Project 22:
Combination Twist Compression and Four-Ball Test Device

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ABSTRACT

Existing tribometers are not affordable for most research labs. In addition, many of these devices are limited to conducting only one specific tribology test. Our sponsor proposes that the Four-Ball and Twist Compression tests can be integrated into a single machine at a considerably lower cost. This will provide our sponsor with the flexibility of conducting a larger variety of tests without the need to acquire multiple expensive devices. This machine will be designed to maintain the operating range and result accuracy, as compared to current market standards.

EXECUTIVE SUMMARY

The aim of this project is to design a Combination Twist Compression and Four-Ball Device, sponsored by Professor Gordon Krauss. In essence, we are to design a test device that combines the capabilities of the Twist Compression and Four-Ball Test Devices. In the Twist Compression Test Device, a hollow cylinder is rotated while being pressed against a flat plate. As for the Four-Ball Test Device, three balls are placed together with a fourth ball sitting above. Similarly, the two layers of balls are pressed against each other while one of them is being rotated.

The main problem faced by the sponsor is the high cost involved to obtain these machines. This is possibly due to the high mechatronics involved and additional features that these machines offer. However, our sponsor is often more concerned with basic information such as the coefficient of friction, and buying these machines is not cost effective.

To be assured of designing a prototype that would meet the customer's requirement, engineering specifications were developed, through literature reviews, market research and feedback from the sponsor. The key engineering specifications are shown in Table 1 below.

Loading Range	Up to 56,700N	Size of machine	3 by 3 by 4 ft
Angular align. Tolerance	$\pm 0.25^o$	Outer ∅ of cylinder	1 in
Translational align. tolerance	± 1 cm	∅ of balls	0.5 in
Temp. Range	0^oC to 100^oC	RPM Range	To 3600 RPM
Cylinder wall thickness	0.05 – 0.13 in		

Table 1: Key Engineering Specifications

The design space was explored extensively in both width and depth. This eventually led to the creating of a few designs and ultimately, one emerged as the final design. Based on the final design, the alpha prototype was developed. During the process of materializing the alpha prototype, changes had to be made also. Eventually, the final prototype emerged.

However, the final prototype could not be completed due to logistical and time constrains. While the components have been manufactured, there still remain a few components to be added, mostly with regards to the electronics of the machine. In view that the machine will be improved by future teams, a "what went wrong" segment was added to aid the troubleshooting required by future teams.

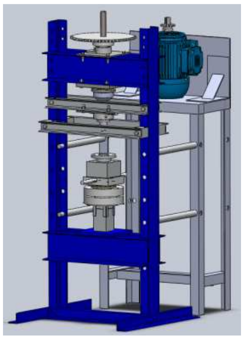

Figure 1: Full Assembly

INTRODUCING THE TWIST COMPRESSION AND FOUR-BALL TRIBOMETERS

In the field of Tribology, the Twist Compression Test Device and Four-Ball Test Device are two of the many other types of equipments used to test the performance of lubricants. This section serves to give basic background information on the two test mechanisms to aid the reader in understand the team’s current situation.

Twist Compression Test

In the Twist Compression Test, a hollow cylinder is pressed against a stationary flat plate. A downward force is applied to the cylinder as it is rotated, producing a lateral force due to the friction occurring at the contact surface. Generally, the downward force and lateral force are measured. The test is conducted in a lubricant bath, with the lubricant filled to above the contact surface. This is shown in Figure 2.

