Twist Compression and Four-Ball Test Device Redesign

ME 450, Winter 2010, Team 6
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ABSTRACT

In countless industries and applications, lubricants and their combinations with additives are vitally important to machine performance, functionality, and safety. Two industrial tests that are used to study lubricants in various environments and scenarios are the Four-Ball Test and the Twist Compression Test. Each test involves applying a normal load to a lubricated surface and turning a tool against the surface at varying speeds to measure lubricant response and conditions at seizure. This project serves to combine the two tests into a single, safe, and interchangeable machine with the speed capability of a Four-Ball Test and the loading capability of a Twist Compression Test. This design will also incorporate measurement of the environmental conditions around the lubricant, as well as a robust data acquisition system with the capability to control turning speed, lubricant temperature, and test volume relative humidity.

EXECUTIVE SUMMARY

In both industry and research, tribologists use a variety of different machines to simulate different lubrication applications. The two tests being focused on in this project are the Four-Ball and Twist Compression tests. Our project sponsor, Professor Gordon Krauss, has requested the development of a single, cost-effective machine that can run each test interchangeably. The capabilities of our machine stem from Tribsys Inc. industry standards combined with a half to double extension of Four-Ball ASTM standards D2266, D2596, D2783, D4172, and D5183 as desired by our sponsor. Alongside desired environmental control capabilities of 0-150ºC and 0-100% humidity, our machine needs to run at speeds of 2 to 3600 rpm with normal loads extending up to 100kN. Data acquisition capabilities are required so that monitored rotational speeds, normal loads and resultant forces can be used to calculate coefficients of friends and other parameters for various tests.

Beginning with a very detailed functional decomposition, we brainstormed and researched numerous concepts to perform our functions. Pugh charts were used to perform rigorous comparisons and concept selection for each function. Keeping in mind our time constraints, our alpha design focused on only specific changes to the existing prototype, encompassing improvements in functionality, safety and data acquisition. Several key engineering concerns arose during concept compilation and inspection. These included the gear ratios, lateral force beam and beam bracket bending, compressive blocks stress, specimen cup and specimen cup teeth stress, heat transfer from the heating coil, and driveshaft and alignment plate deflection. Calculations of the appropriate gear ratios, bending, shear, and compressive stresses, and amount of conductive heat transfer allowed us to approve the majority of our concepts, while a few others required changes before our final design. Due to the cost of the required gear ratios for our load and speed ranges, we were forced to change the power transmission system in our final design from a gearbox to an open gear transmission. We also adjusted our gear ratios slightly due to availability and long manufacturing time. This reduced the maximum normal loads for the Twist Compression and Four-Ball tests to 85,000 N and 13,500 N, respectively, and also reduced the maximum driveshaft speed for the Four-Ball test to 2100 rpm. Although these numbers do not meet our initial engineering specifications, they still capture the entire range of speeds and loads from the ASTM and industry standards for each test. Other concept changes to our final design included using a heating cartridge instead of a heating coil to control the temperature of the lubricant and implementing a tachometer instead of an optical encoder for shaft speed measurement. These changes were mainly driven by improved or equal functionality and lower cost of the replacement part. A combination of data acquisition instrumentation and hardware was also chosen for accurate and precise load, speed, temperature, and humidity readings.

Over the course of the purchasing, manufacturing and assembly phases of our redesign, we adjusted numerous aspects of our Final Design as it evolved into our current prototype. Our goals in any adjustment were to evaluate what can be removed with the least impact on capability. Mechanically, our prototype is in good working order, as is the software programming on the data evaluation end. Electronically, the data acquisition has proven to be the most troublesome. Though tachometer measurement and motor control work fine, climate control has been scaled back to simply humidity logging, temperature measurement and interface cooling capabilities and strain measurement capabilities are limited to the Lateral Force Beams. Where our prototype stands, we still satisfy our sponsor's ranking of the most important prototype functions as well as keeping consistent with what we, as a team, feel is most important to provide for this project as it moves forward, potentially still in this class. For everything that we either couldn't complete for hardware reasons, for time constrains, or for interference with another aspect of the prototype, we provide ample information and reasoning behind the choices we made.

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INTRODUCTION

In the field of Tribology, the science and technology of interacting surfaces in relative motion, there are many test procedures used to study lubrication and wear properties. We have been requested to focus on two standard tests: the Twist Compression Test (TCT) and Four-Ball Test. Currently, two separate machines are used to perform the tests in industry. Furthermore, both machines are extremely expensive. This poses a significant problem since little funding is available for University of Michigan professor Gordon Krauss, the sponsor of this project. With this limited funding, our goal is to spend it as effectively as possible. It was requested of a previous mechanical engineering student group to design an efficient and inexpensive machine that incorporates both the Four-Ball test and the TCT. The team developed a preliminary prototype but it is inoperable at this time due to safety concerns and lack of completeness. The student team has since moved on and the project has been passed on for re-design and enhancement. The goal of this project is to re-design, prototype, and test a machine that will perform all standard test requirements and in some cases outperform current standards in a safe and cost effective manner. If this project is completed and meets the required specifications, this apparatus could be used in the industry.

Twist Compression Test

The Twist Compression Test is used to measure friction and evaluate lubricants and die materials for application in metal forming processes [9]. Shown in Fig. 1 below, a rotating hollow cylinder has an applied pressure and is forced on to a plate that stays stationary. The friction that is produced at the contact area of these two pieces produces a lateral force. During the test the stationary plate is submersed in the lubricant and data is taken of both the downward and lateral forces. This test runs at 2-30 rpm and up to a normal load of 100,000 N.

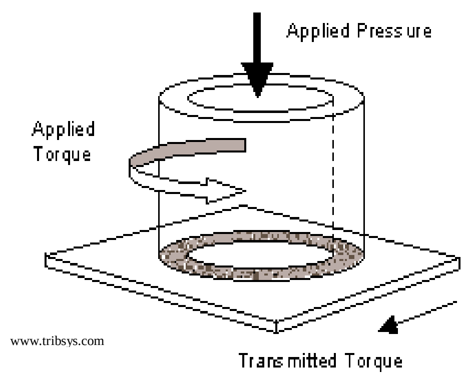


Figure 1: Twist Compression Test

Four-Ball Test

The Four-Ball Test is used to simulate bearing applications. In this test, four balls are placed in a tetrahedral geometry. The three balls on the bottom of this setup are stationary, while the top ball applies the load and spins at various set speeds. This is shown below in Fig. 2, p. 10. The test area is submersed in the lubricant during the process. Unlike the Twist Compression Test, the Four-Ball Test runs at 300-3600 rpm and 30-16,000 N loads.

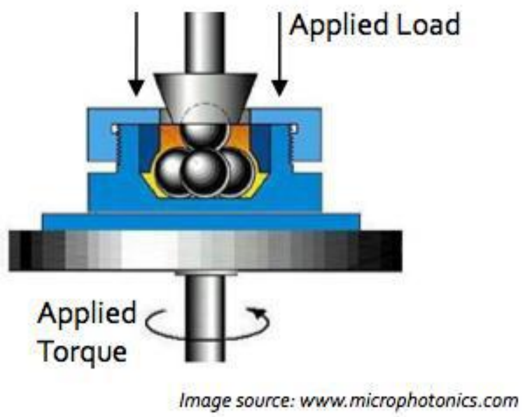


Figure 2: Four-Ball Test

ENGINEERING SPECIFICATIONS

To begin determining the engineering specifications for the project, an initial meeting was organized with our sponsor, Professor Krauss. In this meeting, Professor Krauss shared his concerns with the existing model, requirements and his desires for the redesigned model. By combining our customer's requirements and desires with the ASTM standards and technical benchmarking results from the literature search, non –safety engineering specifications were established. The customer's safety concerns with the existing model were then broken down into quantifiable engineering parameters, and further literature search allowed these parameters to be translated into additional engineering specifications. The level of importance of each requirement and specification was also gauged for later use in the formation of the Quality Function Deployment (QFD) diagram. A complete table of the engineering specifications for the redesigned model can be seen in Table 1, p. 12.

Customer Requirements

The customer's main concerns with the existing model focused on its unsafe operational state. Specifically, the customer was concerned with the stability of the shaft that turns the interchangeable test fixture (hereafter referred to as the shaft), the number of moving parts that are covered from contact, and the ability of these covers to withstand catastrophic failures. In the current model, the shaft experiences deflections and oscillation due to the lack of constraints holding it in place and its pinned, multi-shaft design. These effects are very undesirable because they inhibit torque transmission along the shaft and could be extremely dangerous if the deflections become large enough to cause the shaft to yield or fracture. In this case, the current model lacks any safety covers around the shaft that might control the damage. Safety covers are also missing from the gear and motor setup, and the safety covers that are present surrounding the test area seem to lack the required fracture toughness to reduce the effects of failure. Besides containing failure, adequate safety covers over these moving parts would also protect foreign objects from interrupting the operation of the model.

The customer also had both qualitative requirements and quantitative desires for the redesigned model. The qualitative requirements included meeting the industry-accepted testing standards for both the four-ball and twist compression tests in the areas of shaft rotational speed, applied normal load, test fixture geometry, and test area temperature control and measurement. In addition, the customer required that the redesigned model be designed at low cost and be able to measure humidity in the test area. With the hope of being able to outperform industrial machines, the customer also had desires for both the redesigned model and the specific four-ball and twist compression tests. The redesigned model should be able to measure and control test surface temperatures of 0-150 ºC, measure and control relative humidity from 0-100% at 10% intervals, and cost a maximum of 10% of the cost of an industrial four-ball or twist compression tester. The four-ball test should be able to operate at rotational speeds up to 3600 rpm and normal forces of 0.5-2 times the accepted testing standards, while the twist compression test should be able to operate at rotational speeds up to 30 rpm and normal forces up to 100 kN.

Establishing Testing Standards and Benchmarking

To translate the qualitative customer requirements into quantitative engineering targets, a literature search was performed to obtain the industry-accepted testing standards for the two tests. As detailed in the Information Sources section (p. 78), four ASTM standards that gave shaft speed, normal load, and test surface temperature ranges were found relating to the four-ball test. Two industrial four-ball test machines, one manufactured by Koehler Instruments and one by Falex Corporations, were also studied to obtain their normal load and shaft speed capabilities. Because the twist compression test is relatively new to the tribology industry, no complete and published ASTM standards were available for study. Instead, because Tribsys is the only company that produces twist compression test machines, the shaft speed, normal load, and test surface temperature capabilities of their machines were taken to be the testing standards of the twist compression test. A table of the test standard and benchmarking results can be seen in Table 2, p. 17.

Safety Quantification

Each of the customer's safety concerns with the existing model was looked at individually to assess the engineering principles behind each issue. The main reason for the instability of the shaft on the existing model is the amount of beam deflection that it undergoes from the force of the motor. This beam deflection will be minimized in the redesigned model, with a target specification to follow, after an analysis of the forces and moments applied to the shaft from the gear and motor setup. Concerning the moving components of the existing model, two quantifiable specifications were created. Because the moving components are likely to transfer their high temperatures to the external parts of the existing model, the redesigned model will have external temperatures of less than 38 ºC (100 ºF). This temperature will allow for a safe enough time period of human contact with the external components to avoid burning. Also, because of the dangers of failure of the moving components and their interruption while operating, 100% of the moving parts of the redesigned model will have safety coverings. These safety coverings will also require high fracture toughness so that they contain any dangerous shrapnel in case of failure. Target specifications for fracture toughness will follow after force analyses of each component after potential failure are undergone.

Target Engineering Specifications

Based on our sponsor's requirements, literature search, benchmarking, and safety quantification, we have developed a complete list of engineering specifications. Some specifications are specific to only the Four-ball test or the TCT, while others are indicative of the device itself. The complete list of specification targets can be seen in Table 1, p. 12.

The target shaft speed for the Four-ball test is based on ASTM standard speeds ranging from 600 to 1760 rpm [1-5], and the fact that our customer desires that the device be capable of producing speeds that are half to double those of the standard tests. Therefore, the speeds range from 300 to 3600 rpm fort the Four-ball test. For the TCT, the shaft speed target of 2 to 30 rpm was set by standard tests used by Tribsys, Inc. and our customer requirements [9].

Similar to shaft speed target specifications, normal force target specifications for the Four-ball test were obtained from ASTM standard normal forces ranging from .059 to 7.84 kN [1-5] and our customer requirement of producing half to double the normal forces used in standard tests. Therefore, our target for normal force in the Four-ball test is from 0.03 to 16 kN. For the TCT, the normal force target of up to 100 kN was set by standard tests used by Tribsys, Inc. and our customer requirements [9].

Temperature targets to be used for both tests were predominantly set by our sponsor, who desires to be able to test lubricants at a variety of different temperatures, simulating many applications. This desired temperature range is between 0 and 150 °C. Similarly, our sponsor primarily set our humidity specification goal of being able to control the test conditions from 0 to 100 % relative humidity by intervals of 10 %.

Geometries of the annulus and balls used in the TCT and Four-ball test respectively were set by ASTM [1-5] and industry standards [9, 20]. The annulus for the TCT typically has an outer diameter of 25.4 mm and an inner diameter of 19.0 mm. The balls for the Four-ball test must have a 12.7 mm diameter.

The target specification for the percentage of moving components shielded was set chiefly by the fact that our customer requires the utmost safety of the device in order to test the device. It is our obligation as engineers to ensure the safety of anyone using the device or around the device at all times, so maximum shielding, able to withstand catastrophic failures, must be implemented around the moving components of the device.

For further safety of the device, a maximum temperature was set for the unshielded components of the device so as to prevent injury from burning. After researching the subject of skin burns, a maximum temperature of 44 °C was set for the unshielded components as this has been cited as the threshold of pain temperature for humans and above this temperature burning may begin to occur [21].

A crucial component of the device is the fact that it must be able to measure lateral forces and shaft speed of the test at all times. These pieces of information are needed to characterize the performance of the lubrication in testing throughout the duration of the test. We have set a target specification for data acquisition frequency of 20 kHz, given to us by our sponsor.

Finally, an important consideration for this project is the budget. Given the potential cost of components that may need to be purchased in order to complete this project (such as a gearbox, heating coils, safety shield material, strain gages, and an optical encoder), it is unrealistic to expect that the cost will remain within the $400 budget provided to all ME 450 teams. We have looked at potential costs and money spent by the previous team on the current prototype and expect to pay much more than this for the entire project.

Table 1: Target Engineering Specifications

	Four-ball Test	Twist Compression Test
Shaft Speed (rpm)	300 – 3600	2 – 30
Normal Force (kN)	0.03 – 16	Up to 100
Lubricant Temperature (°C)	0 – 150	
Test Control Volume Humidity (%)	0 – 100	
Annulus Outer Diameter (mm)	N/A	25.4
Annulus Inner Diameter (mm)	N/A	19.0
Ball Diameter (mm)	12.7	N/A
Moving Components Shielded (%)	100	
Temperature of Unshielded Components (°C)	< 44	
Lateral and Normal Force Data Acquisition Frequency (kHz)	20	
Shaft Speed Data Acquisition Frequency (kHz)	20	
Project Cost ($)	< 1500	

Quality Function Deployment (QFD)

The Quality Function Deployment (QFD) diagram, shown in Fig. 3, p. 14, was used as a tool to prioritize each engineering specification target we have set (bottom of QFD). It can be seen from the Weight/Importance row at the bottom of the QFD that the shaft speed measurement output is extremely important to the device. Without the shaft speed output, no information about the performance can be validated due to the fact that the speed wouldn't be able to be verified. Along with the measurement of the shaft speed, the ability of the device to reach high shaft speeds is crucial. Without the capability of reaching high speeds, the Four-ball test would be inoperable and thus we would have failed at combining the two tests into one device. The next most important specifications deal with the safety of the device. It is necessary that the moving components are shielded and the temperature of any unshielded (static) components is within the threshold of pain and burning of 44 °C. Of midrange importance for the device is accuracy of the geometries of the annulus and balls for the TCT and Four-ball tests, the low speed rpm for performing the TCT, producing low loads used in the Four-ball test, and obtaining the lateral force output. These midrange importance target specifications are still quite important in order to say that we have created a device that successfully integrates both the TCT and Four-ball tests, but these get trumped by the sheer necessity of safety, shaft speed measurement and producing high normal force. The remaining target specifications of controlling and measuring the temperature of lubricants, keeping costs low, and controlling the humidity of the test control volume, although desirable, are not critical to the end result of the project. If we are unable to implement the environmental control precisely or keep the costs below our budget, but achieve all other target specifications, the device would at least be functional and further improvements could be made relatively easily.

The QFD is also a valuable tool in that difficulty ratings can be placed within it for each target specification. It can be seen that the most difficult specification to achieve for this project is the temperature control of the lubricant. This control involves the use of a computer controller, precise implantation of both heating and cooling elements, and careful monitoring. It will be very hard to accurately adjust temperature and achieve the extreme ends of the temperature range. Slightly less difficult than temperature control will be achieving the appropriate shaft speeds and controlling them, as well as controlling humidity. It is very important to us and our sponsor that the motor is correctly connected to our driveshaft for proper functionality. Without a proper connection, it will be extremely hard to safely run the device and obtain the correct shaft speeds. The humidity control will be difficult to control due to the fact that we are simply heating water to form vapor in the test volume and are not continuously measuring and adjusting the heating coil for a specified humidity. Another fairly difficult target specification to achieve will be the maximum normal forces needed for the TCT. These loads are extremely high and introduce large stresses to the entire device. It is very important that we design for structural integrity and safety so that the device will be sturdy during high load conditions.

Other specifications, such as obtaining correct lateral force outputs, shaft speed outputs and keeping the cost under budget will be of medium difficulty with careful planning and wise use of resources on and off campus. Easier yet will be keeping the temperature of unshielded components low (since we don't expect much other than the lubricant and water for humidity control to get hot), obtaining low normal loads for the Four-ball test since the press should be very sturdy with low loading, machining the annulus with high accuracy and shielding all moving components.

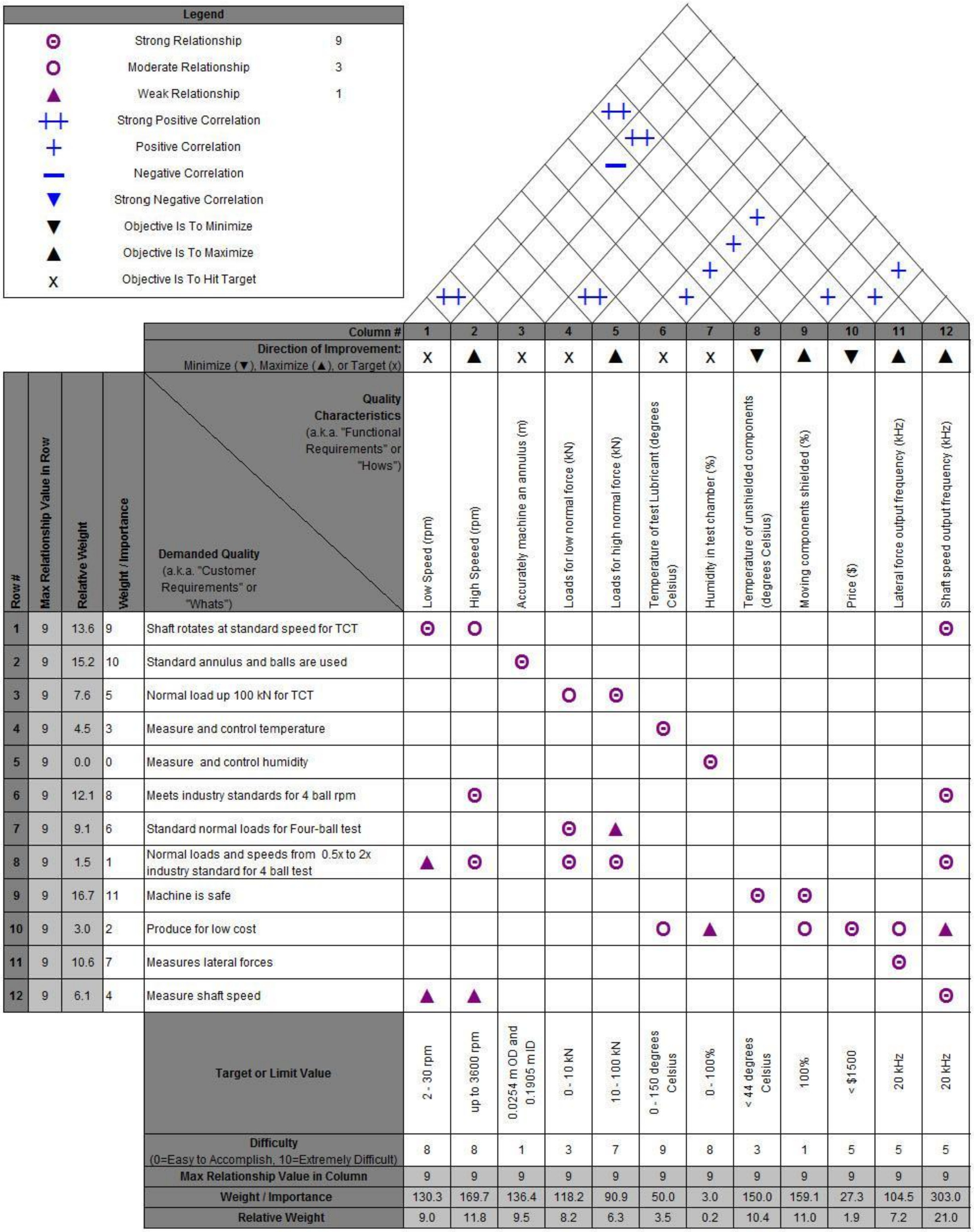


Figure 3: Quality Function Deployment (QFD) Diagram

CONCEPT GENERATION AND SELECTION PROCESS

After establishing our main engineering specifications from our sponsor's customer requirements and ranking their relative levels of importance in the QFD, we began the concept generation and selection process of our prototype redesign. The first step in this process was to create a functional decomposition of our desired prototype to expose the full list of functions that we needed to account for in our concept generation. We based this loosely on the current prototype, but made sure to list the functions generally so that we would not be biased towards the current methods of satisfying them. After we had developed a list of these functions, we began to brainstorm ideas to fulfill them. This stage included disassembly of the current prototype to more closely examine their implemented concepts, as well as research of other methods to satisfy the functions. Because our list of functions was extensive, we decided to develop several concepts and choose the best one for each function individually. The best concepts were judged using weighted selection criteria, and after checking to ensure that they could all coexist in a full design, they were put together to form our alpha design. This section will explain our concept generation and selection process in greater detail, as well as provide renderings of our individual chosen concepts and the assembled alpha design.

Functional Decomposition

In order to identify the functions that the device needs to be able to perform, a functional decomposition was developed. Through literature reviews, in-depth research, and sponsor meetings, the functional decomposition, Fig. 4, p. 16, was made to capture the major functions that the device will eventually need to be able to do. The inputs (energy, material, or signal) are listed on the left, and likewise the outputs are listed on the right. The center box contains the many functions that the device needs to be able to perform. The functions that are highlighted yellow are the functions that required concept generation selected solutions for and will describe in further detail throughout this report. These functions are mainly mechanically based or deal with sensing the forces, shaft speed, and climate conditions that are necessary outputs from the device. The five functions that have not been highlighted are not functions that we can choose at this point. Four of them deal with computer control and will need to be analyzed further at a later date; these include sensing failure of the lateral force sensor, stopping the motor in the case of failure, and checking the temperature and humidity (in a closed loop feedback system). The fifth is the function of actually applying thermal energy to the lubricant. Because applying thermal energy is more of an effect than an actual function, we have incorporated this function into the convert external energy to thermal energy function.

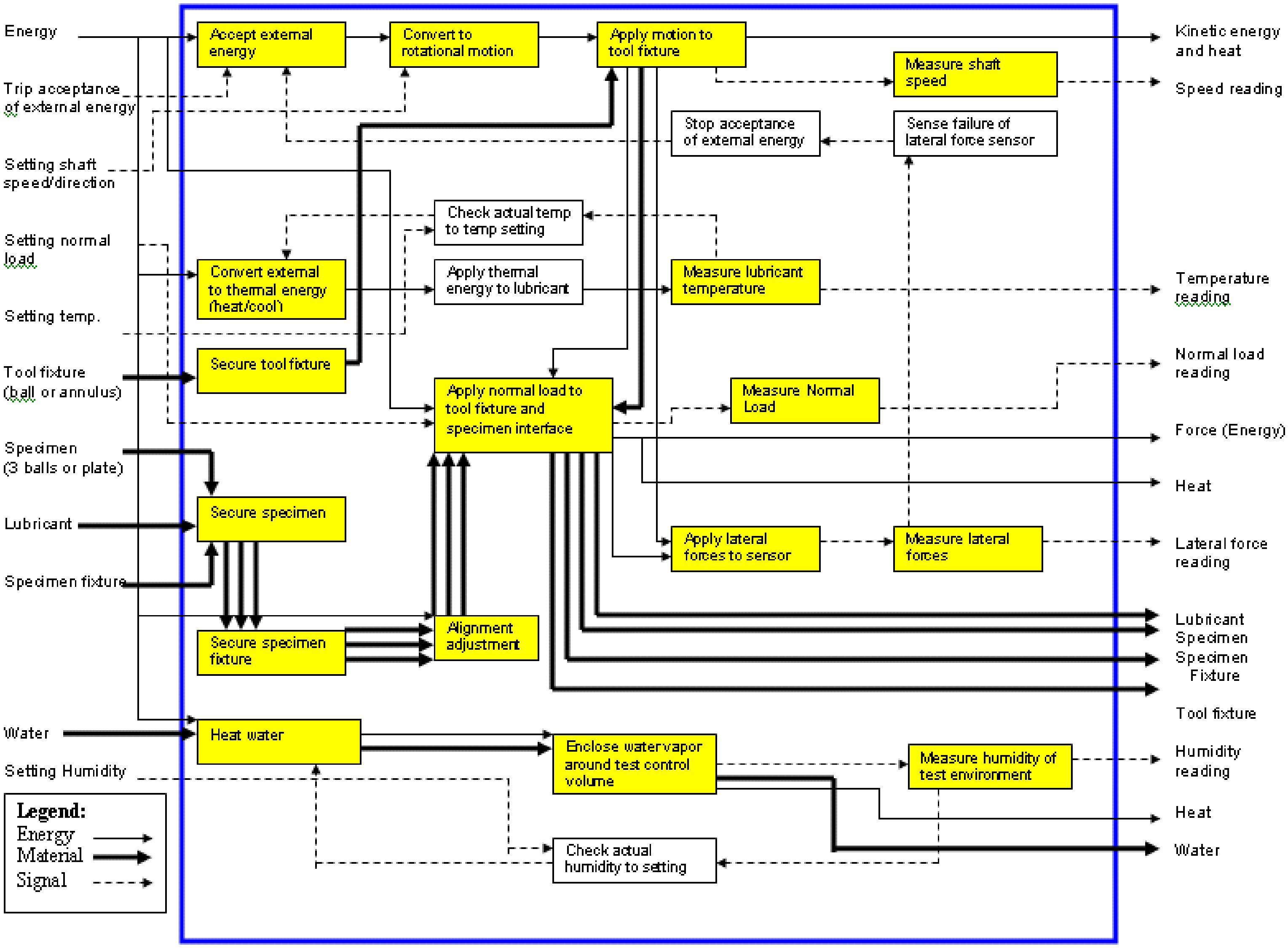


Figure 4: Functional Decompsition for Twist Compression and Four-Ball Tester

Concept Generation

The functional decomposition shows that we have generated 22 total functions for our device. Out of these, 17 were highlighted and researched for this report. However, two of these 17 are actually combined functions with two parts. The first is the conversion of external energy to thermal energy. This is really two functions because we need the device to be able to cool the lubricant down to 0°C and heat the lubricant to 150°C. The second combined function is the alignment adjustment. There are two functions involved due to the fact that we need to account for x-y plane adjustment for the TCT (making sure the annulus is flush with the plate) and z-axis adjustment for the Four-ball test (making sure the center of the bottom three balls is aligned with the center of the top ball). Because of these two combination functions, we are essentially dealing with 19 total functions, excluding the computer programming and application of thermal energy to the lubricant. Potential concepts that can satisfy these 19 functions conveniently split up into two distinct categories: new concepts that we must design and manufacture and existing concepts that we must research and purchase.

New Concepts: The functions that needed to be satisfied by the design of new concepts were the ones that required components specific to our prototype. A list of these functions is shown in Table 2, below. Each team member developed and sketched a few concepts for these functions, and after gauging their feasibility of manufacturing and functionality in our application, we compiled a list of the realistic concepts.

Table 2: Functions Requiring Design of New Concepts

Function	Component on Current Model	Function	Component on Current Model
Apply motion to tool fixture	**Keyed shaft**	**Alignment of specimen tool (xy-plane)**	**Ball joint**
Secure tool fixture	Removable rod through tool fixture	**Alignment of specimen tool (z-axis)**	**Four slots**
Secure specimen	Square cut and ball groove	Apply lateral forces to sensor	Beam on beam (sensor on support structure beam)
Secure specimen fixture	Bolts through cup		

Note: bolded items denote components that were deemed suitable for keeping in redesign

Our conception of ideas was helped greatly by our disassembly of the current prototype, which allowed us to view the concepts implemented for some of our functions by the last team. Viewing the pros and cons of the current designs played a large role in our selection of which concepts we should keep for our redesign and allowed us to produce improve concepts for the other functions. The current designs that we felt were suitable for keeping in our redesign are bolded in Table 2, above. Several design sketches for methods to satisfy the other functions are shown in Figs. 5-8.

Secure Tool Fixture*:* The design for securing the tool fixture on the current model has a through hole in the shaft and the tool fixture into which a rod is placed to hold the fixture in place until the device is loaded. Because we felt that this requirement of removing the rod while the device was loaded was unsafe, we designed other concepts that did not implement this feature. Most of the designs center around securing the tool fixture into the shaft using threads, whether they be in a collar outside the shaft or on the tool fixture itself. Another idea was a variation of the current model that used a screw and bolt to hold the tool fixture in the shaft, with the main difference being that the bolt remained in place during the test.

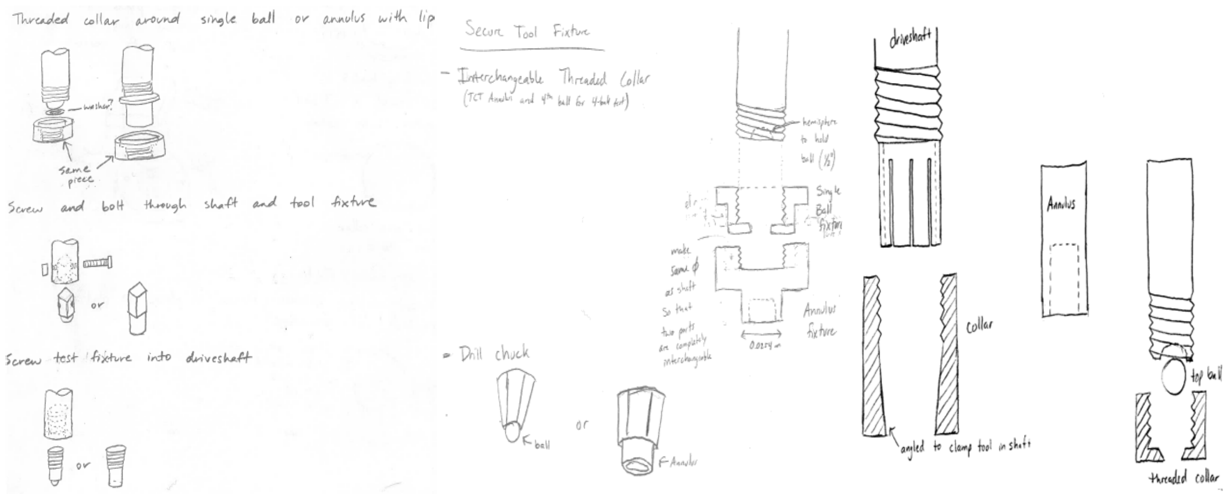


Figure 5: Concept Sketches for Securing Tool Fixture

Secure Specimen*:* The current model incorporates a square cut and ball grooves to hold the respective twist compression and four-ball specimen. These specimens are then held down by the cup, bolts, and wing nuts that serve to hold the specimen fixture in place. We felt that this was a rather bulky design that was difficult for the user to remove when tests were changed due to its size and weight. Threads were again a common feature in our design concepts. For the four-ball test, ideas included higher quality grooves for the balls that would be held in place by a threaded collar, or an interchangeable collar that fit around the balls and was secured with a uniform threaded second collar. For the twist compression test, designs that were proposed to hold the flat specimen plate in place included using bolts or a threaded column of the plate that would screw into the alignment plates below. Also, the interchangeable collar idea was included, in this case with the inner collar fitting around the plate.

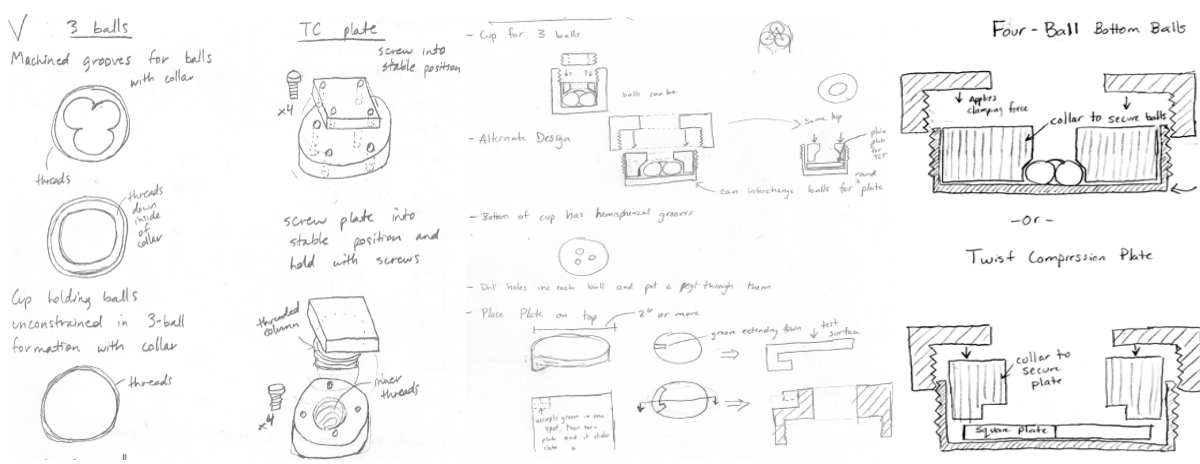


Figure 6: Concept Sketches for Securing Specimen

Secure Specimen Fixture*:* As mentioned above, the current model has four bolts that screw into the alignment plates and hold a cup in place when secured with wingnuts. Similar to our concerns with the current design for securing the specimen, we felt that this design was bulky and inconvenient for the user during testing. Our designs incorporated smaller and lighter pieces that could be removed more easily to hold the specimen fixture in place. The designs were secured with either multiple set screws, machined teeth on the bottom of the fixture, or a large threaded collar.

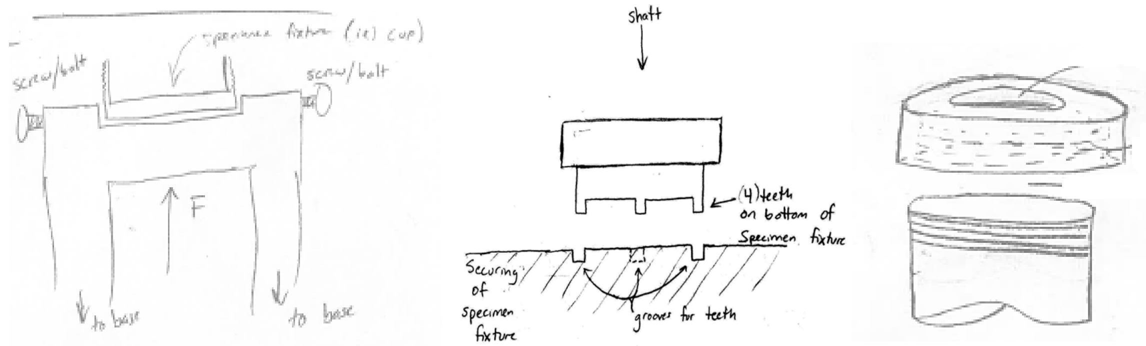


Figure 7: Concept Sketches for Securing Specimen Fixture

Apply Lateral Forces to Sensor*:* The current design applies and measures lateral forces using contact between rods attached to the specimen fixture and blocks of metal attached to cross-beams on the press. The displacement of the blocks of metal is then measured with strain gauges. This design is roughly put together, as the metal blocks are not secured very safely on the cross-beams. Also, measuring the deflection on the metal beams, especially when they are supported on only one end, would seem to give very inaccurate results. One idea that we had used the same design as the one currently on the prototype, but substituted in higher quality metal blocks and safety constraints. Displacement measurements occurred on the specimen fixture rods. Other ideas included using enclosed gear or pulley systems to apply tension to a strain gauge.

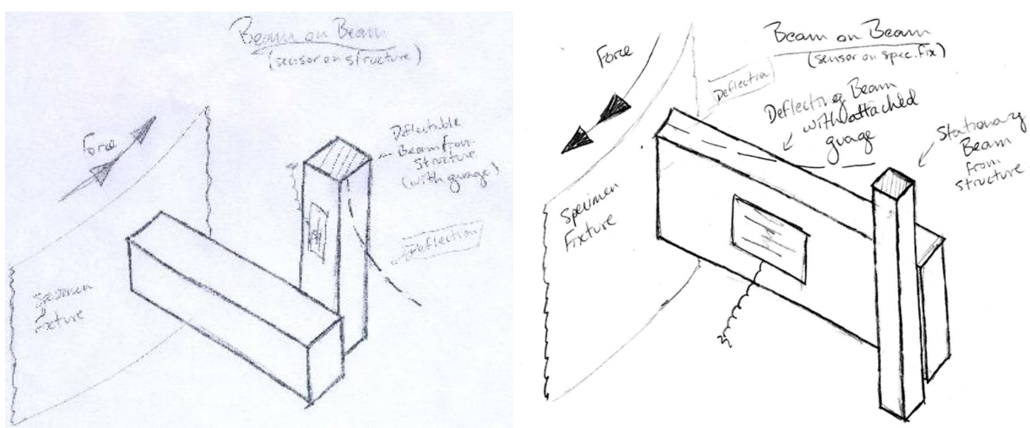


Figure 8: Concept Sketches for Applying Forces to Sensor

Existing Concepts: The remaining 12 functions from our functional decomposition will all require existing components that we will have to purchase, borrow, or use from the existing prototype. As described in the Information Sources section on p. 78, we researched methods of satisfying these functions to establish several potential concepts for each function. A table showing each of the functions requiring existing components, as well as the concepts that we developed for each function, is shown in Table 3, p. 20. Further description of each functions' concepts, as well as comparisons of their positive and negative attributes, can be seen in the following Concept Generation section.

Table 3: Potential Concepts for Functions Requiring Existing Components

Function	Concept 1	Concept 2	Concept 3	Concept 4	Concept 5	Concept 6
Accept external energy	AC motor	DC Motor	Internal combustion engine	Hand crank	Wind mill	
Convert energy to rotational motion	Chain drive	Belt drive	Continuously variable transmission	Gear-to-gear (spur)	Gear-to gear (helical)	Gear box
Apply normal load to specimen	Hydraulic/ Pneumatic press	Free weights	Thermal expansion			
Measure lateral forces	Strain gauges	Load cell	Piezoresistive strain gauges			
Measure shaft speed	Optical encoder	Contact tachometer	Laser Tachometer			
Cool lubricant	Peltier coolers	Refrigeration cycle	Pump gas/liquid			
Heat lubricant	Peltier coolers	Fire	Resistive heat coil	Conductive material (heat recycling)		
Measure lubricant temperature	Thermocouple	Mercury thermometer	Thermistor	Resistive temperature detector		
Heat water (humidity)	Fire	Resistive heat coil	Reverse Peltier coolers			
Enclose water vapor	Transparent box	Wooden box	Metal box			
Measure humidity	Hygrometer	Humidity meter				
Measure normal force	Compressive material	Internal beam deflection	Compressive load cell			

Pugh Charts

For selecting which idea would be best, we had to use some sort of system for selection that allowed us to analyze each individual function. The most logical way we knew of to accomplish this was to use Pugh Charts for each of the 19 functions.

Our Pugh Charts consist of a column of selection criteria that are weighted to the needs of the project and columns of each idea for the function in question. The ideas in the columns were scored and the scores were multiplied by the weight of each selection criteria. The sums of the weighted scores for each idea were compared and the highest scoring idea was deemed the desired solution to the function. A sample Pugh Chart outline is shown in Table 4, p.21. The selection criteria and scores are listed.

Table 4: Sample Pugh Chart with Idea #5 Selected as Solution to Function

Function						
	Weight	Idea #1	Idea #2	Idea #3	Idea #4	Idea #5
Safety	9	-2	-1	0	1	2
Cost	7	-2	-1	0	1	2
Manufacturability	7	-2	-1	0	1	2
Functionality	7	-2	-1	0	1	2
Ease of Operation	5	-2	-1	0	1	2
Maintenance	3	-2	-1	0	1	2
Environmental Impact	1	-2	-1	0	1	2
Total Score		-78	-39	0	39	78

As seen in this sample Pugh Chart, we developed seven selection criteria: Safety, Cost, Manufacturability, Functionality, Ease of Operation, Maintenance, and Environmental Impact. A weight was given to each of the selection criteria and this weight indicated the relative importance of the criteria to the overall scope of the project. Due to the fact that the device is practically useless if it is not safe, we gave safety the top priority on the criteria list. The point of the project is to produce a machine that can perform industry quality tests for a fraction of the price, therefore, the cost of the device was considered to be the second most important criterion. Tied with cost are manufacturability and functionality of the device with a weight of seven each. Since we are all students and have limited experience in manufacturing and machining, and that time is very limited for the project, it is imperative that the device is easy to manufacture and assemble. Furthermore, along with designing for manufacturability, we are very concerned with designing for functionality. We have set ambitious goals (along with our sponsor) for the performance of the device and in order to successfully replicate industry tests, the device must be designed with functionality in mind. Following cost, manufacturability and functionality in the list of selection criteria is the ease of operation of the device. Since the device is being designed and built by a student group for a fraction of the price of industry built machines, ease of operation is of lesser importance than the functionality and manufacturability. Some things that industry machines have are rather luxurious and simply unnecessary for our goals to be met. Some intricacies can be sacrificed on our design and therefore we gave ease of operation a weight of 5, in the middle of the list. Along those same lines, the maintenance and environmental impact of the device are of little importance to us since we are only building one prototype machine. Since there will only be one device, it won’t be an insurmountable task to repair it and the environmental impact will not have to be optimized since we are not mass producing it. Therefore, we gave maintenance a weight of 3 and environmental impact a weight of 1, making it the least important criterion.

After coming up with our criteria and weights, we had to think of a way to score the various ideas listed for each function. To do this, we decided to rank each idea on a scale of -2 to 2 for each of the listed criterion. A -2 indicates that the idea negatively meets the criterion to a high degree. A -1 indicates that the idea somewhat negatively meets the criterion. A 0 indicates that the idea neither positively nor negatively meets the criterion. A +1 indicates that the idea somewhat positively meets the criterion. A +2 indicates that the idea positively meets the criterion to a high degree. The ranking was then multiplied by the weight of the criterion and the sum of all rankings and criteria weights were added and totaled on the bottom of the chart beneath each idea. The idea with the highest total was chosen as the best solution for each function that needed to be met. In the example Pugh chart in Table 4, above, the 5 rankings are demonstrated for five ideas and the idea with the highest total is highlighted green, meaning it is the selected idea. In the example, the chosen idea is Idea #5.

Concept Selection
The example Pugh chart described in the previous section was used in selecting all of our 19 function's solutions. This section will list all of the functions associated with our device and explain how we arrived at our chosen solution and how that solution works to achieve the desired function.

Accept External Energy: Last semester's team used an AC motor to accept their external energy and using the Pugh chart below, we compared this motor to other possible sources of energy. The internal combustion engine, hand crank, and wind mill were our extravagant ideas. The internal combustion engine is too unsafe for our working environment and hard to operate and maintain. The wind mill would be too expensive, unsafe for our working area and lacking in functionality. Even though the hand crank came in second place, we concluded it wouldn't be easy to produce the loads we needed by cranking it by hand. It would also cost money and take time to design and manufacture it. Between the AC and DC motor, we decided to use the AC motor. The main deciding factor was that the AC was already being used by the previous team and it worked and met the specifications that it needed to (speed, etc.). Changing the AC motor to a DC motor would cost us lots of time, money and energy. As long as the AC motor still works there is no need to change it. Figure 9, below shows the current AC motor.

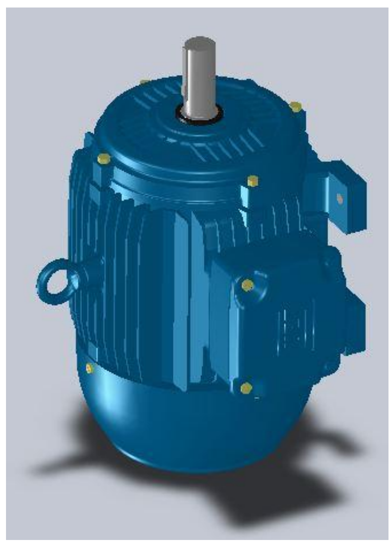


Figure 9: AC Motor CAD Model

Table 5: Pugh Chart for Accepting External Energy

	Weight	AC Motor	DC Motor	Internal Combustion Engine	Hand Crank	Wind Mill
Safety	9	-1	-1	-2	1	0
Cost	7	-1	-1	0	1	-1
Manufacturability	7	2	2	2	0	1
Functionality	7	2	1	1	-1	0
Ease of Operation	5	2	2	-2	1	2
Maintenance	3	2	2	-2	1	1
Environmental Impact	1	1	1	-2	2	2
Total Score		29	22	-15	19	15

Convert Energy to Rotational Motion: During our brainstorming process, we developed six different ideas for converting the energy produced by our power source into rotational motion to be applied to the tool (top ball of Four-ball test or annulus of TCT). The previous team's prototype utilizes a chain drive system, so we considered this as a contender for our solution. We thought of switching out the chain for a belt to make a belt drive system, using a Continuously Variable Transmission (CVT) system, using conventional gears with either vertical or slanted teeth, or using a pre-fabricated gear box. After ranking all of these ideas in a Pugh chart, shown in Table 6, below, we came to the conclusion that the gear box would be the best way to get rotational motion from our energy source since it would be safe, easy to manufacture due to the fact that we would purchase it, and function well. An example of a gearbox is shown in Figure 10, below. However, after researching gear boxes for a configuration such as our device, we are finding that obtaining a gear box that is able to withstand the testing torques at the specific gear ratios is nearly impossible unless we have one custom made for our application. There are also issues in mounting the power source and gear box to the device if a gear box were to be used. Because of the potential problems that are arising, we find that it may be more beneficial to go with our second place idea for this function, using slanted tooth gears without a drive chain or belt.

Figure 10: Example of a Gearbox

Table 6: Pugh Chart for Converting Energy to Rotational Motion

	Weight	Chain Drive	Belt Drive	CVT	Gear to Gear (vertical teeth)	Gear to Gear (slanted teeth)	GearBox
Safety	9	-1	-1	-1	0	0	1
Cost	7	1	1	-1	1	1	0
Manufacturability	7	1	1	-1	1	1	2
Functionality	7	1	0	2	0	1	1
Ease of Operation	5	1	1	-1	1	1	0
Maintenance	3	-1	-1	-2	1	1	0
Environmental Impact	1	-1	0	-1	-1	-1	-1
Total Score		13	7	-21	21	28	29

Apply Motion to Tool Fixture: Some component is needed to translate the rotational motion to the tool fixture, specimen, and specimen fixture which are the important components during testing. Design concepts that were considered included a shaft, a system of pulleys, and multiple ropes, all of which would serve to twist the tool fixture. A Pugh chart showing how these concepts performed when judged against our selection criteria is shown in Table 7, below.

