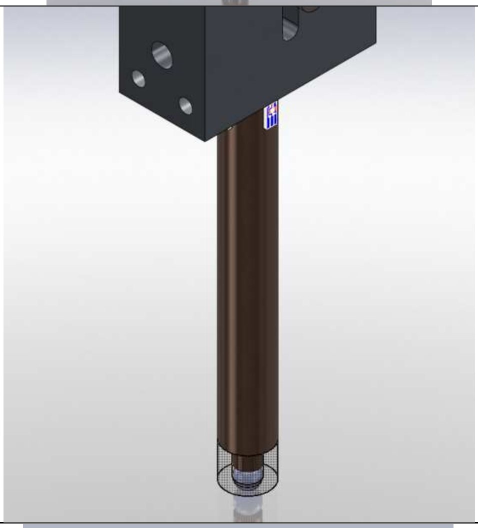
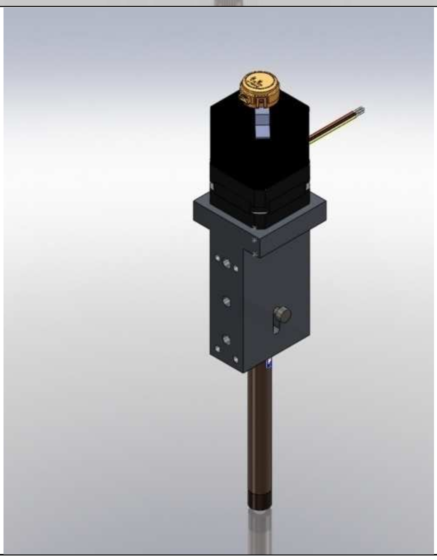
8	Slide the other side into place and push the other end of the rails into place.	
9	Bolt the other side down using the side reinforcers. Again use ½" long 10-32 bolts.	
10	Push drive pulley system into approximate place and mount the motor using 5/16" threaded rod. Use nuts to keep it in place.	

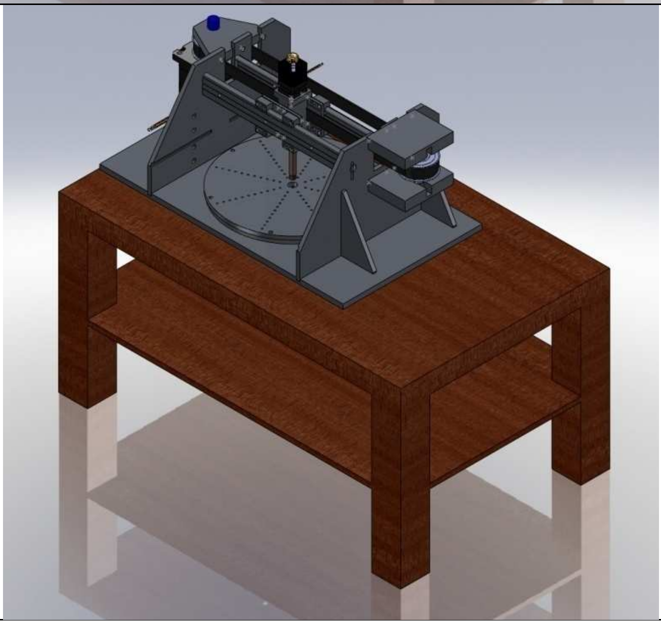
11	Mount bottom of idler mount using 4 1" long ¼"-20 bolts.	
12	Press fit the bearing into the idler pulley. Slide it onto the idler shaft.	
13	Slide belt around the 2 pulleys.	

14	Mount the top idler mount. Place idler tensioner bolts into place(3/8"-24 2"long).	
15	Press bearing into motor mount plate, then slide over motor shaft and mount with 4 1" long ¼-20 screws. Mount potentiometer into place.	
16	Mount the three strain gauges using proper techniques from appendix L.	

17	Select a spring to utilize for a test and set on top of the shaft.	
18	Slide the shaft assembly up into the pin gantry chassis. Do not hit the strain gauges.	

19	Insert and tighten down the shoulder bolts through the slots in the chassis to the holes in the shaft.	
20	Insert 3 1/8" long 4-40 set screws onto the shaft then thread the shaft down into the pin gantry chassis.	

21	Install the motor using 4 20mm long M3 bolts with washers.	
22	Set ball into ball holder and thread onto end of shaft. Tighten with wrench.	
23	Install the encoder mount and encoder onto the end of the shaft using 2-56 bolts on the endcoder, and M3 bolts to attach the mount to the motor.	

24	Follow the instructions to build the ikea coffee table.	
25	Cut a hole in the coffee table to clearance for the motor. Mount the pin gantry which clamps the belt to the linear slider. Mount the complete tribometer with three bolts and rubber feet to the table.	
26	Mount the motor controllers to the bottom shelf. Wire the motors. See appendix R for wiring instructions.	