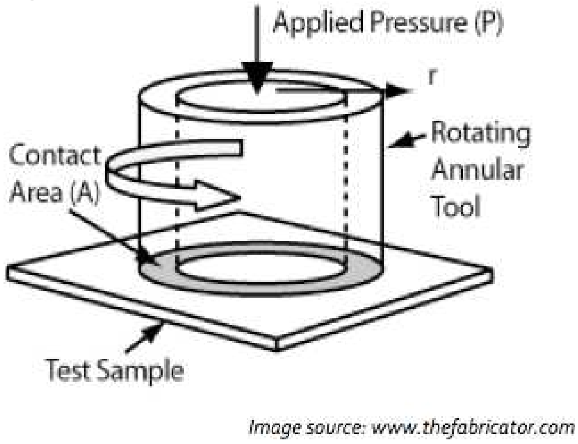

Figure 2: Twist Compression Test

Four-Ball Test

In the Four-Ball Test, three balls are placed together in a “triangle array” with a top ball placed on top of them. Similar to the Twist Compression test, one layer rotates. While conventionally it is the top ball that spins, a few machines have done the opposite. Like the Twist Compression test, the test is conducted in a test cup that is filled with lubricant that rises above the contact points between the balls. This is shown in Figure 3.

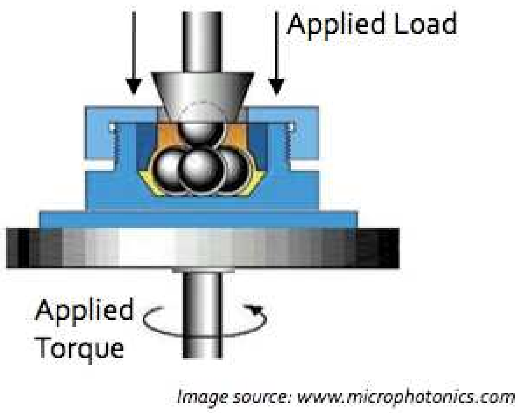

Figure 3: Four-Ball Test

UNDERSTANDING THE CUSTOMER

In order to understand more about the project, it was necessary for us to be informed of the problems faced by our sponsor in relation to the Twist Compression and Four-Ball tests. In addition, the requirements and preferences of our sponsor had to be determined in order to define the scope of our project. Several meetings with our sponsor, Professor Gordon Krauss, were held to accomplish this.

Problems Faced by Sponsor

Our sponsor faced several problems regarding the devices available in the market. Hence, the aim of our project was to target these issues and to come up with a device that can effectively handle the requirements as specified by the sponsor.

Cost

The machines currently available are not cost effective for our sponsor's needs. A large portion of the high cost is attributed to the high level of automation and economic profit. While the automated process and additional features allows for easier operation of the device, it is unnecessary to the sponsor, hence making the extra expenses unnecessary and not cost-effective.

Complications of External Testing

Currently, to run the two types of test on lubricant samples, they have to be sent to different areas for testing, which is a logistical problem. In addition, it is difficult to ensure identical operating conditions for the two different testing, hence resulting in a higher uncertainty in the test results. External testing is also not cost effective if tests are to be conducted frequently.

Inconvenience of Multiple Machines

According to our sponsor, buying two separate machines for the two types of tests (which are very similar) is a strain on resources as they are both very expensive. While there are some companies that offer modular test devices which allow one machine to conduct two types of tests, they are still priced beyond the range that our sponsor is willing to set aside for acquiring this equipment.

Requirements Set by Sponsor

In the process of establishing the problems our sponsor faces, we were able to determine the requirements he had for our project, which are discussed below in order of importance.

Cost

The most important objective of our project is to lower the cost of the machine. We are required to produce a device that could conduct both the Twist Compression and Four-Ball tests together at an affordable price. Safety is crucial, but automation and other extraneous features are not a necessity.

Operating Capabilities

The operating capabilities of our product will have to be comparable to the machines already available on the market. These include the range of operating speeds and normal forces that will be applied on the system. In addition, the test results must have a precision similar to the current standards available.

Integration of Both Tests into One Device

Our sponsor believes that it would be more cost effective if a single product could be made to encompass the functions of both tests, instead of creating two separate devices. As a result, he has requested that we integrate both tests into one device.

Size

Though not a crucial requirement, our sponsor specified a maximum size of 3’x3’x4’ for our product.

DETERMINING THE ENGINEERING SPECIFICATIONS

Based on our sponsor’s requirements, we sought to determine the engineering specifications of our desired single test machine. Before this could be done, we needed to research on the operational range and cost of these.