Table 7: Pugh Chart for Applying Motion to Tool Fixture

	Weight	Shaft	Pulleys	Rope
Safety	9	0	-1	-2
Cost	7	0	1	2
Manufacturability	7	1	-1	2
Functionality	7	2	-1	-2
Ease of Operation	5	2	-1	-2
Maintenance	3	1	0	-1
Environmental Impact	1	0	0	0
Total Score		34	-21	-17

The Pugh chart clearly shows that the shaft is the best concept for our device. The shaft will be made out of metal, which will make it more expensive than the other two concepts; however, this expense will be offset by the shaft being much safer, more functional, and easier to operate. A metal shaft will be more resistant to yield and fracture than pulleys or ropes, and will translate a much higher percentage of rotational motion to the tool fixture than either of these concepts. The metal shaft will also require no additional inputs for operation once it is installed, and the current model of this device uses a metal shaft that we may be able to use or incorporate into our design. An illustration of our shaft is shown in Figure 11, below.

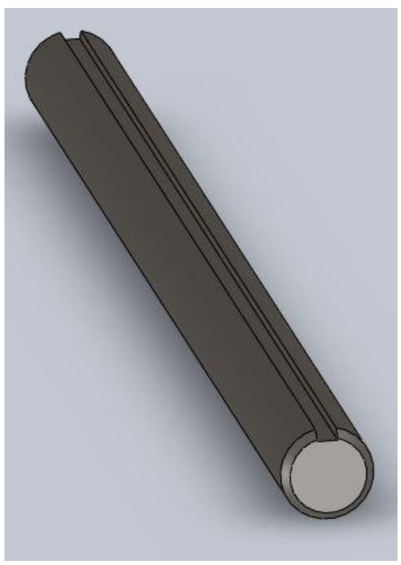


Figure 11: Shaft Model

Secure Tool Fixture: After choosing a shaft as the component to translate rotational motion to the tool fixture, a method for securing the tool fixture into the shaft must be chosen. Design concepts that were considered included a removable rod through the tool fixture, a screw and bolt through the tool fixture, a chuck, an interchangeable threaded collar, and a threaded tool fixture that screwed into the shaft. When running the four ball test, each of these designs except for the interchangeable threaded collar would have a threaded cap piece to hold the single ball in place. A Pugh chart showing how we evaluated these concepts using our selection criteria is shown in Table 8, below.

Table 8: Pugh Chart for Securing Tool Fixture

	Weight	Removable Rod through Tool Fixture	Screw and Bolt through Tool Fixture	Chuck	Interchangeable Threaded Collar	Threaded Tool Fixture that Screws into Shaft
Safety	9	0	-1	1	1	-1
Cost	7	2	1	-1	1	1
Manufacturability	7	1	1	-1	0	1
Functionality	7	1	0	1	1	1
Ease of Operation	5	0	1	1	1	1
Maintenance	3	0	-1	1	0	0
Environmental Impact	1	0	0	0	0	0
Total Score		28	7	10	28	17

The results of our Pugh chart show that the interchangeable threaded collar is the best concept for our device followed closely by the removable rod through the tool fixture and the threaded tool fixture that screw into the shaft. Although the machining and material costs of the interchangeable threaded collar will be slightly higher than the other two competitive designs, its safety grade separates it from them. We reasoned that the collar will face the least stress from rotation under a load due to its placement outside the shaft and tool fixture, giving it the best chance to keep the tool fixture aligned with the shaft. Also, we designed for the chance that the collar might come undone from the shaft to be minimized by threading the shaft in the opposite direction from that of the rotation. A sketch of the interchangeable threaded collar design is shown in Fig. 12, below.

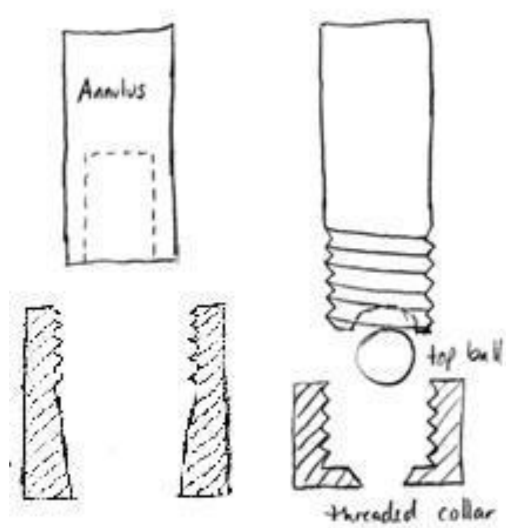


Figure 12: Interchangeable Threaded Collar Sketch

Secure Specimen: Securing the specimen, either the bottom three balls for the Four-ball test or the plate for the TCT, is a crucial function. The specimen needs to be securely fixed in order for the test results to be valid, so when it comes time to actually manufacture the specimen fixture, the utmost precision must be achieved. In brainstorming ideas for the specimen fixture, we kept in mind that the component needs to be very functional but also able to be produced in a timely and effective fashion. The current prototype has a cup with a square cut out on the bottom that can house either the plate for the TCT or a plate with three ball indents for the Four-ball test. A large collar fits over the cup and is bolted down to it to ensure that the plate or three balls are securely fastened. In a similar fashion, we have developed an iteration of this idea that is very similar to what is used on industry Four-ball testers. We call it the Three Piece Cup with Ring and Outer Cylinder, seen in Fig. 13 for the Four-ball test and Fig. 14, for the TCT. Essentially, it consists of a similar bottom cup that will house the square plate and three bottom balls, an interchangeable ring (one to secure the plate, and one to secure the three balls), and a top collar that will be screwed onto the cup itself, eliminating the time consuming task of bolting down the collar on the current device. After ranking the ideas in a Pugh Chart, shown in Table 6 below, we came to the conclusion, that indeed we were going to scrap what is present in the device and manufacture a new specimen fixture as described above.

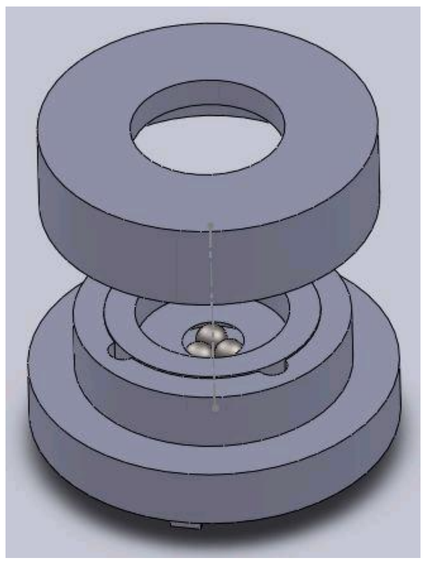


Figure 13: Specimen Fixture with Four-Ball Test Setup

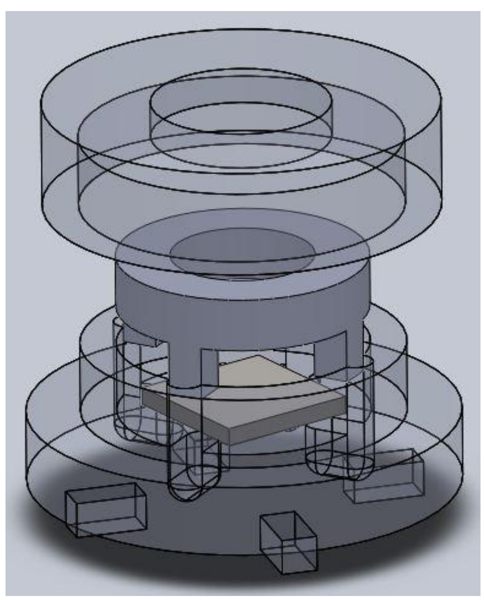


Figure 14: Specimen Fixture with TCT Setup

Table 9: Pugh Chart for Securing Specimen

	Weight	Square Cut and Ball Groove	Three Piece Cup With Ring and Outer Cylinder	Four Piece Cup With Ring, Outer cylinder and Top Plate (for TCT)	Grooved Plate (4-ball) and Four Screw TCT	Grooved Plate (4-ball) and Screw Column TCT
Safety	9	1	1	0	-1	0
Cost	7	0	1	0	1	-1
Manufacturability	7	1	1	-1	0	-1
Functionality	7	2	2	1	1	1
Ease of Operation	5	1	2	1	1	1
Maintenance	3	1	1	1	1	1
Environmental Impact	1	1	1	1	1	1
Total Score		39	51	9	14	2

Secure Specimen Fixture: Along with securing the specimen's themselves via the specimen fixture, the entire specimen fixture itself has to be secured so that no unwanted motion of the specimen occurs during either of the tests. In the current prototype, the same bolts that keep the collar to the specimen cup secure extend down into a block of steel that is placed on a roller bearing. The bolts connect the specimen fixture to the block keeping it all secure to the roller bearing. (The block currently has two rods sticking out of it that were to be used as deflection rods for measuring lateral forces. The rods prevent undesired rotational motion during the tests.) Some new ideas we generated during brainstorming included using set screws, a clamp, or teeth on the specimen fixture that would sit in grooves on a bottom securing plate. After ranking the ideas in a Pugh Chart, shown in Table 10, p. 28, the specimen fixture with teeth was determined to be the frontrunner for the design, shown in Fig. 15, below. This design may not be the easiest to manufacture (although with further design development we are finding innovative ways to make manufacturing easy), but the design will make it very easy for the user to remove the entire specimen fixture and set up either of the two tests, then place the fixture back onto the device.

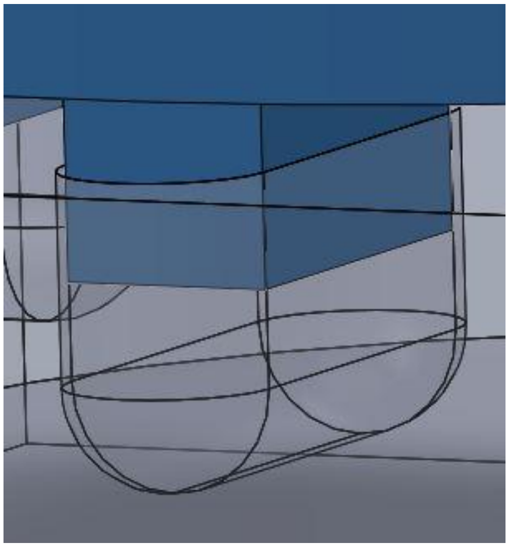


Figure 15: Example of tooth from speciment fixture (blue) fitting into groove of bottom plate (grey and transparent)

Table 10: Pugh Chart for Securing Specimen Fixture

	Weight	Set Screws	Clamp	Teeth
Safety	9	-2	0	0
Cost	7	1	0	0
Manufacturability	7	1	1	-1
Functionality	7	-2	-1	2
Ease of Operation	5	-1	1	1
Maintenance	3	0	0	0
Environmental Impact	1	0	0	0
Total Score		-23	5	12

Alignment of Specimen to Tool (x-y plane): For the TCT, it is critical that the specimen plate face is aligned to the face of the annulus. Due to imperfections in manufacturing, it cannot be guaranteed that the alignment will be perfect, so some sort of alignment mechanism must be implemented. The current prototype uses a two plate design with a ball joint in between them in order to allow for x-y plane alignment. The plates are secured together at the desired position using four bolts. Other ideas we thought of for x-y plane alignment included a three plate, two hinge system, and a complicated four rod system that would allow for the specimen fixture to be adjusted freely and screwed down onto four rods. The ball joint system was the clear choice after ranking the ideas in a Pugh Chart, shown in Table 11 below. A drawing of the design is shown in Fig. 16 below. Given that the previous team had the plates and ball manufactured for them, it makes sense to use what they already have, leaving more time for us to focus on other components for re-design. However, if we find that modifications have to be made to the plates or ball in order to support the large normal loads achieved during testing, we may have to vary the design and have the plates and ball re-manufactured.

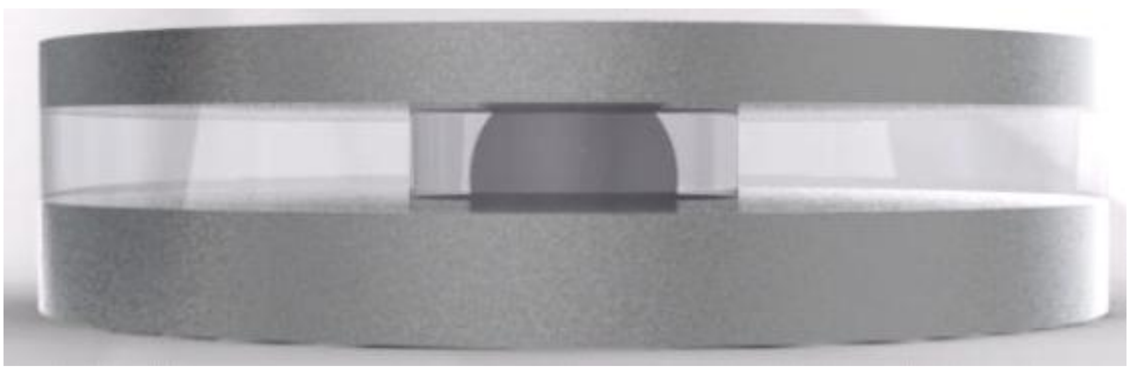


Figure 16: Ball Joint and Two Plates for x-y Plane Alignment

Table 11: Pugh Chart for Aligning the Speciment to the Tool in the x-y plane

	Weight	Ball Joint	Hinges	Four Rods
Safety	9	0	-1	-1
Cost	7	1	1	1
Manufacturability	7	1	0	-1
Functionality	7	1	1	0
Ease of Operation	5	1	1	-1
Maintenance	3	1	0	1
Environmental Impact	1	0	0	0
Total Score		29	10	-11

Alignment of Specimen to Tool (z-axis): For the Four-ball test, it is critical that the center (z-axis) of bottom three balls is aligned exactly to the center (z-axis) of the tool. Similarly to the x-y alignment of the TCT, it cannot be guaranteed that the two axes will be aligned without some sort of alignment device. The current prototype uses a two plate system to allow for z-axis alignment during the Four-ball test. Each plate contains four machined slots that allow for the insertion of bolts that are used to hold the two plates together. The slots on one plate are aligned on the x-axis and on the other plate they are aligned on the y-axis so that the plates can be positioned in such a way that the center of the top plate is aligned with the tool. Fig. 17 below shows the design and allows one to visualize how the two plates can be positioned within a 360° circle, the diameter of with is the length of one of the slots. Another idea we came up with was essentially another two plate system, but much simpler than the four slot idea. It would just be two flat plates with no additional features that would be clamped together. After evaluating the two ideas in a Pugh Chart, shown in Table 12 below, we decided that the current four slot plate design would suite our needs most appropriately.

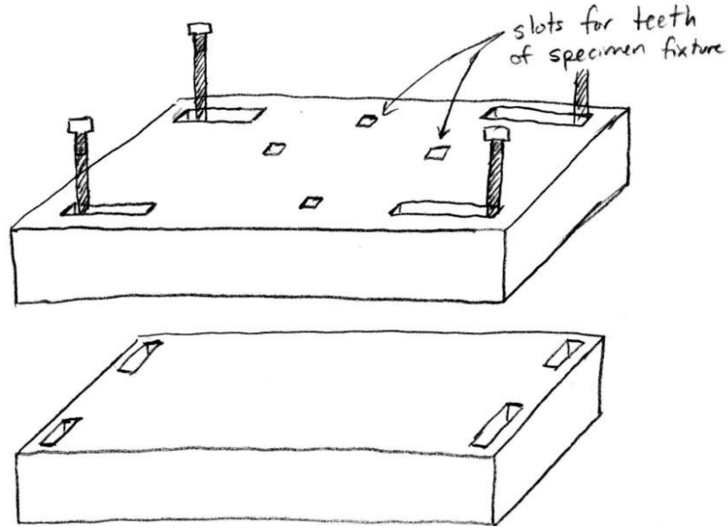


Figure 17: Two Plates with Four Slots for z-axis Alignment

Table 12: Pugh Chart for Aligning the Speciment to the Tool in the z-axis

	Weight	Four Slots	Moving Plate Clamp
Safety	9	1	0
Cost	7	0	0
Manufacturability	7	-1	-1
Functionality	7	1	0
Ease of Operation	5	1	0
Maintenance	3	0	1
Environmental Impact	1	0	0
Total Score		14	-4

When brainstorming these ideas for the x-y plane and z-axis alignment, we originally assumed that we would design two separate plate systems, one for the TCT, and one for the Four-ball test. However, we have recently disassembled the current prototype and we realized that manufacturing two separate systems would be unnecessary. The current two plate system incorporates both the ball joint and four slot designs. When performing the TCT, the ball is inserted in between the two plates and the plates are bolted together. When performing the Four-ball test, the ball is removed, allowing the 360° movement for z-axis alignment, and the plates are once again bolted together. Fig. 18 below shows how the two functions are combined.

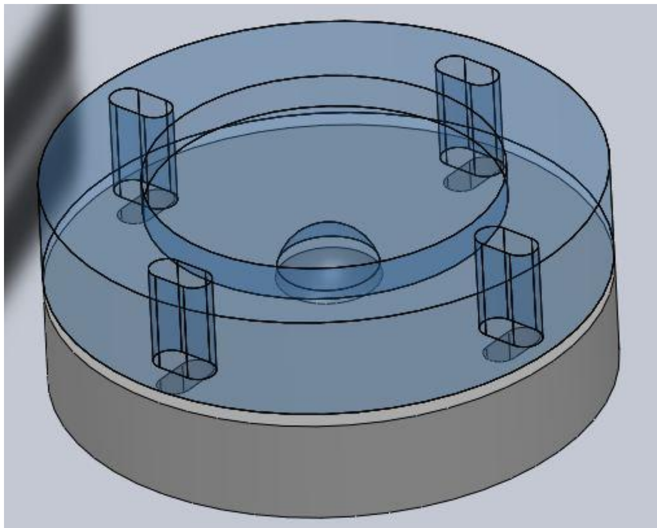


Figure 18: Combined Two Plate Alignment Fixture

Apply Normal Load to Tool Fixture/Specimen Interface: Based both on our Pugh Chart, Table 13 p. 31, and our choices with remaining with most of the current model, we elected to continue using the Omega 625 Shop Press to apply a normal load to the tool and fixture interface. Replacing this would be a waste of money and time, and this structure is more than sufficient for our test needs. Fig. 19 below shows the shop press from the current prototype.

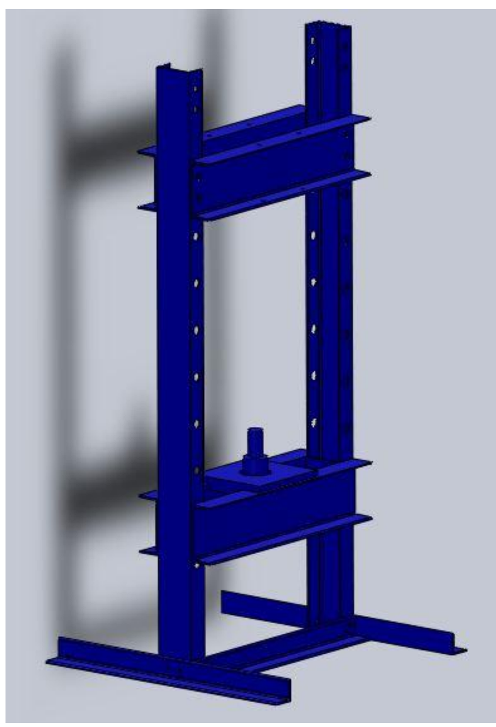


Figure 19: 25-ton Omega Shop Press

Table 13: Pugh Chart for Applying Normal Load to Tool Fixture/Specimen Interface

	Weight	Hydraulic/ Pneumatic	Free Weights	Thermal Expansion
Safety	9	1	-1	-1
Cost	7	0	1	-1
Manufacturability	7	2	2	1
Functionality	7	2	2	-1
Ease of Operation	5	2	-1	-1
Maintenance	3	0	0	0
Environmental Impact	1	0	0	-1
Total Score		47	21	-22

Apply Lateral Forces to Sensor: In the same vein as the current model, we are using a deflecting beam with an attached strain-gauge to measure the lateral forces that arise as a result of friction. However, based on our Pugh Chart, Table 14, p. 32, we are going to rearrange the current structure and use a non-deformable block to cause deflection on a beam attached near the specimen fixture. The previous design placed the strain gauge on a beam attached to the press structure. That beam was deflected by another one attached to the fixture. We figured that this would cause undue losses since both beams could potentially deflect. We are also going to make our deflecting beams interchangeable so we can insert different materials and thicknesses to account for normal forces across the entire test range. Fig. 20 below shows a beam attached the plate that will secure the specimen fixture. This beam will house a sensor (talked about in following section) that will measure the lateral forces.

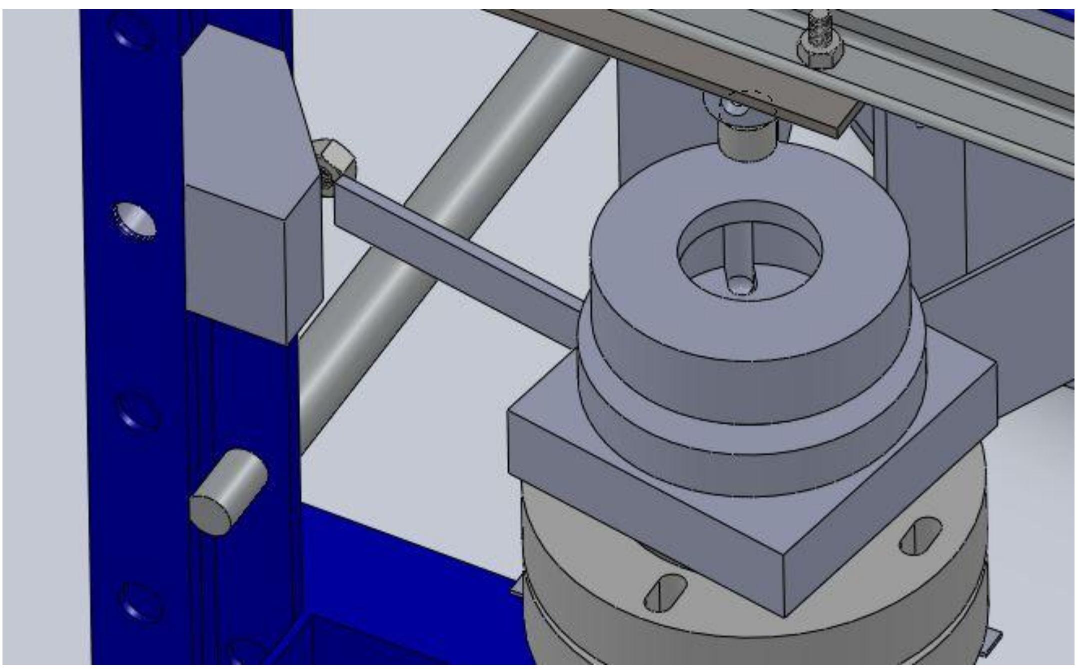


Figure 20: Plate with Deflecting Beam for Measuring Lateral Forces

Table 14: Pugh Chart for Applying Lateral Forces to a Sensor

	Weight	Beam on Beam (sensor on support structure beam)	Beam on Beam (sensor on specimen fixture beam)	Pulley attached to sensor	Gear system
Safety	9	0	0	0	0
Cost	7	1	1	0	0
Manufacturability	7	1	2	0	0
Functionality	7	0	0	1	1
Ease of Operation	5	1	1	0	0
Maintenance	3	0	0	0	0
Environmental Impact	1	0	0	0	0
Total Score		19	26	7	7

Measure Lateral Forces: As mentioned and implied above, we are going to stick to using a strain gauge to measure lateral forces. The previous team used them, and they are cheap and simple to affix and implement. Proper consultation and preparation will be sought to ensure proper functionality and electrical implementation of each gauge. The Pugh Chart used to select a device for measuring the lateral forces is shown in Table 15, below. Fig. 21, below, shows a typical strain gauge.

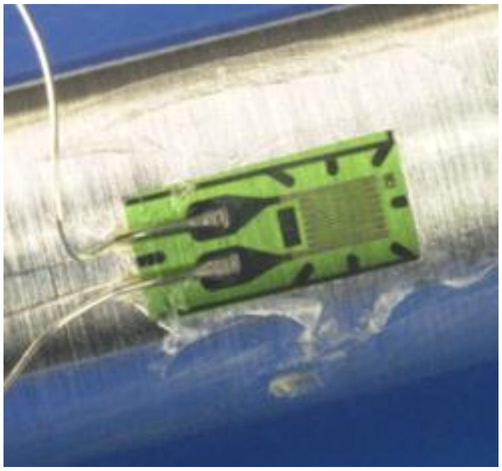


Figure 21: Typical Strain Gauge Attached to a Beam

Table 15: Pugh Chart for Measuring Lateral Forces

	Weight	Strain Gauge	Load Cell	Piezoresistive strain gauge
Safety	9	2	2	2
Cost	7	2	-2	0
Manufacturability	7	1	2	0
Functionality	7	1	2	1
Ease of Operation	5	1	2	0
Maintenance	3	1	2	1
Environmental Impact	1	0	0	0
Total Score		54	48	28

Measure Shaft Speed: We are going to continue using an optical encoder to measure shaft speed. Based on our evaluations in our Pugh Chart, Table 16, below, they are cheaper, safer and easier to implement than a contact tachometer or laser tachometer. However, we can't reuse the previous team's encoder as it was shorted sometime during their prototyping. Fig. 22, below shows a typical optical encoder for measuring shaft speeds.

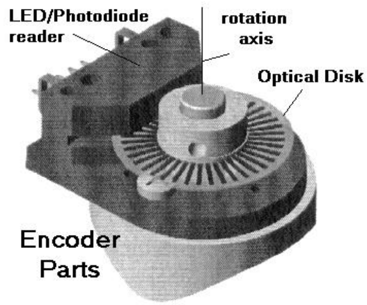


Figure 22: Typical Optical Encoder

Table 16: Pugh Chart for Measuring Shaft Speed

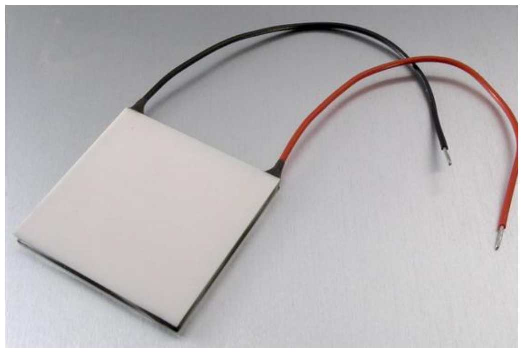
	Weight	Optical Encoder	Contact Tachometer	Laser Tachometer
Safety	9	2	1	2
Cost	7	2	2	0
Manufacturability	7	1	0	0
Functionality	7	1	0	2
Ease of Operation	5	1	1	1
Maintenance	3	0	-1	1
Environmental Impact	1	0	0	-1
Total Score		51	25	39

Cool Lubricant: To achieve the lower temperatures in our engineering specifications in the lubricant, specifically the range below room temperature (0-20 ºC), a means of cooling the lubricant must be implemented. Because it will be extremely difficult to uniformly cool only the lubricant, the component that accomplishes this function will likely be attached to either the specimen or the specimen fixture, and will have losses into these components. The design concepts that were considered included Peltier coolers, a refrigeration cycle, and pumping cold gas or liquid around the test volume. A Pugh chart showing how these concepts scored against our selection criteria is shown in Table 17, below.

Table 17: Pugh Chart for Cooling Lubricant

	Weight	Peltier Coolers	Refrigeration Cycle	Pump Gas/Liquid
Safety	9	0	-1	-1
Cost	7	0	-2	-1
Manufacturability	7	1	-1	-1
Functionality	7	2	2	2
Ease of Operation	5	1	1	1
Maintenance	3	0	-1	-1
Environmental Impact	1	0	-2	0
Total Score		26	-21	-9

The Pugh chart shows that Peltier coolers are the best concept for cooling the test volume of our device. The main advantages of Peltier coolers are shown in their safety, cost, and manufacturability scores. Although the hot sides of Peltier coolers can get very warm, heat sinks with fins will be used to dissipate this heat and decrease this temperature. Additionally, this elevated temperature is safer than the toxic chemicals that would likely be required for cooling in the other two concepts. Peltier coolers are also relatively inexpensive (most between $10-$20), so even a system with several of them will still be less expensive than the refrigeration cycle or the gas/liquid pump, which both would include multiple components and chemicals. Finally, Peltier coolers require only connection to a power supply for use, while the other two concepts would require the assembly of multiple components and therefore had lower manufacturability scores. Five Peltier coolers are also available for our immediate use, as they were purchased for the current device but never implemented due to time constraints. A picture of one of the Peltier coolers is shown in Fig. 23 below.



http://www.kryotherm.ru/imagez/LCB.jpg

Figure 23: Peltier Cooler

Heat Lubricant: Similar to cooling the lubricant to reach the lower temperatures of our engineering specifications, the lubricant must also be heated to reach the higher temperatures of our specifications. The component that accomplishes this function will likely be attached to the test fixture, specimen, or specimen fixture, and likewise will have thermal losses into these components. The concepts that were considered for heating the lubricant were 'reverse' Peltier coolers (where the Peltier device is power inverted so that it acts as a heater), fire, resistive heat coils, and conductive material to recycle the heat from friction. The results of how these design concepts performed against our selection criteria are shown in Table 18, below.

Table 18: Pugh Chart for Heating Lubricant

	Weight	Reverse Peltier Coolers	Fire	Resistive Heat Coils	Conductive Material for Recycling Friction Heat
Safety	9	0	-2	-1	-1
Cost	7	0	2	1	-1
Manufacturability	7	1	1	1	0
Functionality	7	-1	-1	1	0
Ease of Operation	5	1	-1	0	1
Maintenance	3	0	-1	0	1
Environmental Impact	1	0	-2	0	2
Total Score		5	-14	12	-6

The results from the Pugh chart show that resistive heat coils are the best option for heating the lubricant. Although Peltier coolers will already be used in our device to cool the lubricant, the Pugh chart shows that they are only the second best option for heating the lubricant. The main advantage to using resistive heat coils is the potential for the coils to reach the highest temperatures of our engineering specifications, while the reverse Peltier coolers can only reach about half of the desired maximum temperature (about 70 ºC). Although reverse Peltier coolers might be somewhat safer and slightly easier to use than resistive heat coils, these small deficiencies are easily made up for in the lower cost and increased temperature range of the heat coils. A picture of a potential resistive heat coil that we will use is shown in Fig. 24, below.

Figure 24: Resistive Heat Coils

Measure Lubricant Temperature: We are going to stick to using a thermocouple to measure the temperature of the lubricant during testing. In creating our Pugh Chart, Table 19, p. 36, we determined that thermocouples are cheap and the best option over the other possibilities. Of the different varieties of thermocouples, we will most likely use a thin, adhesive-backed thermocouple that would be the easiest to implement and least obtrusive to the setup of the remainder of our model. Fig. 25, below, shows a typical thermocouple.

Figure 25: A Typical Thermocouple

Table 19: Pugh Chart for Measuring Lubricant Temperature

	Weight	Thermocouple	Finger Test	Mercury Thermometer	Thermistor	Resistor Temperature Detector
Safety	9	1	-2	0	1	1
Cost	7	1	2	1	1	-1
Manufacturability	7	2	2	2	2	2
Functionality	7	2	-2	0	1	2
Ease of Operation	5	2	2	2	2	2
Maintenance	3	1	2	2	1	1
Environmental Impact	1	0	0	-1	0	0
Total Score		57	12	36	50	43

Heat Water for Humidity Control: Measuring humidity is one of the functions we are adding to the project this semester per our sponsor's request of adequate environmental control. In order to increase humidity we will be placing a boiling container of water into the environmentally controlled area, which will be explained in the next section. The resistive heating coils will fit around the container and heat it until desired. The other ideas we came up with were heating the water with fire and a peltier. The main reasons fire wasn't chosen were due to its lack in safety and its overall lack in functionality and ease of use. It is also not environmentally friendly. The peltier wasn't chosen due to the fact that it doesn't heat to the temperature that is needed and it's more expensive. An example of a resistive heating coil can be seen in Fig. 24, p. 35.

Table 20: Pugh Chart for Heating Water for Humidity Control

	Weight	Fire	Resistive Heat Coils	Peltier
Safety	9	-2	-1	0
Cost	7	2	1	0
Manufacturability	7	1	1	1
Functionality	7	-1	1	-1
Ease of Operation	5	-1	0	1
Maintenance	3	-1	0	0
Environmental Impact	1	-2	0	0
Total Score		-14	12	5

Enclose Water Vapor Around Test Control Volume: In order to regulate and measure humidity we need to enclose the test area. We have decided that the casing we chose needs to enclose and keep the humidity in, be clear so we can see through it, and be a durable material. Ideally we will be able to have a door that would allow us to get in and out of the test area easily. The other options we thought of were a wooden box and an aluminum foil tent. The wooden box was cheap but instead of containing the humidity, the wood would soak up the water, which would prevent it from functioning correctly. The other large disadvantage is that wood is not see through, which prevents from visually regulating the humidity and reading the humidity meter. The aluminum foil tent has a lot of the same downfalls. Even though it is relatively cheap, aluminum is not see through which prevents us from reading the humidity meter inside. Fig. 26, p. 37, shows an example of a plastic case enclosure.

Figure 26: Clear Plastic Case

Table 21: Pugh Chart for Enclosing Water Vapor around the Test Control Volume

	Weight	Clear Plastic Case	Wooden Box	Aluminum Foil Tent
Safety	9	1	1	0
Cost	7	0	1	1
Manufacturability	7	0	0	1
Functionality	7	1	-1	-1
Ease of Operation	5	0	0	0
Maintenance	3	0	-1	-1
Environmental Impact	1	0	-1	-1
Total Score		16	5	3

Measure Humidity of Test Environment: Per the request of the sponsor, Professor Krauss, we need to be able to measure the humidity of the test area. Through the use of the Pugh Chart, Table 22, p. 38, we have concluded that a humidity meter is the best option to use. Fig. 27, below, shows a typical humidity meter. A humidity meter is safe, inexpensive, easy to use and very functional. We compared the humidity meter to a hygrometer and the main difference between the two is that the hygrometer takes a lot more work by the user. The hygrometer involves two separate thermometers, one wet bulb and one dry bulb; whereas the humidity meter is placed into the test area and outputs the humidity.

Figure 27: Humidity Meter

Table 22: Pugh Chart for Measuring Humidity of Test Environment

	Weight	Hygrometer (bulb test)	Humidity Meter
Safety	9	1	2
Cost	7	1	1
Manufacturability	7	1	2
Functionality	7	2	2
Ease of Operation	5	0	2
Maintenance	3	0	1
Environmental Impact	1	2	2
Total Score		39	68

Measure Normal Force: Throughout both the twist compression and four-ball test we need to be able to measure the normal force. Last semester's team decided to use a block of compressive material with a strain gauge attached and it functioned well. Through the use of a Pugh Chart, Table 23, below, we have decided that keeping this implemented in the design is the best decision. This compressive material is very safe, inexpensive and easy to use. In the figure below, the compressive material can be seen directly above where the load will be applied and directly below where the alignment will be assessed. As the load is applied the strain gauge will measure the normal force. Our other options were using an internal beam deflection, or a compressive load cell. We decided not to use the internal beam deflection due to the fact that it was going to add a lot of time and money to design and implement. The other option, the compressive load cell, would be easy to manufacture and safe, but it costs too much.

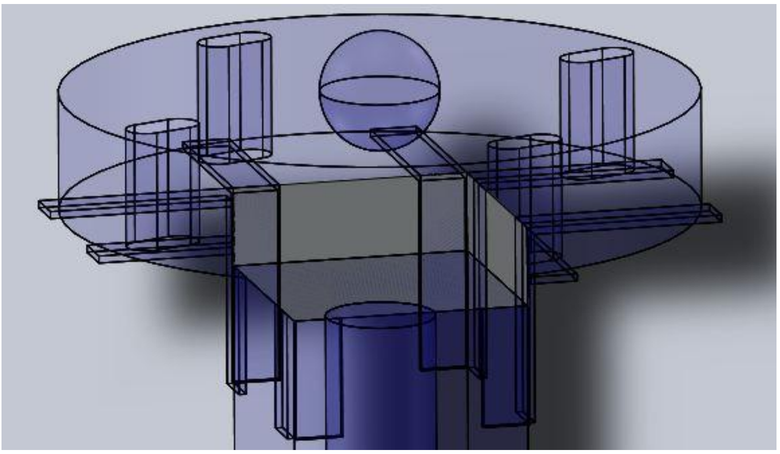


Figure 28: Compressive Material Between Hydraulic Press and bottom Alignment Plate

Table 23: Pugh Chart for Measuring Normal Force

	Weight	Compressive Material	Internal Beam Deflection	Compressive Load Cell
Safety	9	2	1	2
Cost	7	2	1	-1
Manufacturability	7	1	-1	2
Functionality	7	1	1	1
Ease of Operation	5	1	0	1
Maintenance	3	1	-1	2
Environmental Impact	1	0	0	0
Total Score		54	13	43

REASSESSMENT OF SELECTED CONCEPTS

After presenting our initial chosen concepts for each function to our colleagues and our sponsor, it was recommended to us that we reassess which components of the current device actually required redesign. This stemmed from the number of additions that we needed to make to the current device so that it incorporated some of the engineering specifications that were new to our redesign, including environmental control and improved safety shielding. From this high-level overview, we concluded that two of our chosen concepts could be adjusted to different concepts that would require less time and work and allow us to focus on the necessary additions to our device. The reasons for changing each concept and the new proposed concepts for each function are detailed below.

Convert Energy to Rotational Motion: As mentioned in the previous section, a gearbox may not be the most feasible solution to the problem. The Pugh Chart, Table 6, p. 23, shows that the second highest rated idea (highlighted in yellow) is the gear system with slanted teeth. At this point, after some research has been done, we are thinking that an open gear system may be more feasible given limited time and resources for producing a working device. A sample of a helical gear system that may be implemented on our device is shown in Fig. 29, below.

Figure 29: Helical Gears

Secure Test Fixture: The proposed concept for securing the tool fixture was changed from the interchangeable threaded collar to the removable rod through the tool fixture, which was a close second in our Pugh chart for securing the tool fixture. The main factor in our initial choosing of the interchangeable cup over the removable rod was safety, but because the removable rod concept is the one used on the current device, we were able to physically study it more closely. Upon closer inspection, the tool fixture in the removable rod design was manufactured to be a press fit into the shaft, providing additional stability and security. This gave us confidence that the removable rod design could be used again in our redesign, which would also eliminate the need to manufacture a new tool fixture. An illustration of the removable rod through the shaft design of our tool fixture is shown in Fig. 30 below.

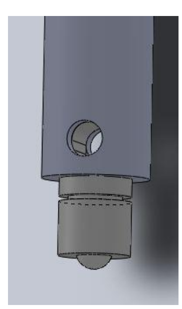


Figure 30: Tool Fixture

ALPHA DESIGN: FOUR BALL AND TWIST COMPRESSION TEST DEVICE

The entire device is made up of many different components and must perform all functions listed and described in the functional decomposition and in this report. It would be impossible to try to design the device without the breakdown of these functions and careful design of each individual component to perform each function. The complete device is shown in a 3-D CAD drawing in Fig. 31, below. Not all dimensions are final since in depth engineering analysis must be performed in order to ensure the safety and functionality of the design, but the model shows the basic structure and overall function of the device. The next step after Design Review #2 will be to perform the engineering analysis and a main focus of this investigation will be to ensure that the device will meet the required engineering specifications. To reiterate from Design Review #1, the machine should be able to meet the standard Four-ball test requirements given by ASTM standards and meet the industry standards of the TCT set by Tribsys Inc. Furthermore, our sponsor would like the machine to be able to exceed these standards. Ideally, we would like to see the Four-ball test run at 0.5 to 2 times the normal loads and speeds of the ASTM standard tests. Our sponsor would also like us to exceed the standards for environmental control of the tests. We would like the device to be able to vary the temperature of the lubricant from 0 – 150°C and the humidity of the test enclosure from 0% to 100% in increments of 10%.

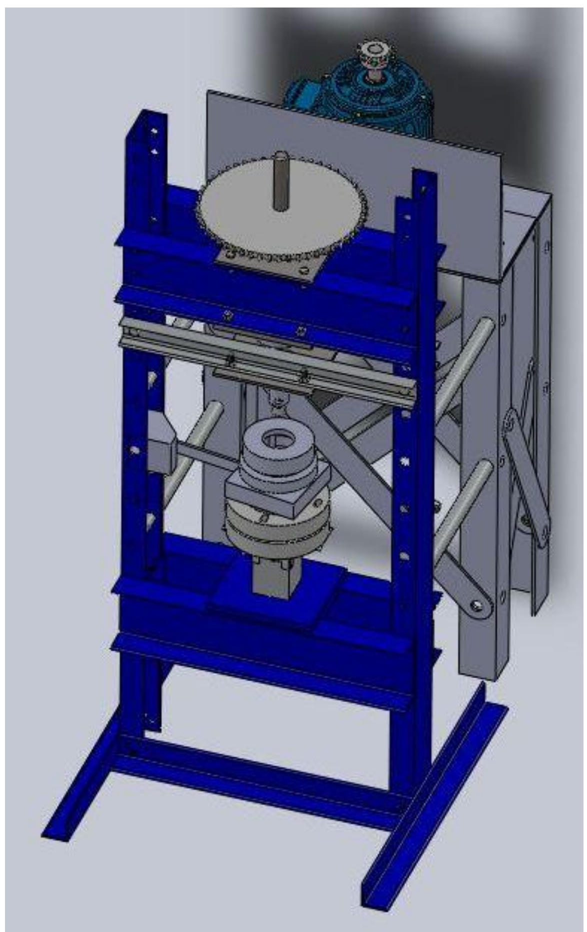


Figure 31: Full Prototype

ENGINEERING ANALYSIS

The construction and basic operation of our project encompasses four different fundamental fields of mechanical engineering: statics, dynamics, heat transfer and materials. In each of the four fields, we cover what applications they apply to as well as what equations and considerations we will make with regards to specific engineering functions of our design.

Statics

Under operation, there will be a number of loads, stresses and strains present and acting throughout the entire prototype. All of these forces result from the load application to the test interface. The hydraulic press will exert a load on the interface, resulting in pressures at the test interface. In regards to the Twist Compression Test, the pressure will be equal to:

$$P = F/A$$

P is the pressure in N/m^2, F is in N and A is in m^2.

In order to sense what normal load is being applied to the interface, a compressive block will be affixed with a strain gauge in order to provide a force reading. The sensitivity of the measurement will be determinant on what kind of material is used in the application. When we measure the lateral loads as a result of friction, a deflectable beam (sensitivity will be width-dependent) will be affixed with a strain gauge to provide an analogous range of force detection.

Because of the extent of normal loads being applied, coupled with friction-induced torques, residual moments and stresses will result throughout the prototype. FEA analysis procedures available to us through SolidWorks will allow us to see how severe these moments and stress concentrations will be and allow us to design additional supports and mechanical constraints to ensure that failure is preventable and accounted for.

Dynamics

The principle function of this prototype is the rotation of a tool fixture on the test interface. This is driven by an AC motor which has been provided for our use. The motor, by itself, is not capable of producing the complete range of speeds for our model, so we have to delve into gear ratios to complement the connection between the motor and drive shaft. The gear ratios that will be established will relate both speed and torque between the drive shaft and motor according to:

$$T_{output} \text{ (or } \omega_{output}) = N_{Driven}/N_{Driving} * T_{input} \text{ (or } \omega_{input})$$

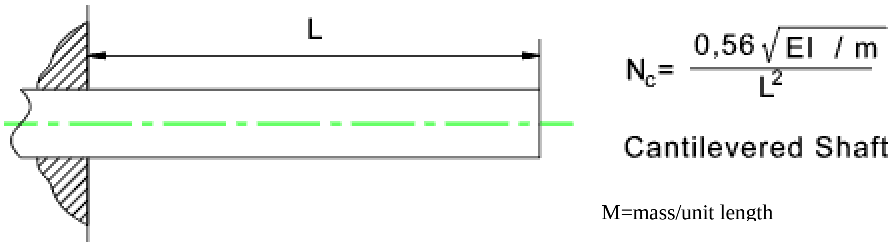
N is the number of teeth on either the driving or driven gear in the gearing setup. By having different gear ratios available, our motor will be able to alternate between overdriving in order to produce 3600 RPM at the drive shaft for the 4-Ball Test, or a greatly amplified Torque output for the Twist Compression Test.

These two tests that we are designing operate around Friction. One aim of each test performed with these kinds of machines would be to evaluate coefficients of friction across load and speed ranges. Frictional force is evaluated using the equation:

$$F_{Friction} = F_{Normal} * \mu$$

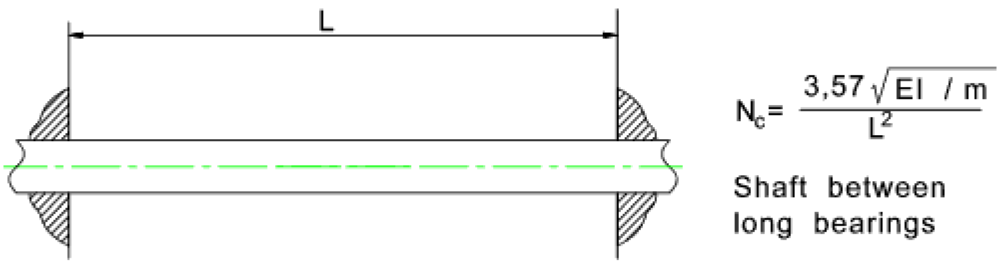
Additional overhead will be accounted for under force sensing because fluids like the oils and greases testable will no doubt be somewhat speed-dependent.

Under rotational motion, any small deviation of the center of mass of the rotation structure and the centroid of the structure will begin to induce what is called ‘whirling’. There are numerous variations of this concept that apply to the rotation of our drive shaft. What follow are examples and equations [18] which will be evaluated on our design and configuration as more thorough and accurate measurements are taken. Our aim is to dimension our model so that the maximum operating speed of 3600 RPM is outside the critical speed where whirling and severe vibrations would occur. N_c = critical speed (rev/s)



M=mass/unit length

Figure 32: Cantilevered Shaft



M=mass/unit length

Figure 33: Shaft Between Long Bearings

Heat Transfer

To evaluate and estimate the cooling and heating of our test interface, we will evaluate the heat transfer properties of the test environment using these equations:

$$Q = UA\,\Delta T$$

Where:

Q is the amount of heat transferred, W

A is the area for heat transfer, m²

ΔT is an effective temperature difference, ºK

U is the overall heat transfer coefficient, W/m².°K

$$U = \frac{1}{\frac{1}{h_1} + R_{f1} + R_w + \frac{1}{h_2} + R_{f2}}$$

Where:

h_1 and h_2 are the partial heat transfer coefficients, W/m².°K.
R_w is the thermal resistance of the wall, m².°K/W.
R_{f1} and R_{f2} are the fouling factors, m².°K/W.

The number of iterations of this calculation will be dictated by how the heating and cooling are ultimately implemented into the design.