Appendix T: Electric Resistance Diagram

Figure T1: Electric Resistance Wiring Diagram [25]

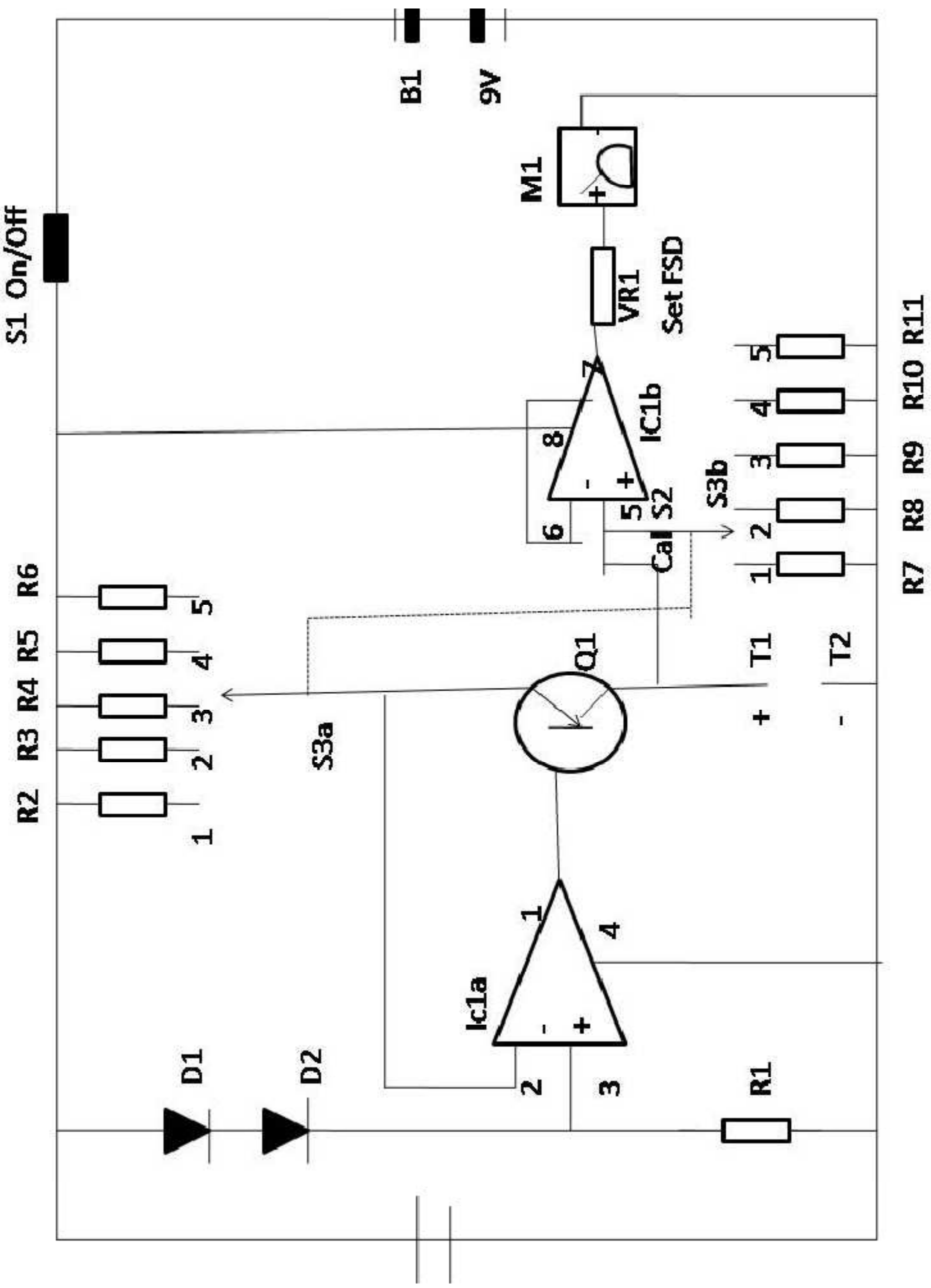


Table T1: Wiring Components

R1	1K8		IC1	CA3240E
R2	560R		Q1	BC179
R3	5K6		D1	1N4148
R4	56K		D2	1N4148
R5	560K			
R6	1M6		C1	100nF Polyester
R7	1K0			
R8	10K		S1	Push to make Red
R9	100K		S2	Push to make Black
R10	1M0		S3	2 pole 6 way
R11	10M0		M1	100µA FSD
VR1	47K LIN		B1	PP3 or equiv.
All Resistors	1% Metal Film		T1	4mm Socket Red
			T2	4mm Socket Black

S2 Range	FSD Reading
Pos 1	1K
Pos 2	10K
Pos 3	100K
Pos 4	1M
Pos 5	10M

Appendix U: DesignSafe

DesignSafe was used to look at the risks of the completed tribometer prototype, and the summary from the design safe program can be seen on the next page. Our group discussed many safety issues including the weight, the fatigue of materials being tested, pinch points, and computer/electrical errors. The weight of the system will be a big problem for moving around as it will weigh around 50 pounds, so two people should work together to move it. There may also be software errors which could cause unexpected starts, or the machine to go crazy and run off the limits set in it. When this happens, there should be a secondary safety box protecting the user from any crashes that may occur.