Broaden Awareness of Specifications

To determine our engineering specifications, information on test standards, existing market benchmarks, and operational specifications had to be obtained. These were done through reviewing published patents, obtaining official standards, and market research.

Patents

We researched on existing patents. However, information from these sources was largely specific to certain test conditions and test specimens. We plan to incorporate the positive aspects of the information we gathered from these patents and journals into our design. These include methods of achieving the huge forces and torques we require in our tests and the engineering specifications we are attempting to meet.

We found eight patents where each patent was essentially updates of existing patents. With the advancement in technology, the later patents showed that more automation was introduced into the test machines to improve ease of use and precision. However, they were essentially using different methods of achieving the same purpose. For example, Patent 2,370,606, which was published in 1943, describes a test machine in which much of the test procedure is carried out manually with an analog method of data acquisition [3]. On the other hand, Patent US 2003/0101792, which was published in 2003, describes a test machine where automation and motion sensors play a big role and the method of data acquisition is digital [4]. Since we look to decreasing the costs by decreasing our reliance on highly sophisticated automated methods, the older patents would serve as very good information sources in the design phase. We also used the newer patents as reference for improved methods of transmitting the huge forces and torques. All these patents were crucial in our design phase as we brainstormed on methods for accomplishing specific tasks as part of designing a fully operational prototype.

Official Standards

Conducting tests that comply with official standards is important to us. This is because they serve as a benchmark for the comparison and verification of our tests results. Based on our findings for the Four-Ball test, the most common standards are the ASTM standards. The two standards which are most applicable to us are ASTM D4172 and ASTM D2783. The operational standards which would affect the specifications of our test machine are briefly summarized in the paragraphs below.

ASTM D4172 [5]: Standard Test Method for Wear Preventive Characteristics of Lubricating Fluid (Four-Ball Method). This standardized Four-Ball test specifies the size of the testing balls to have a diameter of 12.7 mm, and to be made of chrome alloy steel (ANSI standard steel No. E-52100). These balls follow the standards described in ANSI B3.12, with the addition of extra-polish finish (Grade 25 EP) and a Rockwell C hardness of 64-66. The axial load applied to the balls is to be 392 N, and the top ball is to be rotated at a speed of 1200 RPM for 60 minutes. The temperature of the lubricant is to be regulated at 75°C. To compare the lubricants, scar diameters on the three lower balls are examined and measured.

ASTM D2783 [6]: Standard Test Method for Measurement of Extreme-Pressure Properties of Lubricating Fluids (Four-Ball Method). The steel balls in this standardized test are of the same specifications as the ones previously described for ASTM D4172. However, the top ball will be rotated at a higher speed of 1760 RPM, and the lubricants will be kept in the temperature range of 18°C to 35°C. Instead of a fixed axial load as that described above, a series of tests are to be done at increasing loads, for a duration of 10s. This is done until welding between the steel balls occurs.

At this point in time, there are no established standards for the Twist Compression test. Hence, our capabilities to conduct Twist Compression tests were based on existing market benchmarks.

<u>Market Benchmarks</u>
Research on market benchmarks enabled us to specify a wide range of testing capabilities to suit our sponsor's needs. We managed to obtain technical specifications of Four-Ball test machines from four companies, and our findings are as summarized in Table 2.

Manufacturer	**Ball Diameter**	**Test Speed**	**Load**
Koehler Instrument	12.7 mm	300-3000 RPM	up to 12 kN
IST	12.7 mm	1800 RPM	up to 7.8 kN
PTI	12.7 mm	3-3600 RPM	up to 10 kN
Tribotesters	12.7 mm	60-3000 RPM	up to 10 kN

Table 2: Market benchmarks for the Four-Ball Test Machine

As the Twist Compression test is a relatively new test standard, we were only able to find one company that manufactures Twist Compression test machines. The specifications are shown below in Table 3.