PARAMETER ANALYSIS

When designing a mechanical device with many interacting components, it is important to analyze strength and structural integrity, as well as the optimum ways for accomplishing specific functions using an engineering approach. Without scrutinizing each part and function, it would be unknown whether or not the device would be safe or able to achieve its desired capabilities. The analysis we performed for each component or system of our device is detailed in this section. Each component or system required its own approach of analysis and this will be described for each. The components and systems that required evaluation were the following:

- Power Transmission (Gear system)
- Lateral Force Beams
- Lateral Force Beam Brackets
- Compressive Block
- Specimen Cup
- Heating Coils (Temperature and Humidity Control)
- Peltier Coolers (Temperature Control)
- Shaft Deflection
- Alignment Plates
- DAQ system
- Safety shield

Power Transmission
When analyzing the transfer of power from the motor to the driveshaft, the first thing that needed to be determined was the necessary gear ratios. In order to calculate these gear ratios, one for the TCT and one for the Four-ball test, we needed to calculate the expected maximum torques needed to drive the tool, and based on the torque output of the motor, determine a gear ratio. Obtaining the maximum torques needed was a simple procedure that required knowledge of the maximum normal force, F_n, for each test and an assumption of the maximum expected coefficient of friction, μ. Using these two parameters, the force of friction, F_f, was found. To find the torque, T, needed from the force of friction, we needed to know the radius, r, from the center axis of the tool to the point of contact with the test specimen. Multiplying this radius with the force of friction yields the torque needed to turn the tool for the test. See Appendix A for calculations.

Twist Compression Test
For the TCT, the torque was the limiting factor in calculating the gear ratio since the speed of the tests are so low (maximum of 30 rpm). Therefore, no matter what the ratio, the motor would have no problem turning fast enough since its full load speed is rated at 1765 rpm. Knowing that we want to test up to F_n of 100 kN for the TCT and expecting a maximum μ of approximately 0.15, we have determined that the maximum torque needed to run the test is 166.5 Nm. Knowing that the motor generates 11.8 Nm of torque, we have determined that the optimal gear ratio for the TCT is 1:14, meaning that the driveshaft gear should have 14 times as many teeth as the pinion, or motor gear. See Appendix D for all calculations.

Four-ball Test

For the Four-ball test, speed and torque were limiting factors in calculating the gear ratio since the test requires moderate normal loads, up to 16 kN, and very high speeds, up to 3600 rpm. For a first guess to the appropriate gear ratio, speed was used as the limiting factor. Knowing that the torque of the motor declines sharply after about 75% of the full load speed of 1765 rpm, we used the 75% speed in calculating the gear ratio to make use of the full motor torque. We divided the maximum test speed by the 75% full load speed to yield an approximate gear ratio of 2.7:1, meaning that the pinion, or motor gear, would have 2.7 times as many teeth as the driveshaft gear. However, we had to know how much torque the test would require to make sure the ratio would work. Using the same method as we did for the TCT, we determined that the maximum torque needed was 8.7 Nm. At a ratio of 2.7:1, the maximum torque generation would only be 4.35 Nm, half of what is needed. In order to achieve maximum torque, a gear ratio of 1.35:1 would be required, but then the maximum speed would only be 1800 rpm. Clearly one or both engineering specifications, the maximum speed or maximum normal force, had to be adjusted since both could not be achieved at the same time. Since both goals, maximum normal force of 16 kN and maximum speed of 3600 rpm are two times that of the maximums for ASTM test standards, see Table 34, p. 79, we decided to select a gear ratio of 2:1 in order to achieve a compromise of the maximum speed and normal force while still exceeding the highest ASTM standards. With a 2:1 ratio, the maximum obtainable speed is 2650 rpm and the maximum normal force is about 10.75 kN. See Appendix D for all calculations.

Lateral Force Beams

The lateral force beams serve as deflecting beams in which to measure lateral forces. Their main purpose is to bend, and therefore strain, so that an applied strain gage can record the data and be used to calculate the lateral forces produced. Logically, one would think to make the beams as thin as possible to create the largest amount of strain and therefore allow for more accurate readings from the strain gage. This, however, poses two distinct issues. One, if the beam was too thin, it would just bend to the point of losing contact with the deflecting block and the entire test cup assembly would be free to rotate, leaving a dangerous situation and no lateral force measurement. Second, if the beam was thicker to the point where it could maintain contact with the deflecting block, safety becomes a major concern due to the fact that the beam could yield and fracture if it was not thick enough. Analysis had to be performed to find the appropriate thickness to not only maintain safety but achieve readable strain gage measurements. After meeting with Dr. Bress, we learned that the minimum strain needed to produce an accurate reading for a strain gage is approximately 10^{-6}, so we made sure that we kept strain above this value when analyzing what thickness the beams should be.

The maximum expected torques of 166.5 Nm and 8.79 Nm for the TCT and Four-ball tests respectively that were calculated for the gear analyses were also used for the lateral force beams analyses. Because we modeled the system as a cantilever beam with a moment at the fixed end, we were able to find that the reactant force on the end of the beam was equal to the moment divided by the distance from the location of the moment to the end of the beam. Given the maximum moment (torque from the 100 kN TCT) of 166.5 Nm, the maximum reactant force at the end of the beam is 728.4 N. The maximum reactant force for the Four-ball test is 38.5 N, and the minimum reactant force (for the lowest normal force Four-ball test) is 0.07 N. Basic beam bending stress equations were used to determine what the ideal beam thicknesses should be given fixed a fix length and width of the beams. In addition to calculating the stresses from the maximum normal loads, it was important to calculate what kinds of strains to expect under the lowest Four-ball test normal loads to make sure an appropriate beam thickness was used in that case too. At 30 N of normal force, a strain of 1.9×10^{-6} would be accomplished with a 0.125” thick beam. Fig. 34, p. 45, shows the model used for the lateral force beams as seen from above, where *F* is the reactant force. See Appendix D for all calculations.

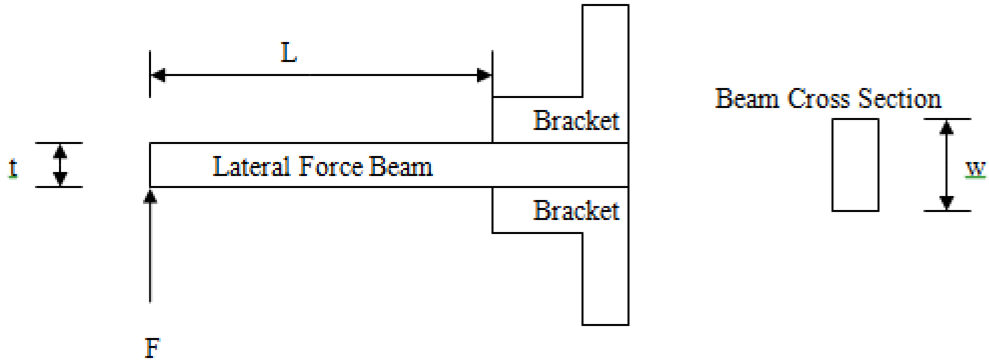


Figure 34: Lateral Force Beam Model

Lateral Force Beam Brackets

Similarly to how we calculated whether or not the lateral force beams themselves would be safe under maximum normal load conditions, we modeled the brackets supporting the beams as cantilever beams under bending stress. The force applied to the brackets is equal to that of the force applied to the beam itself. This maximum force is equal to 728.4 N. The bending stress equation was used in this case to verify that the chosen thickness of the brackets would be sufficient to withstand the maximum forces expected. In the case of these brackets, we were not concerned with strain since the only purpose of the brackets is to hold in place the lateral force beams that will be the measurement tool of lateral force output. Fig. 35, below shows the model used for the brackets, where *F* is the reactant force. See Appendix D for all calculations.

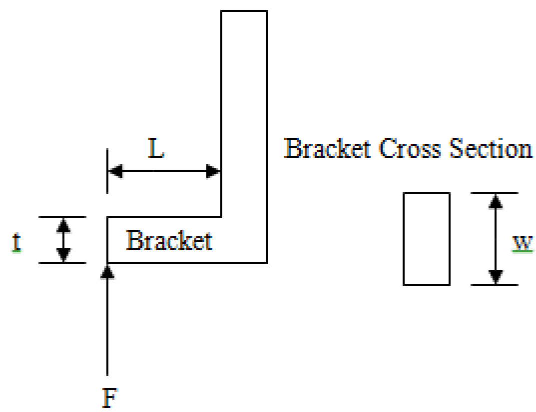


Figure 35: Lateral Force Beam Bracket Model

Compressive Block

The compressive block's purpose is to compress, and therefore strain, in the vertical direction so that an attached strain gage can measure the normal force applied to the device. Like the lateral force beams, safety and strain are key issues for the compressive block. Using the very basic stress and strain equations for tension or compression, we were able to determine the maximum stress and minimum strains for the tests and determine the most appropriate material for the block. Under the maximum normal load of 100 kN, the block material must be capable of withstanding a compressive stress of 17.2 MPa. Under the minimum normal load of 30 N, stress is not an issue for most materials, but a material had to be chosen that would compress by a strain of 1.0 x 10^{-6} or more. Fig. 36, p. 46, shows the model used for the compressive block, where F_N is the applied normal force. See Appendix D for all calculations.

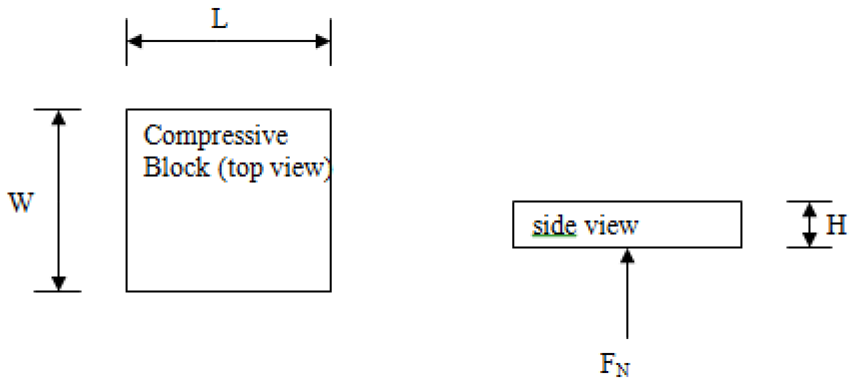


Figure 36: Compressive Block Model

Specimen Cup
In order to complete the final design of the specimen cup, we had to verify that it would be able to withstand the maximum expected forces of the two tests. The key concerns for the cup design were the compressive stresses put on the bottom of the of the cup from the TCT and the shear stresses applied to the teeth on the bottom of the cup that secure it to the steel plate that rests on a roller bearing. A similar model to the compressive blocks was used for the compressive forces applied to the bottom of the cup, but strain was not of importance, just the stresses applied. The maximum compressive stress was calculated to be 38.8 MPa based on the 100 kN of force applied to a 2" square steel plate placed in the bottom of the specimen cup for the TCT.

For the teeth, a cantilever beam model was used and basic shear stress calculations were used to ensure that the thickness of the teeth was sufficient under maximum load conditions. The shear stress was found from dividing the maximum torque from the TCT (166.5 Nm) by the distance from the center of the axis of the device to the center of the teeth. This force is 3277 N, but once this force is divided equally by the four teeth in the cup, the maximum shear stress applied to any one tooth is 819 N. If the teeth were to shear off from the bottom of the cup, there would be nothing holding the cup to the device and it would be free to rotate off, causing a catastrophic failure of the device, so it is crucial that the teeth can withstand the shear stress during the tests. Fig. 37, below, shows the model used for the teeth shearing, where V is the applied shear stress. See Appendix D for all calculations.

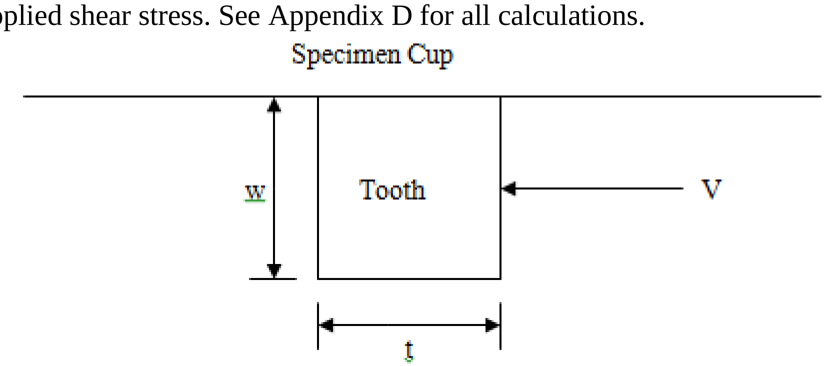


Figure 37: Specimen Cup Teeth Model for Shearing

Heating and Cooling
The heating and cooling elements that we decided on using in our alpha design would have to be able to output enough energy to sufficiently heat and cool the bottom of the test cup, and therefore the lubricant, to temperatures ranging from 0 to 150°C. Since we hope to place the heating and cooling devices within the bottom of the specimen cup, just below the test surface, a simple conductive heat transfer analysis

was all that was needed to determine whether or not a chosen heating or cooling element would be able to reach the temperatures necessary to get the test surface to its minimum of maximum temperature. If the element was placed directly in the middle of the 1" thick bottom of the specimen cup, the heating or cooling energy of the device would have to travel through 0.5" of solid aluminum in order to heat or cool the lubricant. Using a basic conductive heat transfer equation, we determined the cold and hot temperatures that the devices needed to reach in order to get the lubricant to 0 or 150°C. The coolest temperature needed to get the lubricant to 0°C is -8.0°C. The highest temperature needed to get the lubricant to 150°C is 165.3°C. Fig. 38, below, shows the model used for the heat transfer of the heating and cooling of the lubricant. See Appendix D for all calculations.

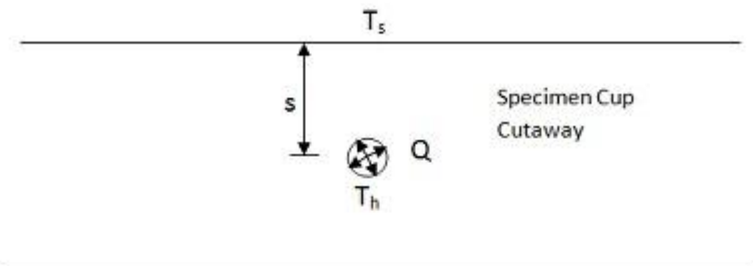


Figure 38: Heat Transfer Model

Alignment Plates

We chose to use the alignment plate system that existed on the current prototype for our prototype and final design because of its innovative design and immediate availability. However, our sponsor did express concern over the structural integrity of the alignment plates when they were set up for the twist compression test. As seen in Figure 39, below, the twist compression test alignment plate design utilized a spherical metal ball placed into grooves in each of the alignment plates.

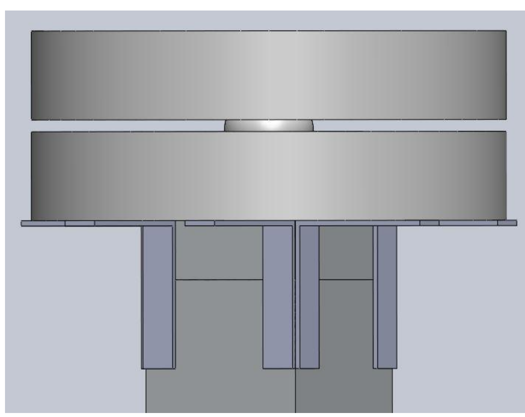


Figure 39: Metal Sphere in Twist Compression Test Alignment Plate Setup

This design was meant to allow for a slight offset from horizontal in the angle of contact between the annulus and the test plate, as seen in Fig. 40a, p. 48. The concern with this design was that, at the maximum offset angle, the ball might be ejected from between the two plates. However, after inspection of the design, another concern arose regarding the stability of the components above the test plate. If the vertical axis through the annulus is too far offset from the vertical axis of the metal sphere (Fig. 40b, p. 48), a moment could occur in the alignment plates due to the misalignment of the vertical forces in the prototype. This could lead to bending moments in the component structures above and/or below the alignment plates, which would be a catastrophic failure at high load.

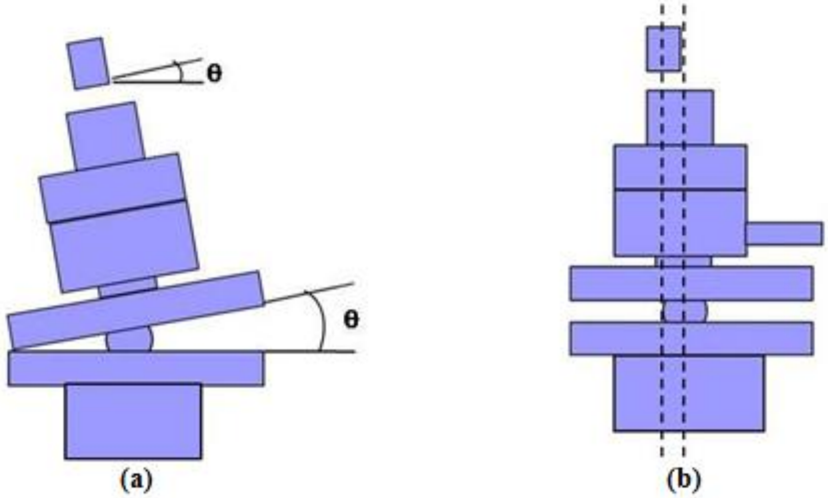


Figure 40: Twist compression test alignment plate stability concerns showing (a) maximum annulus offset angle and (b) maximum vertical axes offset

Two separate analyses were done to study the stability in each scenario. In the first, geometry was used to examine the maximum angular offset of the annulus and the possibility of the ball ejecting. A simplified schematic of the alignment plates system showing the plates when they are parallel and when they are at their maximum angular offset is shown in Fig. 41, below. As seen in this portrayal, the maximum opening between the two plates (when their opposite sides come in contact) is 0.8 in., while the diameter of the ball is 1.5 in. Using geometry, it can be seen that the maximum offset angle of the plates (and the annulus) is only 5.71º. Additionally, the grooves holding the ball in place are 0.55 in. deep into the alignment plates, so we can say with high confidence that the ball will never be ejected from between the plates.

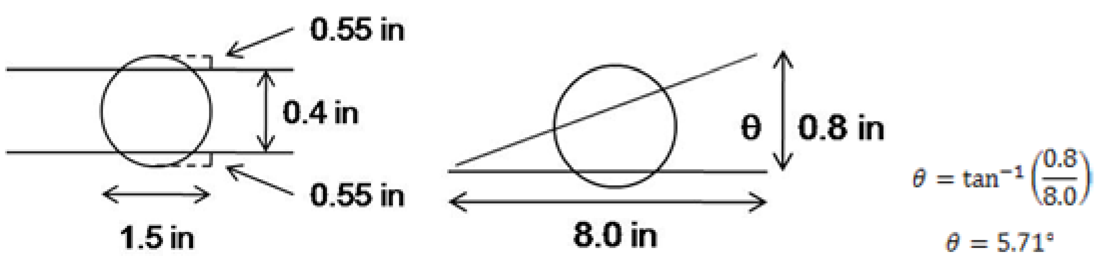


Figure 41: Maximum opening between alignment plates and offset angle

In the second stability analysis, the maximum offset of the vertical axis of the annulus from the vertical axis of the ball was calculated to see if a large enough moment would be created to cause failure. As seen in Fig. 42, p. 49, the 2.0 in. diameter specimen cup restricts the vertical axis of the 1.0 in. annulus to move a maximum of 0.5 in. from the axis of the ball. When the vertical axis of the annulus is at this maximum 0.5 in. offset, it still passes through the 1.5 in. diameter ball at 0.25 in. inside the surface of the ball. Also, at this 0.5 in. offset, the vertical axis of the annulus passes through the compressive block and aluminum inverted cup that stabilizes the ram of the press. These two components form a 3 in. by 3 in. by 7 in. block centered on the axis of the ball that not only stabilizes the ram of the press, but also should serve to cancel the maximum moment of 1270 N-m that can be produced by a 0.5 in. axis offset and 100 kN of force. This maximum moment was calculated in the same manner as the maximum torques when calculating the gear ratios. This information leads us to believe that it is highly unlikely that a bending moment would lead to failure in the components near the alignment plates.

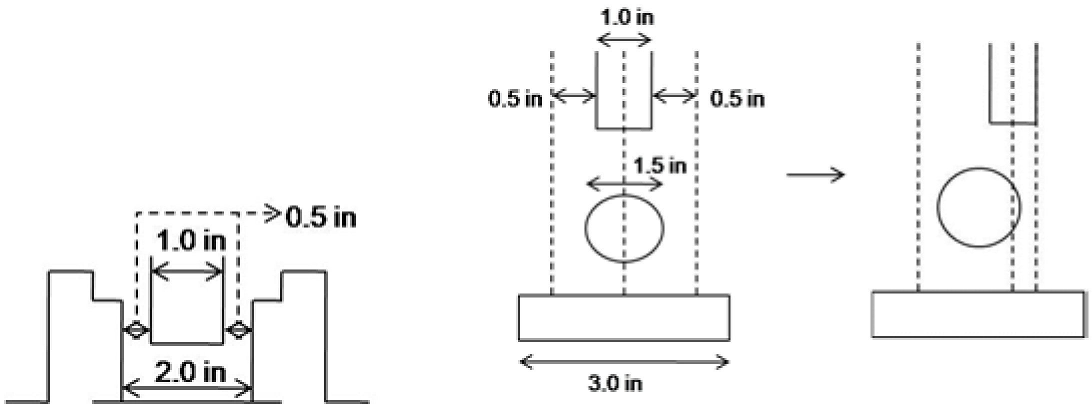


Figure 42: Maximum offset of annulus axis from ball axis

Shaft Deflection

One collection of concerns of both our team and of our sponsor is the possibility of deflection or buckling of the three long shafts that are present in our design. Under load, there is always the possibility that any slight deflection could have a potentially catastrophic result. In order to evaluate the severity of deflection in a particular section of beam, we performed worst-case calculations. The four segments are 1) the ram of the shop press, 2) the segment of the main driveshaft that encompasses the tool fixture, 3) the segment of the main driveshaft that is attached to the gears, and 4) the shaft extending from the motor that will be affixed with gears. Fig. 43, p. 50 shows the shafts. In order to establish worst case scenarios, a number of assumptions will be made. For the ram and tool fixture segments, it will be assumed that a lateral force will be introduced by the available tilting of the alignment blocks during a twist compression test. In order to underestimate the capabilities of the prototype and exaggerate the forces that will be present, simplifications to complex forms will amount to taking a single shaft of the smallest diameter present and low-end estimates of Young's Modulus. In addition as previously mentioned in the analysis of the alignment plates, the maximum travel of the top block would create a 5.7° angle, and therefore a resultant lateral force. For deflection evaluation, the beams are modeled as cantilever beams. Shaft deflection and buckling equations were used to evaluate deflection (δ) and the critical force required to instigate buckling in a beam (F_c). In the equations, K is a factor based on the end supports of the beam. In this case, $K = 2.0$ because one end of the beam is fixed while the other is free to move laterally. For the deflection in beams 3 and 4, the situation evaluated would be the case where beam 3 is frozen, when climbing would be most severe. The resultant force on the cantilevered-beam model is derived from the maximum torque we're designing to output evaluated as a point force at the top end of the dimensions of the gear. Table 24, below, shows the resultant deflections on each of the beams.

Table 24: Results of Shaft Deflection Calculations

Givens				Results		
Shaft Segment	D (m)	L (m)	E (GPa)	F_R (N)	δ (m)	F_c ($\cdot 10^6$ N)
1	0.0349	0.1207	210	$9.93\cdot10^3$	0.0011	2.5901
2	0.0254	0.0762	70	$9.93\cdot10^3$	0.0031	6.0776
3	0.0254	0.0762	210	728	$2.2243\cdot10^{-5}$	
				77.4	$1.4712\cdot10^{-6}$	
				($2.0783\cdot10^5$)	0.00635	
4	0.0191	0.0762	210	728	$6.9566\cdot10^{-4}$	
				77.4	$7.3962\cdot10^{-4}$	
				($6.6452\cdot10^4$)	0.00635	

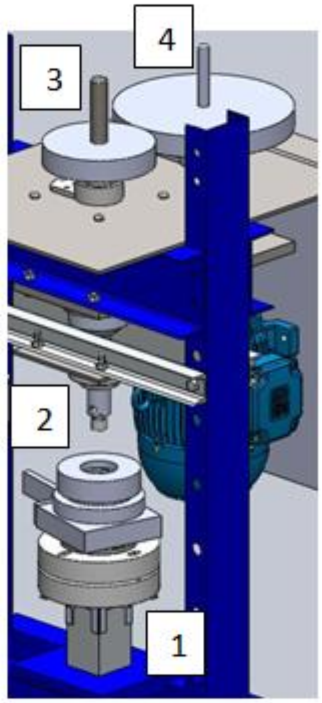


Figure 43: Numbered shafts that could possibly deflect

As is evidenced in the results given gross overestimates of the possible forces that will be present under loading, we do not need to worry about beams deflecting to a point where catastrophic failure will occur.

Safety Shield

The safety shield acts as a protective barrier against any broken parts that may fly off the device in the case of failure and therefore must be able to withstand high forces. In order to safely choose a material for the safety shield, an analysis of the maximum stress the material would have to endure was necessary. To do this, we looked at the maximum force of impact the shield would ever have to tolerate. We took into account that the specimen cup acts as a shield for the balls or plate, meaning we didn't have to worry about those impacting the safety shield, but in the case that the stress on the lateral force beams became too much, we had to look at the maximum force one could impact the shield with. To do this, we determined the maximum mass and speed of the beam if it were to break off, the momentum they would have and the impulsive force they would hit the shield with. Then, we divided this force by the smallest contact area of the beam and found the stress that the shield would have to endure. From this calculation we were able to choose a material for our safety shield. See Appendix D for all calculations.

What We Learned

Over the course of this project, we as individuals and as a team have learned a tremendous amount about the design process, including material and manufacturing process selection, designing for environmental sustainability, and practicing the safest possible work habits. In this section we will outline briefly the key points that we have learned about each one of these categories.

Material and Manufacturing Process Selection

Analyzing material and manufacturing process selection was in fact very beneficial to us since we chose two critical components in which to choose material for. When we were designing the compressive block and lateral force beam systems, and choosing materials to use, we had not completed the assignment, seen in Appendix C, and used multiple iterations of math in Microsoft Excel to determine the appropriate materials and thickness beams to use for the force measurement systems. We ended up using three different materials for the compressive block and four different lateral beams made out of two different materials. After completing the selection assignment, we realize the benefits of using a software program and determining material indices to choose materials. After completing the assignment we see that we could have gotten away with just one compressive and two lateral beams block (able to withstand the highest forces and produce the minimum strains at all times). Starting a material selection process requires some thought and careful planning in order to properly constrain the component and choose a correct material index, but the benefits are clear. You can spend less time solving equations to find the right material and less time machining if you choose the best one.

The manufacturing process selection assignment was only marginally beneficial to us in comparison to the material selection assignment. This is because we were looking at production processes for a production run of 100 instead of producing our one-off prototype, and the fact that for the time we had available to us, machining the parts using the mill in our shop was the smartest decision. In terms of our long term careers as engineers however, we have all learned the benefits of using the CES software for assisting in choosing appropriate, cost effective manufacturing solutions for individual components.

Design for Environmental Sustainability

Before taking ME 450 and delving into this project, we had little knowledge of how to know if our design choices and material selections would affect the environment. After completing the assignment, we now understand, at least to a degree, how to use the SimaPro software package to compare material choices. The software makes it very intuitive to analyze the material choices and their impacts on the environment. Given the variety of different charts the program outputs, one can chose a specific emission to analyze or the environmental impact on a whole. With more time using the program, we understand that the possibilities are endless when designing for the environment and that determining impact for life cycles of products can also be performed. We also know that it is difficult to choose a material based on the SimaPro software alone because there are so many other factors that go into choosing materials like manufacturing processes and cost which may outweigh the benefits of using a slightly more environmentally friendly material.

Design for Safety

Safety in engineering and manufacturing is stressed to students from the beginning of their college careers, but the safety reporting that is a staple of ME 450 takes safety to a new and necessary level. At first, we collectively thought that the need for safety reports was excessive and disliked having to do them. We wrote three throughout the course of the semester, one for disassembly, one for manufacturing and fabrication, and one for testing. As we progressed into our fabrication of parts and assembly of the device, it became apparent that the reports were beneficial to our progress and kept us safer due to the fact that they had to be approved by not only our instructor and graduate student instructor, but by Bob Coury, our machine shop expert whose primary concern is safety. Getting a report approved assured us that our methods for completing tasks were going to be very safe if carried out with care and caution. Although the safety reports were long and often tedious, they do have the potential to speed up the actual process of fabrication or testing since they force team members to know exactly how to perform a given task. This rigorous planning and writing will only help all of us in the professional world when we have to perform potentially hazardous and time consuming tasks.

FINAL DESIGN DESCRIPTION

During the parameter analysis of all components and systems, we began selecting and parts for purchase, materials for fabrication and final dimensions and specifications of all components. The engineering examination made it possible for us to intelligently design all new parts and purchase parts that we know will work for our device. This section of the report will describe how we arrived at the appropriate solutions and conclusions for each piece of our device, including power transmission, test cup design, lateral and compressive force measurement, temperature and humidity control, safety shielding and Data Acquisition (DAQ) of forces, shaft speed, temperature, and humidity. It encompasses everything that is necessary for all engineering specifications to be met to the highest degree possible. A full description will be included for each piece or system of the device and where applicable, engineering drawings will be shown.

Power Transmission

As described in the Concept Selection section, we first selected to use a gearbox to transfer the rotational motion of the motor to the driveshaft. With the two gear ratios calculated, we went on to trying to select an appropriate gearbox that would allow us to run either the TCT or Four-ball test. As we searched, it became apparent that a 2:1 ratio gearbox was readily available in a variety of styles including parallel input/output shafts, right-angle, and bevel, but a 1:14 ratio gearbox was uncommon and usually would only be available in a worm gear style due to the large ratio. This meant that there would be no way to have both gear ratios housed in one box and we would have to interchange the two depending on what test was being run. As we searched more, we realized that not only were the correct gear ratios hard to find, but we have a set driveshaft diameter of 1 in. and a motor shaft diameter of 1.125 in. and the gear box would need to be able to house these diameters, or we would have to purchase the right size shafts to fit the gear box, and couple the shafts to the motor and drive shafts. We decided to inquire about ordering a custom gearbox but were informed that companies would not be willing to invest the time and money into designing a custom gearbox and only produce one of them to be used in our prototype.

Open Gear Selection

Due to the fact that finding an appropriate gearbox or even two separate gearboxes that matched our gear ratios, input/output shaft diameters and power ratings was proving to be extremely difficult, we began exploring the option of replacing the current chain drive system with an open gear system using interchangeable gears to match the required gear ratios as first described in the Concept Selection section. In this case, we would eliminate the complexity associated with gearbox parameters but add several components that would be necessary in order to support open gears. Since the search for a gearbox was extremely difficult, and stock gears of many sizes are readily available, we focused our attention on choosing appropriate gears for the implementation of an open gear transmission.

Determining what gears to use was no easy task, but needed close attention to detail to ensure the device will work as expected. We want to mount the motor in the upright position as it is on the current prototype and make sure that it remains in a fixed position for each of the two tests. This means that the center distance between the two interchangeable sets of gears must be the same. Additionally, meeting the 1:14 gear ratio is limited by the amount of space available between the shop press supports where the driveshaft gear will be placed. Lastly, the bore size, or inner diameter of the gears had to meet the shaft diameters of the motor and drive shafts. These three constraints had to be considered when selecting gears and led to slight deviations in the gear ratios we had originally chosen. After modifications had been made to the ratios so that we could select stock gears in order to keep costs lower, we ended up with ratios of 1:12 and 1.6:1 for the TCT and Four-ball tests respectively. See Appendix Afor calculations. After receiving a quote from several companies and weighing the costs and benefits between the open gears and finding a gearbox, we decided the extra work of implementing the open gears would be our best option to ensure that we are able to produce a working prototype and test it by the Design Expo on April 15. The three largest gears are made out of cast iron, while the smallest gear (pinion for the TCT) is made out of steel. Two gears are manufactured by Martin Sprocket and Gear, Inc. and two are manufactured by Browning Gears. Table 25, p. 53, gives specifications for all of the gears.

Table 25: Gear Specifications

	TCT Pinion	TCT Gear	Four-ball Pinion	Four-ball Gear
Teeth	12	144	96	60
Bore (in.)	0.75	1	0.75	1
PD (in.)	1.5	18	12	7.5
Pressure Angle (°)	20			
Center Distance (in.)	9.75			

In order to successfully implement the open gears, several other components have to be purchased and/or fabricated. First of all, since AC motors do not like lateral forces applied to their shafts, the pinion gear will have to be attached to a separate shaft that will be fully supported with two ball bearings. The new shaft and motor shaft will have to coupled using a flexible coupler to be sure that no lateral force is applied to the motor shaft. In order to support the bearings, steel plates will be attached to the shop press support structure and flange mounted ball bearings will be used to support lateral forces from the interactions of the gears. Finally, key stock will be needed to secure the gears to their respective shafts, guaranteeing that they cannot slip while rotating. A bill of materials for the gears and accessory parts needed to implement them is given in Table 26, below.

Table 26: Bill of Materials for Gear System

Component	Manufacturer	Part Number	Cost
TCT Pinion Gear	Browning Gear	YSS812	$28.86
TCT Driveshaft Gear	Martin Sprocket, Inc.	TC8144	$314.27
Four-ball Pinion Gear	Martin Sprocket, Inc.	TC896	$208.5
Four-ball Driveshaft Gear	Martin Sprocket, Inc.	YSS860	$136.57
Flange Mount Ball Bearing (x2)	Import	05550637	$49.34
Pinion Gear Shaft	McMaster-Carr	1497K956	$25.96
Flexible Coupler	DieQua Corporation	EK2-60	$132.00
Key Stock	McMaster-Carr	98535A150	$3.55
Steel Sheet	McMaster-Carr	1388K131	$219.58
Total Cost		**$1130.43**	

Normal Force Measurement System

For our alpha design, we chose to use a compressive block with a strain gage attached to it in order to measure the normal forces applied to the tool/specimen interface. We have determined the maximum stress that the material needs to endure (for the TCT) is 17.2 MPa. We originally wanted to use PVC as the material for the compressive block material, which would be appropriate due to the fact that PVC has a compressive strength of about 55.5 MPa [36]. However, if subjected to this force for any length of time, we were informed that creep may become an issue, so PVC may not be the material of choice for the high normal force tests, and that aluminum or steel would be better for those applications. Since 6061 aluminum has a compressive stress of 249 MPa, we knew it would be fine under the highest loads, but were concerned that it may not strain enough to be detected by standard foil strain gages, which require a minimum strain of 1.0×10^{-6}. Therefore, we had to analyze the force range that the aluminum block could be used for and still detect the minimum required strain. At 100 kN of normal force, the aluminum block would compress to a strain of 2.5×10^{-4}, well above the minimum required. We wanted to use the aluminum block for the widest possible range, so we calculated how low the normal force could be so that aluminum was still suitable. This low normal force was found to be 401 N. We decided to start the use of aluminum at normal forces of 1000 N or more. At 1000 N of normal force, the aluminum would compress to a strain of 2.5×10^{-6}. We also checked to verify that the strain of the PVC would be above the minimum required at the lowest normal force of 30 N. At this normal force, the PVC would compress to a strain of 1.9×10^{-6}, above the minimum required. Also, at the maximum normal force we would use PVC for, the stress on the block would be 0.172 MPa.

As mentioned, foil strain gages would be used to detect the strain of the compressive blocks, and they need a minimum of 1.0×10^{-6} strain to be effective. These strain gages will be discussed below under Lateral Force Measurement.

Specimen Cup Assembly

The specimen cup assembly design includes four 6061 alloy aluminum components: specimen cup, TCT ring, and outer collar. The cup itself houses the bottom three balls of the Four-ball test or the 2 in. square steel plate of the TCT. It also bears the compressive normal loads of each test. Fig. 44, p. 55 shows the parts of the four parts associated with the specimen cup assembly. We originally designed the cup to have a 1 in. thick bottom surface and after determining the maximum compressive stress on the cup to be 38.8 MPa, we saw no need to change the design, since the compressive strength of 6061 aluminum is 249 MPa [36], giving us a safety factor of 6.4. Along with the compressive stresses from the normal forces, we analyzed the teeth design against shear stress. The four teeth are located underneath the bottom of the cup and fit into machined grooves in a steel plate that sits below the cup. This plate rests on a roller bearing and houses the brackets for the lateral force beams as well. We determined that the maximum shear stress the teeth would have to endure would be 3.8 MPa. The maximum allowable shear stress for 6062 aluminum is 120.5 MPa [36], half of the yield stress. Due to a safety factor of 31.7, we felt no need to change the design.

The rings and outer collar work in conjunction with one another to secure the three balls or steel plate to the bottom of the cup depending on the test. Both rings simply rest in the cup on top of the specimen. They are designed to be secured to the cup by pressure applied by the outer collar, which screws onto the cup. As the collar is tightened, it presses down onto the top of the ring and the pressure is applied to the specimen, ensuring that it doesn't move during testing. The Four-ball collar sits over the bottom three balls and relies on the pressure from the outer collar alone to secure the balls. The TCT ring however, has built in teeth that sit into milled grooves on the bottom surface of the specimen cup, disallowing rotational motion of the steel plate used in the TCT. The rings and outer collar do not support the lateral force of the tests and therefore no analysis was needed to make sure that they would not fail under various test conditions.

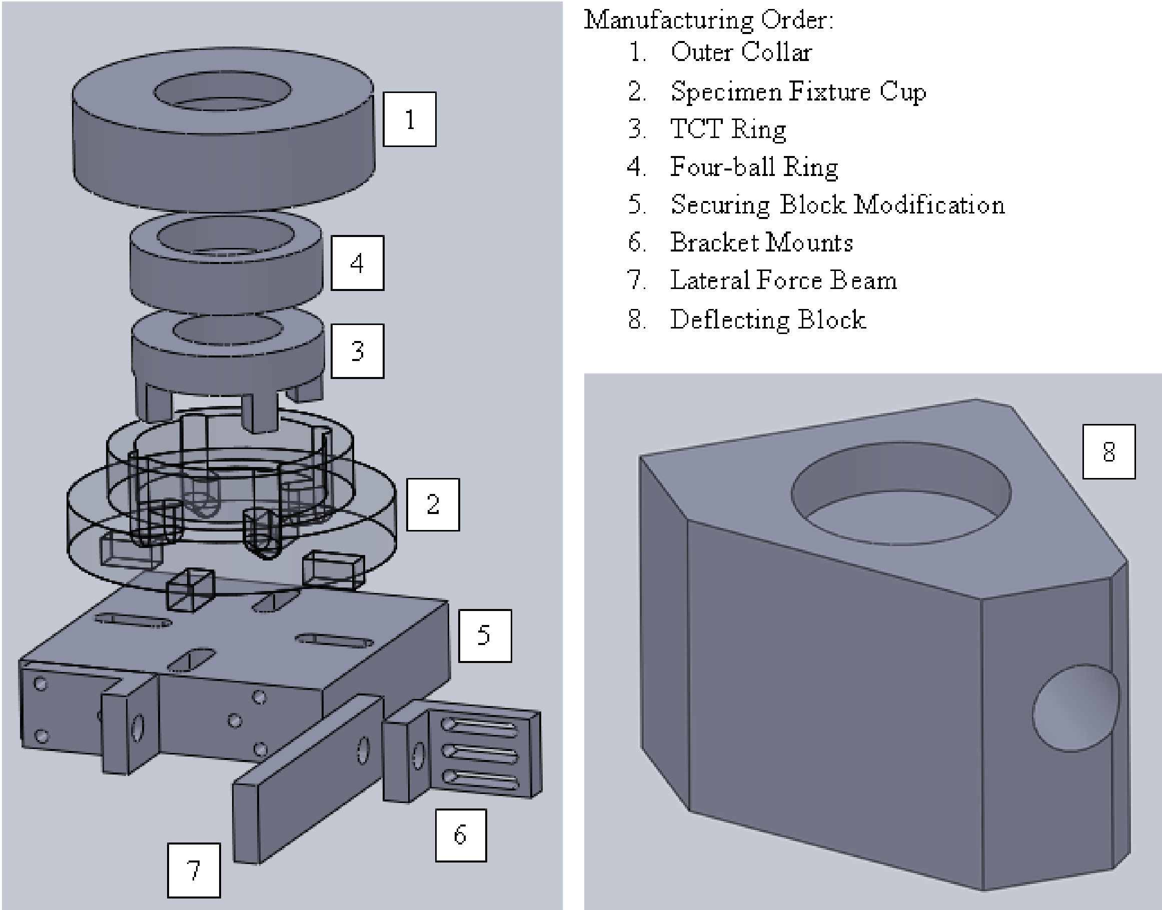


Figure 44: Specimen cup and lateral force measurement system assembly

Lateral Force Measurement System

The lateral force system we developed for our alpha design includes a 1 in. thick, strain gage equipped beam connected to a steel plate that supports the specimen cup via brackets, and a deflecting block that allows the beam to be bent so that strain can be measured. The two brackets are attached to the steel plate and securely hold the beam in place. Figure 44 above shows the parts associated with the lateral force measurement system. In order make sure that our design would be adequate for safety and strain detection, we performed bending stress and strain analysis on the beam and brackets which were modeled as 0.5 in. thick cantilever beams. For the TCT, a maximum force of 728.4 N would be put on the end of the beam. With a 1 in. thickness, the maximum bending stress would be 20.3 MPa. 6061 aluminum has a maximum compressive stress of 249 MPa, giving a more than adequate safety factor of 12.3. We next checked to ensure that the beam would strain enough for the foil strain gage to record an accurate reading at the lowest normal force setting of the Four-ball test of 30 N. For this low normal force, the reactant force at the end of the beam would be 0.07 N, yielding a strain of 2.9×10^{-8}, much below the necessary minimum strain. Using the Four-ball test normal force of 2 kN, the 1 in. thick beam would deflect enough to output a strain of 2.0×10^{-6}, above the necessary minimum strain.

Because we need to be able to record lateral forces at normal loads less than 2 kN, another beam will need to be used. This second beam has a thickness of 0.125 in. so that it will be more sensitive to strain at lower forces. At the low strain of 30 N for the Four-ball test, the thin beam will give a strain of 1.9×10^{-6}, more than what is required for the strain gages. It also needed to be verified that the thin beam would be safe operating at normal forces up to 2 kN, so the stresses were calculated at 2 kN of normal force. At this normal force, the maximum reaction force of 14.6 N on the beam would occur during the

TCT. Under this force, the maximum bending stress on the beam would be 26 MPa, giving a safety factor of 9.6 for the stress.

For the brackets themselves, they will have to endure the same TCT maximum force of 728.4N. At this force, the brackets would undergo 18.1MPa of stress, giving a safety factor or 15.5. A bill of materials for the normal force measurement, specimen cup, and lateral force measurement systems is given in Table 27, below.

Table 27: Summary of Material and Cost

Component	Manufacturer	Part Number	Cost
PVC block	McMaster-Carr	8788K57	$26.39/foot
6” diameter by 6” length Aluminum round	Alro Steel Corporation		$93.23
4” diamteter by 6” length Aluminum round	Alro Steel Corporation		$46.52
6” length x 4” square Aluminum block	Alro Steel Corporation		$56.90

We have decided to use the same method of measuring normal and lateral forces as the previous team by using foil strain gages. We are implementing four Vishay C2A-13-250lw-350 strain gages total [24]. One on each of the compressive material blocks and one of each of the lateral force beams. The strain gages are approximately .25 in. long and have a resistance of 350±.6% ohms [24]. The strain level of ±1700με with a range of ±3% [24] is enough to be able to detect the strain levels we are expecting. In order to measure strain with a bonded resistance strain gage, it must be connected to an electric circuit that is capable of measuring the minute changes in resistance corresponding to strain [32]. Strain gage transducers usually employ four strain gage elements electrically connected to form a Wheatstone bridge circuit [32]. To implement this bridge system we are purchasing a bridge module that has these bridges already built into it. The module NI 9237 from National Instruments has the desired specifications to meet our needs [23]. It has 4 individual channels, 24-bit resolution, ±25 mV/V analog inputs with RJ50 connectors, 4 simultaneously sampled analog inputs; 50 kS/s maximum sampling rate and a programmable half- and full-bridge completion; up to 10 V internal excitation [23]. Each one of the 4 channels will be occupied with its own strain gage. This module will then be placed into the DAQ chassis and programmed using LabVIEW.

Temperature Measurement and Control

Our alpha design for temperature measurement and control included using three components: a heating coil to raise the temperature of the lubricant from ambient to the maximum temperature of 150°C, Peltier coolers to lower the temperature of the lubricant from the ambient to the minimum temperature of 0°C, and a thermocouple placed within the lubricant to measure the temperature. Based on the parameter analysis we developed and the final design of the specimen cup, we were able to finalize the temperature measurement and control system by choosing the heating and cooling elements needed and a functional thermocouple that would fit into our design.

Based on the high and low temperatures of 165.3°C and -8.0°C needed to get the lubricant to the highest and lowest temperatures, we chose to use a small, 25 Watt cartridge heater to increase the temperature of the lubricant and the Peltier coolers that were purchased by the previous team to decrease the temperature of the lubricant. The cartridge heater can get up to 760°C while the Peltier coolers can reach a low temperature of -15°C. With the components chosen and verified to work, we chose a thermocouple that would be able to read the full temperature range and input the information into LabVIEW. We have decided to use the same method of measuring the testing area’s temperature as the previous team using a thermocouple. We are implementing an Omega TJ96-CASS-18G-12 thermocouple. This thermocouple is ‘K’ type, has a diameter of 1/8 in. and a length of 12 in [26]. They have an accuracy of ±1.1° of accuracy (or .4% of temp) and a temperature range of -200°C to 1250°C [26]. These thermocouple specifications fit into our needs of temperature expectations. In order to measure the temperature using the thermocouple, one of the wires needs to be kept at a set (reference) temperature while the other wire measures the temperature of the test area. This will output a voltage difference which is temperature dependent. To implement the thermocouple, we are using another module to insert into the DAQ chassis. The module NI 9211 from National Instruments has the desired specifications to meet our needs [25]. It has 4 individual channels, 14 S/s, 24-bit resolution; 50/60 Hz noise rejection, ±80 mV analog inputs, and an operating temperature range of -40 to 70°C [25]. This module will be placed into the DAQ chassis and programmed using LabVIEW.

Humidity Measurement and Control

In our alpha design, we selected to use a humidity meter that would be placed within the limits of the safety shield and allow for data acquisition into LabVIEW for the measurement of humidity. For humidity control, we selected to use a small beaker of water as the source for more humidity. Our thoughts were to place the small beaker on top of the deflecting block of the lateral force measurement system and place a second small 25 Watt cartridge heater within the block to heat and therefore evaporate the water in the beaker, creating water vapor within the limits of the safety shield and increasing the humidity of the test. We have changed our method of measuring humidity in our Alpha Design from a simple humidity meter that only displays the current humidity to product that measures humidity and relays the information to LabVIEW. We have decided to use a Vaisala INTERCAP HMP50 Humidity and Temperature Probe [31]. This probe has a humidity measurement range of 0 to 98% relative humidity [31]. The accuracy from 0 to 90% relative humidity is ± 3% relative humidity, and from 90 to 98% relative humidity the accuracy is ± 5% relative humidity [31]. To implement this humidity probe we will be purchasing a third module to insert into the DAQ chassis. This National Instruments 9201 module is an 8-channel, ± 10 V input range, 500 kS/s aggregate sampling rate, 12-bit resolution analog input module [30].