The safety box can also be used to protect the user from test samples that can not withstand the high rotational rates, and fatigue during testing. We recommend the user wear safety glasses to protect their eyes. The other nice thing about the safety box is that it blocks all of the pinch points. We will be sure to put up warning signs to clearly mark all pinch points on the system, and the user should be educated to stay away from those points until a power lock out procedure has occurred. There may also be other power problems including wiring and over current issues, and we will have to make sure that we take our time during the electrical wiring process of the prototype.

The biggest take away from design safe was the need for an overlying safety box that can contain the system in on itself in case of catastrophic failure. The failure could come from any number of sources, but as long as it is contained within the safety box, then the user will be safe.

tribometer - designsafe

File Edit View Reports Library Help

Identify Hazards | Assess and Reduce Risk

	Item Id	User	Task	Hazard Category	Hazard	Cause/Failure Mode
1	1-1-1	All Users	All Tasks	mechanical	pinch point	a pulley system is in basically the wide open.

	Task	Hazard Category	Hazard	Cause/Failure Mode	Severity	Exposure	Probability	Risk Level	Reduce Risk	Severity	Exposure	Probability	Risk Level
1	All Tasks	mechanical	pinch point	a pulley system is in basically the wide open.	Serious	Frequent	Probable	High	fixed enclosures / barriers, warning sign(s), video	Minimal	Remote	Unlikely	Low
2	All Tasks	mechanical	unexpected start	a programming error, or a spuratic test could cause an unexpected start. should be powered down before opening case.	Slight	Occasional	Unlikely	Moderate	audible alarm or sounds, lock out / tag out, instruction manuals	Slight	Remote	Possible	Moderate
3	All Tasks	mechanical	fatigue	the device will see a lot of cyclic loading on all parts because it does long endurance rotational and linear reciprocating wear tests	Slight	Remote	Possible	Moderate	safety glasses, fixed enclosures / barriers	Slight	Remote	Possible	Moderate
4	All Tasks	mechanical	break up during operation	the sample or pin could break up during a test and fly off.	Slight	Remote	Possible	Moderate	substitute less hazardous material / methods, fixed enclosures / barriers	Slight	Remote	Unlikely	Low
5	All Tasks	electrical / electronic	improper wiring	this is a prototype and it will probably have poor wiring	Minimal	Remote	Unlikely	Low	warning sign(s), gloves	Minimal	None	Negligible	Low
6	All Tasks	electrical / electronic	water / wet locations	lubricant and water could splash around	Slight	Remote	Possible	Moderate	gloves, warning sign(s), fixed enclosures / barriers	Slight	Occasional	Unlikely	Moderate
7	All Tasks	electrical / electronic	unexpected start up / motion	programing could cause an unknown startup.	Slight	Occasional	Possible	Moderate	other design change, audible alarm or sounds, lock out / tag out	Slight	Remote	Possible	Moderate
8	All Tasks	electrical / electronic	software errors	this is a prototype, so there will be programing errors, and some could be serious.	Slight	Occasional	Possible	Moderate	warning sign(s), special procedures, other design change	Serious	Remote	Negligible	Low
9	All Tasks	electrical / electronic	overvoltage /overcurrent	we are using some non-production parts, and custom circuits which could be built wrong	Minimal	Remote	Negligible	Low	prevent energy buildup, warning label(s)	Slight	None	Negligible	Low
10	All Tasks	ergonomics / human factors	excessive force / exertion	to tighten the belt, or the ball bearing, it might take a lot of force. also lifting and moving the test device	Minimal	Frequent	Probable	High	warning sign(s), standard procedures, video	Slight	Occasional	Unlikely	Moderate
11	All Tasks	ergonomics / human factors	duration	during long tests, users may get bored and make mistakes due to long durations	Minimal	Occasional	Possible	Moderate	audible alarm or sounds, standard procedures	Slight	Occasional	Possible	Moderate
12	All Tasks	fire and explosions	static electricity	not sure, but the cyclic movement could cause static build up depending on test materials	Slight	Occasional	Possible	Moderate	prevent energy buildup, warning sign(s), standard procedures	Serious	Occasional	Unlikely	Moderate
13	All Tasks	noise / vibration	fatigue / material strength	again the test material will undergo fatigue and have the potiental to break.	Slight	Occasional	Probable	High	fixed enclosures / barriers, prevent energy release, standard procedures	Serious	Occasional	Possible	High
14	All Tasks	material handling	excessive weight	the tribometer will weight over 50lbs, and require 2 people to lift.	Slight	Frequent	Probable	High	special procedures, video	Slight	Remote	Possible	Moderate
15	All Tasks	environmental / industrial hygiene	corrosion	the tribometer will operate in tough enviroments which could lead to corrosion	Minimal	Frequent	Probable	High	fixed enclosures / barriers, safety glasses	Slight	Occasional	Unlikely	Moderate

Hazard Information

Hazard Category: mechanical

Hazard: pinch point

Cause/Failure Mode: a pulley system is in basically the wide open.