Manufacturer	**Cylinder Diameter**	**Test Speed**	**Pressure**	**Load**
Tribsys	25.4 mm	2-20 RPM	up to 35 ksi	120 kN

Table 3: Market benchmarks for the Twist Compression Test Machine

ENGINEERING SPECIFICATIONS

Setting the engineering specification was a highly iterative process due to the multiple conflicting situations between our sponsor's requirements and our engineering analysis. The engineering specifications were set mainly from the sponsor's requirements, with a few exceptions where external literature search and calculations were involved. Specifically, the loading range, temperature range, temperature fluctuation, inner and outer diameter for both ball and cylinder, RPM range and size of machines were set by the sponsor. The precision targets were obtained from studying the cost-effective level of precision offered by the commercial sector. The alignment tolerance were calculated from the maximum allowable alignment errors that would still enable the machine to operate safely. Eventually, the specifications were set and are shown in Table 4.

Loading Range	up to 35 ksi
Loading Measurement Precision	± 1%

Temperature Range	$0^{o}C$ to $100^{o}C$
Temperature Measurement Precision	± 3%
Temperature Fluctuation	± $5^{o}C$

Size of machine	3 ft by 3 ft by 4 ft
Outer diameter of cylinder	1 in
Cylinder wall thickness	0.05 – 0.13 in
Diameter of balls	0.5 in

RPM Range	To 3600 RPM
RPM Measurement Precision	± 2%

Angular alignment	± 2.86^{o}
Translational alignment	± 0.5 in

Table 4: Engineering Specifications

EXPLORING THE DESIGN SPACE

To ensure that we explore the design space as thoroughly as possible, we sought to develop a systematic approach of producing ideas to fulfill all our component functions. We first created a functional deployment chart to understand all the functions of the system and how they interact with each other. Thereafter, we explored various methods we could implement to fulfill the requirements of each function in the machine. Evaluations were then done using Pugh charts to compare our ideas and decide on the individual components that would best fit our sponsor requirements.

Functional Decomposition

The entire machine was decomposed into components that are functionally independent of each other. Each component was explored for methods of achieving the required function. Each method is explained and analyzed in the section. Upon further discussion, our team considered manufacturing and safety issues which are integral in combining all the functional components in the assembly of the entire machine. These manufacturing and safety issues are also discussed and evaluated in this section. The functional decomposition is found in Appendix P.