Shaft Speed Measurement

We have decided to use an infrared tachometer to measure the shaft speed. The previous semester’s team used an optical encoder that was short-circuited during a test. Due to the fact that tachometers cost less than optical encoders and still have the function we need of measuring shaft speed, we are now implementing a tachometer into our device. The tachometer is mounted and its infrared light is directed at the shaft that has a piece of reflective material on it. As the shaft spins, the light senses the reflective material and measures the speed. This Monarch Instrument, IRS-P infrared sensor measures speed from 1-999,990 rpm, has an operating distance of .5 to 1 in., is able to operate at a temperature range of -23° to 100°C and produces an analog voltage output [28]. To implement this infrared tachometer we are using last semester’s NI USB-6009 Multifunctional DAQ system [27]. This DAQ system has 8 analog inputs (14-bit resolution, 48 kS/s), 2 analog outputs (12-bit resolution, 150 S/s); 12 digital I/O; 32-bit counter and is compatible with LabVIEW [27]. This DAQ system will be hooked up to and programmed using LabVIEW.

Motor Control

We have decided to use the same method of controlling the motor as last semester’s team using the Inverter Drive ABB ACS150 motor control [29]. This motor control is 0.5-3 HP, 200-240 V, 3 phase motor compatible [29]. It has 1 analog input, 5 digital inputs and 1 relay contact set [29]. This motor control and its specifications have not changed since our Alpha Design. To implement this motor control we are using the same DAQ system as above in shaft speed measurement. This NI USB-6009 will also be compatible to control the motor through the use of LabVIEW [27].

Heating Cartridge
To control the temperature of the testing area and the water for regulating humidity, we have decided to use a McMaster-Carr Miniature High-Temperature Cartridge Heater [33]. One heater will be inserted into the specimen cup and one into deflection block. These cartridges are made for very small hot plates or spot heating of small dies [33]. Their diameter is 1/8 in., and has a maximum wire lead temperature of 249°C [33]. These product specifications fit our engineering specifications and sponsor requirements of controlling the temperature from 0 - 150°C. To implement the heating cartridges we will be purchasing a temperature controller from Thermal Corporation [34]. This Basic Unit – No Alarm device, model number 5040-10, accepts temperature and process inputs and offers a choice of three kinds of outputs to meet a wide variety of needs, whether it's a single or dual outputs include relay, SSR driver, or 4-10mA [34].

Data Acquisition System
The DAQ system we have decided to implement is the NI cDAQ-9174 [22]. This system holds up to 4 C Series I/O modules, runs analog input modules at different rates with multiple timing engines, has four general-purpose 32-bit counter/timers built into the chassis and is compatible with LabVIEW [22]. Based on our need of controlling normal and lateral force using a strain gage and temperature control using thermocouples, this is the best option that we have decided to implement. We are also implementing a DAQ card that the previous team used last semester, the NI USB-6009 [27]. This has the capabilities of measuring the shaft speed using the tachometer and controlling the motor using the AC drive speed motor controller. It has 8 analog inputs (14-bit, 48 kS/s), 2 analog outputs (12-bit, 150 S/s), 12 digital I/O, 32-bit counter and is compatible with LabVIEW [27]. Table 28, below, summarizes the components and their prices.

Table 28: Summary of Data Acquisition Products

Function	Module	Price ($)	Product	Model	Price ($)
Normal and Lateral Force Measurement	NI 9237	1149.00	Strain Gage (Vishay)	C2A-13-250lw-350	Pk. of 10 for 71.25
Temperature Measurement	NI 9211	329.00	Thermocouple (Omega)	TJ96-CASS-18G-12	33.00
Shaft Speed Measurement	NI USB-6009	Already Own	Infrared Tachometer (Monarch)	IRS-P	200.00
Motor Control	NI USB-6009	Already Own	AC Drive Speed Controller (Inverter Drive)	ABB ACS150	Already Own
Humidity Measurement	NI 9201	379	Humidity and Temperature Probe (Vaisala)	HMP50	240
Heating Cartridge	Basic Unit – Temperature Controller	179	2 Miniature High-Temperature Cartridge Heater	8376T21	27.35 each
Total Price		2036			598.70
Grand Total Price	**$2634.95**				

Safety Shields
As part of the alpha design, we selected to use a clear plastic as an outer shell for the test area that serves two purposes: one is to retain any flying objects that may break in the case of failure of the device keeping the user safe, and two is to retain the water vapor produced by the heated beaker of water to maintain constant humidity for the tests. From the procedure described in the Parameter Analysis section, p. 43, we calculated that the Four-ball beam produced the maximum force of 160.9 N that the shield would have to withstand. Even though the TCT beam is thicker and therefore more massive, the

speeds of the TCT are much slower than the Four-ball, so the forces produced by failure of the TCT beam are considerably less, at a maximum of 10.7 N. Dividing the maximum force by the smallest possible contact area of one of the beams onto the shield, we were able to find the maximum stress induced by a breaking beam of approximately 51 MPa. We originally wanted an all lexan shield so that the test would be visible, but the maximum tensile stress at fracture for lexan is 65 MPa, which doesn't leave much room for increased forces. Therefore, for our final design, we will use aluminum sheets as safety shield protection with a small lexan window so that the user can still see some of the test.

PROTOTYPE DESCRIPTION

The purpose of our completed prototype will be to test the most important parts of our final design. Therefore, due to time constraints and the need for testing certain concepts from the final design to validate their functionality, we will not be able to implement everything from the final design into the prototype. From discussions with our sponsor, we have established that his most important objective is to have a functioning prototype that can run safely under various speeds at a wide range of loads. Therefore, we have determined that the power transmission system (gears), shaft speed sensing system, test cup, normal and lateral force measurement systems, and safety shielding are the main points of interest for our prototype. Optimizing these components will allow for a functional prototype for testing the important parts of the final design.

After deliberation within our group and an additional meeting with our sponsor, we have chosen temperature and humidity control to be the two components that will be adjusted from our final design to our prototype. While it would be very beneficial to have these two functions in the design of our prototype, they are the least important to our sponsor as we finalize our purchasing, manufacturing, and assembly plans. Also, the financial cost of purchasing the hardware and the man-hour cost of designing the software to implement full temperature and humidity control do not balance the additional increase in functionality of the prototype gained by their addition. The specific adjustments made to the temperature control and humidity control components are detailed further below.

Temperature Measurement and Heat-Control

Although measuring the temperature of the specimen during testing was required by our sponsor, the control of the specimen temperature is not as important for developing a functional prototype. Therefore, we have decided to remove the ability to reduce the temperature of our test specimen by not implementing the Peltier coolers in our prototype. We chose not implementing the Peltier coolers (and the cooling system) over not implementing the heating cartridge (and the heating system) based on the requested temperature range of 0-150 ºC from our sponsor. Assuming an ambient temperature in the lab of 20ºC, removing the Peltier coolers would result in a loss of only 20 ºC from the requested temperature range (0-20 ºC). On the other hand, removal of the heating cartridge would result in a loss of 130 ºC from the requested temperature range (20-150 ºC); even when the temperature of the specimen increases from friction during testing, the range of temperature lost by removing the heating cartridge would still likely be much larger than 20 ºC. Additionally, the Peltier coolers are rated at 336 W power consumption for operation, and the prototype would likely require more than one to cool the specimen to 0 ºC. In comparison, the heating cartridge is rated at only 25 W power consumption, and because it can reach 760 ºC at full power, only one cartridge would be required. Although the Peltier coolers are available to us from the previous group's work, we feel that this amount of saved energy justifies the purchase of a heating cartridge for around $30.

Humidity Measurement

While our sponsor still requires the ability to measure humidity in our prototype, the control of the humidity near the test specimen is the least vital function for us to manufacture a functional prototype. Therefore, we will be adjusting our humidity control system to a humidity measurement system. In doing so, we will also be changing the instrument and data acquisition hardware that we specified for our final design to a simpler and much less expensive instrument. The Vaisala HMP50 Temperature and Humidity Probe and NI 9201, which allowed for relative humidity measurement and interaction with LabView but cost a total of $619.00, will be replaced by an NI MCC USB-502 Low Cost USB Temperature and Humidity Logger. This product is also capable of reading the relative humidity of the test area at the required interval of 1 second and sending this information to a computer, but it cannot interact with LabView. However, this device is also significantly less expensive, costing only $82.00. Based on discussions with our sponsor about the importance of humidity measurement and control of the test area, this seems like a more appropriate dollar amount to be allotting to this task.

Each of the other designs and components detailed in the Final Design Description will be implemented as stated in our prototype. Because our prototype will very nearly be the exact physical representation of our final design, we have high confidence that it will accurately evaluate the feasibility and performance of the components that we implement from our final design.

FABRICATION PLAN

Manufacturing

The parts of our prototype that we chose to re-design and manufacture or modify were all evaluated under the auspices of making our prototype ergonomic, user-friendly, and adjustably flexible.

We chose to re-manufacture portions of the previous prototype because we felt that there was significant room for improvement in the model’s functionality, ergonomics and user-friendliness. In creating these parts, our primary concern was ease of manufacturing, knowing that our skills in manufacturing are not perfect. The considerations that we took during manufacturing allowed us to machine to tolerance rather than design to tolerance. This way, we were able to achieve the fits and tolerances we wanted to by working up to them rather than potentially over-doing one little process that would have drastic effects on the finish of our product later on. Likewise, we did not have the time to re-do any of our parts, so by designing extra material and allowing ourselves some flexibility in machining, we avoided having to take the time to start a part over from scratch. For each part that we manufactured, respective engineering drawings, feeds and speeds can be found in Appendix F.

The first part manufactured was the Outer Collar. This part was manufactured solely in the lathe. Using a facing tool, a 6 in. round of aluminum stock was turned down to the desired outer diameter, and the bottom of the stock faced off until the desired length of the collar was achieved. At this point, a boring bar was used to take out the top hole of the collar and take out the inside of the collar to where the desired inner diameter of the threads would lie. In order to thread the collar, a threading tool was put in place and the threading operation was commenced with the assistance of the automatic feeding capabilities of the lathe.

The second part we manufactured was the Specimen Cup. This part is made from the other half of the 6 in. round of aluminum stock that the Outer Collar was made from. The first operation was to face the entire round of stock down to the overall desired thickness of the entire piece. The second operation was the boring of the inner cup of the Specimen Cup. This operation is not critical in interfacing with the Outer Collar, but rather is critical down the line in producing the TCT and 4-Ball Collars. Once the inner cup was bored out, the outer diameter of the cup was taken down to that of the outer diameter of the threads that would be created in order to fit in with the Outer Collar. Threading, again, was performed using the automatic feeding capabilities of the lathe in order to ensure proper fit. Once turning operations on the lathe were completed, the part was then moved to the mill to complete the remainder of the operations on it. The first operation, taking out the grooves that the TCT Collar’s teeth would eventually interface with, was done using a ½ in. ball-nosed end mill. The grooves were taken out slightly from the initially-designed width of 0.5 in. in order to accommodate for both deflection in the end mill while machining the grooves, but also for inconsistencies in machining out the teeth on the bottom of the TCT Collar. On the underside of the cup, four teeth were milled out that would interface with four grooves in the Lower Block. From there, two more holes were machined in order to provide a setting for our resistive heating cartridge, and also for our thermocouple. A final little modification to the original design was a slight, ¾ in. indentation at the center of the cup using a flat end mill. This provides a small resting area for the three specimen balls used in the 4-Ball test so they don't easily separate and inconveniently fall into a groove.

Third in the process order is the 4-Ball Collar. Although this piece could have been done in tandem with the TCT Collar, we were only on one lathe at the time, and this one required the most work. Working down from the 4 in. round of aluminum, the outer diameter was taken down to what the inner diameter of the Specimen Cup was measured to once that aspect of the machining was done. Boring out the through-hole was done next, followed by the bottom, 45° chamfer. The part was flipped around and the entire block was faced down until the piece was of desired length. The inner recess was then created, again using the boring bar, until the opening of the through-hole was just inside of 1 in. in diameter.

The TCT Collar followed the 4-Ball collar in the lathe, and was started off in a similar fashion. The outer diameter was turned down until it fit in the Specimen Cup, and the through hole was created using a boring bar. Another chamfer was done, though not to the same extent as the 4-Ball Collar. The round was then flipped around and faced on the other side to the desired length of the piece. Once this length was established, the piece was taken to the mill to have the teeth cut out from the excess stock at the end of the round. Once the teeth were milled out, the Specimen Cup and TCT Collar were both touched up until the collar fit down into the teeth at full engagement in each possible orientation. Touching up involved both filing down burrs on the teeth and grooves and also taking the width of the grooves out slightly, compensating for the deflection that occurred in the end mill during those operations.

During the time that was spent on the lathe, the Lateral Force Beam Brackets were being machined on the mill. In order to ensure proper alignment between the two brackets, a coincident origin was established from which to center all measurements. Facing was done using a ½ in. end mill, as were the large-diameter holes. The slots and smaller holes were done using a ¼ in. end mill to ensure uniformity between the two brackets.

Deflecting Beams are cut individually out of a length of appropriate material stock, and then milled down to exact size and finish. Where the ½ in. drill was initially design, a slot is additionally taken down from the hole in order to allow easier changeability between tests.

The two bearing alignment plates were designed to facilitate proper alignment between the two shafts in our system. The top of the pair replaced a plate that was already present in the previous prototype, and the bottom fit between the shop press and the thrust bearing plate. Because the plates were made out of steel, extra attention had to be exercised in order to get the machining done right the first time. From larger plates of the steel, the two plates were cut down to relative size on the band saw. The remaining operations, those being drilling, slotting and edge-finishing were performed on a mill to ensure precise measurement techniques. The slotting of fastener holes and bearing mounts was necessary to allow the implementation of these plates to be flexible. Because we had to adjust to inconsistencies in the drilling of the previous prototype, slotting of these plates would allow us to make small adjustments during final assembly in order to create a well-aligned system.

Because of the shaft alignment redesign we incorporated to accommodate our new power transmission system, we need a shorter shop bench to hold our motor. The shorter bench was drilled and assembled in the same fashion that the previous one was.

The gear enclosure box was bent and bolted together out of a flat piece of 0.025 in. aluminum sheet stock. Holes were drilled on the top side of it to accommodate the ends of the driveshafts, and brackets were mounted along the side to be bolted to the outer C-Beams of the shop press. The Humidity Enclosure was designed out of clear ABS plastic, secured together with brackets.

Modification and Adjustment

The parts of the previous prototype that we chose to modify or append to were done so in order to make the prototype slightly adjustable in key areas. Because we performed these modifications, we were able to accommodate slight errors both in what we manufactured and what was left for us on the previous prototype.

Just like we had to machine adjustability into the bearing alignment plates, we had to machine some adjustability into the parts we were left with. The thrust bearing plate and the bottom-most bearing plate both required slotting in order to allow for complete flexibility along the main drive shaft.

We had to make some slight adjustments to the shop press structure in order to eliminate already-present alignment issues. By loosening and re-tightening the bolts that secured the horizontal C-Beams to the vertical ones, we were able to level the mounting surface of each of the driveshaft bearing plates.

The Lower Block required a number of modifications to accommodate our redesigns. On the top side of the block, four grooves were milled out in order to interface with the four teeth on the underside of the Specimen Cup. On one side of the block, six small holes were drilled and tapped where the Beam Brackets would bolt down to. The peltier setup, covering two other sides of the block, involved using Arctic Silver thermal adhesive to bind the peltiers to the block, and their respective heatsink and fan assemblies on top of that.

Assembly of Final Prototype

There are two halves to the assembly process involved with this prototype. The first, and most important, is the aligning of the motor driveshaft to the tool-fixture driveshaft. The second half, and the one that is inherently more flexible, is the assembly of the Specimen Fixture down through the bottle jack of the shop press.

In assembling the driveshaft interface, we started from the tool fixture and worked our way back to the motor. By leaving every fastener loose by a turn or two, we could put everything together, and then tighten down each piece from the bottom up as we spun the tool-fixture driveshaft to ensure that the bearings were properly in line.

The same process was applied to the motor driveshaft. The motor, once secured to its mount, was moved into roughly the proper place underneath its bearings. Sitting loosely in untightened and loosely-secured bearings, the smaller shaft was then coupled to the motor's driveshaft using the flexible coupler. Once this coupling was tightened down, final adjustments were made to the positioning of the motor mount relative to the shop press in order to adjust tooth engagement in the gears. We set this engagement using the TCT gear ratio since the 1.5 in. sprocket did not create undue forces on the driveshaft. Once tooth engagement was in place, the motor shaft was turned, by hand, so that the motor driveshaft bearings could be tightened down while still ensuring that the flexible coupler was not under a significant amount of misalignment stress.

Putting together the bottom assembly was kept simple. From the bottle jack on up, the aluminum block is placed over the jack, followed by a compressive block of a desired material, followed by the pair of steel alignment plates. Depending on the test being run, either the large steel ball or the 4 bolts are put in place. The modified Lower Block, complete with the peltier setup, is placed over the bearings on the top side of the upper steel alignment plate. The block is oriented so that the affixed Deflecting Beam interfaces with the Deflecting Block. The Deflecting Block is bolted to the left C-Beam of the shop press.

Once the specimen assembly, tool fixture selection and gear ratio selections are complete, the safety shields can be bolted down at their respective locations. The environmental enclosure only requires that the specimen cup and tool fixture be close to engagement, so necessary preparations in per-experiment setup are required before initial loading of the shop press can commence.

Changes between Final Design and Prototype

Two changes between the Final Design and Prototype were the use of an aluminum safety shield instead of the large humidity chamber, as well as the use of steel dowels instead of teeth to interface the Specimen Cup and Lower Block.

The new safety shield, an environmental enclosure, was bent around the specimen assembly, and bracket-mounted to the motor mount, creating a thin wall that will prevent errant human contact with the interface during testing.

The steel dowels replaced the teeth that would have been used underneath the Specimen Cup. The dowels were press-fit into the steel Lower Block, and slip-fit into the Specimen Cup. The slip-fit holes in the bottom of the Specimen Cup had a chamfer added to make placing were additionally reamed out in order to provide flexibility in what direction the cup could fit over the dowels nicely. The dowels in the Lower Block were glued into place to ensure rigidity. Over the course of curing, one cured off-angle, so it was removed. The three dowels that are currently in place provide more than adequate shearing protection and security for the Specimen Cup.

VALIDATION PLAN

Fundamentally, we will have 4 major preliminary tests that will prove/disprove our mechanical and electrical systems before we test the entire device and everything all at once. These tests will occur as assembly is in progress. The 4 preliminary tests can be broken down as:

1. Force Measurement Systems
2. Motor Control
3. Temperature Control
4. Transmission System

The rest of this section presents tables that breakdown the specifics about each of the four preliminary tests as well as one that describes the nature of the full prototype tests where all systems will be working in conjunction with one another.

Table 29: Force Measurement Systems

Specification to be Validated	Step	Procedure
Normal Load: 0 – 85 kN Lateral force measurement	1	Plug strain gages of beam/block being tested into the **appropriate** strain gage half-bridge circuit
	2	Calibrate each force measurement component using known forces (applying known loads and measuring the resultant voltage) and our LabVIEW program for data acquisition
	3	Create force vs. voltage curves from the calibration data
	4	Compare calibration data to known forces to check accuracy
Normal Load: 0 – 85 kN	5	Validate shop press's ability to hold compressive loads by placing aluminum compressive block (rated up to 100 kN) into compressive block slot and pump hydraulic press up to 85 kN
	6	Record strain gage data for one hour
	7	Analyze results: if press does not hold loads, possibly order a new bottle jack ($357.00)

Table 30: Motor Control

Specification to be Validated	Step	Procedure
Shaft speed: 0 – 2100 rpm	1	Verify motor is secured to its current stand and disconnected from chain drive or coupler
	2	Using LabVIEW, turn the motor
	3	Set speed to 100 rpm in LabVIEW
	4	Verify accuracy of motor speed using optical sensor
	5	Once verified, increase speed by 100 rpm
	6	Given that the motor remains stable and accurate to the set input speed for 5 minutes, continue increasing speed by 100 rpm intervals
	7	Once motor speed has reached 1300 rpm, increase speed by 24 rpm to 1324 rpm since this is the maximum speed required for testing (achieves maximum driveshaft speed of 2100 rpm with a 1.6:1 gear ratio for the Four-ball test)
	8	After 1324 rpm has been achieved, slow motor to 1300 rpm
	9	Decrease motor speed by 100 rpm intervals for 2 minutes until it reaches 100 rpm
	10	Stop motor and disconnect motor controller
	11	Unplug motor

Table 31: Temperature Control

Specification to be Validated	Step	Procedure
Temperature: Ambient – 150°C	1	Place heating cartridge into test cup
	2	Fill bottom of test cup with Olive Oil (MSDS in section 8.3)
	3	Set up the four ball test in the test cup
	4	Using LabVIEW, set a temperature for the lubricant to 25°C
	5	Verify that the cartridge heater warms the lubricant up to the set temperature and wait to ensure that if the temperature increases over the set temperature that the Peltier fans will come on and the Peltiers cool the lubricant until it reaches the set temperature again
	6	After this test has been verified, increase the temperature to 30°C and wait again
	7	Upon each successful temperature, increase temperature by intervals of 10°C until the maximum desired temperature of 150°C has been reached
Temperature: 0°C – Ambient	8	Set temperature to 15°C and observe Peliters cool the lubricant to this temperature
	9	If successful, continue cooling to 0°C by 5°C intervals
	10	Turn off both power supplies

Table 32: Transmission System

Specification to be Validated	Step	Procedure
Shaft speed: 2 – 30 rpm for TCT	1	Attach 1:12 gear ratio for TCT
	2	Connect the pinion gear driveshaft to the motor shaft via the flexible coupler
	3	Plug in motor
	4	Input gear ratio to LabVIEW and set driveshaft speed to 2 rpm
	5	Verify that the measured shaft speed is 2 rpm
	6	Increase shaft speed to 5 rpm
	7	Continue increasing speed by 5 rpm increments until max speed of 30 rpm is reached
	8	Turn off motor
	9	Unplug motor
Shaft speed: 0 – 2100 rpm for Four-ball test	10	Swap gears for the 1.6:1 ratio of the Four-ball test
	11	Plug in motor
	12	Input new gear ratio to LabVIEW and set driveshaft speed to 100 rpm
	13	Verify that the measured speed is 100 rpm
	14	If verified, continue increasing speed by 100 rpm intervals until 2100 rpm (max speed of Four-ball test) has been reached
	15	Turn off motor
	16	Unplug motor and disconnect motor controller

Table 33: Full Tests

Specification to be Validated	Step	Procedure
TCT Specifications	1	Attach 1:12 gear ratio for TCT
	2	Place Polyethylene compressive block beneath alignment plates
	3	Place steel plate and olive oil into bottom of test cup
	4	Place TCT ring into test cup
	5	Screw collar down to secure the steel plate and ring
	6	Attach the Polyethylene lateral force beam to the steel plate and put test cup on the steel plate
	7	Align the x-y plane using the alignment plates and tighten the plates
	8	Place Humidity meter within the device and start data
	9	Attach the safety shield over test cup assembly
	10	Attach gear safety shield over transmission
	11	Load the shop press to 100 N
	12	Input the temperature setting and speed (start at 25°C and 2 rpm)
	13	Start the motor
	14	Verify that the whole system is working (normal force, lateral force, speed, temperature readings)
	15	Stop test. Check that lubricant is still able to be tested
	16	Replace the lateral force beam with 0.125” aluminum beam
	17	Keep the Polyethylene compressive block
	18	Repeat steps 7 – 15 with 1000 N and 5 rpm
	19	Replace the lateral force beam with 0.25” aluminum beam
	20	Replace the compressive block with the PVC block
	21	Repeat steps 7 – 15 with 10000 N and 15 rpm
	22	Replace lateral force beam with 0.5” aluminum beam
	23	Replace the compressive block with the aluminum block
	24	Repeat steps 7 – 15 with 85000 N and 30 rpm (maximum TCT capabilities)
	25	Take out test cup and remove the TCT ring and steel plate
Four-ball Test Specifications	26	Place 3 bottom balls of Four-ball test into test cup
	27	Place Four-ball ring into test cup
	28	Screw collar down to secure the 3 balls and ring
	29	Place polyethylene compressive block beneath the alignment plates
	30	Attach the Polyethylene lateral force beam to the steel plate and put test cup on the steel plate
	31	Align the z-axis using the alignment plates and tighten the plates
	32	Place Humidity meter within the device and start data
	33	Attach the safety shield over test cup assembly
	34	Attach gear safety shield over transmission
	35	Load the shop press to 30 N
	36	Input the temperature setting and speed (start at 25°C and 600 rpm)
	37	Start the motor
	38	Verify that the whole system is working (normal force, lateral force, speed, temperature readings)
	39	Stop test. Check that lubricant is still able to be tested
	40	Increase load to 392 N (ASTM 5183) and keep speed at 600 rpm
	41	Repeat steps 37 – 39

	42	Replace lateral force beam with 0.125" aluminum beam
	43	Replace compressive block with PVC block
	44	Load shop press to 7840 N and set speed to 1770 rpm (ASTM D2783 and D2596)
	45	Repeat steps 37 – 39
	46	Increase load to 13500 N and set speed to 2100 rpm (max Four-ball capabilities)
	47	Repeat steps 37 – 39
	48	Stop the motor, disconnect power to motor and power supplies

Of course, the fifth table of complete tests is just scraping the surface of the amount of tests that can performed but we feel that these will give us a great start and we can build upon these preliminary tests.

VALIDATION RESULTS

In order to validate that our design works, we needed to start with small, individual functions and work our way in to more complex and complete configurations within our prototype. There are three primary regimes where validations are required: Mechanics, Electronics and Software.

Within the mechanical regime of our prototype, our primary goal is to make sure our prototype would be capable of running a normal test for both the 4-Ball and TCT setups. An initial mechanical validation was that the individual shafts rotated without undue friction on the bearings. By getting the shafts in line, and spinning them by hand, we could adjust the bearing seating when a spot of friction came around. Ensuring that the motor ran by itself was step two. Running the motor through local control on the motor drive, we were able to run the motor at a few different speeds and concluded that the motor runs very smoothly and free of vibrations. Step three was to ensure that the two shafts rotated together well when combined with the two gear ratios. Turning the gears and shafts by hand allowed us to again check for undue friction in the bearings. Once the motor shaft was connected and properly aligned with the motor driveshaft, we tested, again, to make sure that there were no undue frictional forces present in the bearings as the motor turned. Manual turning again allowed us to determine whether undue frictional forces were present during rotation. The final validation we could do was put a low load on the hydraulic press to evaluate how it moved while being loaded. The press moved the assembly up easily, and the pressure release valve worked properly.

The software side of the project was the next group of items that could be evaluated separate from the other aspects of the model. All of our code was written in LabVIEW, and initially split up into four separate loops. One is responsible for controlling the motor, the second controls the Tachometer, the second reads in and evaluates the strain gauge data, and the fourth is the encompassing temperature control system. To evaluate the motor control and tachometer loops, we were able to run those with the motor and tachometer in order to explicitly test that the specified device worked with its respective LabVIEW loop. In order to validate the logic and capabilities of the strain loop, we used a dummy data input to replicate how data from the DAQ would be supplied to the loop. We tested the Temperature Control loop in the same fashion, supplying dummy values and making sure the logic in the system is sound. Every loop functioned flawlessly separately. Completing the validation for this will be to run each loop in parallel in the singular, encompassing LabVIEW program. A diagram of the encompassing program can be found in Appendix H.

The electrical portion of our project was validated as portions of it were made available to test. Validation of the motor control implementation was done in part, as the electrical wiring necessary to go straight from the DAQ to the motor drive was able to be implemented and tested successfully, but the full amplification desired to reach full load was not yet wired into the system. The same regard holds for the Tachometer implementation. The Tachometer was successfully wired into the DAQ and the loop run successfully inside the prototype. The encompassing strain gauge circuit covers the strain gauge pairs for both the lateral beam and the compressive block. The half governing the lateral beam was capable of registering changes in strain when the attached beam was deflected.

As it stands, our prototype is well on the way to working properly enough to run full tests. Mechanically, this prototype is close to ideal functionality, the electrical side, however, needs some work in order to effectively bridge the mechanical side with the software side. This is a very robust system, designed with very high safety factors. Once some key mechanical considerations are taken care of, this prototype will function very well mechanically. Although the electronic side didn't work out well for us up through this point, once it does function easily and properly, the software side is very well set already in order to immediately allow for full tests to run properly.

DESIGN CRITIQUE

Now that we have reached the conclusion of our prototype redesign, we have compiled a list of things that we would have done differently. A common theme in the tasks that we would have done differently is starting them earlier, which would have given us more time to work on implementing our designs and dealing with the unforeseen problems that arose. Other reasons that we would have chosen different ways to do tasks include misjudging of our sponsor's highest priorities and underestimating the time that non-mechanical tasks would take. Several of the things that we would have done differently are explained further below.

Disassembly: Conducting the disassembly of the previous prototype earlier in the redesign process and doing so more completely were two things that would have greatly improved our chances of completing our redesign. Because we conducted the disassembly later than we should have, we got a late start on analyzing the engineering parameters of our prototype that needed improvement. This delayed our product research, purchasing, and assembly, and did not allow us time for full system debugging. Because we did not conduct the disassembly as fully as we should have, we were not aware of some serious design flaws on the previous prototype until too late in the redesign process that we could not correct them. These design flaws included the misaligned axes of the two connecting drive shafts above the test cup and the inability of the bottle jack of the shop press to move the specimen cup up high enough to engage with the tool fixture. Because the failure of the shop press to engage the tool fixture with the specimen cup specifically was a major design flaw, it severely hindered our ability to undertake validation testing of our complete system.

Power Transmission System: Starting to contact vendors for product quotes on gears and bushings earlier would have given us a much better chance of implementing a gearbox, our sponsor's first choice for a power transmission system. Finding a vendor that was both reliable and willing to work with a student group took much longer than we expected, and getting an actual quote on a gearbox from our vendor did not happen until a stage in the redesign process where we would not have had a reliable power transmission system to present at the Design Expo. If the search for a vendor had begun and concluded earlier, more time would have been left for negotiation on a gearbox and there is a high likelihood that we could have gotten one made for our prototype.

Peltier Cooling System: Putting more work into implementing the Peltier coolers, heat sinks, and CPU fans as a reliable lubricant cooling system would have almost certainly allowed us to build a functional cooling system. All of the components for the cooling system, including the Peltier coolers and thermal adhesive, were provided to us from the previous prototype, so the only additional work for us to implement the cooling system was to wire the correct circuit diagram. Although selecting components for the electrical system and wiring them together took a far greater amount of time than anticipated, starting to do this for the Peltier cooling system earlier would have given us more time to debug the system and test the limits of its cooling ability. Because of a brief elimination of the Peltier cooling system from our design and not beginning to install it until later in the assembly process, our system does not function although it is seemingly wired correctly.

Software and Electronics: The two parts of our redesign whose required workloads and times to complete we underestimated the most were the compiling of the LabVIEW program for data acquisition and instrument control, and the purchasing and connecting of the electrical components and instruments. Although our LabVIEW code is nearly complete for each of the separate tasks of our data acquisition and control system, it has not been optimized to function together correctly in one block diagram. The individual LabVIEW codes also need to be adjusted slightly after the calibration of their respective measurements to provide the desired outputs from the codes. Starting the compiling of our LabVIEW code earlier would have allowed us to combine each of the separate codes into one block diagram, but calibration testing would still have been required to finalize the entire code. A discussion of what we would change about our calibration and validation testing plans is provided in the next section.

The purchasing and implementing of our electronic system was a struggle throughout the assembly process and took much longer than expected. Our very minimal knowledge of electronics and their connections required us to seek constant instruction and led even small tasks such as wiring circuits to take far longer than we expected. Because of the multitude of electronic components in our redesign (seen in the Power and Electronics Diagram in Appendix I), we were able to connect all of the components in our system but were left with very minimal time to debug and troubleshoot. This resulted in only the motor controller and optical encoder working functionally out of all of the electronic components in our prototype. Starting the installation and connection of these electronic components earlier would have given us more time to debug and troubleshoot, but because of our lack of knowledge on the topic even this would not have guaranteed a fully functional system.

Force Measurement System Assembly: Earlier assembly of the necessary components to conduct calibration for several of the instruments on our prototype would have been one of the most crucial changes towards running validation tests of our full system. Specifically, attaching the strain gages to both the deflecting beams and the compressive blocks earlier would have allowed for more time for troubleshooting the strain gauge amplification circuit, which appeared to be faulty during attempted calibration testing. Because the strain-gage-based force measurement system was not functional and calibration testing could not be completed, the LabVIEW program could not be finalized to output the desired quantities and no validation testing was able to be undergone of our entire system. Although the outsourcing of the attachment of the strain gages to their respective blocks and beams took longer than expected, we could have ordered the parts earlier so that this process was begun and completed earlier. Having time to troubleshoot the strain gage amplification circuit and get it to work functionally would have allowed us to at least run a test of our complete system besides the environmental control, which was the most prominent desire of our sponsor as our project progressed.

Prototype Strengths

The strengths of our prototype lie mainly in its mechanical components. Vast improvements were made in many of the mechanical systems of the prototype that made it safer, more stable, more functional, and easier to operate by the user. The functioning aspects of the software and electrical systems also add important capabilities to the prototype that make it both more convenient for the user and more precise in its operation. Specific strengths of our prototype are outlined below.

Interchangeable Gear System: The interchangeable gear system is much more stable and has the potential to be much more effective towards achieving our sponsor's desired specifications than the chain drive on the previous prototype. It allows for quick and easy interchangeability between the Twist Compression and Four-Ball Tests by simply removing and replacing two sets of gears. Because each set of gears fits into the set distance between motor shaft and drive shaft, a much safer and more robust stability system for the shafts, gears, and motor was able to be implemented (which will be detailed further below). The gear ratios for the two sets of gears were assigned so that each test is able to be run at speeds and normal loads that meet or exceed the ASTM standards (Four-Ball Test) or industry standards (Twist Compression Test). The interchangeable gear system can also handle larger levels of stress than the chain drive and is likely quieter while in operation.

Redesigned Motor and Drive Shaft Stability System: As mentioned briefly in the discussion of the interchangeable gear system, our prototype features a vastly improved motor and drive shaft stability system over the previous prototype. The drive shafts are held in place by two 0.25 inch thick steel plates, which are fixed to a cross beam of the shop press. The steel plates serve as constraints on the drive shafts when they want to bend from the interaction of the gears, and are structurally stable enough to handle the stresses of the drive shafts at the maximum speeds and torques that the sponsor desires the prototype to run at. Bearings are affixed to these steel plates to ensure that the shafts are able to rotate smoothly, and the bearings are proven to be aligned properly by the ease with which the shafts can be slid up and down inside them. As additional stress relief for the motor shaft, a flexible coupler is used as a connector between the motor shaft and the drive shaft directly on top of it. Finally, the motor has been relocated so that it is positioned at a lower level, closer to the rest of the prototype, and with its supporting wall facing the prototype. Having the motor positioned closer to the rest of the prototype reduces the distance that torques have to be transmitted between the drive shafts and allows the rods connecting the shop press and motor table to provide additional stability to the motor table.

Safety Shielding: A main concern that our sponsor had with the previous prototype was its lack of safety shielding around the chain drive power transmission system and its inadequate safety shielding around the test area. We have implemented aluminum sheet safety shielding around each of these areas of our prototype. The box-shaped shielding around our gear system covers all dangerous moving parts, prevents objects from getting into the gears while they are operating, and provides some level of containment or restraint of dangerous flying pieces in the case of fracture. The aluminum sheet was bent into a half-cylindrical shape to fit around the test area and is held in place by bolts through the motor support wall. It prevents objects from interfering with the rotating tool fixture and the potentially high-temperature heat sinks of the Peltier coolers, as well as provides some resistance in case of fracture.

Redesigned Test Cup: We manufactured a completely new test cup that is lighter, less bulky, and much easier to use than the test cup from the previous design. It implements a three piece design, where one of the three pieces is interchangeable depending on which test is being run. The appropriate interchangeable piece fits inside the test cup and is held in place by the outer collar, which simply screws on to the test cup. This design is much for convenient than the design in the previous prototype, which required the user to blindly thread four holes through the test cup and lower steel block. The redesigned test cup is smaller than the previous test cup and is made of aluminum so that it is lighter and easier to remove from the lower block. It is held in place on the lower block by three steel dowels which also contribute to the easy removal and replacement.

Drive Shaft Speed Measurement and Motor Control: The drive shaft speed measurement system and the motor control system provide great convenience and higher confidence to the user than the systems implemented in the previous design. Our motor control system has been connected so that a user can input a desired test fixture shaft speed into our LabVIEW program and the motor will run at this speed, instead of requiring the user to manually turn the dial on the motor controller. A control of this inputted shaft speed is also in place using an optical encoder to measure the shaft speed. The measurements from the optical encoder are fed back into the LabVIEW program to ensure the shaft is rotating at its inputted speed and to adjust for any variations. The feedback system allows for higher user confidence that the test is running at the desired speed.

Prototype Weaknesses and Solutions

The weaknesses of our prototype lie mainly in the electronic components and their interactions with the data acquisition system and the LabVIEW software. As spoken about in the What Could Have Been Done Differently section, our lack of knowledge of electronics led us to struggle mightily with their implementation throughout the duration of our prototype assembly. Two components that some of these electronics were connected to, the strain gage and thermocouple amplifier circuit boards, were experimental parts that we used in our data acquisition system but that had never been tested or debugged. Finally, although we feel that our LabVIEW software program is adequate and functional, we were never able to test its robustness because of our combined uncertainty with the electronics and the data acquisition circuitry. Because the electronics, the data acquisition circuits, and the software all must function properly both independently and together for the measurement and control systems of our prototype to work, the chance for errors was great. This was unfortunately the case in our prototype, and is the main reason that this part of our prototype produced the majority of the weaknesses. Though the mechanical portions of our prototype tended to be much more functional, there were also a few weaknesses in our mechanical design. These stemmed both from design flaws of the previous prototype that we found too late to change and oversights in potential problems by our group in our designs. The main weaknesses of our prototype are described below, along with direction towards possible next steps and solutions make them better.

LabVIEW Software: As mentioned several times in the preceding paragraphs, we have high confidence that our LabVIEW program for data acquisition and instrument control is nearly complete and functional. However, the two major weaknesses of the program right now are that each function is in its own block diagram and that the functions do not provide the appropriate outputs. The first weakness could be solved fairly quickly by implementing each of the block diagrams into one main program for our DAQ to run and troubleshooting this program. The issue of the functions not producing the correct outputs requires slightly more work to solve, as it relies on several other non-functional parts of the prototype to be fixed before it can be resolved. To get the proper algebra that can be inputted into the LabVIEW program so that the outputs are correct, several of this instruments must be calibrated to correlate their voltage outputs to the appropriate outputs. These instruments include the strain gages on the lateral and normal force sensing blocks and the thermocouple, all of which currently have their own issues that must be debugged first. These will be explained in further detail below, but once they are resolved then the instruments simply require calibration for the LabVIEW software to be finalized.

Temperature Measurement System: The temperature measurement system (Appendix I), comprised of a thermocouple, a cold-junction amplification circuit, and an input to our DAQ module, was never found to work functionally. As we purchased the thermocouple from a reliable vendor and tested to confirm that the appropriate power level was reaching the cold-junction amplification circuit, we suspect that the circuit itself is the location of the problem. After several hours of work with the creator of the circuit, John Baker, we were unable to achieve a successful temperature reading. Recent further research by John Baker has led us to believe that the circuit indeed is where the malfunction is occurring, but that there should be a simple fix as long as the circuit is not blown. Replacing the 20 kohm resistor on the circuit with a 50 kohm or higher resistor should bring the output current of the thermocouple to a low enough level that it can be read accurately by the DAQ module.

Force Measurement System: The temperature measurement system (Appendix I), comprised of two sets of two strain gages (two each on the lateral force sensing beam and the compressive block), two half-bridge strain gage circuit boards, and a strain gage amplifier board, was found to work on occasion but was very unreliable. Similar to the thermocouple circuit board, we believe that the combination of the half-bridge circuit boards and the strain gage amplifier board are the location of the problem in this system. The lateral force measurement system was found to work on occasion, proving that the strain gages and LabVIEW program for collecting this data work. However, the data from collected from the lateral force measurement tests was very sporadic and depended on unreliable factors like the connection between the half-bridge circuit board and the strain gage amplifier circuit board, the orientation of the boards, and the amount of noise in the room. Also, the compressive force blocks (using the same strain gages) were found to not work during testing of up to 80 kN on an Instron machine. This is near the maximum compressive loads that the block should be able to sustain and give readings on in our prototype, but in this testing the data showed only noise. The poor results of attempted tests with both the lateral force beams and compressive force blocks are our justification that the unreliability of the circuit boards is where the error occurs in the force measurement system. Further work with John Baker, who also designed these chips, should be undertaken to troubleshoot this problem.

Motor Controller Operational Amplifier: Although the LabVIEW program and motor controller allow the user to control the shaft speed through a computer interface, an operational amplifier is required to get the signal from the DAQ to the appropriate level for the motor controller to run the motor at its full range of speeds (Appendix I). The DAQ module (which receives the desired shaft speed) outputs signals between a range of 0 and 5V, but the motor controller requires a 0 to 10V range to run the motor from 0 rpm to 1765 rpm (the motor's top speed, which we need to achieve the ASTM standard for the Four-Ball test). We created an operational amplifier on our breadboard through which the DAQ output signal could be run to get to this level, but due to the amount of time we spent on other electronics we were unable to test the operational amplifier that we made. We are unsure whether the resistors that we used in our op amp will allow for the signal to get amplified to the full 0 to 10V range, but testing could be done to see if this is the case. We are confident in the op amp circuit that was created, so if the full 0 to 10V range is not achieved the problem could be easily solved by testing different combinations of resistors in place of the ones that we used.

Peltier Cooling System: As mentioned above in the What Could Have Been Done Differently section, we did not get the Peltier coolers, heat sinks, and CPU fans to function appropriately. The circuit for this system proved to be very complicated (Appendix I), but with guidance from John Baker we feel that the series-parallel circuit that we wired should work to operate all four Peltier coolers and all four CPU fans. In attempting to troubleshoot this circuit, we found that the digital DAQ output to the optical isolator was not providing enough power to close the relay and turn on these eight devices, so we believe an issue may be occurring in the optical isolator. Upon further review by John Baker, it was recommended to us that two obscure elements in the circuit diagram for the optical encoder need to be added to our circuit. These are a 270 ohm resistor between the digital signal from the DAQ and the anode pin on the optical encoder, and a 0.1μF capacitor between the +12V and COMM lines from the power supply. Considering the placement of these circuit elements, there is also a chance that the optical isolator is blown, so that may also need to be replaced.

Cartridge Heating System: The cartridge heating system (Appendix I), comprised of a cartridge heater and a solid state relay, was the last electronic element that was considered when we had a very short time frame left before our prototype was completed, so we do not have a complete understanding of its implementation. However, we do know that the cartridge heater requires AC power, which is different from all of the other electronic systems in our prototype. Although the connection between the solid state relay and the cartridge heater seems fairly simple, we are unsure how to safely connect the cartridge heater to the AC power coming out of the wall. This task should be considered with extreme caution, as wall outlets can provide 120 VAC if their connections are short circuited. Consultation with John Baker or another electronics expert would be greatly advised before undertaking the installation of the cartridge heaters that are currently available on the prototype.

Stability of Bottle Jack When Shop Press is Highly Loaded: As can be clearly seen if the shop press is pressurized when attempting to compress the tool fixture against the specimen cup, the current setup of the cross beams that support the bottle jack do not securely hold the bottle jack in place as it extends. This likelihood of the bottle jack not having the same axis as the tool fixture is magnified when the specimen cup and all of the pieces below it are placed in their positions on top of the bottle jack, because the additional weight throws the bottle jack off of its vertical axis even more quickly. Although this problem can be solved manually by guiding the specimen cup into place as it engages the tool fixture, this could present a safety hazard. Fortunately, a fairly easy alternative is available. The cross beams of the press that are supporting the bottle jack are currently only sitting on two bolts for vertical support; if these beams were also horizontally bolted to the shop press frame, the bottle jack would seemingly lose its ability to wobble in the horizontal direction and become much more stable. Ensuring that the bottle jack and the tool fixture share the same vertical axis could then by done by using washers around the newly-implemented horizontal bolts holding the cross beams in place.

Strength of Test Cup Material to Indentations: The final major design weakness of our prototype was brought to our attention very late in the design process and would have required extensive time to correct in the remaining time period that we had for our project. The issue involves the setup of the bottom test specimen (either three balls or a flat plate) inside the specimen cup. Because the bottom test specimen, which is made of steel, will be compressed inside the specimen cup, which is made of aluminum, there is concern that either the three balls or the flat plate will deflect into the aluminum under high loads and leave indentations. Using the equation, shown below, for the Brinell hardness of a material, we found that the steel balls would indent 4.5 mm into the aluminum, which is unacceptable for our sponsor's testing.

$$BHN = \frac{2P}{\pi D(D - \sqrt{D^2 - d^2})}$$

A material with a Brinell hardness greater than steel would need to be applied over the areas of the aluminum specimen cup that touch the steel balls or plate to prevent these deflections. If time permitted, our idea going forward would have been to use our same test cup and bottom test specimen design, but to add a thin layer of a material harder than steel to cover the aluminum in the spots where it interacted with the steel balls or plate.

FUTURE RECOMMENDATIONS

After assessing the strengths and weaknesses of our final prototype, we have compiled a list of recommendations for future tasks and points of emphasis in the continued redesign and advancement of our project. The recommendations include changes in areas that we considered both strengths and weaknesses of our prototype. The weaknesses obviously need to be improved for the prototype to have better functionality and for it to meet our sponsor's requests and specifications. We also feel that some of the strengths of our prototype could realistically be improved to makes its functionality comparable to that industrial machines. Many of these recommendations have been mentioned in the discussion of the prototype's strengths and weaknesses, but they will be reiterated again below.

Gearbox

Although we are extremely confident that our interchangeable gear system for power transmission will be suitable for the full range of required tests, implementing a gearbox for power transmission has many additional benefits. The gearbox eliminates the need for the user to switch the sets of gears when changing between test types by housing all four gears together and utilizing a shifter to engage the different sets (similar to a car's transmission). This leads to increased convenience for the user. Because the gearbox would likely have to be professionally machined, its structural stability would likely be higher than the steel plates used in the current interchangeable gear system. This would allow for better protection against the stresses from the gears' interactions for both the drive shafts and the motor. Finally, the gearbox would house the gears in an enclosed structure, so it would be safer than the current design and would eliminate the need for an external safety shield. We feel that the additional benefits provided by implementing a gearbox would easily outweigh the moderate amount of additional design work and time searching for a manufacturer.

Tool Fixture Drive Shaft Modification

Although we found the misalignment of the axes of the two pieces of the drive shaft above the tool fixture too late in the assembly stage to fix this problem, we strongly recommend that this issue get fixed early in the next stage of the redesign of this prototype. The misalignment is not a huge issue at low shaft speeds and low normal loads, but once the device is ready to perform tests over the full range of speeds and forces this design flaw presents the potential for severe bending of the drive shaft and possible ejection of the tool fixture. Our suggestion for resolving this issue is to bore out the hole in the flared section of the lower of piece where the top piece slides in until the misalignment has been completely removed, and then to add a bushing to this newly bored hole so that it returns to the diameter of the top piece. Care needs to be taken to ensure that the boring occurs on the same axis as the bottom piece so that design flaw is not repeated. Once the bushing is placed in so that the axes of the two pieces will be aligned, the shafts and bushed hole will both also need to be re-keyed.

Hard Material Layer Around Engaging Surface of Test Cup

As mentioned above, the Brinell hardness of the steel bottom test specimen (whether it is the three balls or the steel plate) is much higher than the Brinell hardness for the aluminum specimen cup that is compressed around it. To prevent indentation into the aluminum, we recommend that a thin layer of material with a Brinell number higher than the steel used in the bottom test specimen pieces be machined to fit over the parts of the specimen cup that interact with the bottom test specimen. Although it will likely be expensive to purchase a material with a higher Brinell number than steel and very tedious to machine this material to the desired shape, this was the best solution that we could think of that would definitively solve the problem.

Alternate Heating Source

Because the heating source that we purchased is the only electronic device in our prototype that requires AC power, we recommend purchasing a different heating cartridge or a different heating source altogether that can run on DC power. There are two DC power supplies (one low-voltage, one high-voltage) already being used to power other electronics on the prototype, so using a heating source that could operate off of either of their power outputs would greatly increase its ease of installation. Implementing a heating source that could operate off DC power would also be much safer than working with AC power, as the fluctuating current in AC power can produce voltages up to 120 VAC if shorted. Additionally, the AC power would need to be drawn from a wall power supply, which would require the power to be turned off to the test room for safe wiring and could lead to surges and other interruptions in power during testing if other devices are plugged into the same wall's AC power line.

Improved Safety/Environmental Enclosure Around Test Area

One of our sponsor's requirements that we deemed to have a low level of importance relative to the other requirements was the ability to control humidity in the test area. This would require a fully sealed enclosure around the test area, as well as a means of creating or eliminating water vapor in this enclosure. Because this enclosure would be similar in size, shape, and location to the safety shielding around the safety area, we recommend designing an enclosure that acts as both an improved safety shield and a humidity-controllable enclosure. Though the safety shield around the test area on our prototype is an improvement over the previous prototype's safety shield, it provides only shielding only around the sides of the test area and leaves the test area exposed and accessible on top of the shield and underneath it. The incorporation on an improved safety shield with the design of a sealed humidity chamber would solve combine two functions into one component, and also would improve the maintenance and control of the temperature around the test area.

Horizontal Constraints on Bottle Jack Cross Beam

To improve the stability of the bottle jack of the shop press, we recommend installing bolts horizontally through the cross beam supporting the bottle jack and the frame of the press. Installing these bolts as horizontal constraints would greatly reduce the tendency of the bottle jack, especially when loaded with the weight of the specimen cup and all of the components beneath it, to deflect horizontally so that its vertical axis was not aligned with the vertical axis of the tool fixture. This would also be a fairly easy process to complete, as it requires only measuring, drilling accurate and appropriately sized holes through the frame and the cross beams, and aligning the vertical axes of the tool fixture and the bottle jack with washers.

DAQ System

Because we believe that the crucial electronic failures of the strain gage and thermocouple amplification systems on our prototype are directly due to the unreliable circuit boards that we used for data acquisition, we recommend purchasing externally made, industrial-quality DAQ modules to collect this data. We presented a large package of National Instruments data acquisition hardware (Appendix E) as part of a previous design review, but this instrumentation was not approved due to its high cost of $2634.95. Though the NI 9211 Thermocouple Input Module ($329), 9237 4-Channel Simultaneous Bridge Module ($1149), and NI cDAQ 9174 4-Slot Chassis ($699) required for strain gage and thermocouple data acquisition constituted a large portion of this cost, no validation testing of any sort can be run on the prototype without functional strain gage and thermocouple data acquisition. We believe that our thermocouple, strain gages, and lateral and normal force measurement systems all function properly and that with the implementation of these industrial-quality DAQ modules we would have been able to measure compressive force, lateral force, and temperature, and therefore would have been able complete at least some validation testing. Having these DAQ modules also would have allowed us to spend more time troubleshooting and debugging our other electronic systems that did not function properly, which would have presumably led to further improvement of our prototype. We strongly recommend the reconsideration of the purchase of National Instrument DAQ modules for strain gage and thermocouple data acquisition.

CONCLUSIONS

The study of tribology allows mechanical engineers to study friction and wear, as well as various lubrications using devices known as tribometers. Two of the many types of tribometers perform what are known as the Twist Compression Test (TCT) and the Four-Ball Test. These two tests can model real world applications such as bearing performance and sliding contact which are of great importance to Professor Gordon Krauss at the University of Michigan. In the fall of 2009, Professor Krauss sponsored an ME 450 project that involved developing a single device capable of performing both the Twist Compression and Four-Ball tests. Both tests require a rotating tool (annulus for TCT and secured steel ball for Four-ball test) interfaced with a stationary specimen (steel plate for TCT and three secured balls for Four-ball test) while a set amount of compressive force is applied between them. The challenging part of developing the machine is the fact that required rotational speeds and compressive forces are vastly different for the two tests, creating many mechanical issues in the design including the power source, transmission for the driveshaft, and force measurement systems (compressive and resultant lateral) required for data analysis. In addition to these requirements, the machine also requires thermal control for the broadest range of uses. The ME 450 from the fall of 2009 was able to build an unusable prototype that could be used a starting point for a redesign to another team.

Our team acquired the project in January 2010 as a redesign project and it was our goal to take the basis of the previous design and develop a machine capable of achieving the requirements set by Professor Krauss's desires, industry standards and ASTM standards. We quantified these requirements into engineering specifications and target values were given as goals of the final prototype to achieve. From that point, we produced a functional decomposition of the device and generated a list of functions that the device needed to perform. This allowed us to generate concepts for each of the functions and compare them to the methods that were currently integrated onto the previous team's prototype. Using Pugh charts as a selection tool, we chose methods of achieving each of the functions and modeled an Alpha Design of what we wanted our prototype to be. This design utilizes a 3 HP AC motor for power to the driveshaft, a gearbox with two gear ratios for the two separate tests, a modular driveshaft with the ability to interchange the annulus and top ball for different tests, a modular specimen cup able to hold the lubricant and both the steel plate and bottom three balls, a cantilever beam lateral force measurement system, a compressive block normal force measurement system, and a 25-ton shop press used to generate the compressive forces.

To ensure that our Alpha Design would adequately meet the pre-described specification targets, we went on to perform an extensive analysis of the entire device. Due to inability to acquire a gearbox and physical constraints with interchangeable gears (chosen as our second option for transmission), the final design is limited to a maximum compressive load of 85 kN during TCT's instead of the desired 100 kN. Furthermore, the maximum compressive load is limited to 13.5 kN during Four-ball tests due to a lack of power in the motor to achieve the desired loading of 16 kN. Along with this reduction in compressive load came a reduction in maximum speed capabilities for the Four-ball test. It was determined that the maximum achievable speed could be 2100 rpm, below the desired speed of 3600 rpm. We also researched various options for recording data and determined what instruments we would use to record forces, temperature, and driveshaft speed. Our initial plan was to purchase instruments that could be directly plugged into one or several central National Instruments DAQ cards, but after analyzing the cost of such a system and our abilities to incorporate the instruments into a single, cheap DAQ card with the use of signal conditioning, we decided to dismiss the notion of purchasing the very expensive central DAQ system.

With analysis complete, we began fabrication and assembly of the prototype, the final phase of the project. The first physical work produced was the fabrication of the specimen cup and lateral force measurement system components. As this work progressed, we simultaneously developed a plan for the electrical system and designed a LabVIEW program capable of recording all necessary and desired data as well as allow the user to input a desired driveshaft speed and temperature. When we completed the fabrication of the initial components, physical work shifted towards fabricating an open, interchangeable gear system. The hopes of a gearbox as part of the prototype dwindled as we progressed due to odd required gear ratios as well as available time and money. Therefore, we had to design a system that would be able to allow for easy interchangeability of gear ratios while offering ample support under extreme test conditions. The work load necessary for completing the project was extremely heavy so the work was divided amongst our team. We worked diligently on completing a working mechanical prototype, electrical system, and LabVIEW program in the final days leading up to the Design Expo, held on campus.

The result of our efforts was a complete physical prototype with a working transmission system, specimen cup assembly, and lateral force measurement system assembly. Strain gages were attached to the three chosen compressive blocks (made of Aluminum, PVC, and LDPE Polyethylene in order to achieve requested resolution of force) and the four chosen lateral force beams (three made of Aluminum

(0.5”, 0.25”, and 0.125” thicknesses) and one made of HDPE Polyethylene (0.125” thickness) in order to achieve requested resolution of force). Peltier coolers were attached to the steel plate that supports the specimen cup by way of high strength thermal adhesive. The tachometer that we chose was mounted to the shop press and a strip of optical tape was attached to the driveshaft. The motor was wired to our ABB motor controller. The thermocouple, Peltier Coolers (with heat sink fans), strain gages, and tachometer were wired to appropriate circuits and power supplies. We were unable to wire the heating cartridge that was going to be used as a heat source for high temperature tests due to unforeseen AC circuit requirements that we ran out of time to implement. Finally, two safety shields, one covering the test region and one covering the open gears, were produced to provide protection against moving parts of the device.

After a tireless effort, we were unable to achieve a working status of the prototype as a whole, but were able to validate that the mechanical aspects of the device including the specimen cup, lateral force measurement system, and transmission are all mechanically sound and safe to use. The motor controller is connected and working, albeit only able to output have the desired speed at this point due to the fact that we still have to connect an amplifier to the circuit in order to achieve full speed range usage. Furthermore, the tachometer was able to record a rotational speed when turning the gears and driveshaft by hand. We achieved a working lateral force strain gage circuit, but are unsure of the compressive force circuit, and were unable to calibrate the blocks and beams, but are hoping to perform this task before final completion of the prototype. We ran into a variety of problems when connecting the thermocouple circuit to the DAQ card, and will be investigating ways to remedy this issue. The most incomplete part of the prototype other than the heating cartridge connection is the Peltier cooling system. We have a very potent power supply able to run the four Peltiers and heat sink fans, but experienced difficulties in wiring a working circuit and relay switch that allows for control of the devices.

Overall, the qualities of the mechanical aspects within the prototype are very high and we hope to completely validate their capabilities prior to delivering a final prototype. Certain components of the electrical system are working to at least some degree, but others are not working at all and the whole circuit system and wiring can certainly be improved upon. We hope to be able to implement a complete working model by April 27, but at a minimum, want to complete strain gage calibration and full validation of all mechanical aspects of the design. The ability of the machine to run at the specified speeds and loads, while recording both compressive and lateral forces as well as outputting the driveshaft speed, is far more important and essential than the ability to control test temperatures. These systems are the first priority of our group to improve upon and should be the primary focus of future student groups or faculty members who work on this project.

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- Jeff Emerson and the workers at American Gear and Engineering for their expertise in gears and machining
- Todd Wilber for mounting all of our strain gauges for us, and assisting us in finding Instron machines to test our samples with

INFORMATION SOURCES

When first assigned this project, little was known about how a Twist Compression Test and Four-ball Test were run or what they were used for. To begin to fully understand the Twist Compression Test and Four Ball Test, it was necessary for our team to research each of the tests and meet with the sponsor. Once a basic understanding of the tests was achieved, the team began obtaining customer requirements and benchmarking what would become the engineering specifications and targets.

Google Web Search/YouTube

The internet was used to find the basic definitions of tribology as well as information about the individual Four-Ball and Twist Compression tests. The Google search engine was primarily used for all of online research. After preliminary definitions were found, YouTube was searched in the hopes of finding any videos of the tests that would further reinforce an understanding of them. One video for Twist Compression Test and two videos for the Four-Ball Test were found. The videos were very helpful in the beginning stages of research in that they allowed for easy visualization of how the tests are conducted. [15-17]

Company Websites

Based on information obtained by the project sponsor and internet searches, companies that lead the industry were determined and their websites were scrutinized for information. Tribsys is currently the only company that manufactures a Twist Compression Test machine. Two of the industry leaders of the Four-ball test machine are Falex Corporation and Koehler Instruments.

Official Standards

In addition to the information gathered by company websites, ASTM standards used for the Four-ball test were analyzed. The standards were obtained using the University of Michigan's Library services. There are five main standards that pertain to the Four-ball test: ASTM D2266 [1], titled "Standard Test Method for Wear Preventive Characteristics of Lubricating Grease (Four-Ball Method)," ASTM D2595 [2], titled "Standard Test Method for Measurement of Extreme-Pressure Properties of Lubricating Grease (Four-Ball Method)," ASTM D2783 [3], titled "Standard Test Method for Measurement of the Extreme-Pressure Properties of Lubricating Fluids (Four-Ball Method)," ASTM D4172 [4], titled

"Standard Test Method for Wear Preventive Characteristics of Lubricating Fluid (Four-Ball Method)," and ASTM D5183 [5], titled "Standard Test Method for Determination of the Coefficient of Friction of Lubricants Using the Four-Ball Wear Test Machine."

Technical Benchmarks

The technical benchmarks compiled from our Google web searches, company websites, and official standards are summarized below in the table.

Table 34: Benchmarking Standards

Resource	Speed Range	Normal Load Range	Temperature Range
ASTM Standards* (4-Ball)	600 - 1760 rpm	59 - 7840 N	18 - 75°C
Falex [6] [7] (4-Ball)	100 - 3600 rpm	0 - 9800 N	T_{amb} - 175°C
Koehler [8] (4-Ball)	300 - 3000 rpm	NA	NA
Tribsys [9] (Twist Compression Test)	5 - 30 rpm	Up to 60,000 psi	NA

*A complete and detailed list of ASTM standards [1-5]

Information Gaps

As seen in the table above, a few of the ranges for both tests are missing. The Four-Ball test specifications can be found in the ASTM standards [1-5]. Because Tribsys is the only company that produces the TCT, and there are no current ASTM standards available on the TCT, only the information given on the Tribsys website and sponsor requests have been used to determine TCT benchmarks. We have been able to locate a standard in development but it is not available for viewing at this time.

ADDITIONAL RESEARCH

For each of our functions we have done additional research to aid in our decisions on which concepts to pick. This research was primarily done on the internet, contacting companies and interviews with people knowledgeable in the field.

AC Motor

Due to the fact that the previous team's motor was donated, they had not received the proper torque vs. speed and current vs. speed curves. They also did not have information about the motor's full load torque and full load speed. These curves allow us to determine what gear ratios will be implemented in the motor. To obtain this information we contacted both the manufacturer (A.O. Smith) and the distributor (Packard Inc.)

Gearbox

This is an area of research that is still ongoing. We have primarily been using the internet to research different gearboxes with specific gear ratios and testing torques. At first glance we thought obtaining a gearbox with these specific requirements was going to be impossible and require us to purchase an expensive, custom made one. We switched our focus to using gears and researched lecture slides from our ME 350 class [11] but decided this was not going to be a big improvement from the chain drive that is currently being used. After more research looking at a specific custom gearbox website [10], we have decided that using a gearbox is feasible.

Optical Encoder

The previous team used an optical encoder which was very functional. To solidify our decision to keep this we used the internet to search optical encoders and laser tachometers to compare pricing and functionality. [12] These two options were our top two when using a pugh chart to decide which one to use. After further research we decided to keep the optical encoder but did discover we need to purchase a new one due to the previous team's being short-circuited and inoperable.

Thermocouple

The previous team had purchased a thermocouple and had anticipated using it, but due to time constraints never got the opportunity to implement it into the design. To make sure this was the best way to measure the lubricant temperature research using the internet was performed [13].

Humidity Meter

The internet was used to research different humidity meters to find one that measured humidity up to 100% as well as being inexpensive [14].

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APPENDIX A: Bill of Materials

Quantity	Unit	Item Number	Source	Item Description	Unit Price ($)	Total ($)
1	Ea	00371946	Motion Industries, Inc.	YSS812 20 Spur Gear	28.86	28.86
1	Ea	00375736	Motion Industries, Inc.	YSS860 20 Spur Gear	136.57	136.57
1	Ea	00372003	Motion Industries, Inc.	TC896 20 Spur Gear	208.56	208.56
1	Ea	00374906	Motion Industries, Inc.	TC8144 20 Spur Gear	314.27	314.27
2	Ea	05550637	MSC Industrial Supply Co.	2 Bolt Mounted Flange Bearing: Diam. = 0.75”	24.67	49.34
1	Ea	8975K423	McMaster-Carr	3’ x 1.5” x 0.5” Aluminum Block	15.81	15.81
1	Ea	8975K563	McMaster-Carr	3’ x 1.5” x 0.25” Aluminum Block	9.45	9.45
1	Ea	8975K34	McMaster-Carr	6’ x 1.5” x 0.125” Aluminum Block	8.80	8.80
1	Ea	8671K19	McMaster-Carr	4’ x 1.5” x 0.125” Polyethylene HDPE Block	2.40	2.40
1	Ea	8975K313	McMaster-Carr	1’ x 3” x 1” Aluminum Block	17.86	17.86
1	Ea	8788K57	McMaster-Carr	1’ x 3” x 1” PVC Block	26.39	26.39
1	Ea	8588K17	McMaster-Carr	1’ x 3” x 1” Polyethylene LDPE Block	5.46	5.46
1	Ea	1497K956	McMaster-Carr	18” Long by ¾” Diameter Driveshaft with 3/16” Wide by 3/32” Deep Key	25.97	25.97
1	Ea	98535A140	McMaster-Carr	3/16” x 3/16” High-Carbon Plain Steel Key Stock	2.74	2.74
1	Ea	98535A150	McMaster-Carr	¼” x ¼” High-Carbon Plain Steel Key Stock	3.55	3.55
2	Ea	8376T21	McMaster-Carr	25W, 120 VAC, 1 ¼” Cartridge Heater	27.35	54.70
1	Ea	EK2-60-A-1	DieQua Corporation	1.125” and 0.75” Flexible Coupler with Standard Keys	132.00	132.00
1	Ea	1388K131	McMaster-Carr	Steel Sheet	219.58	219.58
1	Ea		Alro Steel Corporation	6” Diameter by 6” Length Aluminum Round	93.23	93.23
1	Ea		Alro Steel Corporation	4” Diamteter by 6” Length Aluminum Round	46.52	46.52
1	Ea		Alro Steel Corporation	6” Length x 4” Square Aluminum Block	56.90	56.90
2	Pkg.	C2A-13-250lw-350	Vishay	Strain Gage	71.25	142.50
1	Ea	TJ96-CASS-18G-12	Omega	Thermocouple	33.00	33.00

1	Ea	6180-056	Monarch	ROS-W Optical Sensor	144.00	144.00
2	Ea	8376T21	McMaster-Carr	2 Miniature High-Temperature Cartridge Heater	27.35	54.70
1	Pkg.	91201A031	McMaster-Carr	3/8” ID by 7/8” OD Washers	11.55	11.55
1	Ea	8973K63	McMaster-Carr	36” x 24” x 0.025” Aluminum Sheet	16.52	16.52
1	Ea	8973K53	McMaster-Carr	48” x 36” x 0.025” Aluminum Sheet	28.69	28.69
1	Ea	46715T31	McMaster-Carr	24” x 18” Medium Duty Table with 30” Height (Stationary)	121.39	121.39
2	Ea	9517K106	McMaster-Carr	24” x 8” x ¼” Steel Plate	86.08	172.16
16	Ea	218	Bolt Depot	Hex Bolt 3/8-16 x 1	0.14	2.24
4	Ea	232	Bolt Depot	Hex Bolt 3/8-16 x 5 ½	0.52	2.08
1	Ea	7071K153	McMaster-Carr	Continuous-Flex Miniature Wire, 30 Awg, 0.031” OD, 600 VAC, Red, 30 Ft	20.40	20.40
1	Ea	2103960	Jameco Electronics	Mean Well Net-35B 35W Power Supply (±12 V)	26.95	26.95
1	Ea	G3MC-201PL	Omron	DC5 Steady State Relay	4.36	4.36
1	Ea	8443K41	McMaster-Carr	A2 Tool Steel Oversize Rod, 2" Diameter, 1" Length	10.39	10.39
1	Ea	8443K51	McMaster-Carr	A2 Tool Steel Oversize Rod, 3" Diameter, 1" Length	13.44	13.44
1	Pkg.	99604A101	McMaster-Carr	Silicone Sealing Washer, No. 6 Screw Size, 1/4" Od, .093" Thick	11.04	11.04
1	Ea	5852T28	McMaster-Carr	A2 Tool Steel Tight-tol Flat Stock W/ Cert, 1/4" Thick, 2" Width, 18" Length	67.19	67.19
1	Pkg.	98381A714	McMaster-Carr	Alloy Steel Dowel Pin, 1/2" Diameter, 1-1/4" Length	8.37	8.37
1	Ea		RadioShack	16-Ft. USB 2.0 Cable with A-B Male Connectors	42.99	42.99
1	Ea	278-1218	RadioShack	90-Ft. UL-Recognized Hookup Wire (22AWG)	6.59	6.59
2	Ea	278-1611	RadioShack	Heat-Shrink Tubing Set (36-Pack)	3.49	6.98
1	Ea	MCC USB-502	National Instruments	Temperature and Humidity Logger from Measurement Computing	82.00	82.00
1	Ea		Meijer	Olive Oil	18.99	18.99
2			American Gear & Engineering	Work on Gears	100.00	100.00
Total						**2607.48**

APPENDIX B: Description of Engineering Changes since Design Review #3

Normal Force Measurement System

The changes to measuring the normal forces included adding a third compressive block. This block was added to be able to measure the smallest strains from the smallest loads. The block is made out of polyethylene. Depending on what forces the user is going to expect, that is the compressive block they will need to implement into the design. A table of the three compressive blocks with their corresponding loads is shown below in Table 35.

Table 35: Summary of Compressive Blocks and their Corresponding Forces

Compressive Block Material	Calibration Forces
Polyethylene	Up to 449.6 lbs
PVC	449 to 9216.8 lbs
Aluminum	9216 to 19108 lbs

Lateral Force Measurement System

The changes to measuring the lateral forces including adding a 1/8" thick polyethylene beam, a 1/4" thick aluminum beam and a 1/2" thick aluminum beam. These different sized and material beams allow for a more accurate strain measurement. Depending on what forces the user is going to expect, that is the beam they were need to implement into the specimen cup brackets. A table of the four lateral force beams with their corresponding loads is shown below in Table 36.

Table 36: Summary of Lateral Force Beams and their Corresponding Forces

Lateral Force Beam Material/Thickness	Calibration Forces
Polyethylene (1/8")	Up to 1.13 lbs
Aluminum (1/8")	Up to 9 lbs
Aluminum (1/4")	4.5 to 16.41 lbs
Aluminum (1/2")	16.19 to 164.10 lbs

Strain Gage

The method our team wanted to implement in the final design was using a data acquisition system. Due to cost issues this was not implemented. Instead, we worked with John Baker (University of Michigan) and his own design of a circuit board that included a half-bridge amplifying system for both the normal and lateral force measurement systems. This circuit board was connected to a Mean Well Net-35B 35W Power Supply as well as the National Instruments USB-6009 DAQ system [27].

Thermocouple

The method our team wanted to implement in the final design was using a data acquisition system. Due to cost issues this was not implemented. Instead, we worked with John Baker (University of Michigan) and his own design of a circuit board that included a cold junction amplifying system for our thermocouple. This circuit board was connected to a Mean Well Net-35B 35W Power Supply as well as the National Instruments USB-6009 DAQ system.

Humidity Measurement

The method our team wanted to implement in the final design was using a Vaisala INTERCAP HMP50 Humidity and Temperature Probe [31], along with its own National Instruments module. Due to cost issues this was not implemented. Instead, we purchased a National Instruments MCC USB-502 Temperature and Humidity Logger from Measurement Computing. When a humidity chamber is completed in future work, this device will sit inside and be able to measure the humidity. This device can store 16,382 temperature and relative humidity measurements. It supports 0 to 100% relative humidity and -35 to +80 °C (±1 °C accuracy). It can also be programmed to log humidity at rates of every 10 seconds, 1 minute, 5 minutes, 1 hour, 6 hours and 12 hours.

Safety Shields

Due to time restraints, we were unable to implement the small Lexan window into our safety shield that's implemented over the specimen cup area. We were able to manufacture a safety shield for both the specimen cup area and over the gear system. Both shields are easy to put on and remove.

Specimen Cup

Our team's final design wanted to manufacture teeth out of the base of the specimen cup. Due to the amount of time that we were told it was going to take to manufacture these teeth, we changed the teeth design to three steel dowels. Three holes were drilled into the base of the specimen cup (replacing where the teeth would have been) as well as the top face of the securing block that holds the brackets and lateral force beams.

Peltier Coolers

In our team's final design we did not implement the Peltier coolers. For the design expo we did end up attaching four Peltier coolers to the securing block; two were placed on the front face (facing away from the motor) and two on the face that is one side counter-clockwise from it. Along with the Peltier coolers, Masscool 5C12B3 CPU Cools and Fans were also attached on top.

APPENDIX C: Design Analysis Assignment from Lecture

1. Material Selection Assignment (Functional Performance)

Two major components of our final design include the compressive (normal) and lateral (frictional) force measurement systems. These systems are rudimentary in concept and design in that they both involve a block of material that can be deformed and strained due to the applied force and two strain gages attached in order to measure the strain. This is a very simple method for measuring forces but the blocks of material for both the compressive and lateral forces must be able to be strained to a level that is able to be detected by our strain gages with a 1% resolution error (equivalent to a strain of 0.0001) and withstand the stresses with a minimum safety factor of 2. Therefore, it is vitally important to select an appropriate material based on given constraints. Table 37, below summarizes the function, objective and constraints for each force measurement block of material.

Table 37: Function, Objective, and Constraints of Selected Components

	Compressive Force Block	**Lateral Force Beam**
Function	Column in compression	Beam in bending
Objective	Maximize strain	Maximize strain
Constraints	Shape and volume fixed; maximum force fixed; cannot fail in compression	Shape and volume fixed; cannot fail in bending

Compressive Block

Selecting the material index for both of these components is somewhat unique since the goal is to maximize the strain while disallowing stress failures. For the compressive block, we know that the strain is equal to the normal stress, σ, divided by the Young's modulus, *E*, of the material, shown below.

$$\varepsilon = \frac{\sigma}{E}$$

Furthermore, we know that the normal stress is equal to the compressive force divided by the normal surface area of the block of material, *A*, shown below.

$$\sigma = \frac{F}{A}$$

The surface area has been set to 0.00580644 m^2 (a 3" x 3" block), and the maximum compressive force that the machine is designed to be capable of withstanding is 85,000 N. Using the stress equation, we know that the block must be able to withstand 14.64 MPa and with a safety factor of 2, the block must be able to withstand approximately 30 MPa at a minimum. Therefore, its compressive strength must be greater than 30 MPa. However, we must also look at the minimum force applied to ensure that that the material can produce enough strain for the strain gages to pick it up with no more than a 1% resolution error. The minimum compressive force is set to be 30 N, which yields the minimum compressive stress of 5.166 kPa. Now that we know the minimum expected stress and minimum acceptable strain, we can solve for the maximum Young's modulus by rearranging the above equation. Dividing the minimum stress by the minimum strain gives a maximum Young's modulus of 5.166 GPa. Knowing this as well as the range of expected stresses, we used CES software to determine applicable materials. The only other constraint that we decided to put onto our selected material was a maximum cost of $5/kg.

With all parameters set, we were given a list of 12 materials in which to choose from. The five front runners were chosen and we took down information about price, Young's modulus, and compressive strength. We also calculated the minimum strain expected for each material just as a check. This information is presented in Table 38, below.

Table 38: Characteristics of Five Possible Materials to be Used as a Compressive Block

	Price ($/kg)	**Young's Modulus (GPa)**	**Minimum Strain**	**Compressive Strength (MPa)**
Chlorosulfonated Polyethylene	4.90	0.01	0.000517	30
Natural Rubber (unreinfoced)	1.80	0.0018	0.00287	30
SEBS (Shore 50D)	3.80	0.02	0.000258	38
PVC (Flexible Shore 65A)	0.95	0.007	0.000738	30
PVC (Flexible Shore 80D)	0.95	0.032	0.000161	30

Based on these five material choices for the compressive blocks, we have decided to pick the PVC (Flexible Shore 80D) as our final choice. This material is by far the cheapest, along with the Shore 80D PVC at $0.95/kg, making it a very attractive choice of material. The distinguishing factor for the Shore 65D that made it the top choice over the Shore 80D was the smaller Young's modulus meaning the strains would be amplified while the compressive strength remains the same. This will allow for smaller resolution errors on force measurements.

Lateral Force Beam

For the lateral force beam, selecting a material is much more difficult due to the constraints and extremely large range of forces. We began by using the strain equation and the bending stress equation for beams where *F* is the frictional force applied, *L* is the beam length, *w* is the beam width, and *h* is the beam thickness.

$$\sigma = \frac{6FL}{wh^2}$$

We have set the beam length to 0.1143 m, the beam width to 0.0381 m and the beam thickness to 0.0127 m. Furthermore, we have calculated the minimum and maximum expected lateral forces on the beam based on the range of compressive forces and an expected maximum coefficient of friction equal to 0.15. Using a similar procedure to the one described for the compressive blocks, we calculated the maximum stress the beam must be able to endure using a safety factor of 2 and the maximum value of Young's modulus based on the minimum strain and bending stress. We determined that the tensile strength could be a minimum of 162.4 MPa and Young's modulus to be a maximum of 0.24 GPa. Plugging these numbers into the CES software under the limits tab, we yielded zero results that matched the criteria. This was not unexpected as the tensile strength requirement is excessive large for any known materials with such a low Young's modulus. Generally speaking, only a metal would have a high enough tensile strength for this application, but its Young's modulus would be far too high in order to give readable results with strain gage measurements under the lowest load conditions. In order for our design to work, it is clear that we needed at least two different kinds of materials, but for the purposes of this assignment, we stuck to the low end of the force spectrum.

In order to successfully chose a material, we lowered the maximum compressive force to 30,000 N, resulting in a minimum tensile strength of 50 MPa (once again, with a safety factor of 2) for the material that we select. With new numbers plugged into the CES software we were given a list of 8 materials. Of the eight, we chose 5 finalists and once again recorded, price, Young's modulus, and tensile strength. We also calculated the minimum strain expected for each material just as a check. This information is presented in Table 39, below.

Table 39: Characteristics of Five Possible Materials to be Used as a Lateral Force Beam

	Price ($/kg)	Young's Modulus (GPa)	Minimum Strain	Compressive Strength (MPa)
Leather	18.00	0.1	0.00024	50
PEBA (Shore 55D)	8.40	0.145	0.000166	55
Polyurethane Rubber	5.10	0.03	0.0008	50
TPU (Shore 50D)	6.10	0.16	0.00015	50
TPU (Shore 60D)	6.10	0.235	0.000102	51

Based on these five material choices for the lateral force beam, we have decided to pick the Polyurethane Rubber as our final choice. This material is the cheapest by $1.00/kg and has the mandatory minimum tensile strength of 50 MPa while having the smallest Young's modulus, meaning it will give the best resolutions in strain over the range of tests.

2. Material Selection Assignment (Environmental Performance)

Since we have set the dimensions set for both the compressive block and the lateral force beam, it is very simple to calculate the needed mass of each of the selected materials. To calculate the mass, we m^3 multiplied the volume, V, of the component by the density, ρ, shown below.

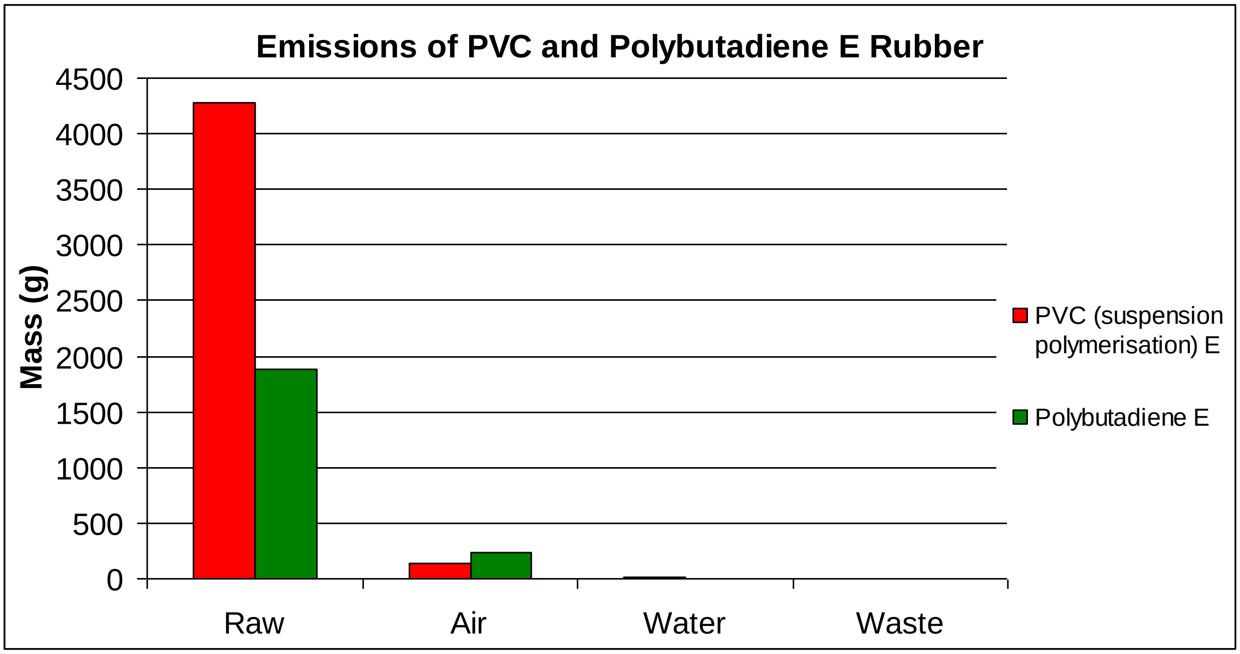
$$m = V\rho$$

Table 40: Mass Data for the Two Selected Materials

	Polyurethane Rubber for Lateral Force Beam	PVC (Flexible Shore 65A) for Compressive Block
Volume (m^3)	0.000055	0.000049
Density (kg/ m^3)	1200	1300
Mass (kg)	0.066368	0.06391

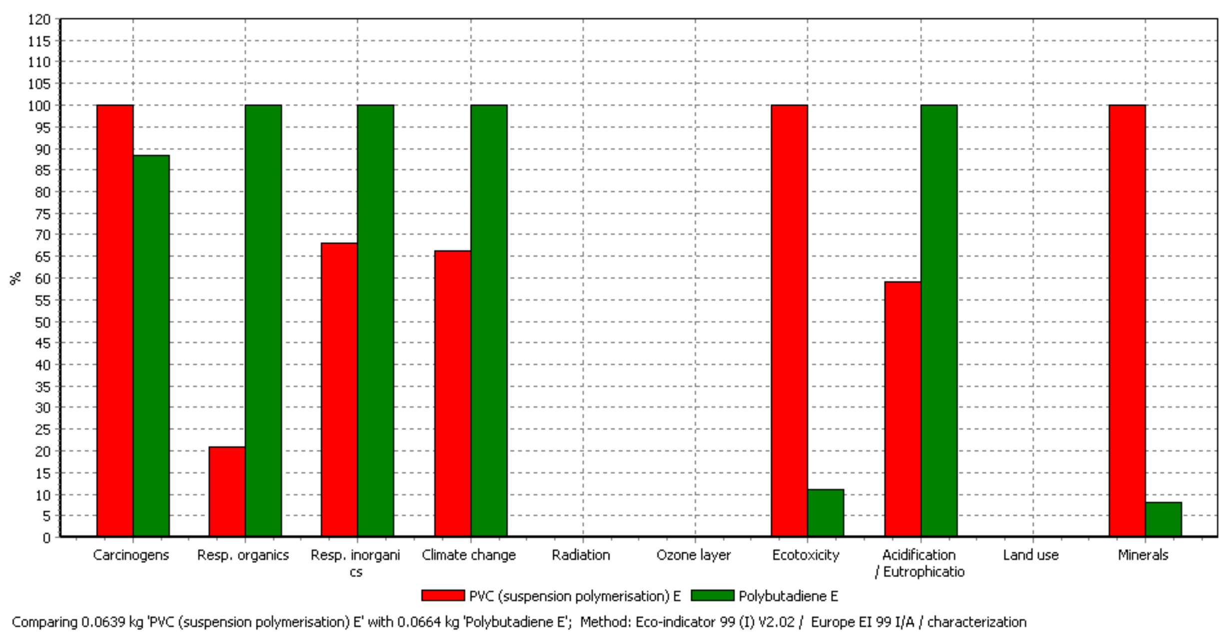
Once we calculated the masses of each component, we used the SimaPro software package to assist us in determining the environmental impacts each component would have. Within SimaPro, it was difficult to find exact matches of the materials we chose in CES, but we were able to choose very similar materials. For the PVC (Flexible Shore 65A), we chose PVC (suspension polymerisation) E. For the Polyurethane Rubber, we chose Polybutadiene E Rubber within SimaPro. By way of comparison in the software, we were able to calculate the total mass of air emissions, water emissions, use of raw materials, and solid waste. This information is shown in Figure 45, below.

Figure 45: Mass of Emissions from PVC and Polybutadiene E Rubber



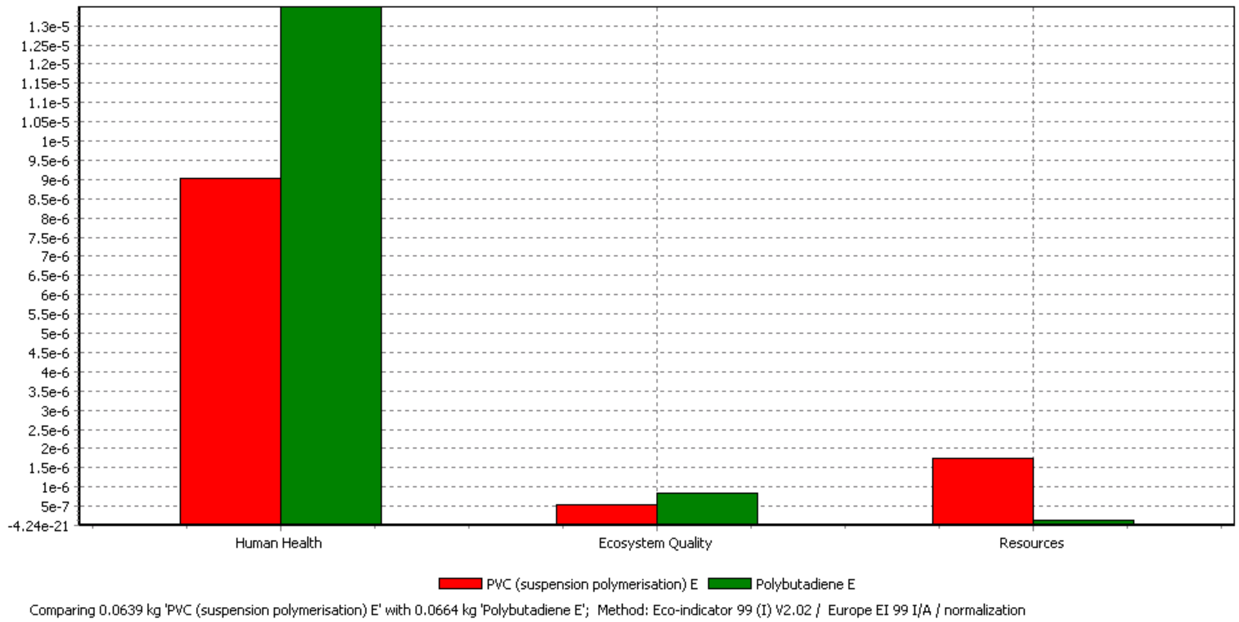
Based on the EcoIndicator 99 damage classifications, shown in Figure 46 below, it is not immediately apparent which material has a bigger impact on the environment. It seems as though Polybutandiene E has a bigger impact based on the data given on the left side of the bar chart, and the category of Acidification/Eutrophication, but in the categories of Ecotoxicity and Minerals, the PVC has a larger effect on the environment.

Figure 46: EcoIndicator 99 Damage Classifications



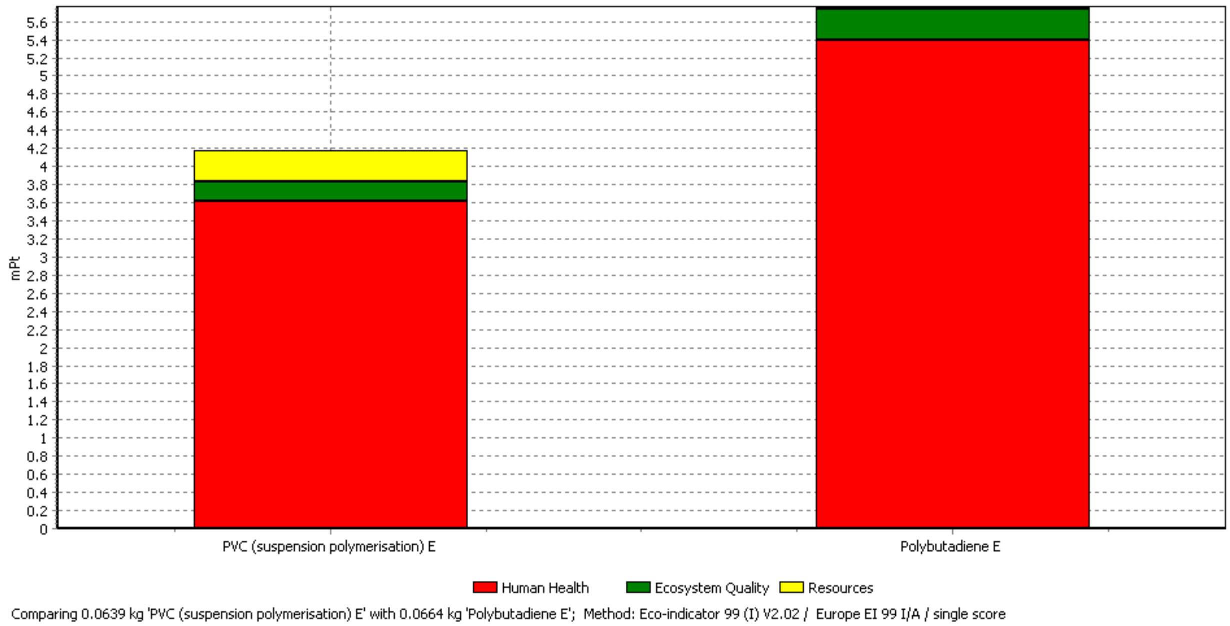
Looking deeper into the matter, we looked at the damage meta-categories of Human Health, Ecotoxicity and Resources. The point values for each of these categories are given in Figure 47, below. From the chart, it is clear that Human Health is the most important category and should be of greatest importance when considering material usage for the two components. The point values of the two materials for Ecosystem Quality are very low and only the PVC has a relatively high value for the Resources category.

Figure 47: Damage Meta – Categories



In the SimaPro software, the total point values were calculated and plotted, seen in Figure 49, p.92. Consistent with our conclusions from the meta-categories chart above, we see that the Polybutadiene E has a higher point value on the EcoIndicator 99 scale. When the life cycle of the whole product is taken into consideration, it is hard to say which component material would have a bigger impact. This is because the Polybutadiene E rubber lateral beam has a larger point value on a one time basis, but over the life of the product, we believe the PVC compressive block would be the component to wear faster due to a longer amount of time exposed to high forces than the lateral beam, and have to be replaced more often than the lateral beam. Because of this increased replacement of the compressive block, it may be more damaging to the environment over the products life cycle, which will likely be several decades.

Figure 48: EcoIndicator 99 Total Point Values



It is hard to say that we would pick different materials based on this environmental assessment. The environmental impact seems relatively low for both materials, making them both attractive to be used. However, if we were to try to eliminate some of the damage caused by using Polybutadiene E, we could switch to a different material, like TPU and spend slightly more money, or use several beams of different low impact materials and thicknesses in order to satisfy the requirements of the tests. This is in fact what we did, using a high density polyethylene for the lowest force range, and three different thickness aluminum beams for the mid to upper range force levels expected on the lateral force beam.

3. Manufacturing Process Selection Assignment

Our product is being developed specifically for the use of our sponsor, Professor Gordon Krauss, but could potentially have a use in other University laboratories for tribology research purposes. Based on the research we conducted early in the project and our knowledge of industry testing, we believe a starting estimate for the production volume is roughly 100 units. Being a relatively small number in the way of producing products as basic as the two components we are examining, we had to come up with manufacturing approaches that would keep costs low but quality high.

Using the CES software once again, we were able to analyze the many different forming and manufacturing processes available for each of the two materials. When we actually made the components, we ordered raw material that had most likely been extruded to stock bars and sheets and machined it using milling machines to the specifications we had set. However, in looking at the processes on CES, we saw that there is a very high equipment cost associated with milling and other options can yield a cheaper unit investment with a limited production run of 100 units. After examining the processes for both materials, we determined that water jet cutting (WJC) the two components as if we were producing multiple prototypes would be a more cost effective and accurate method of manufacturing. The specifications for WJC are given in Figure 50, p.93.

Figure 49: Water Jet Cutting Specifications

Shape				
Flat sheet	✓			
Physical attributes				
Range of section thickness	5e-4	-	0.025	m
Tolerance	0	-	5e-4	m
Roughness	0.8	-	6.3	µm
Cutting speed	0.05	-	0.5	m/s
Minimum cut width	7.5e-5	-	4e-4	m
Process characteristics				
Machining processes	✓			
Cutting processes	✓			
Prototyping	✓			
Discrete	✓			
Continuous	✓			
Economic attributes				
Relative equipment cost	medium			
Relative tooling cost	low			
Labor intensity	low			

It can be seen that WJC is used on flat sheets, which both components are, and can cut up to thicknesses of 0.025 m or 1 inch, which is the thickest component we need to cut. Furthermore, the tolerance is superb for this process, more than good enough for our intended purposes. More important to these components is the surface roughness, since strain gages need to be applied to the material. For WJC, the surface roughness is very small, ranging from 0.8 – 6.3 µm. The cutting speed can be very fast and all process characteristics can be obtained using WJC. Another attractive feature of WJC is the fact that the costs are all relatively low, good for a product with a production run of only 100.