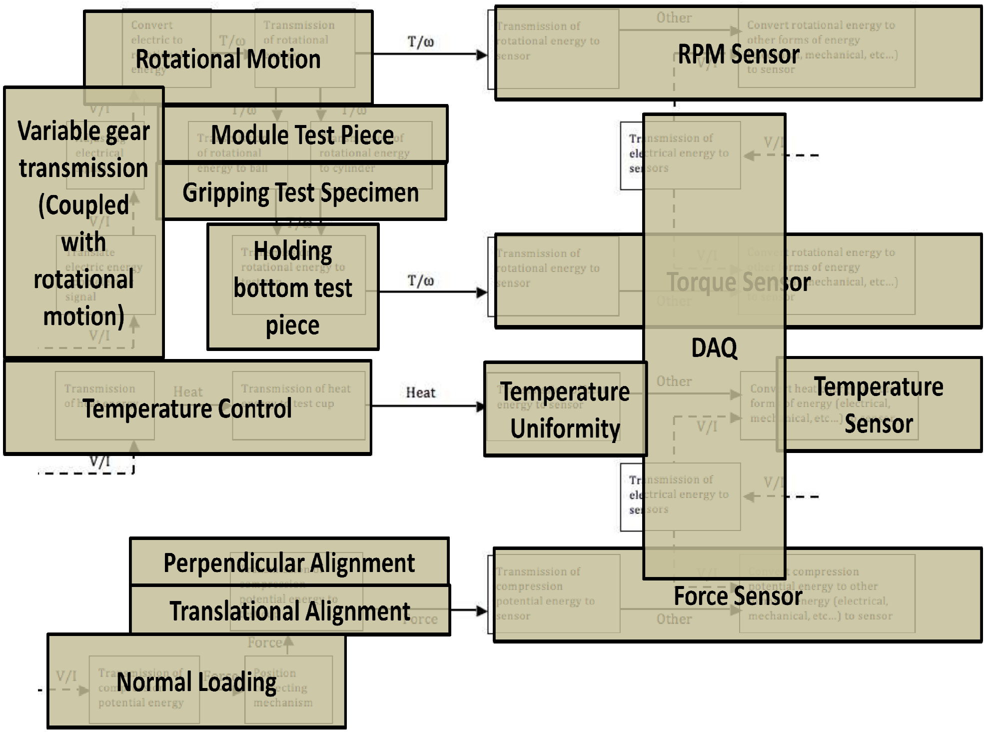

Figure 4: Functional Decomposition showing the categorization of design space coverage

Rotational Motion

The Four-ball and Twist Compression tests both require rotation while the testing surfaces are pressed against each other. Hence, this section will discuss the various concepts considered to build the depth of this breadth. Important quantities are the maximum rotational speeds required of the Four-Ball and Twist Compression tests, which are 3600 RPM and 30 RPM respectively, and the maximum torque required to spin the test piece during testing, which are 5.8 Nm and 175 Nm (See calculations in Appendix C), respectively. A cross-evaluation of the listed designs is conducted at the end.

<u>AC Motor (below 30 HP) With Variable Gear Transmission</u>

AC motors generally have high RPM and efficiency, but work at fixed-speed, constant torque settings. AC motors need controllers such as variable frequency drives or adjustable speed drives to vary torque and speed. There are single-phase and multiphase AC motors, for different voltage inputs. Compared to single-phase motors of the same size, multiphase AC motors generally have higher starting torques, lower current draw, are cheaper, and are better for heavy duty applications [1].

The rationale for the variable gear transmission is to cater to the different gearing ratios, required of a motor with power rating below 30HP (See calculations in Appendix A) to be able to run both types of tests.

AC Motor (above 30 HP) Without Variable Gear Transmission
Using a motor above 30 HP will eradicate the need for any variable gear transmission, which essentially means that the test conditions can be achieved solely by varying the electrical power input into the motor. This is proven by the calculations in Appendix C. However, the main drawback is the high cost incurred in purchasing such motors with high power ratings.

DC Motor With Variable Gear Transmission
DC motors generally have high starting torques and low but consistent RPM. However, DC motors are expensive. Variables of DC motors are easily adjustable by directly changing the input voltage. Due to the high cost of such motors, the option of the DC Motor without Variable Gear Transmission was omitted as that can be achieved more cheaply by using an AC Motor.

Hand Crank
Hand crank costs less than purchasing a motor, but it has a low accuracy for speed and torque control. It requires large amounts of manual labor to generate the huge speed and torque. Even if the speed and torque required for the application is achieved, it will fluctuate too much for results to be considered valid. To provide an estimate, achieving 3600 RPM at a torque of 5.8 Nm and a gear ratio of 1:10 would require one to crank at 6 revolutions per second, at a force of about 12N, which is not feasible.

Overall Evaluations
It is determined that AC motors are the best option for rotational motion because it is accurate yet not very expensive and can meet our requirements of a maximum speed of 3600 RPM and maximum torque of 175 N-m. A Pugh chart is shown below in Table 5.

	Weight	AC Motor (below 30 hp)	AC Motor (above 30 hp)	DC Motor	Hand Crank
Cost	9	+	--	--	+
Size	9	+	+	+	-
Safety	9	++	++	++	--
Precision	9	++	++	++	-
Operation Ease	3	+	++	+	-
Maintenance	3	+	+	+	0
Noise	1	-	-	-	-
Environmental Impact	1	-	-	-	0
Total Score		58	34	31	-31

Table 5: Pugh Chart of Rotational Motion

Loading Techniques
For the Four-Ball and Twist Compression tests, ASTM standards require the contact surfaces to be subjected to compressive axial load. This simulates the surfaces being placed under pressure during use, similar to the conditions lubricants will be used in. This section will discuss the specifications of loading techniques taken into consideration for our Alpha design.