APPENDIX D: Analysis Calculations

GEAR ANALYSES CALCULATIONS

Motor Specifications:
Full load torque = 11.8 N-m
Full load speed = 1765 rpm
75% motor speed = 1324 rpm

TCT Gear Ratio:
Max normal force = 100 kN
Max expected coefficient of friction, $\mu = 0.15$
Max friction force = μF = 15 kN
Radius of Annulus = 0.0111 m
Max moment about center of cup = r*Ff = 166.5 N-m
Gear ratio = Max torque/full load torque of motor = 166.5/11.8 = 14
Gear ratio (pinion:gear) = 1:14

Four-ball Gear Ratio:
Max normal force = 16 kN
Max expected coefficient of friction, $\mu = 0.15$
Max friction force = μF = 2.4 kN
Radius of Annulus = 0.00367 m
Max moment about center of cup = r*Ff = 8.79 N-m
Gear ratio = Max motor speed/75% motor speed = 3600/1324 = 2.7
Gear ratio (pinion:gear) = 2.7:1

STRESS ANALYSES CALCULATIONS

Compressive Block:
Max normal force (TCT) = 100 kN
Compressive block area = 0.0058 m^2
Compressive stress = F/A = 17.2 MPa

PVC (30 – 999 N)
Max Compressive stress = 999 N/ 0.0058 m^2 = **0.172 MPa**
Young's Modulus = 2.74 GPa
Minimum strain = min stress/modulus = (30 N/ 0.0058 m2)/2.74 GPa = **1.9 x 10-6**

Aluminum (1 – 100 kN)
Max Compressive stress = 100 kN/ 0.0058 m^2 = **17.2 MPa**
Young's Modulus = 68.9 GPa
Minimum strain = min stress/modulus = (1000 N/ 0.0058 m2)/68.9 GPa = **2.5 x 10-6**

Bottom of Specimen Cup:
Max normal force (TCT) = 100 kN
Specimen plate area = .0508m x .0508 m = 0.002581 m^2
Maximum stress on cup [F/A] = 100,000/0.002581 N/m^2 = **38.75 MPa**
Compressive strength of 6061 Aluminum (from CES EduPack) = **249 MPa**

Specimen Cup Teeth Shear Stress:
Max normal force (TCT) = 100 kN
Max expected coefficient of friction, $\mu = 0.15$
Max friction force = μF = 15 kN
Radius of Annulus (TCT) = 0.0111 m
Max moment about center of cup = r*Ff = 166.5 N-m
Distance from tooth midpoint to center of cup = 0.0508 m
Max Shear force, V = M/d = 3277 N
Number of teeth = 4
Max shear force per tooth, V = 819 N
Tooth base, b = 0.0254 m
Tooth height, h = 0.0127 m
Moment of inertia, I = $(1/12)bh^3 = 4.3357 \times 10^{-9}$ m^4
Statical moment of area, Q = (bh/2)(h/4) = 5.12096 m^3
Shear Stress = VQ/It = **3.81 MPa**
Max allowable shear stress for 6061 Aluminum = **120.5 MPa**

Lateral Beams:
TCT:
Max moment about center of cup, M1 = 166.5 N-m
Distance from end of beam to center of cup = 0.2286 m
Max Reaction force at end of beam, F = M1/d = 728.35 N
Base of beam, b = 0.0381 m
Height (thickness) of beam, h = **0.0254 m**
Beam length, l = 0.1143 m
Beam moment arm, M2 = Fl = 83.25 N-m
Moment of inertia = $(1/12)bh^3 = 6.50362 \times 10^{-9}$ m^4
1/2 height of beam, y = 0.0127 m
Max bending stress = M2y/I = **20.3 MPa**
Tensile stress of 6061 Aluminum = **280 MPa**

Four-ball:
Max moment about center of cup, M1 = 8.79 N-m
Distance from end of beam to center of cup = 0.2286 m
Max Reaction force at end of beam, F = M1/d = 14.6 N
Base of beam, b = 0.0381 m
Height (thickness) of beam, h = **0.003175 m**
Beam length, l = 0.1143 m
Beam moment arm, M2 = Fl = 4.40 N-m
Moment of inertia = $(1/12)bh^3 = 1.0162 \times 10^{-10}$ m^4
1/2 height of beam, y = 0.0015875 m
Max bending stress = M2y/I = **26 MPa**
Tensile stress of 6061 Aluminum = **280 MPa**

Brackets (modeled as beams in bending):
Beam length, l = 0.0254 m
Beam base, b = 0.0381 m
Beam height, h = 0.0127 m
Force at end of beam, F = 728.35 N
Moment arm on beam, M = Fl = 18.5 N-m
Moment of inertia = $(1/12)bh^3$ =6.50362 x 10-9 m^4

1/2 beam height, y = 0.00635 m
Bending stress = My/I = **18.06 MPa**
Tensile stress of 6061 Aluminum = **280 MPa**

Safety Shield (withstanding broken lateral force beams)
TCT beam (0.0254 m thick)
Length of broken beam, l = 0.1143 m
Angular velocity, w = 30 rpm
Max velocity of broken beam = lw = 0.359 m/s
Mass of beam = volume x density = 0.1143 m x 0.0254 m x 0.0381 m x 2700 kg/m^3 = 0.299 kg
Impulse = mass x change in velocity (when hitting the shield) = 0.107 kg-m/s
Force = impulse/time = 10.7 N
Stress shield needs to endure = force/smallest area of beam = **3.4 MPa**

TCT beam (0.0254 m thick)
Length of broken beam, l = 0.1143 m
Angular velocity, w = 3600 rpm
Max velocity of broken beam = lw = 43.1 m/s
Mass of beam = volume x density = 0.1143 m x 0.003175 m x 0.0381 m x 2700 kg/m^3 = 0.03733 kg
Impulse = mass x change in velocity (when hitting the shield) = 1.609 kg-m/s
Force = impulse/time = 160.9 N
Stress shield needs to endure = force/smallest area of beam = **51 MPa**

HEAT TRANSFER CALCULATIONS

Heating:
Area of heating, A = 0.0001 m^2
Distance from heating cartridge to bottom surface of cup, s = 0.0111 m
Thermal conductivity of 6061 aluminum, k = 180 W/m-K
Rate of heat transfer, Q = 25 Watts
Surface temperature of cup, T_s = 150°C
Needed temperature of heating cartridge = (Qs)/(kA)+ T_s = **165.3°C**

Cooling:
Area of heating, A = 0.002581 m^2
Distance from heating cartridge to bottom surface of cup, s = 0.0111 m
Thermal conductivity of 6061 aluminum, k = 180 W/m-K
Rate of heat transfer, Q = -335 Watts
Surface temperature of cup, T_s = 0°C
Needed temperature of heating cartridge = (Qs)/(kA) + T_s = **-8.0°C**

APPENDIX E: DAQ Specification Sheets

NI 9211 4-Channel, 14 S/s, 24-Bit, ±80 mV Thermocouple Input Module

- 4 thermocouple or ±80 mV analog inputs
- -40 to 70 °C operating range
- 24-bit resolution; 50/60 Hz noise rejection
- Hot-swappable operation
- NIST-traceable calibration

Overview

The National Instruments NI 9211 thermocouple input module for use with NI CompactDAQ and CompactRIO chassis includes a 24-bit delta-sigma ADC, antialiasing filters, open-thermocouple detection, and cold-junction compensation for high-accuracy thermocouple measurements. The NI 9211 contains NIST-traceable calibration and channel-to-earth ground double isolation barrier for safety, noise immunity, and high common-mode voltage range.

Specifications

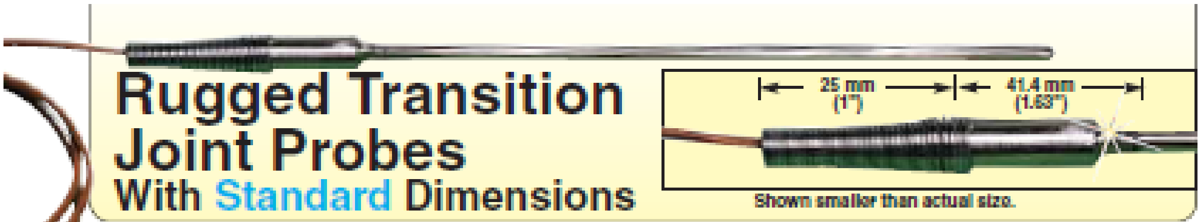
Specifications Documents

Detailed Specifications

Data Sheet

Product Family	Industrial I/O
Form Factor	CompactRIO , CompactDAQ
Operating System/Target	Windows , Real-Time
Measurement Type	Temperature , Thermocouple , Voltage
Isolation Type	Ch-Earth Ground Isolation
RoHS Compliant	Yes
Signal Conditioning	Cold-junction compensation
Analog Input Channels	0 , 4
Single-Ended Channels	0
Differential Channels	4
Resolution	24 bits
Sample Rate	14 S/s
Max Voltage	80 mV
Maximum Voltage Range	-80 mV , 80 mV
Minimum Voltage Range	-80 mV , 80 mV
Simultaneous Sampling	No
Analog Output Channels	0
Bidirectional Channels	0
Input-Only Channels	0
Output-Only Channels	0
Number of Channels	0 , 0 , 0
Counter/Timers	
Counters	0
Physical Specifications	
Length	9 cm

Width	2.3 cm
Minimum Operating Temperature	-40 °C
Maximum Operating Temperature	70 °C
Minimum Storage Temperature	-40 °C
Maximum Storage Temperature	85 °C



Rugged Transition Joint Probes
With Standard Dimensions

Note: See page A-70 for probe terminations. Please see page A-131 for compression fittings.

- 304 SS, 310 SS, 316 SS, 321 SS, Inconel or Super OMEGACLAD® XL Sheath
- Diameters from 1/16 to 1/4"
- 40" PFA Coated Stranded Lead Wire: 20 AWG for 3/16 and 1/4" OD, 24 AWG for 1/16 and 1/8" OD Probes
- Cal-5 Available

OMEGA® heavy duty transition joint probes offer convenient termination to PFA coated lead wire. The transition joint is 1.63" long, with a 1" spring for strain relief. The joint diameter is 3/8" for 3/16 and 1/4" Dia. probes, 1/4" for 1/16 and 1/8" Dia. probes.

Dual Element TJ Probes
To Order, add suffix "-DUAL" to model no. Add to base probe price any options (extra probe length, overbraiding, armored cable), then multiply this price x 1.75.
Ordering Example: TJ36-CASS-14G-12-DUAL, $35 x 1.75 = **$61.25**.

For additional PFA lead wire length, add **$1.00** per 12" over 40" and change "36" in part no. to desired length in inches.
Ordering Example: TJ120-ICSS-14G-12, type J transition joint probe with 120" of PFA lead wire, 1/4 OD stainless steel sheath, 12" length, $35 + 7 = **$42**.

Available as SUPER OMEGACLAD® XL — See page A-43 to A-48.

ANSI color code shown

To order IEC color code see page A-9

310 Stainless Steel Sheaths
To Order, replace "SS" in Model no. with 310SS, no add'l charge. **Ordering Example: TJ36-CA310SS-18G-12**, type K transition joint probe with 310 stainless steel sheath, $28.

Discount Schedule
1-10 units .... Net
11-24 units 10%
25-49 units 20%
50 and up .... Consult Sales

MOST POPULAR MODELS HIGHLIGHTED!

To Order *(Specify Model Number)*

Alloy/ANSI Color Code	Sheath Dia. (")	Model No. 12" Length	Price G*/E	Price U*	Model No. 18" Length	Price G*/E*	Price U*	Model No. 24" Length	Price G*/E*	Price U*	Price/ Add'l 12"
IRON-CONSTANTAN Inconel Sheath J	1/16	TJ36-ICIN-116(*)-12	$28.00	$30.00	TJ36-ICIN-116(*)-18	$28.80	$30.80	TJ36-ICIN-116(*)-24	$29.55	$31.55	$1.55
	1/8	TJ36-ICIN-18(*)-12	28.00	30.00	TJ36-ICIN-18(*)-18	29.80	31.80	TJ36-ICIN-18(*)-24	31.15	33.15	3.15
	3/16	TJ36-ICIN-316(*)-12	29.00	31.00	TJ36-ICIN-316(*)-18	31.20	33.20	TJ36-ICIN-316(*)-24	33.35	35.35	4.35
	1/4	TJ36-ICIN-14(*)-12	35.00	37.00	TJ36-ICIN-14(*)-18	38.80	40.80	TJ36-ICIN-14(*)-24	42.50	44.50	7.50
IRON-CONSTANTAN 304 SS Sheath J	1/16	TJ36-ICSS-116(*)-12	$28.00	$30.00	TJ36-ICSS-116(*)-18	$28.80	$30.80	TJ36-ICSS-116(*)-24	$29.55	$31.55	$1.55
	1/8	TJ36-ICSS-18(*)-12	28.00	30.00	TJ36-ICSS-18(*)-18	28.90	32.90	TJ36-ICSS-18(*)-24	29.85	31.85	1.85
	3/16	TJ36-ICSS-316(*)-12	29.00	31.00	TJ36-ICSS-316(*)-18	30.80	32.80	TJ36-ICSS-316(*)-24	32.15	34.15	3.15
	1/4	TJ36-ICSS-14(*)-12	35.00	37.00	TJ36-ICSS-14(*)-18	37.50	39.50	TJ36-ICSS-14(*)-24	40.00	42.00	5.00
CHROMEGA®-ALOMEGA® ** Inconel Sheath K	1/16	TJ36-CAIN-116(*)-12	$28.00	$30.00	TJ36-CAIN-116(*)-18	$28.80	$30.80	TJ36-CAIN-116(*)-24	$29.55	$31.55	$1.55
	1/8	TJ36-CAIN-18(*)-12	28.00	30.00	TJ36-CAIN-18(*)-18	29.80	31.80	TJ36-CAIN-18(*)-24	31.15	33.15	3.15
	3/16	TJ36-CAIN-316(*)-12	29.00	31.00	TJ36-CAIN-316(*)-18	31.20	33.20	TJ36-CAIN-316(*)-24	33.35	35.35	4.35
	1/4	TJ36-CAIN-14(*)-12	35.00	37.00	TJ36-CAIN-14(*)-18	38.80	40.80	TJ36-CAIN-14(*)-24	42.50	44.50	7.50
CHROMEGA®-ALOMEGA® 304 SS Sheath K	1/16	TJ36-CASS-116(*)-12	$28.00	$30.00	TJ36-CASS-116(*)-18	$28.80	$30.80	TJ36-CASS116(*)-24	$29.55	$31.55	$1.55
	1/8	TJ36-CASS-18(*)-12	28.00	30.00	TJ36-CASS-18(*)-18	28.90	30.90	TJ36-CASS-18(*)-24	29.85	31.85	1.85
	3/16	TJ36-CASS-316(*)-12	29.00	31.00	TJ36-CASS-316(*)-18	30.80	32.80	TJ36-CASS-316(*)-24	32.15	34.15	3.15
	1/4	TJ36-CASS-14(*)-12	35.00	37.00	TJ36-CASS-14(*)-18	37.50	39.50	TJ36-CASS-14(*)-24	40.00	42.00	5.00
CHROMEGA®-CONSTANTAN Inconel Sheath E	1/16	TJ36-CXIN-116(*)-12	$28.00	$30.00	TJ36-CXIN-116(*)-18	$28.90	$30.90	TJ36-CXIN-116(*)-24	$29.85	$31.85	$1.85
	1/8	TJ36-CXIN-18(*)-12	28.00	30.00	TJ36-CXIN-18(*)-18	29.90	31.90	TJ36-CXIN-18(*)-24	31.15	33.15	3.75
	3/16	TJ36-CXIN-316(*)-12	29.00	31.00	TJ36-CXIN-316(*)-18	31.50	33.50	TJ36-CXIN-316(*)-24	33.35	35.35	5.00
	1/4	TJ36-CXIN-14(*)-12	35.00	38.00	TJ36-CXIN-14(*)-18	39.80	41.80	TJ36-CXIN-14(*)-24	43.50	45.50	7.50
CHROMEGA®-CONSTANTAN 304 SS Sheath E	1/16	TJ36-CXSS-116(*)-12	$28.00	$30.00	TJ36-CXSS-116(*)-18	$28.90	$30.90	TJ36-CXSS-116(*)-24	$29.55	$31.55	$1.55
	1/8	TJ36-CXSS-18(*)-12	28.00	30.00	TJ36-CXSS-18(*)-18	29.90	31.30	TJ36-CXSS-18(*)-24	30.50	32.50	2.50
	3/16	TJ36-CXSS-316(*)-12	29.00	31.00	TJ36-CXSS-316(*)-18	31.50	32.60	TJ36-CXSS-316(*)-24	32.15	34.15	3.15
	1/4	TJ36-CXSS-14(*)-12	35.00	37.00	TJ36-CXSS-14(*)-18	39.80	39.50	TJ36-CXSS-14(*)-24	40.00	42.00	5.00
COPPER-CONSTANTAN Inconel Sheath T	1/16	TJ36-CPIN-116(*)-12	$28.00	$30.00	TJ36-CPIN-116(*)-18	$28.90	$30.90	TJ36-CPIN-116(*)-24	$29.55	$31.55	$1.85
	1/8	TJ36-CPIN-18(*)-12	28.00	30.00	TJ36-CPIN-18(*)-18	29.90	31.90	TJ36-CPIN-18(*)-24	31.75	33.75	3.75
	3/16	TJ36-CPIN-316(*)-12	29.00	31.00	TJ36-CPIN-316(*)-18	31.50	33.50	TJ36-CPIN-316(*)-24	33.35	35.35	5.00
	1/4	TJ36-CPIN-14(*)-12	35.00	38.00	TJ36-CPIN-14(*)-18	39.80	41.80	TJ36-CPIN-14(*)-24	43.50	45.50	7.50
COPPER-	1/16	TJ36-CPSS-116(*)-12	$28.00	$30.00	TJ36-CPSS-116(*)-18	$28.80	$30.80	TJ36-CPSS-116(*)-24	$29.55	$31.55	$1.55

NI 9237
4-Channel, ±25 mV/V, 24-Bit Simultaneous Bridge Module

- 24-bit resolution, ±25 mV/V analog inputs with RJ50 connectors
- 4 simultaneously sampled analog inputs; 50 kS/s maximum sampling rate
- Programmable half- and full-bridge completion; up to 10 V internal excitation
- Smart-sensor (TEDS) compatible
- 1,000 Vrms transient isolation
- -40 to 70 °C operating range

Overview

The National Instruments 9237 simultaneous bridge module for use with NI CompactDAQ and CompactRIO contains all the signal conditioning required to power and measure up to four bridge-based sensors simultaneously. The four RJ50 jacks provide direct connectivity to most torque or load cells and offer custom cable solutions with minimal tools. The high sampling rate and bandwidth of the NI 9237 offer a high-quality, high-speed strain or load measurement system with zero interchannel phase delay. With 60 VDC isolation and 1,000 Vrms transient isolation, the NI 9237 has high common-mode noise rejection and increased safety for both the operator and test system.

The NI 9237 can perform offset/null as well as shunt calibration and remote sense, making the module the best choice for strain and bridge measurements.

The NI 9944 and NI 9945 are accessories for use with quarter-bridge sensors. These accessories have a female RJ50 connector on one end and screw terminals on the other end. Purchase these accessories with a kit of RJ50 cables (qty 4).

For screw terminals without the quarter-bridge completion, purchase the NI 9949 and a kit of RJ50 cables (qty 4). This setup exposes all 10 pins for each channel as screw terminals.

Specifications Documents

- Detailed Specifications
- Data Sheet

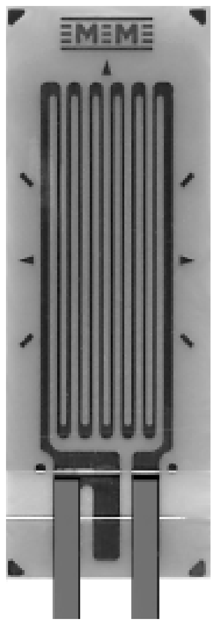
Specifications Summary

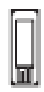
General	
Product Name	NI 9237
Product Family	Industrial I/O
Form Factor	CompactDAQ , CompactRIO
Operating System/Target	Windows , Real-Time
Measurement Type	Bridge-based sensor
Isolation Type	Ch-Earth Ground Isolation
RoHS Compliant	Yes
Signal Conditioning	Bridge completion , Voltage excitation , Anti-aliasing filter

Analog Input	
Channels	0 , 4
Single-Ended Channels	0
Differential Channels	4
Resolution	24 bits
Sample Rate	50 kS/s
Max Voltage	25 mV/V
Maximum Voltage Range	-25 mV/V , 25 mV/V
Maximum Voltage Range Accuracy	0.038 mV/V
Simultaneous Sampling	Yes
Excitation Voltage	10 V , 5 V , 3.3 V , 2.5 V
Bridge Configurations	Quarter Bridge , Half Bridge , Full Bridge
Analog Output	
Channels	0
Digital I/O	
Bidirectional Channels	0
Input-Only Channels	0
Output-Only Channels	0
Number of Channels	0 , 0 , 0
Counter/Timers	
Counters	0
Physical Specifications	
Length	9 cm
Width	2.3 cm
I/O Connector	RJ50
Minimum Operating Temperature	-40 °C
Maximum Operating Temperature	70 °C
Minimum Storage Temperature	-40 °C
Maximum Storage Temperature	85 °C

General Purpose Strain Gages - Linear Pattern

GAGE PATTERN DATA





actual size

GAGE DESIGNATION See Note 1	RESISTANCE (OHMS)	OPTIONS AVAILABLE
L2A-XX-250LW-120	120 ± 0.6%	
L2A-XX-250LW-350	350 ± 0.6%	
C2A-XX-250LW-120	120 ± 0.6%	
C2A-XX-250LW-350	350 ± 0.6%	

DESCRIPTION

Widely used general-purpose gage.

RoHS COMPLIANT

GAGE DIMENSIONS Legend: ES = Each Section; S = Section (S1 = Sec 1); CP = Complete Pattern; M = Matrix

inch / millimeter

Gage Length	Overall Length	Grid Width	Overall Width	Matrix Length	Matrix Width
0.250	0.363	0.100	0.100	0.440	0.170
6.35	9.22	2.54	2.54	11.18	4.32

GAGE SERIES DATA

See Gage Series data sheet for complete specifications.

Series	Description	Strain Range	Temperature Range
L2A	Encapsulated constantan gages with preattached ribbon leads.	±3%	–100° to +250°F [–75° to +120°C]
C2A	Encapsulated constantan gages with preattached ready-to-use cables.	±3%	–60° to +180°F [–50° to +80°C]

Infrared Sensor

1-999,990 RPM. Both the IRS-P and IRS-W Infrared Sensors are proven to be reliable high speed sensors. With a working range of 12 to 25 mm from very high speed equipment or an application providing contrasting light and dark surfaces.

TACHOMETERS	**Rechargers/Power Supplies**		**SENSORS W=Tinned Wires; P = 1/8" Phone Plug**	
	R-5 (115 Vac)	R-6 (230 Vac)	IRS-W	IRS-P
PT99	N/A	N/A		
PLT-200	N/A	N/A		X
Pocket-Tach 100	N/A	N/A		
Pocket-Tach Plus Kit	N/A	N/A		X
Phasar-LCD	X	X		
Phasar-LCD-R	X	X		
Phasar-Laser	X	X		
Phasar-Laser-R	X	X		
Tach-4A	X	X		
Tach-4AR	X	X		X
ACT-1B	12 Vdc or AC Powered		X	
ACT-2A			X	
ACT-3			X	
ACT-3A			X	

STROBOSCOPES	**Sensors - P = 1/8" Phone Plug all SPS products work with Nova-Strobe Series**	**Lamps**		
	IRS 5P	L1902	L1903	L1904
Nova-Strobe AA *		X		
Nova-Strobe AB *		X		
Nova-Strobe BA		X		
Nova-Strobe BB		X		
Nova-Strobe DB *			X	
Nova-Strobe DB Plus	X		X	
Nova-Strobe DA Plus	X		X	
Palm Strobe	X			X
Phaser-Strobe PB	X		X	

Self-Contained DA Plus SC	X		X	
Vibration Strobe			X	
* Obsolete				

Specifications

MODEL	IRS-P IRS-W
Operating Distance	0.5 to 1.0" (12 mm) sensing gap
Speed Range	1-999,990 RPM
Operating Temperature	-10° to 212° F (-23° to 100° C)
Power required	5 Vdc, 10 mA
Output Signal	TTL 5-0 Vdc
Standard Cable	8 feet (2.4 m)
Dimensions	0.625" x 2.90" (16 x 73 mm)

Low-Cost, Bus-Powered Multifunction DAQ for USB – 12- or 14-Bit, up to 48 kS/s, 8 Analog Inputs

NI USB-6008, NI USB-6009

- 8 analog inputs at 12 or 14 bits, up to 48 kS/s
- 2 analog outputs at 12 bits, software-timed
- 12 TTL/CMOS digital I/O lines
- 32-bit, 5 MHz counter
- Digital triggering
- Bus-powered
- 1-year warranty

Operating Systems

- Windows Vista (32- and 64-bit)/XP/2000
- Mac OS X[1]
- Linux®[1]
- Windows Mobile[1]
- Windows CE[1]

Recommended Software

- LabVIEW
- LabVIEW SignalExpress
- LabWindows™/CVI
- Measurement Studio

Other Compatible Software

- C#, Visual Basic .NET
- ANSI C/C++

Measurement Services Software (included)

- NI-DAQmx driver software
- Measurement & Automation Explorer configuration utility
- LabVIEW SignalExpress LE

[1]You need to download NI-DAQmx Base for these operating systems.

Product	Bus	Analog Inputs[1]	Input Resolution (bits)	Max Sampling Rate (kS/s)	Input Range (V)	Analog Outputs	Output Resolution (bits)	Output Rate (Hz)	Output Range (V)	Digital I/O Lines	32-Bit Counter	Trigger
USB-6009	USB	8 SE/4 DI	14	48	±1 to ±20	2	12	150	0 to 5	12	1	Digital
USB-6008	USB	8 SE/4 DI	12	10	±1 to ±20	2	12	150	0 to 5	12	1	Digital

[1]SE = single ended, DI = differential [2]Software-timed

Overview and Applications

With recent bandwidth improvements and new innovations from National Instruments, USB has evolved into a core bus of choice for measurement applications. The NI USB-6008 and USB-6009 are low-cost entry points to NI flagship data acquisition (DAQ) devices. With plug-and-play USB connectivity, these modules are simple enough for quick measurements but versatile enough for more complex measurement applications.

The USB-6008 and USB-6009 are ideal for a number of applications where low cost, small form factor, and simplicity are essential. Examples include:

- Data logging – quick and easy environmental or voltage data logging
- Academic lab use – student ownership of DAQ hardware for completely interactive lab-based courses (Academic pricing available. Visit **ni.com/academic** for details.)
- OEM applications as I/O for embedded systems

Recommended Software

National Instruments measurement services software, built around NI-DAQmx driver software, includes intuitive application programming interfaces, configuration tools, I/O assistants, and other tools designed to reduce system setup, configuration, and development time. National Instruments recommends using the latest version of NI-DAQmx driver software for application development in NI LabVIEW, LabVIEW SignalExpress, LabWindows/CVI, and Measurement Studio software. To obtain the latest version of NI-DAQmx, visit **ni.com/support/daq/versions**.

NI measurement services software speeds up your development with features including:

- A guide to create fast and accurate measurements with no programming using the DAQ Assistant.
- Automatic code generation to create your application in LabVIEW.
- LabWindows/CVI; LabVIEW SignalExpress; and C#, Visual Studio .NET, ANSI C/C++, or Visual Basic using Measurement Studio.
- Multithreaded streaming technology for 1,000 times performance improvements.
- Automatic timing, triggering, and synchronization routing to make advanced applications easy.
- More than 3,000 free software downloads available at **ni.com/zone** to jump-start your project.
- Software configuration of all digital I/O features without hardware switches/jumpers.
- Single programming interface for analog input, analog output, digital I/O, and counters on hundreds of multifunction DAQ hardware devices. M Series devices are compatible with the following versions (or later) of NI application software – LabVIEW, LabWindows/CVI, or Measurement Studio versions 7.x; and LabVIEW SignalExpress 2.x.

NATIONAL INSTRUMENTS™

Specifications

Typical at 25 °C unless otherwise noted.

Analog Input

Absolute accuracy, single-ended

Range	Typical at 25 °C (mV)	Maximum (0 to 55 °C) (mV)
±10	14.7	138

Absolute accuracy at full scale, differential[1]

Range	Typical at 25 °C (mV)	Maximum (0 to 55 °C) (mV)
±20	14.7	138
±10	7.73	84.8
±5	4.28	58.4
±4	3.59	53.1
±2.5	2.56	45.1
±2	2.21	42.5
±1.25	1.70	38.9
±1	1.53	37.5

Number of channels........................... 8 single-ended/4 differential
Type of ADC .... Successive approximation

ADC resolution (bits)

Module	Differential	Single-Ended
USB-6008	12	11
USB-6009	14	13

Maximum sampling rate (system dependent)

Module	Maximum Sampling Rate (kS/s)
USB-6008	10
USB-6009	48

Input range, single-ended.................. ±10 V
Input range, differential..................... ±20, ±10, ±5, ±4, ±2.5, ±2, ±1.25, ±1 V
Maximum working voltage .... ±10 V
Overvoltage protection .... ±35 V
FIFO buffer size .... 512 B
Timing resolution .... 41.67 ns (24 MHz timebase)
Timing accuracy .... 100 ppm of actual sample rate
Input impedance .... 144 kΩ
Trigger source...................................... Software or external digital trigger
System noise.. 5 m V_{rms} (±10 V range)

Analog Output

Absolute accuracy (no load) .... 7 mV typical, 36.4 mV maximum at full scale
Number of channels............................ 2
Type of DAC .... Successive approximation
DAC resolution.................................... 12 bits
Maximum update rate .... 150 Hz, software-timed
Output range .... 0 to +5 V
Output impedance............................... 50 Ω
Output current drive........................... 5 mA
Power-on state.................................... 0 V
Slew rate... 1 V/µs
Short-circuit current........................... 50 mA

Digital I/O

Number of channels........................... 12 total
8 (P0.<0..7>)
4 (P1.<0..3>)
Direction control .... Each channel individually programmable as input or output
Output driver type
USB-6008 .... Open-drain
USB-6009 .... Each channel individually programmable as push-pull or open-drain
Compatibility...................................... CMOS, TTL, LVTTL
Internal pull-up resistor .... 4.7 kΩ to +5 V
Power-on state.................................... Input (high impedance)
Absolute maximum voltage range...... -0.5 to +5.8 V

Digital logic levels

Level	Min	Max	Units
Input low voltage	-0.3	0.8	V
Input high voltage	2.0	5.8	V
Input leakage current	–	50	µA
Output low voltage (I = 8.5 mA)	–	0.8	V
Output high voltage (push-pull, I = -8.5 mA)	2.0	3.5	V
Output high voltage (open-drain, I = -0.6 mA, nominal)	2.0	5.0	V
Output high voltage (open-drain, I = -8.5 mA, with external pull-up resistor)	2.0	–	V

Counter

Number of counters............................ 1
Resolution .... 32 bits
Counter measurements....................... Edge counting (falling edge)
Pull-up resistor................................... 4.7 kΩ to 5 V
Maximum input frequency.................. 5 MHz
Minimum high pulse width................. 100 ns
Minimum low pulse width.................. 100 ns
Input high voltage.............................. 2.0 V
Input low voltage .... 0.8 V

Power available at I/O connector

+5 V output (200 mA maximum)......... +5 V typical
+4.85 V minimum
+2.5 V output (1 mA maximum).......... +2.5 V typical
+2.5 V output accuracy .... 0.25% max
Voltage reference temperature drift... 50 ppm/°C max

[1]Input voltages may not exceed the working voltage range.

ABB ACS150 - 0.75kW & 0.55kW 230V 1ph to 3ph - AC Inverter Drive Speed Controller

Range: ACS150
Part No: ACS150-01E-04A7-2
Description:

ABB Drives ACS 150 AC Inverter for 0.75kW (1.0HP) 230V 3 Ph motor in VxF control to 4.7A and 0.55kW (0.75HP). Simple to set-up and converts single phase 230V input to three phase 230V for a standard AC Induction motor.

R1 Size - 70mm Wide x 144mm Deep x 201mm high (plus cable clamp plate at 38mm) IP20 case.

Overload - 150% x 60seconds.
Speed Control Range - 0/500Hz.
Braking - To 56 Ohm minimum external resistor (not supplied) - use the 'Which Resistor' button on this page.

Features - Front Mounted Potentiometer, 1 x Analogue Input, 5 Digital Inputs, 1 Relay Contact set.
Programmable from a pc via 'Flashdrop'.
EMC Filters to EN61800-3 to the 2nd Environment C3 (Industrial). See linked products below for external 1st Environment (Domestic) EMC Filter.
Can be used with supplies protected by RCD Type A.

Input Current - 11.4A.
Input Voltage - 200/240V single phase +-10% at 50/60Hz.

Wall mount in clean environment or cubicle mount.
Rated at 40C Ambient.
Ventilation space above and below - 75mm.
Ventilation space at sides - 0mm.
Heat Loss at max output - 62W.
Mounting onto symmetrical DIN rail or use the screw fixings for side or rear mounting.

Full part number is - ACS150-01E-04A7-2
View alternatives - search for 0.75kW.

NI 9201 8-Ch, ±10 V, 500 kS/s, 12-Bit Analog Input Module, C Series

- 8 analog inputs, ±10 V input range
- 500 kS/s aggregate sampling rate
- 12-bit resolution, single-ended inputs, screw terminal or D-Sub connectors
- Hot-swappable operation; overvoltage protection; isolation
- NIST-traceable calibration
- -40 to 70 °C operating range

Overview

The NI 9201 is a C Series module for 8-channel analog input at a maximum aggregate rate of 500 kS/s. The NI 9201 represents a good combination of channel count and speed at a low price for an economical multifunction system.

As with the majority of C Series modules, the NI 9201 is protected from harmful voltage spikes of up to 2,300 Vrms. This means that no harmful voltage within the isolation rating can harm other modules in the system, the chassis, or any connected computer equipment. In addition to the absolute protection from the isolation, there is up to 100 V of overvoltage protection for errant signal connection or unexpected outputs to the individual channels.

There are two connector options for the NI 9201 – a 10-position screw terminal connector for direct connectivity and a 25-position D-Sub connector. The industry-standard 25-position D-Sub connector provides low-cost cabling to a wide variety of accessories available from NI or other vendors. A number of vendors also provide custom D-Sub cable fabrication services, and can provide cables with a pin-out that matches your exact application needs.

NI recommends the NI 9932 strain-relief connector accessory for the NI 9201. The NI 9934 (or other 25-pin D-Sub connector) is required for use with the NI 9201 with D-Sub module. The NI 9934 includes a screw-terminal connector with strain relief as well as a D-Sub solder cup backshell for creating custom cable assemblies.

Specifications

General

Product Name	NI 9201
Product Family	Industrial I/O
Form Factor	CompactDAQ , CompactRIO
Operating System/Target	Windows , Real-Time
Measurement Type	Voltage
Isolation Type	Ch-Earth Ground Isolation
RoHS Compliant	Yes
Product Name	NI 9201 D-Sub
Product Family	Industrial I/O

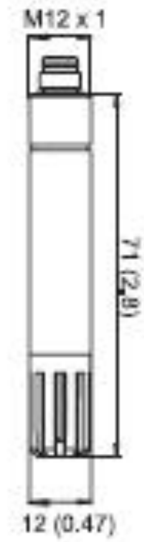
Form Factor	CompactDAQ , CompactRIO
Operating System/Target	Windows , Real-Time
Measurement Type	Voltage
Isolation Type	Ch-Earth Ground Isolation
RoHS Compliant	Yes
Analog Input	
Channels	8 , 0
Single-Ended Channels	8
Differential Channels	0
Resolution	12 bits
Sample Rate	500 kS/s
Max Voltage	10 V
Maximum Voltage Range	-10 V , 10 V
Maximum Voltage Range Accuracy	0.053 V
Minimum Voltage Range	-10 V , 10 V
Minimum Voltage Range Accuracy	0.053 V
Simultaneous Sampling	No
Channels	8 , 0
Single-Ended Channels	8
Differential Channels	0
Resolution	12 bits
Sample Rate	500 kS/s
Max Voltage	10 V
Maximum Voltage Range	-10 V , 10 V
Maximum Voltage Range Accuracy	0.053 V
Minimum Voltage Range	-10 V , 10 V
Minimum Voltage Range Accuracy	0.053 V
Simultaneous Sampling	No
Analog Output	
Channels	0
Channels	0
Digital I/O	
Bidirectional Channels	0
Input-Only Channels	0
Output-Only Channels	0
Number of Channels	0 , 0 , 0
Bidirectional Channels	0
Input-Only Channels	0
Output-Only Channels	0
Number of Channels	0 , 0 , 0
Counter/Timers	
Counters	0
Counters	0
Physical Specifications	
Length	9 cm
Width	2.3 cm
I/O Connector	Screw terminals
Minimum Operating Temperature	-40 °C
Maximum Operating Temperature	70 °C
Minimum Storage Temperature	-40 °C
Maximum Storage Temperature	85 °C
Length	9 cm
Width	2.3 cm
I/O Connector	25-pin D-Sub
Minimum Operating Temperature	-40 °C
Maximum Operating Temperature	70 °C
Minimum Storage Temperature	-40 °C
Maximum Storage Temperature	85 °C

VAISALA

www.vaisala.com

HMP50 Miniature Humidity and Temperature Probe for OEM Applications

The HMP50 features good measurement performance in a small, rugged and simple probe.



Features/Benefits

- Miniature-size humidity transmitter
- Very low power consumption - well suitable for battery-powered applications
- Short power-up time
- Measurement range 0 ... 98 % RH and -10 ... +60 °C (+14 ... +140 °F)
- Cable detachable with standard M8 quick connector
- Rugged metal housing
- Interchangeable Vaisala INTERCAP® Sensor - no need for recalibration
- Optional outputs and cable lengths

The Vaisala INTERCAP® Humidity and Temperature Probe HMP50 is a simple and cost effective humidity transmitter suitable for volume applications or integration into other manufacturers' equipment.

The HMP50 is ideal for a variety of applications such as glove boxes, greenhouses, fermentation chambers, data loggers, and incubators.

Installation flexibility

The probe cable has a screw-on quick connector for easy installation. Two different cable lengths are available, and customers can also use any M8 series cable of their choice.

Several outputs available

The temperature measurement is optional. Three standard voltage outputs are available.

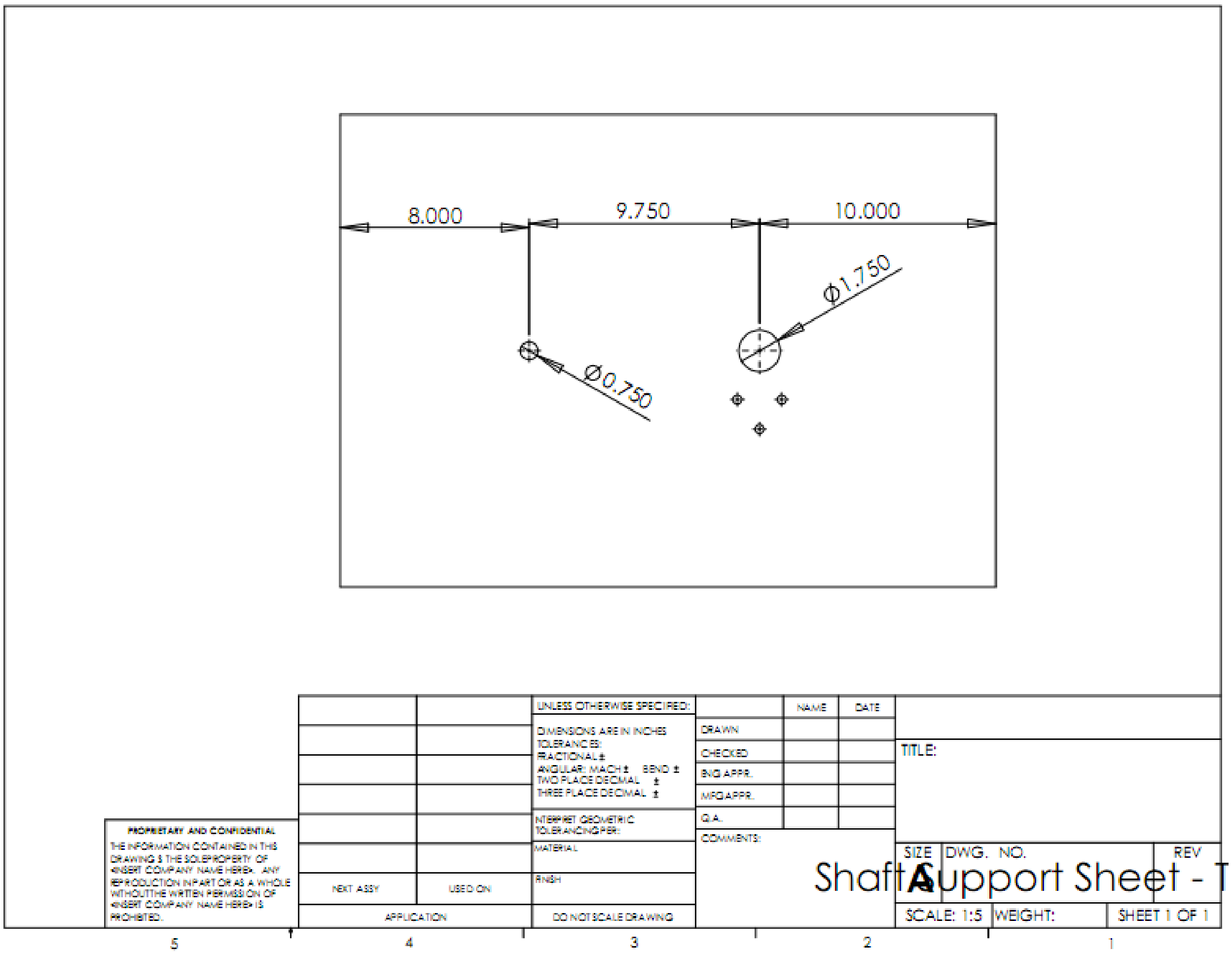
For the RH-only model, a current interface output is available. It can be used to build a 4 ... 20 mA loop-powered current output with external components (the optional current converter kit).

Rugged design

The aluminum body of the HMP50 is IP65-classified. The sensor is protected by a membrane filter and plastic grid, or optionally a stainless steel filter.

APPENDIX F: Additional Engineering Drawings and Basic Manufacturing Plans

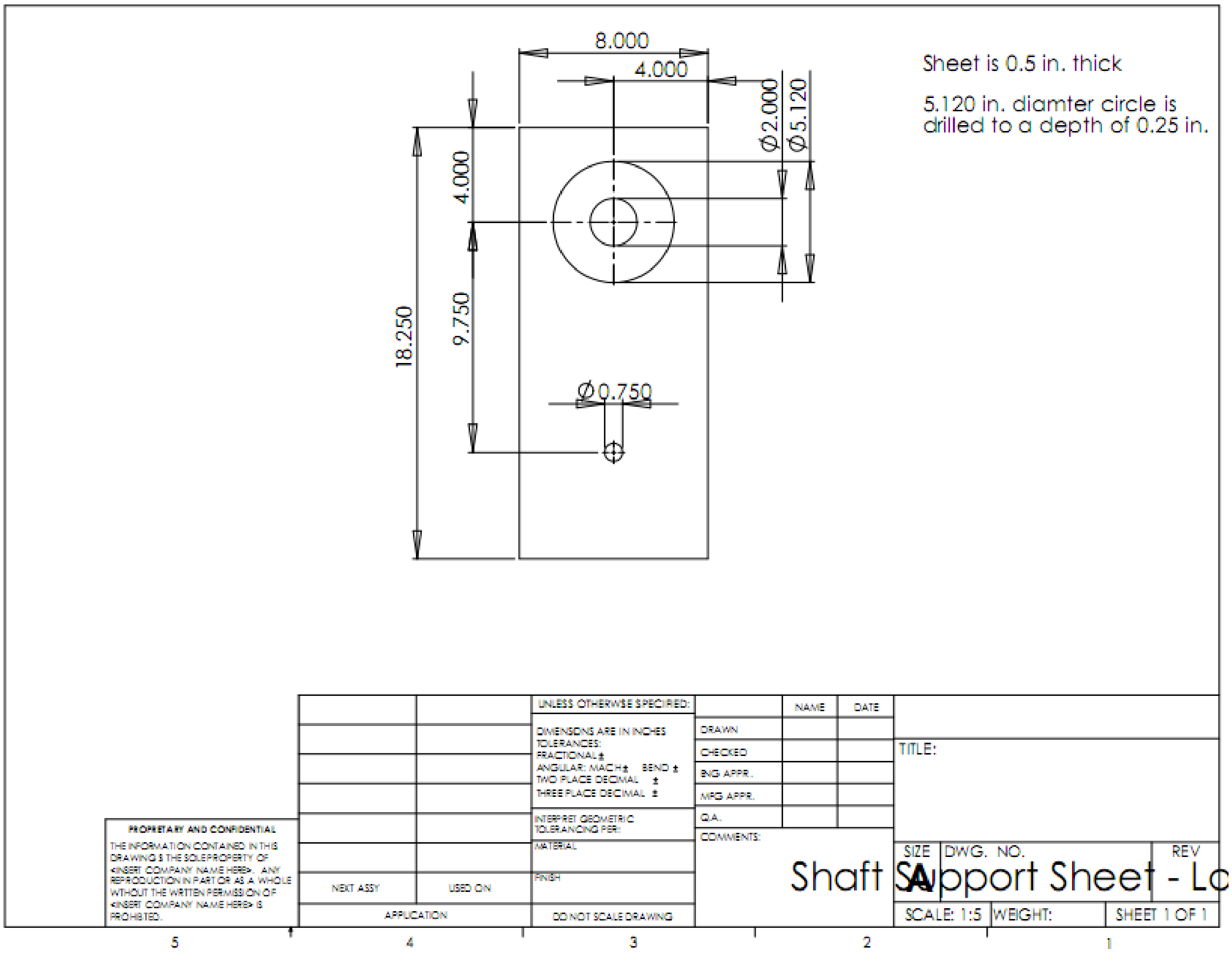
1. Shaft Support Sheet – Top



Task #	Purpose	Machine	Tool	Spindle Speed (rpm)
1	Cut to Size	Band Saw		Fast
2	Drill 2 in. hole	Drill Press	2” Drill	Fast
3	Drill ¾ in. hole	Drill Press	¾” Drill Bit	Fast

The primary dimensions are those between the two through holes, after the distance from the datum edge to the center of the 2 in. hole. Additional drilling will be done to account for the mounting of the bearings. This sheet will be ½ in. thick Low-Carbon Steel

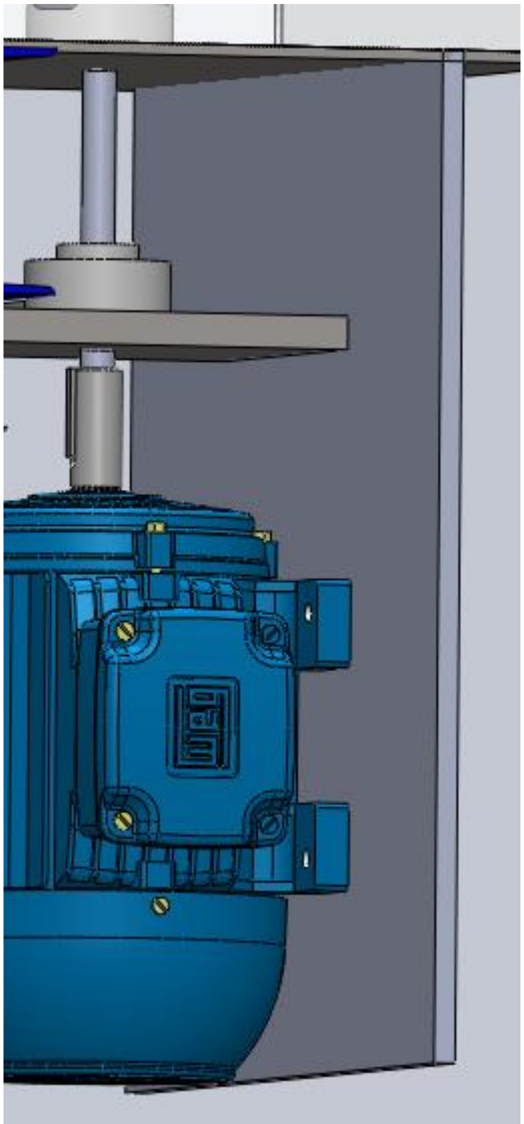
2. Shaft Support Sheet – Lower



Task #	Purpose	Machine	Tool	Spindle Speed (rpm)
1	Cut to Size	Band Saw		Fast
2	Drill 2 in. hole	Drill Press	2" Drill	Fast
3	Drill Counterbore (possibly)	Drill Press		Fast
4	Drill ¾ in. hole	Drill Press	¾" Drill Bit	Fast

The primary dimensions are those between the two through holes. The counterbore might not exist, that will depend on how the Thrust Bearing needs to be mounted. This sheet will be ½ in. thick Low-Carbon Steel

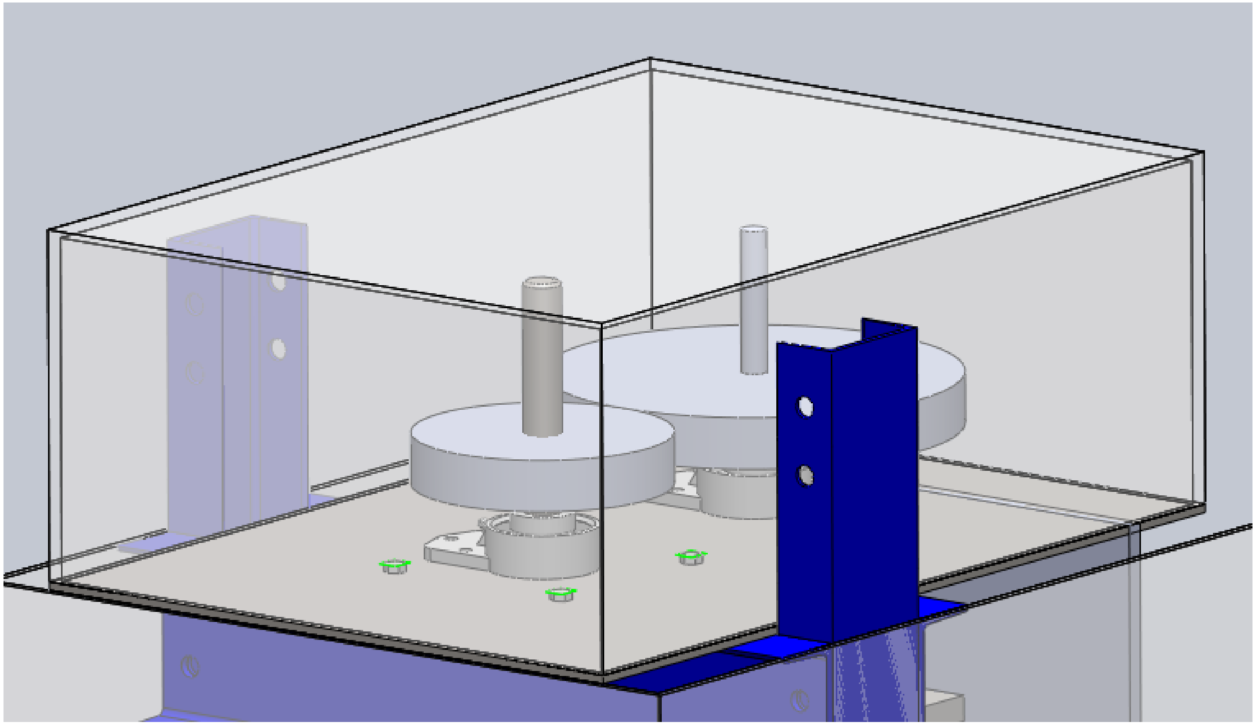
3. Shaft Support Sheet – Rear



Task #	Purpose	Machine	Tool	Spindle Speed (rpm)
1	Cut to Size	Band Saw		Fast
2	Drill Brackets for Top Sheet	Drill Press	½" Drill	Fast
3	Drill Brackets for Lower Sheet	Drill Press	½" Drill	Fast

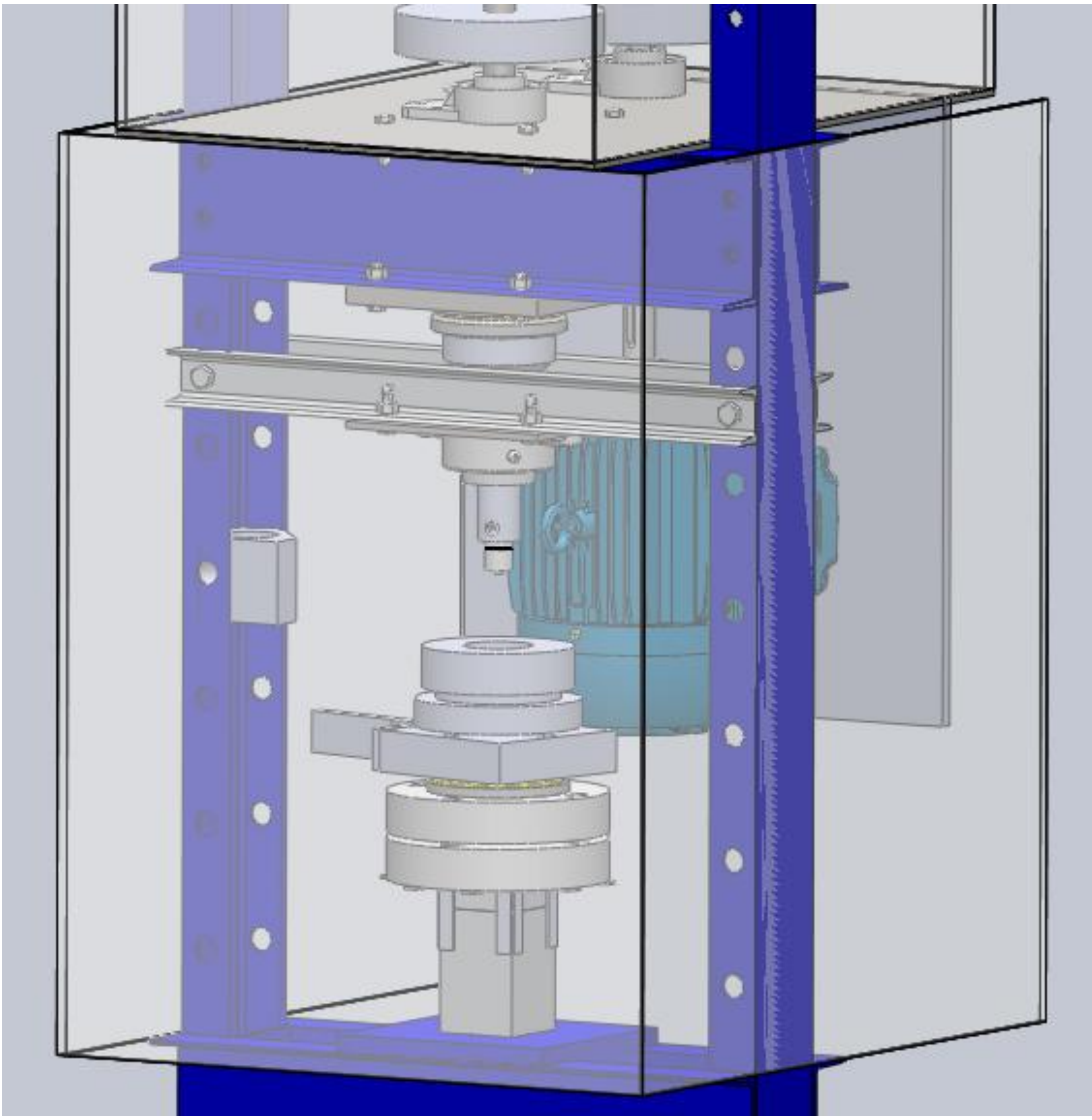
Once the Top and Lower sheets are in place, measurements for bracket placement will be finalized. This sheet will be ½ in. thick Low-Carbon Steel

4. Gear Enclosure



This enclosure is constructed from multiple sheets of ¼ in. material. Once the final shaft lengths are established and the Top Shaft Support sheet is in place, then one inch will be afforded outside of the largest gear dimensions and on top of the shafts.

5. Environment Enclosure



This box is constructed of multiple sheets of ¼ in. material that are fitted to where the final Shaft Support Sheets fall. The sides will be secured to the outer edges of the shop press, and the front will eventually be able to be opened.

APPENDIX G: Supplementary Safety Report for Fabrication and Assembly

ME 450 Safety Reporting: Winter 2010

Project #: ____6_____ **Date:** _____3/10/10______________

Report Version #: ____1.0_____

Project Title: _Twist Compression and Four Ball Test Device Redesign__

Team Member Names:_Justin Hopkins, Rachel Gunderson, Rich Main, Scott Malinowski__

Team Member Uniquenames:_jghop, rachelrg, richacha, scomal__

Attach your Safety Report to this cover page and instructions found on Pages 2 and 3.

The Safety Report is to be completed by your team and must be approved by your section instructor (or approved substitute) prior to any hands-on experimentation, manufacturing or testing of your prototype.

The safety hazards inherent in your experimental plans, component selection, manufacturing methods, assembly techniques, and testing must be expressed and evaluated before any hands-on work with safety consequences will be allowed to proceed.

The purpose of this safety report is to assure that you have thought through your hands-on work before it begins, and that you have shared your plans with your Section Instructor. You may submit more than one version. This will likely be necessary as your project evolves.

APPROVAL:

Name: ___________________________

Signature: ___________________________

Date: ___________________________

1. Executive Summary

To implement our redesign concepts for securing the specimen, securing the specimen fixture, and applying and measuring the lateral forces from the specimen fixture, we need to manufacture new components. This report will present the materials needed to produce our new components. It will also include manufacturing and assembly plans for our redesign concepts, as well as precautions that we will take in the shop to ensure safe and successful machining.

Because the maximum proposed loads for our prototype are very high, the engineering analyses done on our redesigned components studied how each component would respond under maximum stress. Specific areas that were considered included the shear stress on the bottom teeth of the specimen fixture, the compressive stress on the bottom of the specimen fixture, the bending stress on the lateral force beams, and the bending stress on brackets holding the lateral force beams. Using the torques from the maximum proposed loads and the distances from our redesign concepts, we calculated that the yield stresses of our components had a safety factor of at least 2 over the maximum stresses that they could be subject too. This led us to conclude that our redesign concepts were low-risk and ready for manufacture. All calculations are shown in Appendix B. We also conducted a Designsafe risk assessment on our redesign concepts to ensure that we did not miss any design alterations that would make the concepts safer.

To ensure that we do safe and efficient work in the shop, we have developed thorough manufacturing and assembly plans. These include an inventory of purchased material, detailed engineering drawings, cutting speeds and feed rates for each component and material, a machining procedure list, and an assembly procedure list. We have purchased one block and two rounds of aluminum to produce our components. Our machining will be done using a mill, a lathe, and a band saw.

As we will be using large and powerful machinery in the shop, safety precautions will need to be implemented and followed closely to ensure that there are no injuries. We have identified several hazards involved when using this machinery, and even when simply being present in the shop using DesignSafe. These include, but are not limited to, human contact with running machines, tool failure and fracture leading to flying debris, and machine instability or faulty functionality. Several of our proposed safety precautions are rules of the shop, including wearing safety glasses and appropriate clothing. Other precautions that we will follow include double-checking that machines are set to their proper cutting speeds and feed rates for given materials, that the tools are in good condition and fixed securely in the machines, and that the pieces of stock that we are machining are tightly secured in vices. Finally, we will consult with Bob or Marv to ensure that the machines are set up correctly before we begin any manufacturing.

3. Material Inventory

Since we designed entirely new components for securing the specimen, securing the specimen fixture, and applying and measuring the lateral forces from the specimen fixture, we purchased the raw material necessary for fabricating them. The raw material we purchased was 6061 aluminum alloy. Three pieces of various sizes and prices of the alloy were purchased:

1. 4” x 4” x 6” block $56.90
2. 4” diameter x 6” long round $46.52
3. 6” diameter x 6” long round $93.23

From the 4” square block, the lateral force deflecting block and lateral force beam brackets will be fabricated.

From the 4” diameter round, the two rings, one for the Four-ball test and one for the Twist Compression Test (TCT) will be fabricated. The two rings are used to secure either the four balls or flat plate.

From the 6” diameter round, the specimen cup and the outer collar of the cup will be fabricated. The cup houses the lubrication and specimen, while the outer collar secures the ring within the cup.

All components can be seen interacting with one another on page 7.

4. CAD Drawings and DesignSafe Summary for Design Parts

For the components that we will be manufacturing, detailed engineering drawings are provided in this section. In addition to the drawings (beginning on page 7), a safety evaluation has been performed not only on the parts themselves but the manufacturing processes that will be used to fabricate them using the DesignSafe software package. Complete tables summarizing the hazards, risks associated and ways to reduce these risks can be found in Appendix A.

Using the software, it was determined that there were very few hazards involved with the actual parts. None of the components we are producing involve the use of any chemicals or high internal pressures. They are simply external housings and support structures. The three main hazards we recognize as pertinent to the components are cutting/severing, fatigue, and lifting/bending/twisting. The parts could have been designed with many sharp points or edges which could potentially hurt the user or anyone handling them. In order to avoid injury, we have designed the parts with mainly blunt ends and only 90° edges when possible. Unless the user is careless, it should be very low risk that they get cut by any of the components. More important to the design of the components is the risk of fatigue once they are implemented onto the device and experience high loads. If the parts had been designed with insufficient thicknesses in regions that experience the loading, they could yield or fracture resulting in a complete failure of the test region which would be extremely risky and dangerous for anyone in the area. To avoid fatigue, we had to make sure that the specimen cup, beams and brackets would be able to withstand loading. Stress analysis was performed on all parts to ensure that they would not fail using a safety factor of at least 2. See Appendix B for calculations. Finally, since several of the components that we are fabricating will be removed from the device by hand on a regular basis to set up different tests, close attention had to be paid to ergonomics. If too heavy, the parts could injure the user when being lifted out of the device, so we made sure not to use too much metal in places where it was not necessary. Also, we chose to use standard 6061 aluminum over steel due to its lightweight.

For the actual fabrication of the components, there are many more inherent dangers, simply due to the hazards associated with many common machining processes that we will be using. The main hazards associated with fabricating the parts are as follows:

- Cutting/severing
- Drawing-in/trapping/entanglement
- Pinch point
- Unexpected start
- Fatigue
- Break up during operation
- Machine instability
- Impact
- Debris
- Human errors/behaviors
- Deviations from safe work practices
- Interactions between persons

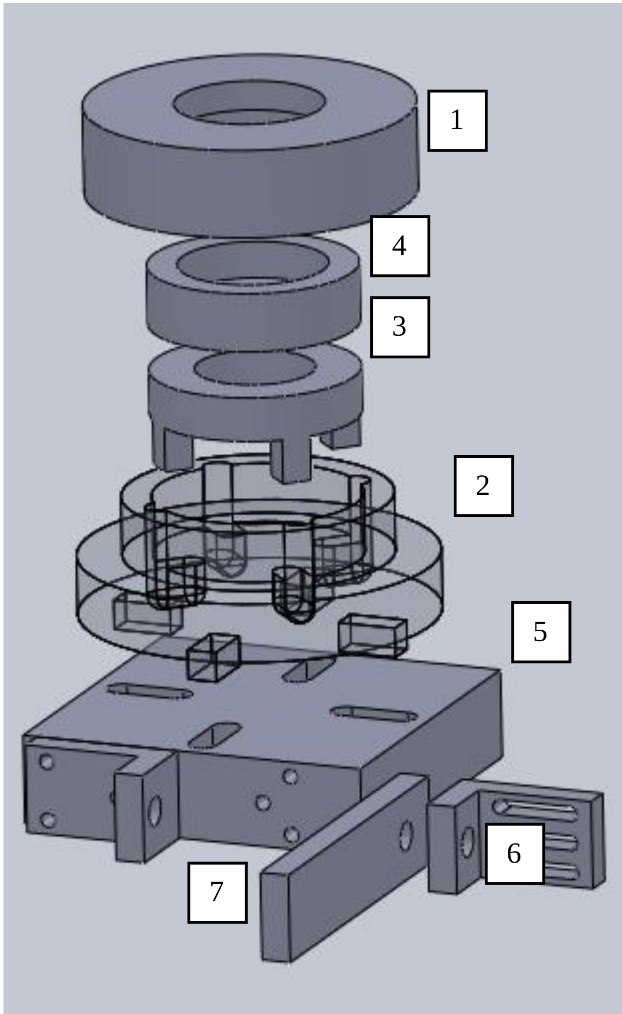
In general, if we have not set up work pieces properly or use incorrect tools/feeds/speeds, become distracted, behave inappropriately or fail to follow safety precautions, serious injury could occur. Most of the possible hazards have low to moderate levels of risk, but are nonetheless possible. Therefore, we have developed a list of precautions that must be taken before any work within the ME 450 machine shop can begin.

- Remain focused on task at hand
- Remove loose articles of clothing such as jewelry, tuck in shirts, put up hair
- Be aware of surroundings including other people and machines, as the shop can become crowded
- Know the expected tool life to avoid tool breakage
- Understand federates and use emergency stop if machine instability occurs
- Ask for help from machine shop supervisors if ever hesitant or unsure about a correct work piece setup or machine operation
- Wear safety glasses at all times and gloves if necessary
- Use brushes to remove shavings from work surface and wood block to remove material from band saw
- Double check setup and machine speeds/feeds before beginning to cut any material

If all of the above procedures and precautions are used at all times throughout the machining process, we should have no issues of safety and the risk level for any hazards will be very low. Once again, for a complete listing and table of all hazards and ways to reduce their risks, refer to Appendix B.a.

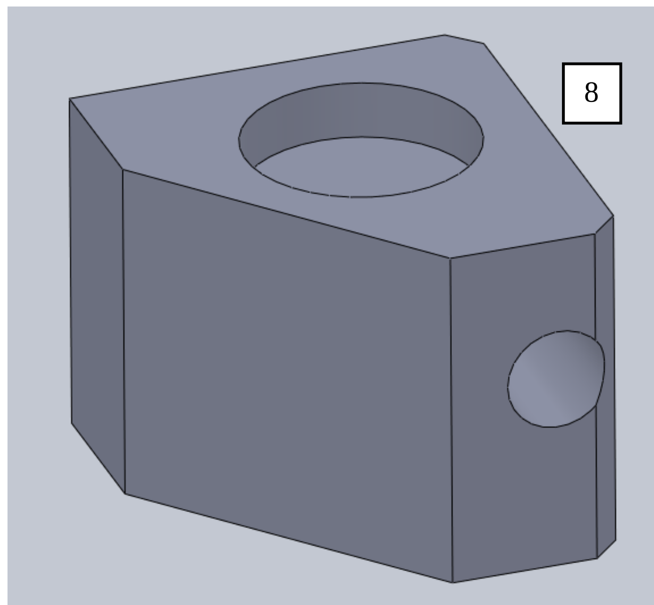
All engineering CAD drawings of components to be manufactured are shown below and on the next several pages.

Complete Specimen Cup Assembly

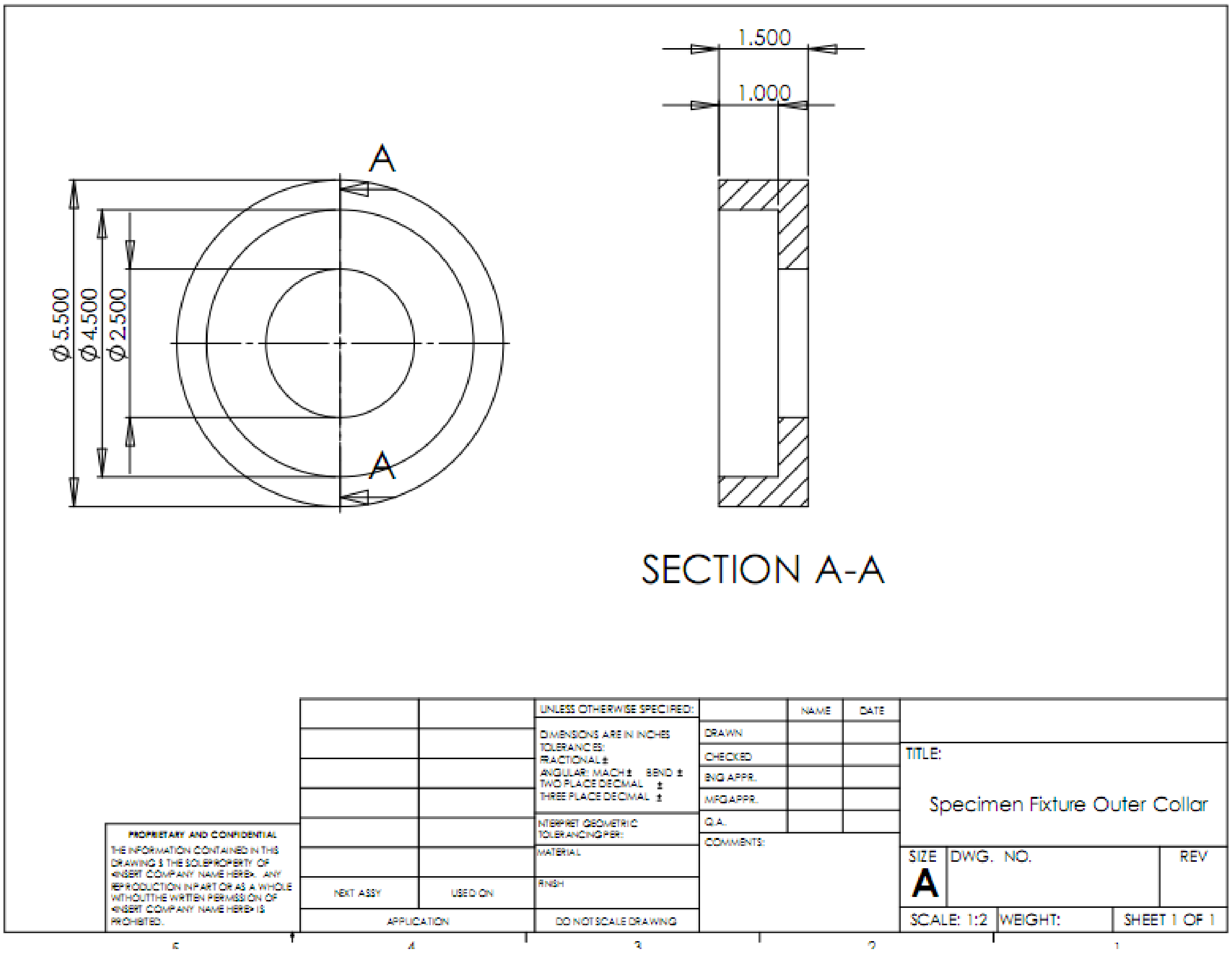


Manufacturing Order:

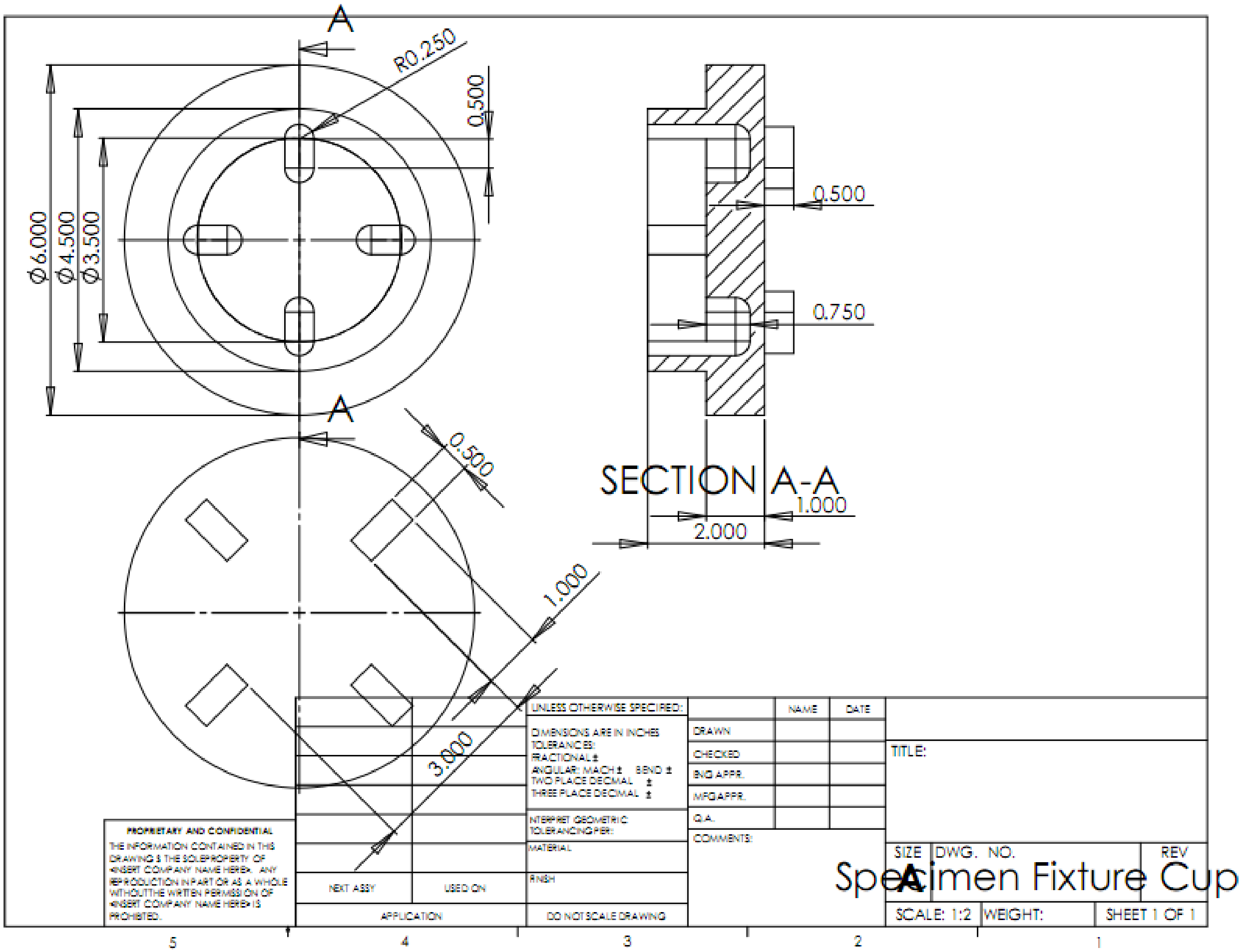
1. Outer Collar
2. Specimen Fixture Cup
3. TCT Ring
4. Four-ball Ring
5. Securing Block Modification
6. Bracket Mounts
7. Lateral Force Beam
8. Deflecting Block



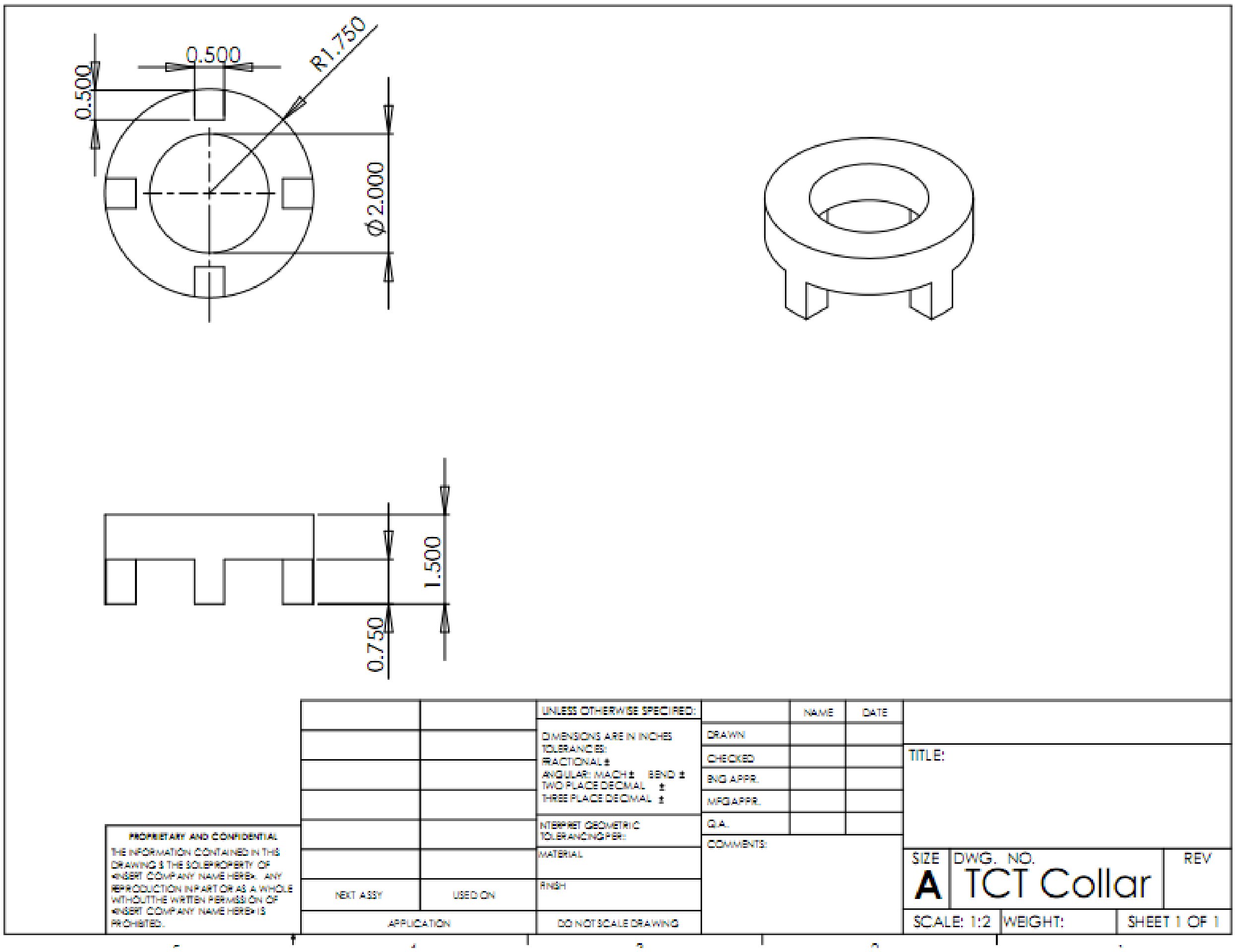
1. Outer Collar



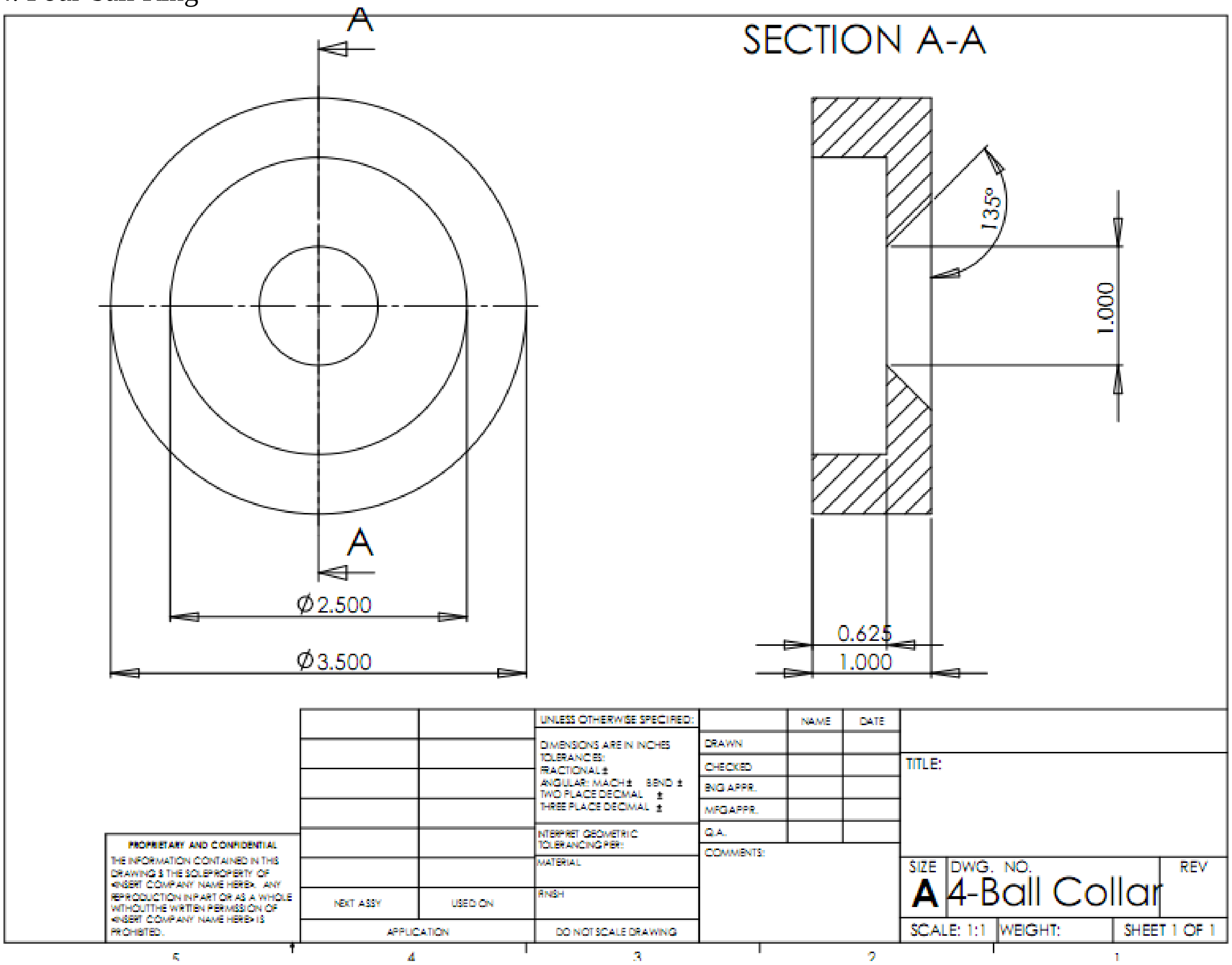
2. Specimen Fixture Cup



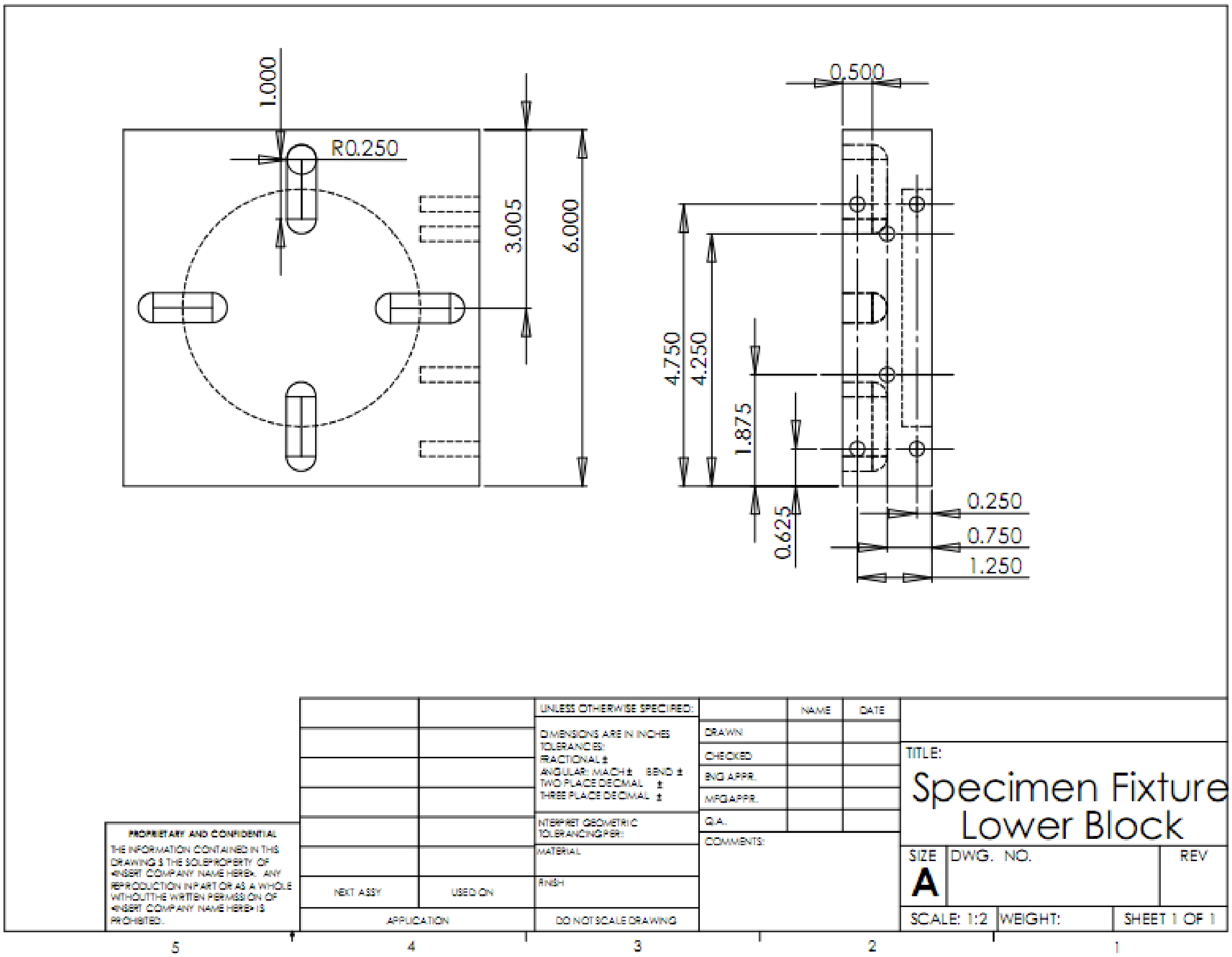
3. TCT Ring


0.500
0.500
R1.750
⌀2.000
1.500
0.750
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±
INTERPRET GEOMETRIC
TOLERANCING PER:
MATERIAL
FINISH
DO NOT SCALE DRAWING
NAME
DATE
DRAWN
CHECKED
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
TITLE:
SIZE
A
DWG. NO.
TCT Collar
REV
SCALE: 1:2
WEIGHT:
SHEET 1 OF 1
NEXT ASSY
USED ON
APPLICATION
PROPRIETARY AND CONFIDENTIAL
THE INFORMATION CONTAINED IN THIS DRAWING IS THE SOLE PROPERTY OF <INSERT COMPANY NAME HERE>. ANY REPRODUCTION IN PART OR AS A WHOLE WITHOUT THE WRITTEN PERMISSION OF <INSERT COMPANY NAME HERE> IS PROHIBITED.

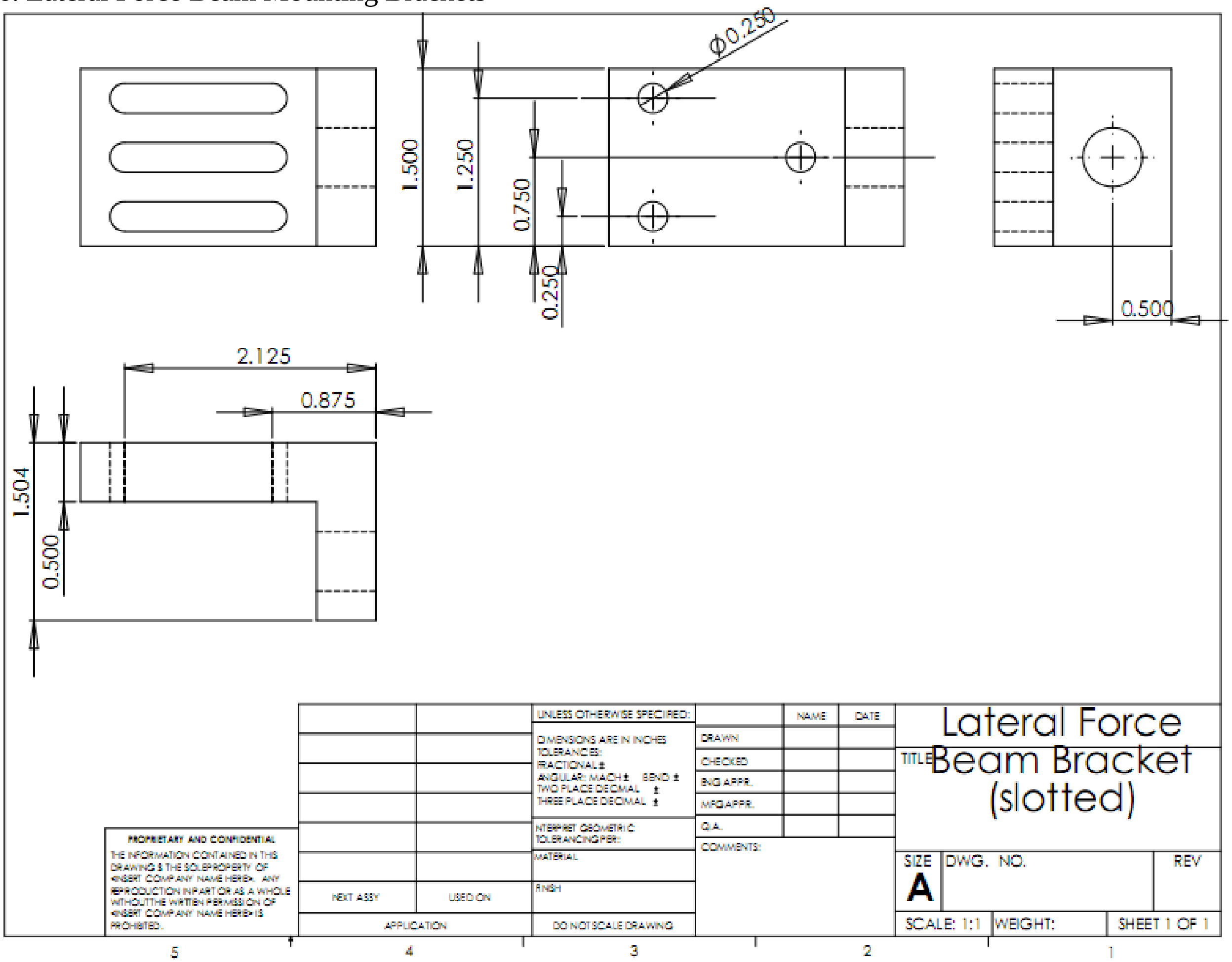
4. Four-ball Ring



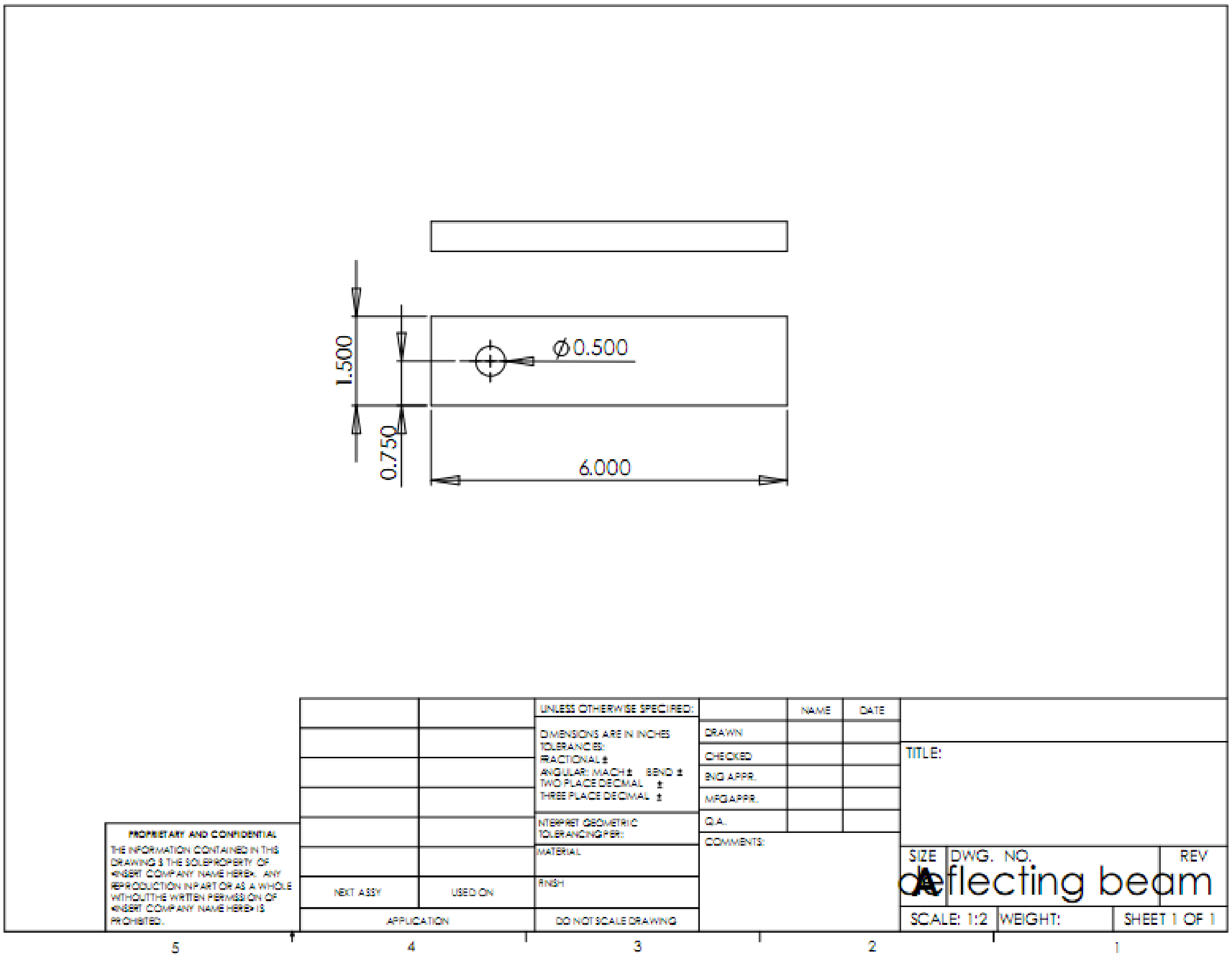
5. Securing Block (Modification)



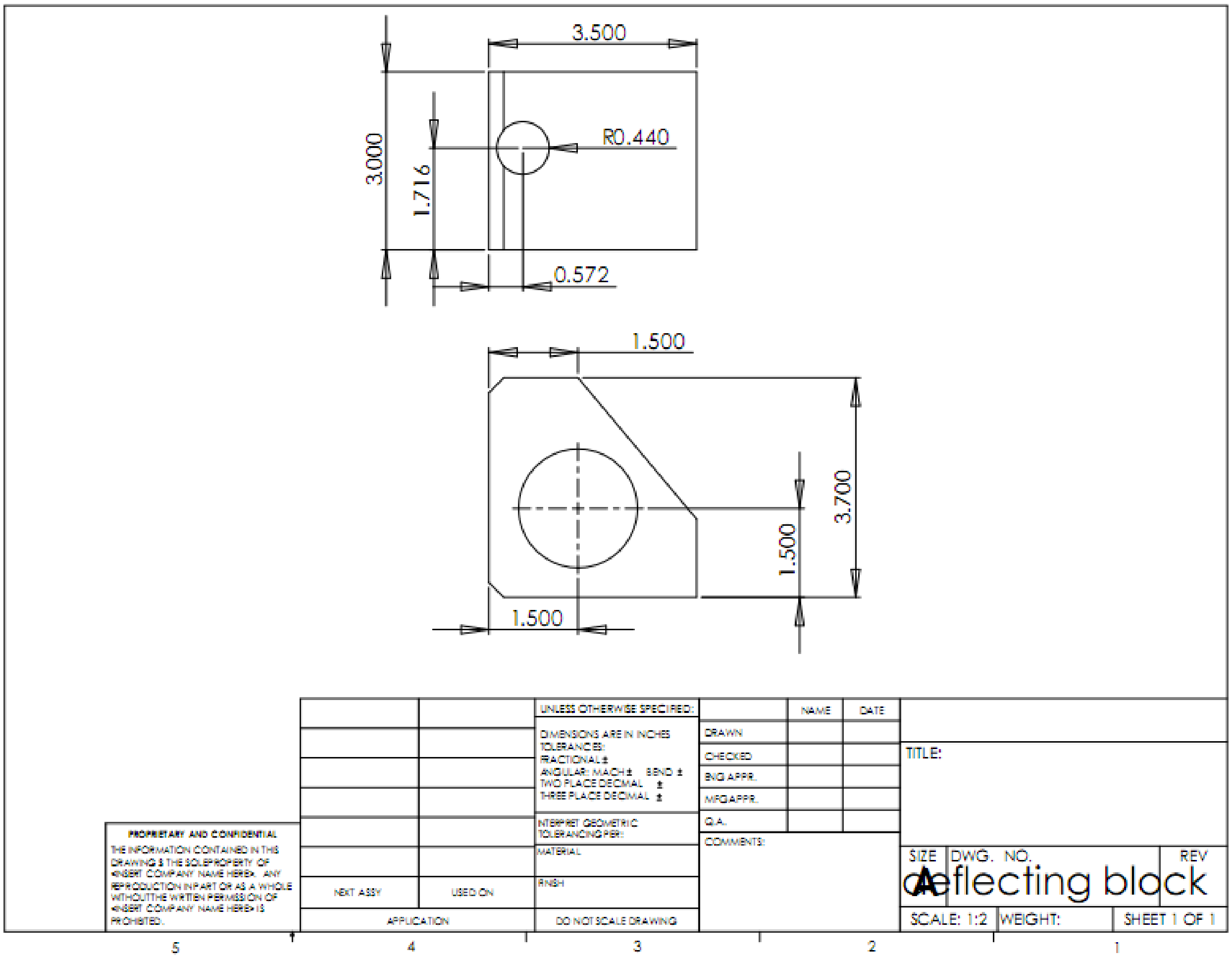
6. Lateral Force Beam Mounting Brackets



7. Lateral Force Beam



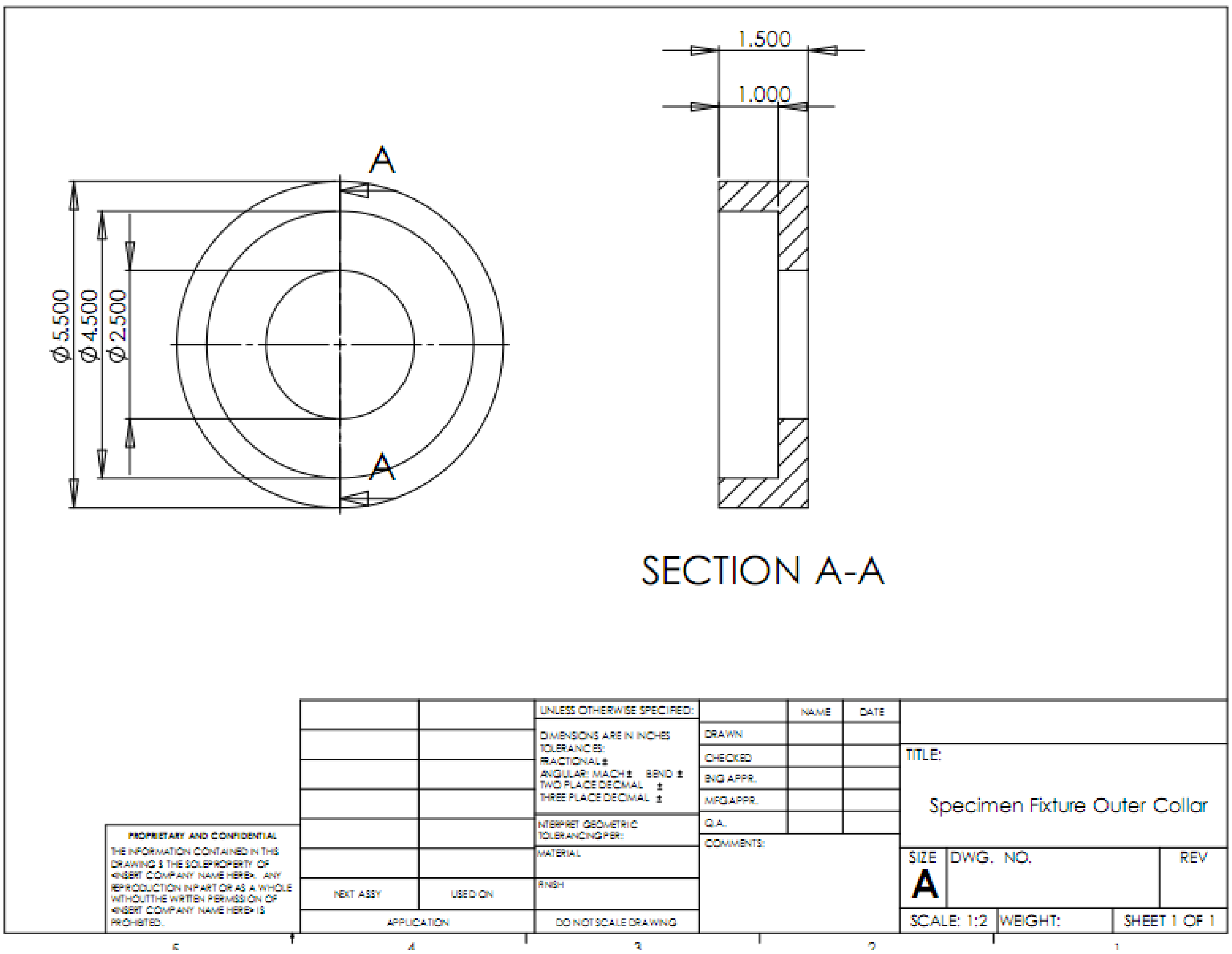
8. Deflecting Block



5. Manufacturing

This section will describe in detail the manufacturing process for each component. All fabrication will take place in the ME 450 machine shop in the G.G. Brown building. Every task listed below each drawing is necessary to achieve the desired shapes and contours of our components.