Hydraulic/Pneumatic

Hydraulic and pneumatic cylinders are both machines that apply load when electrically powered. Both are very accurate and can be controlled via motor drives. However, pneumatics are weaker than hydraulics,, thus cost more to achieve the same load.Hence, hydraulics are our main consideration.

Solid Expansion

Solid expansion applies heat to a material to cause it to expand. If this expansion is restricted, the material applies a force. Temperature control and heat loss is an issue in this load application because it would become difficult to calculate the amount of load it creates if the amount of heat residing in the material cannot be measured or calculated. Assuming a typical expansion coefficient of a metal (10-6m/°C), a Young's Modulus of steel to be 102 GPa and the assumption that the expanding solid does not strain, the temperature required to achieve 120 kN is above 10000°C .

Dead Weight

Dead weight option uses weight to apply load. While it is a very simple design and applies a very consistent, it is very inefficient and impractical. It will require us to move a large amount of weight on and off the machine. As an estimate, the loads required to achieve 120kN is a 12 tons mass or equivalent, which is approximately the weight of six cars.

Pulley

Pulleys do not apply load, but when used, a lower load is needed to achieve a higher resulting load. The pulleys commercially available that can take these loads will be accurate and not deform under the loads. A triple sheave pulley with a max load capacity of 30000lb costs $183.70 [5].

Lever

Levers do not apply load, but when used, a lower load is needed to achieve a higher resulting load. The lever will have to be made to not deform under the load applied on it. This is not possible so we will need to account for the deformation as we apply the load onto it, which easily makes it more inaccurate. The strongest cross section would be an I-beam which we will need to purchase.

Overall Evaluation

It is determined that hydraulics are the best option for exerting axial load because although it is somewhat expensive, it is very accurate, easy to control, and can meet our requirements of a maximum pressure of 35 ksi. A Pugh chart is shown below in Table 6.

	Weight	Hydraulic/ Pneumatic	Solid Expansion	Dead Weight	Pulley	Lever
Cost	9	-	--	--	0	0
Size	9	+	-	--	+	+
Safety	9	++	+	--	+	-
Precision	9	++	-	++	-	-

Operation Ease	3	+	+	-	+	+
Maintenance	3	+	+	-	++	+
Noise	1	0	++	++	+	+
Environmental Impact	1	--	0	0	-	0
Total Score		40	-19	-40	18	-2

Table 6: Pugh Chart of Loading Methods

Force Measurements

The test specimen will be subjected to a compressive force along its vertical axis. We plan to place a dummy material in line with the test specimen. By obtaining the force experienced by this dummy material, we will be able to obtain the normal force experienced by the test specimen since it is the same as that of the dummy material. Therefore, our methods center around finding a sensor that will be able to measure the normal force on the dummy material. Considerations include maximum allowed force, cost, size, operating temperature range and ease of installation.

<u>Strain gage</u>

When a normal force is applied, the length of the dummy material will change. Strain is the ratio of this change in length to the original length. A strain gage which contains conducting material is attached to the dummy material. As the dummy material is distorted, the strain gage will distort and cause a change in resistance of the conducting material. The change in resistance is measured by the DAQ and displayed as a function of strain. The resultant normal force can then be determined using this measured strain. Multiple strain gages can be arranged differently based on the geometry and type of deformation of the dummy material. For example, we may use a Wheatstone bridge arrangement with four strain gages. This will allow us to compensate for changes in strain due to temperature and increase the sensitivity of the measurement.

<u>Piezoresistive Force Sensor</u>

Piezoresistive Force Sensors utilize doped semiconductors that are very sensitive to force changes. Because of their sensitivity, they are very difficult to mount and very easy to damage, making them difficult to maintain. Force changes will result in a positional displacement that will significantly change the electrical measurements within the semiconductors. The typical piezoresistive force sensors operate till about 10 ksi. Since our project require a force sensor to withstand 35 ksi, we would need special piezoresitive force sensors that are expected to be much more expensive than the strain gages available now.