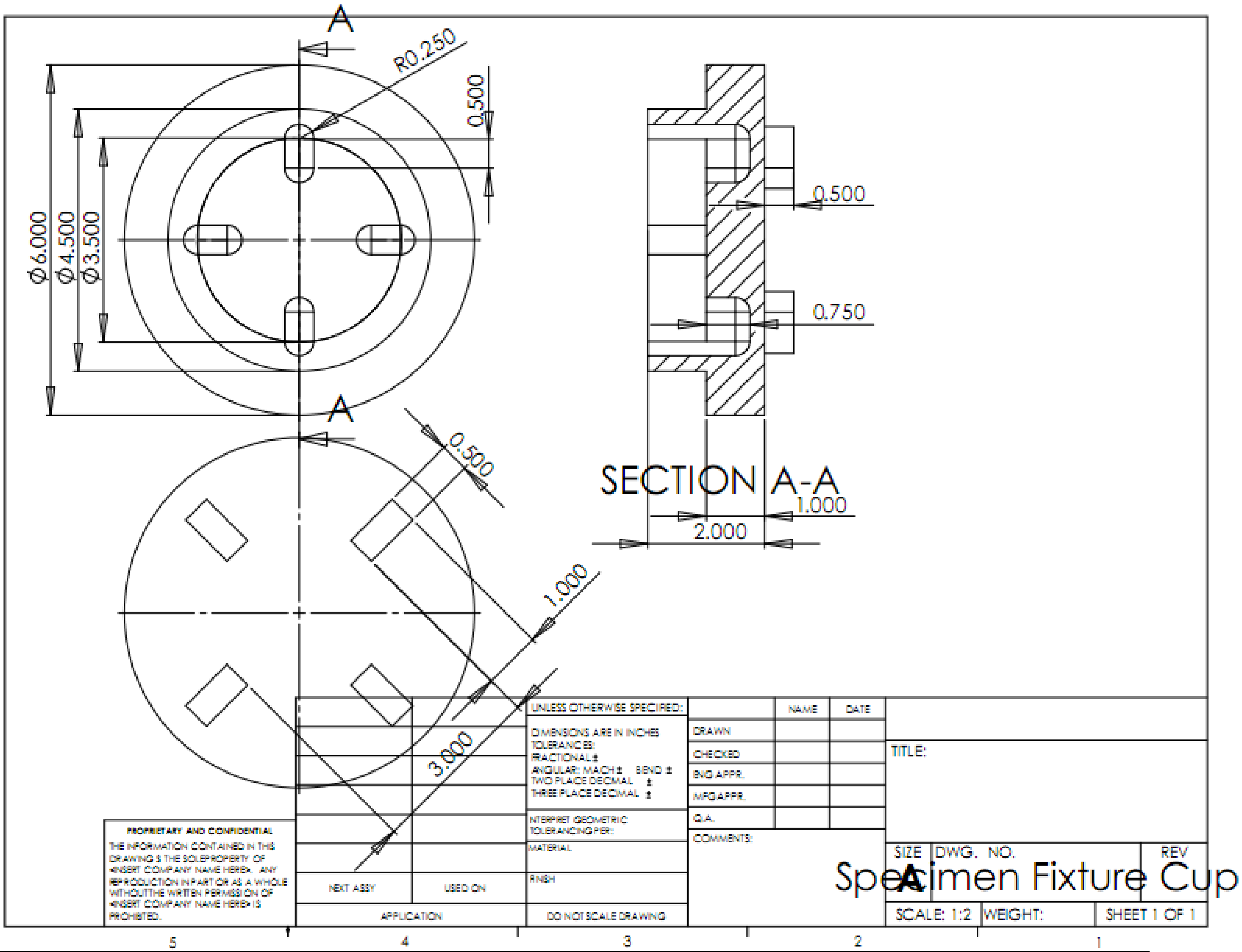
1. Outer Collar



Task #	Purpose	Machine	Tool	Spindle Speed (rpm)
1	Face out inner recession	Lathe	Bore Bar	650
2	Thread inner face	Lathe	Threading tool	15
3	Drill center circle	Mill	Drill Bit	2520
4	Facing of thinner wall	Mill	½" End Mill	1260

Notes: Threading will be approximately 8 threads per inch. This part is manufactured first and the Specimen Fixture Cup is fit to this part. The thin wall facing will be later to ensure adequate travel for securing the inner rings.

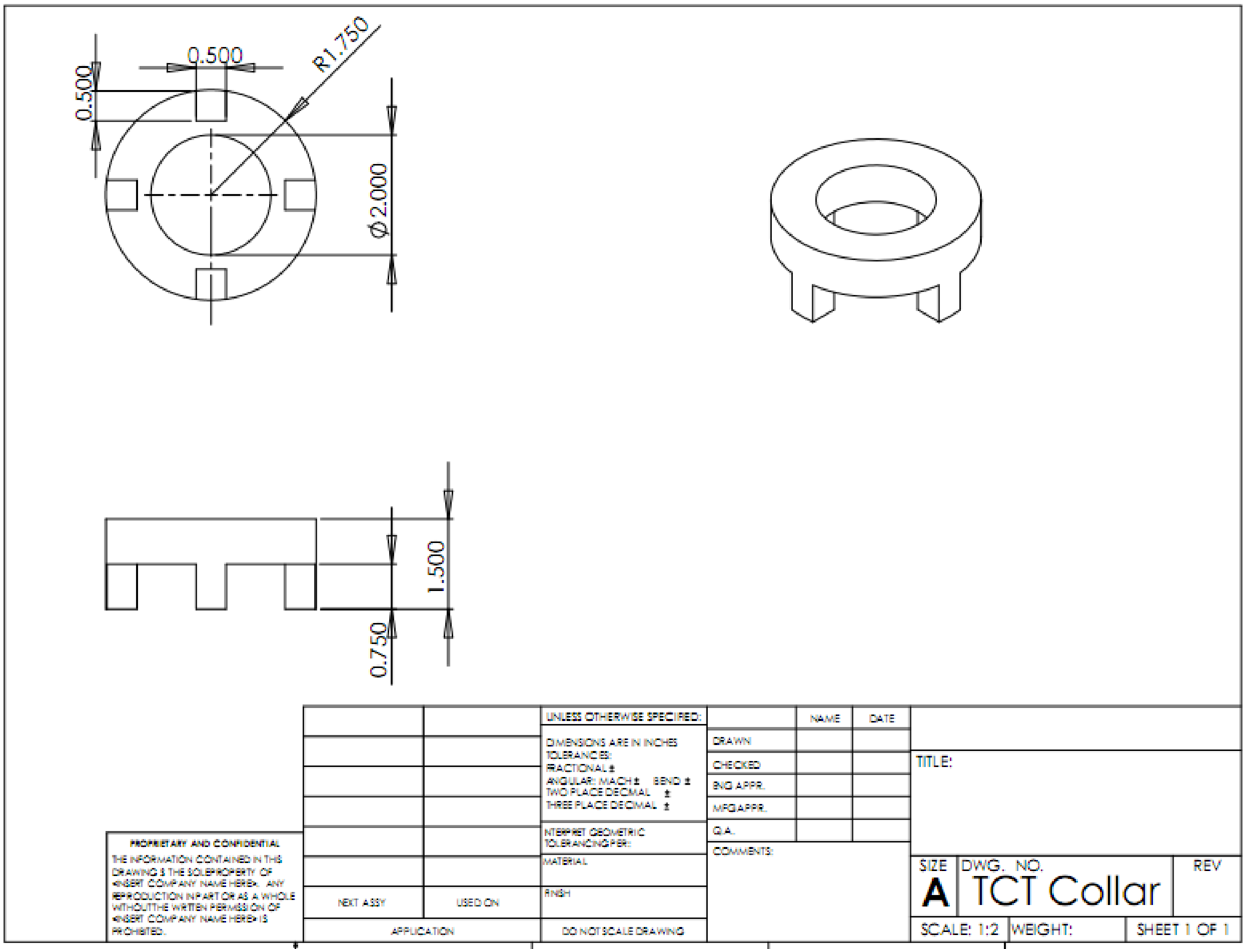
2. Specimen Fixture Cup



Task #	Purpose	Machine	Tool	Spindle Speed (rpm)
1	Face outside of cup	Lathe	Facing tool	380
2	Thread outside of cup	Lathe	Threading Tool	15
3	Face inside of cup	Lathe	Bore Bar	600
4	Mill out grooves inside of cup	Mill	3/8" Ball Mill	1680
5	Mill out teeth	Mill	½" End Mill	1260
6	Face down teeth	Mill	½" End Mill	1260

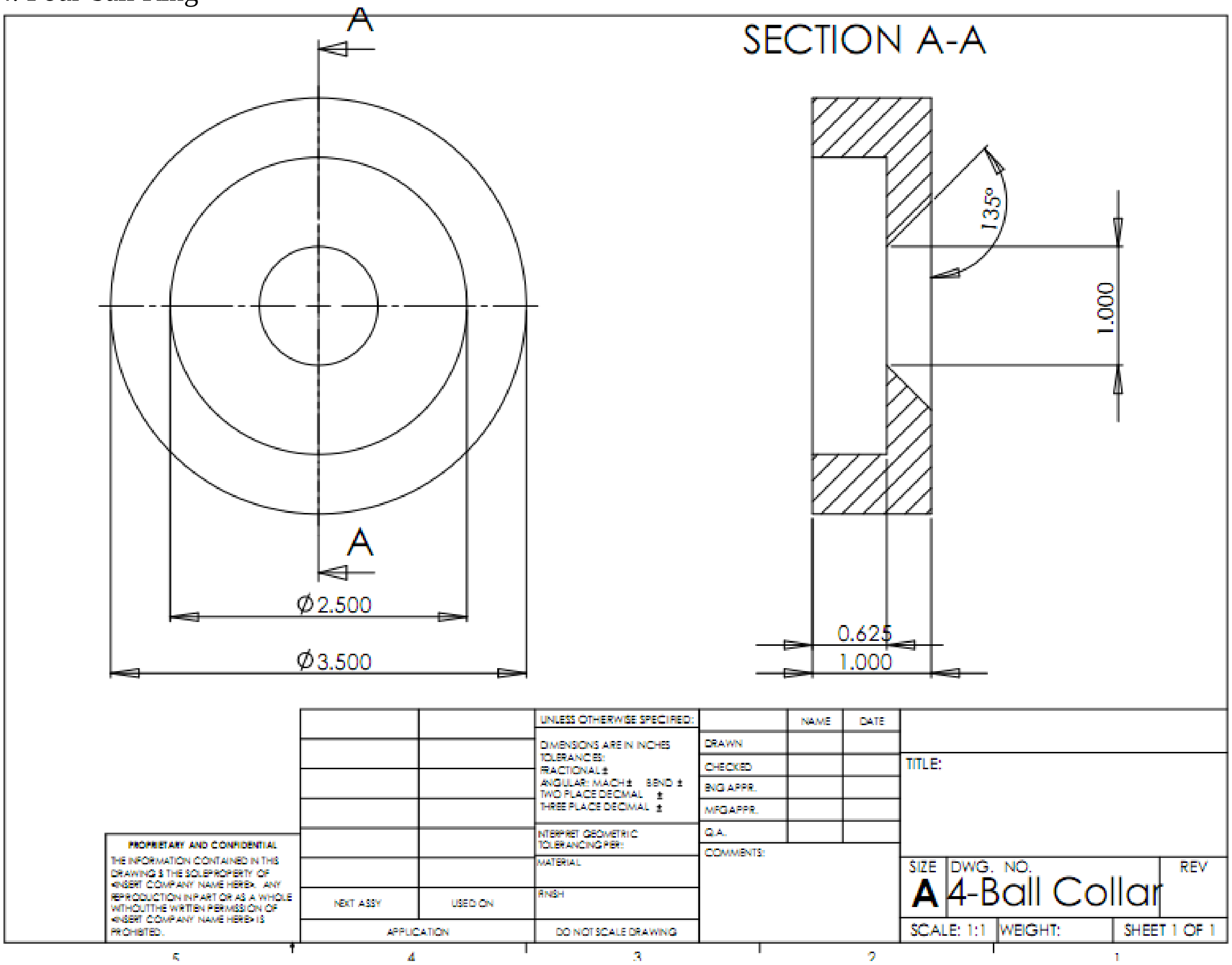
Notes: The facing and threading of the cup will be done to fit inside of the Outer Collar. Using the 3/8" ball mill to do the interior grooves is an adaptation from using a ½" ball mill due to the limited cutting depth possible. It is to be determined into the process whether step 3 will be covered by facing the Specimen Fixture Cup or the Outer Collar.

3. TCT Ring



Task #	Purpose	Machine	Tool	Spindle Speed (rpm)
1	Face outer diameter	Lathe	Facing tool	650
2	Mill out teeth	Mill	½” End Mill	1260
3	Face down teeth	Mill	½” End Mill	1260
4	Face down top surface	Mill	½” End Mill	1260
5	Drill out center hole	Mill	2” Drill	415

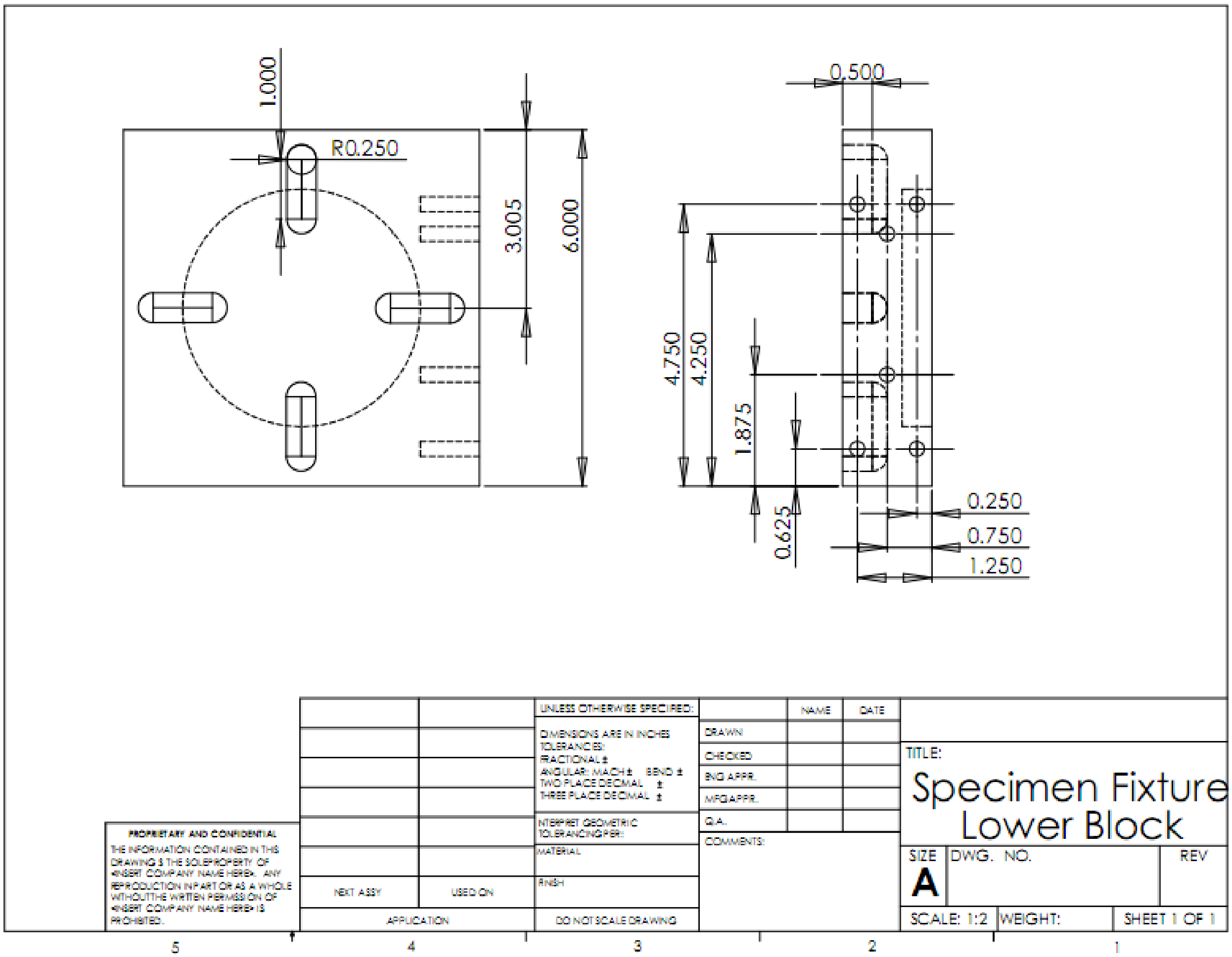
4. Four-ball Ring



Task #	Purpose	Machine	Tool	Spindle Speed (rpm)
1	Face outer diameter	Lathe	Facing tool	650
2	Angular face	Lathe	Angled facing tool	650
3	Face out inner recession	Lathe	Bore Bar	650
4	Face bottom of ring	Mill	½" End Mill	1260
5	Face top of ring	Mill	½" End Mill	1260

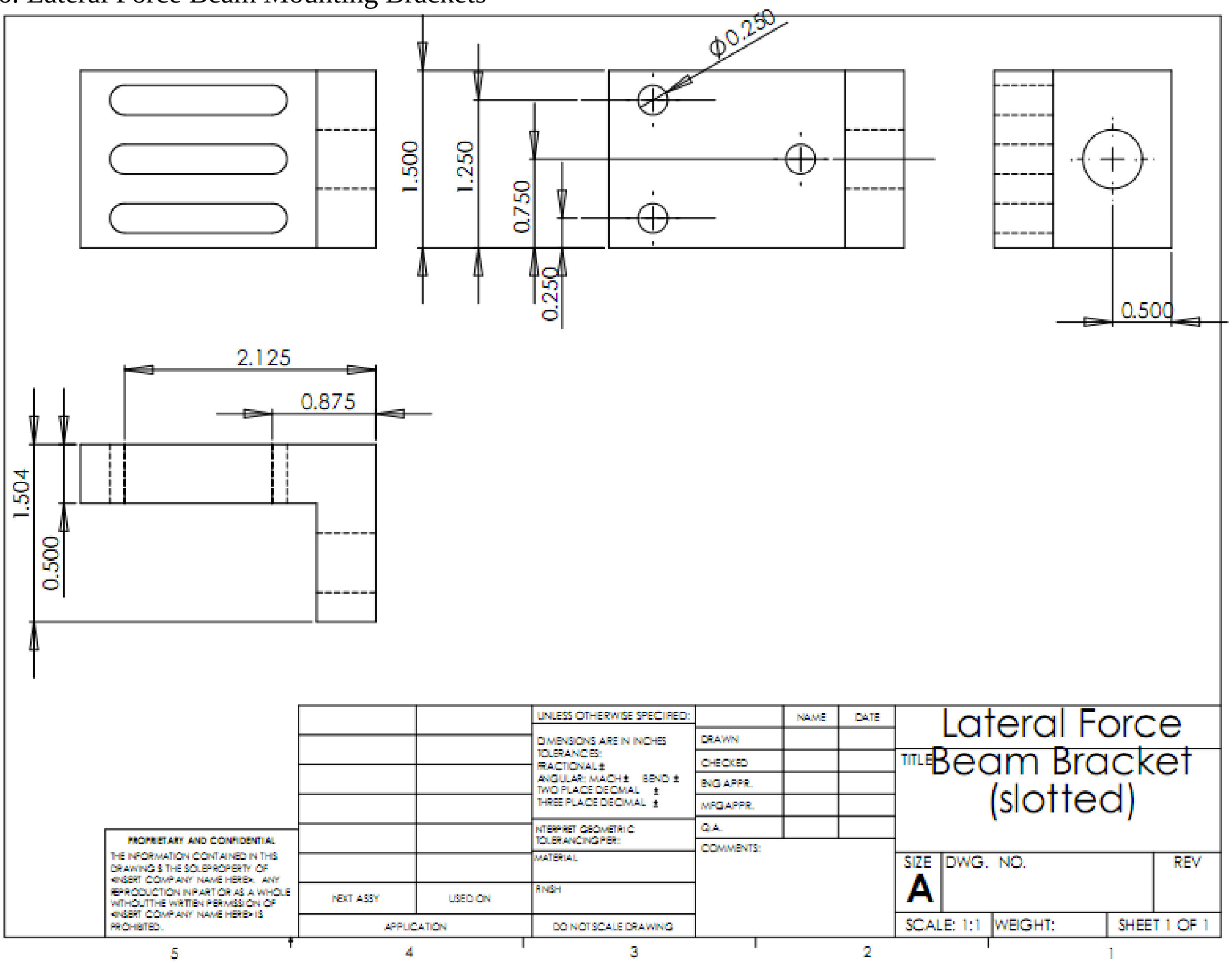
Notes: Facing out the inner recession will occur to allow for easier manufacturing of the part, machining to tolerance, rather than designing to it.

5. Securing Block (Modification)



Task #	Purpose	Machine	Tool (HSS)	Spindle Speed (rpm)
1	Cut top grooves	Mill	½” Ball Mill	1528
2	Drill bracket mounting holes	Mill	¼” Drill	3060
3	Thread ¼”-20	Hand	¼”-20 thread tool	By Hand

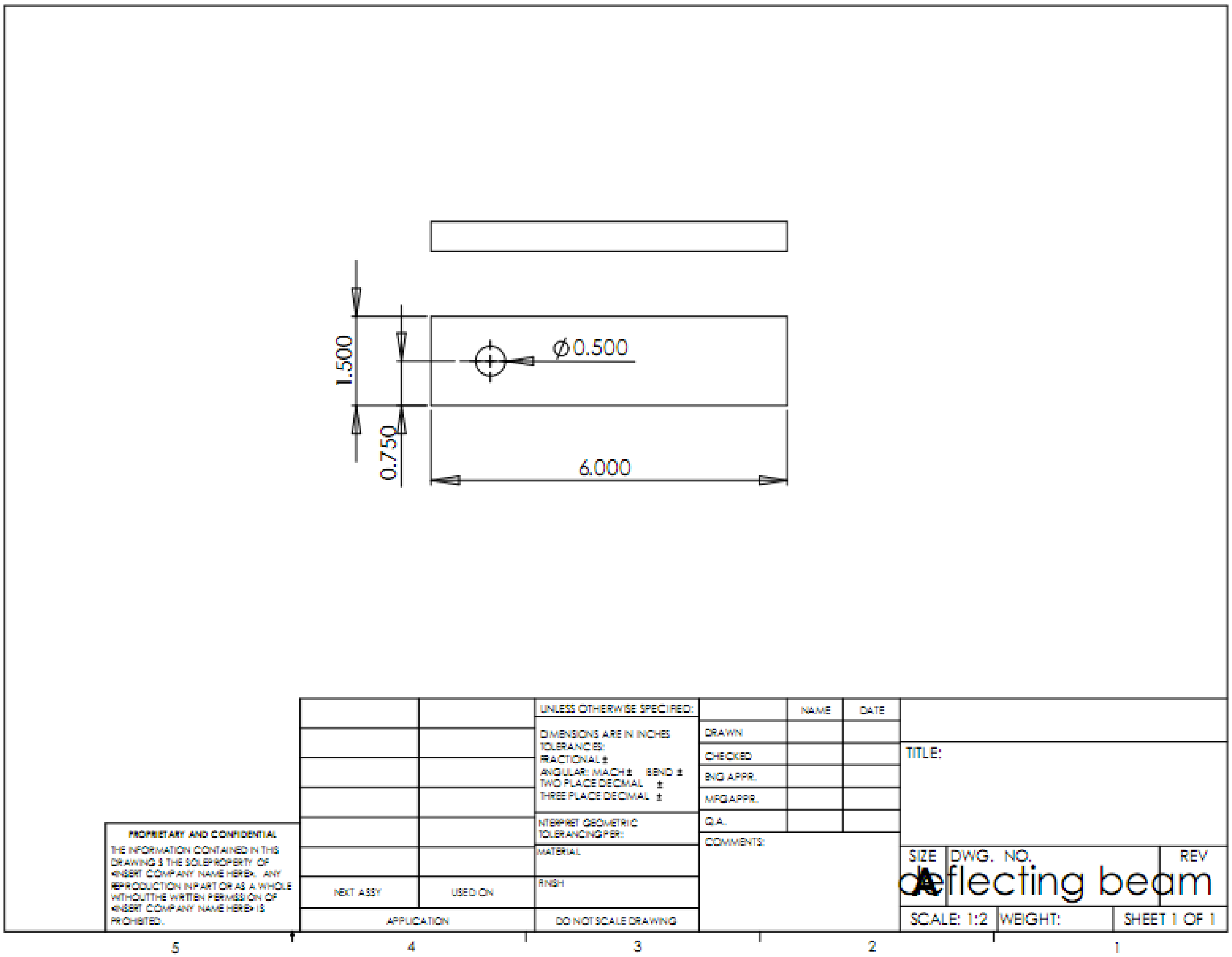
6. Lateral Force Beam Mounting Brackets



Task #	Purpose	Machine	Tool (HSS)	Spindle Speed (rpm)
1	Face width	Mill	½” End Mill	1260
2	Face depths	Mill	½” End Mill	1260
3	Drill holes / slots	Mill	¼” Ball Mill	2520
4	Drill big hole	Mill	½” End Mill	1260
5	Face height	Mill	½” End Mill	1260

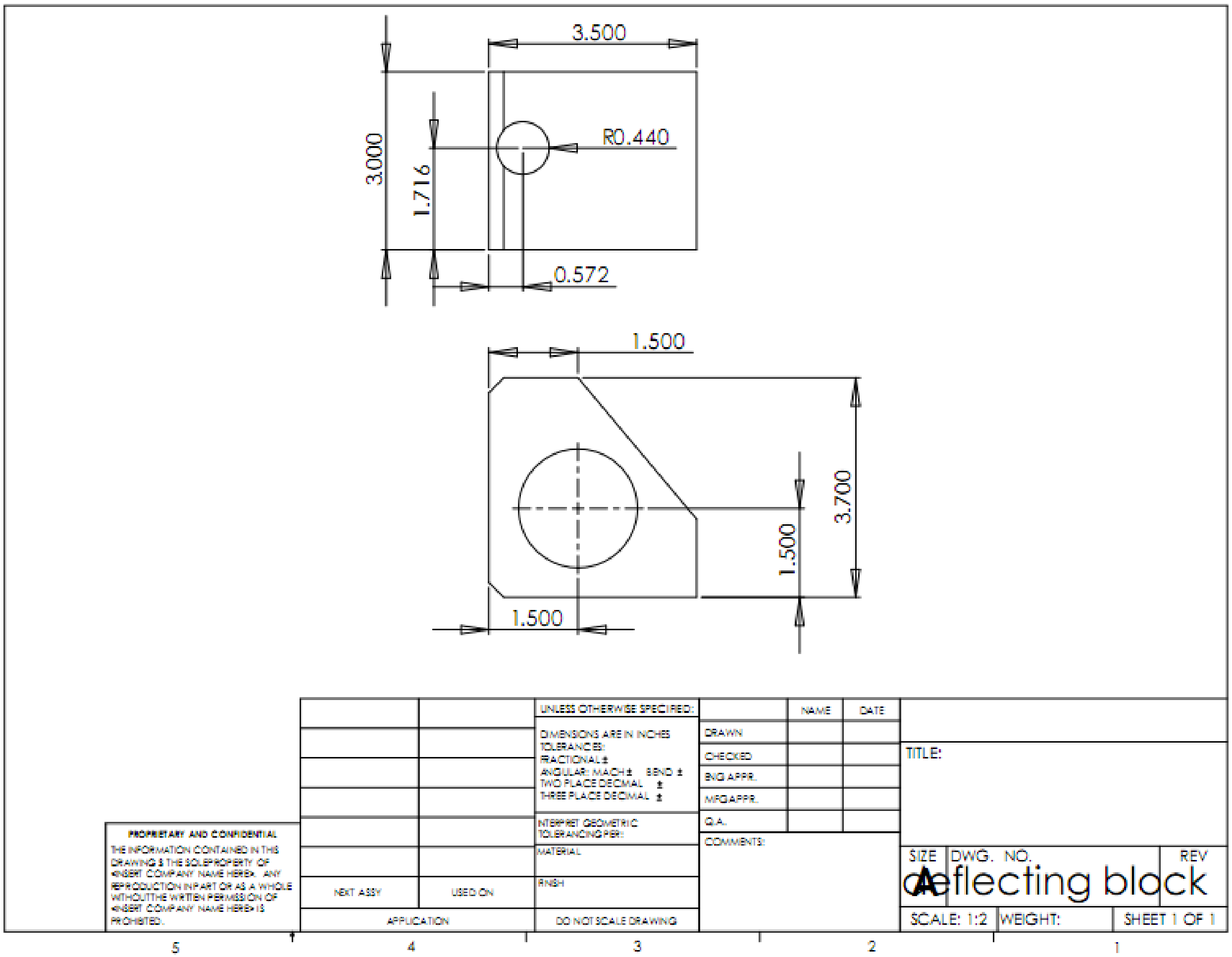
Notes: Measurements will be taken from the bottom face in terms of how they’re oriented, in order to ensure proper alignment once in place on the block.

7. Lateral Force Beam



Task #	Purpose	Machine	Tool	Spindle Speed (rpm)
1	Cut down beam to size	Band Saw		1260
2	Drill bracket mounting holes	Mill	½"	2520
3	Thread ¼"-20	Hand	¼"-20 thread tool	By Hand

8. Deflecting Block



Task #	Purpose	Machine	Tool	Spindle Speed (rpm)
1	Face width (3.7”)	Mill	½” End Mill	1260
2	Drill 0.9” Hole	Mill	0.9” Drill	600
3	Small Chamfers	Mill	½” End Mill	1260
4	Face distance from hole to nearest end of block	Mill	½” End Mill	1260
5	Big Chamfer	Mill	½” End Mill	1260
6	Large hole on top	Mill	2” End mill	650

Notes: Tasks 1 and 4 will be done interspersed with taking the part back to our lab and trying to fit it into the support structure of the shop press. The key with this part is to have it fit snugly but not to the point of over-machining it. The large hole on top may not be drilled out immediately.

6. Assembly

Once the individual components have all been fabricated, they will need to be assembled either to each other or to the shop press in the case of the lateral force deflecting block. Components that will be assembled after fabrication are the specimen cup, ring(s), specimen cup collar, and steel securing plate. The specimen cup will be inserted into the steel securing plate via teeth and grooves, the ring(s) will be placed inside the cup, and the collar will be screwed onto the top of the cup via threads which will safely secure the ring within the cup. This assembly process will not require the use of any tools, powered or otherwise, simply the use of our hands.

Furthermore the lateral force beam brackets will be attached to the steel securing plate and the lateral force deflecting beam will be attached to the shop press. In order to fasten the brackets to the steel securing plate, we will be using 1/4" – 20 thread screws, and a screwdriver to tighten them. In order to fasten the lateral force deflecting block to the shop press, we will be using a 7/8" bolt and a nut that will go through the block and press structure. The bolt and nut will be tightened using a wrench.

These assembly processes will take place within our designated workstation in lab 2190A within the G.G. Brown building. Based on analysis of the individual components and the maximum expected stresses exerted on them, we have designed all components to withstand all forces with a safety factor of at least 2, and are therefore certain that they will not fail under any circumstances, even after testing has begun. However, for the time being we do not plan to test the new components, just fabricate and assemble them.

8. Additional Appendices

APPENDIX B.a: DesignSafe

DesignSafe for Components

	Item Id	User	Task	Hazard Category	Hazard	Cause/Failure Mode	Severity	Exposure	Probability	Risk Level
1	1-1-1	All Users	All Tasks	mechanical	cutting / severing	Parts designed with sharp points	Slight	Remote	Unlikely	Low
2	1-1-2	All Users	All Tasks	mechanical	fatigue	Insufficient thickness of components under stress (specimen cup securing plate, bottom of specimen cup, lateral force beams, brackets for lateral force beams, teeth of specimen cup). This could result in complete failure of the test area	Catastrophic	Frequent	Unlikely	High
3	1-1-3	All Users	All Tasks	ergonomics / human factors	lifting / bending / twisting	Use of heavy or excess material could result in excessive lifting for user of machine	Slight	Frequent	Unlikely	Moderate

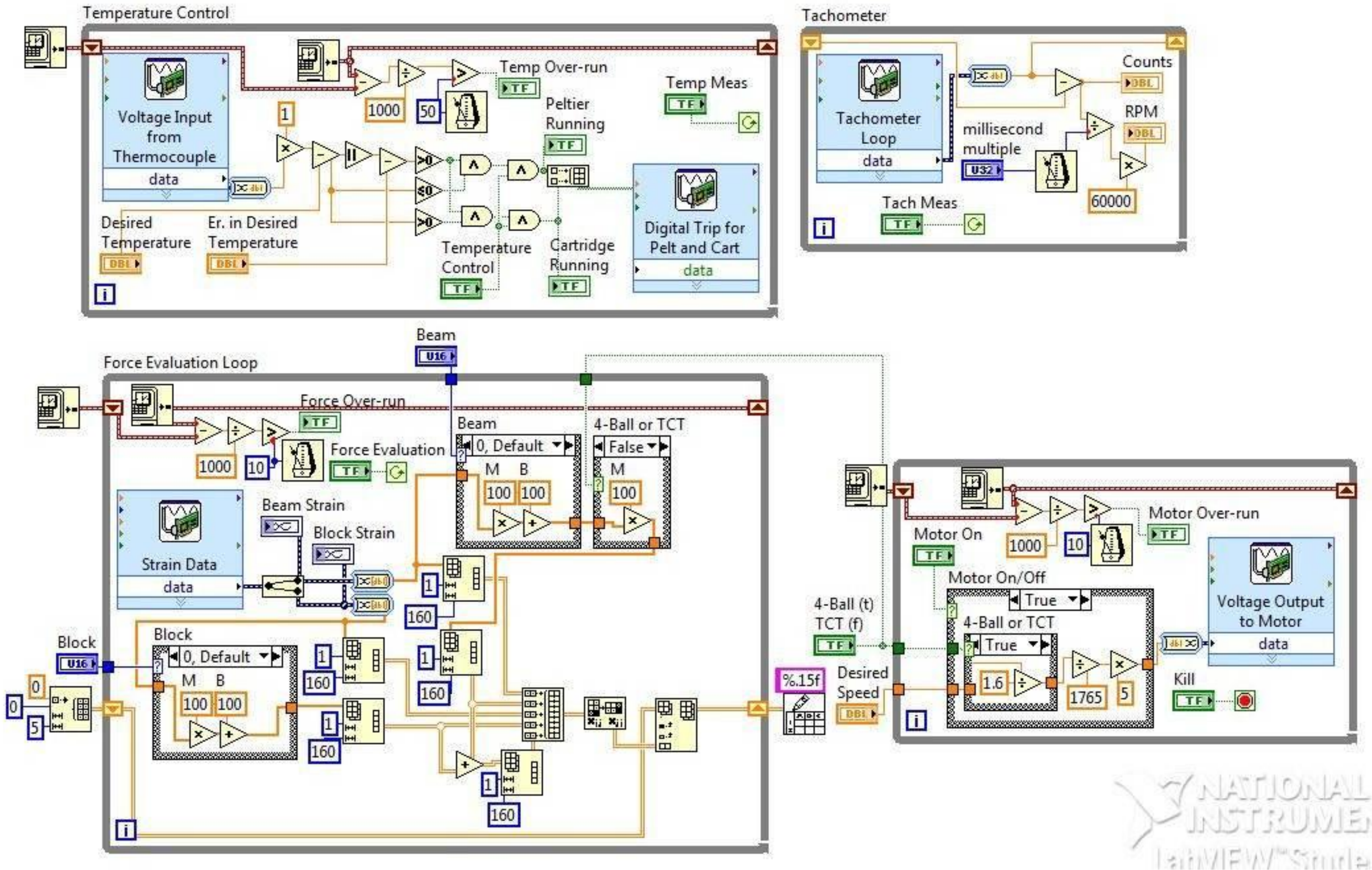
	Item Id	User	Task	Hazard Category	Hazard	Reduce Risk	Severity	Exposure	Probability	Risk Level	Person Responsible	Status
1	1-1-1	All Users	All Tasks	mechanical	cutting / severing	Design parts with only blunt ends and use rounded edges when possible	Slight	Remote	Unlikely	Low	Rich, Scott, Rachel, Justin	Complete
2	1-1-2	All Users	All Tasks	mechanical	fatigue	Analyze stresses in components based on the maximum expected forces, using a safety factor of 2 or greater when determining thicknesses	Serious	Remote	Unlikely	Moderate	Justin	Complete
3	1-1-3	All Users	All Tasks	ergonomics / human factors	lifting / bending / twisting	While designing for strength, not overusing material and using lightweight materials when possible such as aluminum over steel	Minimal	Occasional	Unlikely	Low	Rich	Complete

DesignSafe for Fabrication Processes

	Item Id	Sub-process	User	Task	Hazard Category	Hazard	Cause/Failure Mode	Severity	Exposure	Probability	Risk Level
1	1-1-1-1	All Sub-processes	All Users	All Tasks	mechanical	cutting / severing	not paying attention while using a band saw while cutting material	Serious	Occasional	Unlikely	Moderate
2	1-1-1-2	All Sub-processes	All Users	All Tasks	mechanical	drawing-in / trapping / entanglement	loose clothing or hair can get caught in mill or lathe while rotating	Catastrophic	Occasional	Unlikely	High
3	1-1-1-3	All Sub-processes	All Users	All Tasks	mechanical	pinch point	not paying attention, resulting in hand being pinched inbetween tool and material while using the mill or lathe	Serious	Remote	Unlikely	Moderate
4	1-1-1-4	All Sub-processes	All Users	All Tasks	mechanical	unexpected start	mill or lathe could unexpectedly start if start button is bumped by user or another person	Serious	Remote	Unlikely	Moderate
5	1-1-1-5	All Sub-processes	All Users	All Tasks	mechanical	fatigue	tool may be weakened by excessive use	Slight	Remote	Unlikely	Low
6	1-1-1-6	All Sub-processes	All Users	All Tasks	mechanical	break up during operation	material or tool may have an unknown flaw such as a crack	Serious	Remote	Negligible	Low
7	1-1-1-7	All Sub-processes	All Users	All Tasks	mechanical	machine instability	vibration may occur if an incorrect feedrate is used	Serious	Remote	Unlikely	Moderate
8	1-1-1-8	All Sub-processes	All Users	All Tasks	mechanical	impact	drill press may be used inappropriately	Slight	Remote	Unlikely	Low
9	1-1-1-9	All Sub-processes	All Users	All Tasks	slips / trips / falls	debris	shavings flying off tool or part injuring user, risk is increased if safety glasses are not worn	Serious	Occasional	Possible	High
10	1-1-1-10	All Sub-processes	All Users	All Tasks	ergonomics / human factors	human errors / behaviors	incorrect feed rate, tools, cutting speeds, part setup used	Serious	Remote	Possible	Moderate
11	1-1-1-11	All Sub-processes	All Users	All Tasks	ergonomics / human factors	deviations from safe work practices	not wearing safety glasses, using hands to remove debris from saw or tools on the mill or lathe	Serious	Occasional	Negligible	Moderate
12	1-1-1-12	All Sub-processes	All Users	All Tasks	ergonomics / human factors	interactions between persons	miscommunications on procedure and setup	Slight	Remote	Unlikely	Low

	Item Id	Sub-process	User	Task	Hazard Category	Hazard	Reduce Risk	Severity	Exposure	Probability	Risk Level	Person Responsible	Status
1	1-1-1-1	All Sub-processes	All Users	All Tasks	mechanical	cutting / severing	remained focused on task and wear appropriate safety apparel	Slight	Remote	Unlikely	Low	user	TBD
2	1-1-1-2	All Sub-processes	All Users	All Tasks	mechanical	drawing-in / trapping / entanglement	follow lab safety procedures including tucking in clothes, taking off loose jewerly and putting hair up	Slight	Remote	Unlikely	Low	user	TBD
3	1-1-1-3	All Sub-processes	All Users	All Tasks	mechanical	pinch point	remained focused on task	Slight	Remote	Unlikely	Low	user	TBD
4	1-1-1-4	All Sub-processes	All Users	All Tasks	mechanical	unexpected start	be conscience of surroundings, including people and other machines	Slight	Remote	Unlikely	Low	user	TBD
5	1-1-1-5	All Sub-processes	All Users	All Tasks	mechanical	fatigue	know tool life before starting and calculate how long it can be used for	Slight	Remote	Unlikely	Low	user	TBD
6	1-1-1-6	All Sub-processes	All Users	All Tasks	mechanical	break up during operation	examine material and tool before use	Slight	Remote	Unlikely	Low	user	TBD
7	1-1-1-7	All Sub-processes	All Users	All Tasks	mechanical	machine instability	understand what feedrate should be used for the material, use emergency stop button if it should occur	Slight	Remote	Unlikely	Low	user	TBD
8	1-1-1-8	All Sub-processes	All Users	All Tasks	mechanical	impact	remained focused when using machines and ask for help when needed	Slight	Remote	Unlikely	Low	user	TBD
9	1-1-1-9	All Sub-processes	All Users	All Tasks	slips / trips / falls	debris	wear safety glasses and gloves if necessary, if shavings are unavoidable, use a shield	Slight	Remote	Unlikely	Low	user	TBD
10	1-1-1-10	All Sub-processes	All Users	All Tasks	ergonomics / human factors	human errors / behaviors	remained focused on task and know feedrate, tools, cutting speeds, and part setup. check with shop supervisors to see if setup is correct	Serious	Remote	Unlikely	Moderate	user	TBD
11	1-1-1-11	All Sub-processes	All Users	All Tasks	ergonomics / human factors	deviations from safe work practices	wear safety glasses, use brushes to remove debris and wood block to remove parts from band saw	Slight	Remote	Unlikely	Low	user	TBD
12	1-1-1-12	All Sub-processes	All Users	All Tasks	ergonomics / human factors	interactions between persons	double check setup with shop supervisors and machining partner if applicable	Slight	Remote	Unlikely	Low	user	TBD

APPENDIX H: LabVIEW Encompassing Program (Block Diagram)



APPENDIX I: Power Electronics Diagram