<u>Load Cells</u>

A load cell is a transducer which converts force into a measureable electrical output. There are different types of load cells, such as mechanical load cells and strain gage load cells. For our purpose, a strain gage load cell would work the best. Also, the load cells come in a variety of geometry and is chosen based on how the load cell is implemented into the system. Thus, a load cell is essentially a strain gage with a protective casing and electrical wires to be connected to a DAQ all packaged in. These tend to be more expensive than just purchasing strain gages and mounting them on our own.

Overall Evaluation

Based on the Pugh chart shown in Table 7, the strain gage is the best option. This is largely due to the low cost compared to the other options. The strain gage is also comparable to the other options with regards to size, safety and precision, which are the important factors in choosing a force sensor.

	Weight	Strain Gage	Piezoresistive Force Sensor	**Load Cell**
Cost	9	++	--	-
Size	9	+	+	+
Safety	9	+	+	+
Precision	9	+	++	+
Operation Ease	3	0	+	0
Maintenance	3	-	--	-
Noise	1	+	+	+
Environmental Impact	1	-	0	+
Total Score		42	28	17

Table 7: Pugh Chart of Force Measurement Sensors

Torque Measurements

The test specimen will be subjected to a torque transmitted through a shaft. Torque will be measured by either sensing the actual shaft deflection caused by the twisting force, or by detecting the effects of this deflection. In order to measure the torque experienced by the test specimen, we plan to either measure the torque on the shaft or the torque on the test cup. Considerations include maximum allowed torque, cost, size, operating temperature range and ease of installation. Given the large forces and torques we will be running the tests at, we also want to mount the torque sensor on a part of the machine that does not experience excessive translational and rotational motion. Using some clever design and basic physics principles, we are able to convert the torque into a force that can be measured using the methods described above.

Torque Sensors

It is preferable for a torque sensor to be mounted directly on the rotating shaft for torque measurement to be effective. However, this will introduce many problems with respect to connecting the sensor to an external DAQ safely. Furthermore, the prices of torque sensors range from approximately $800 to over $2000.

Force Sensors (Strain Gage/Piezoresistive Force Sensor/Load Cells)

Using basic physics principles, we are able to calculate the torque by measuring the force exerted at an external point connected to the main system, multiplied by the perpendicular distance. This allows us to use all the methods of force measurement described in the section above.

Overall Evaluation

We will be designing our machine such that we can measure the torque by measuring a corresponding force. We will be using strain gages to measure this force. The benefits of using strain gages are described in the evaluation of force sensors in the section above. Torque sensors are not considered because they are very expensive.

	Weight	Torque Sensors	Strain Gage	Piezoresistive Force Sensor	Load Cell
Cost	9	--	++	--	-
Size	9	-	+	+	+
Safety	9	-	+	+	+
Precision	9	+	+	++	+
Operation Ease	3	-	0	+	0
Maintenance	3	0	-	--	-
Noise	1	-	+	+	+
Environmental Impact	1	0	-	0	+
Total Score		-22	42	28	17

Table 8: Pugh Chart of Torque Measurement Sensors

Temperature Measurements

Our sponsor requires that we build a test machine that will be able to test lubricants over the temperature range of 0 to 100 °C. The temperature in the test cup needs to be measured so that test conditions can be monitored and potentially controlled. Considerations include operating temperature range, cost, size, and ease of installation.

<u>Thermocouple</u>

A thermocouple measures temperature differences based on the voltage that is produced between two different metal probes. The metal probes can be shaped to fit into the test cup as desired. Thermocouples can be as cheap as $17, hence the cost spent on temperature measurement is not an issue. However, the main concern is the accuracy of measurement.

<u>Resistance Temperature Detector (RTD)</u>

RTDs are usually made of platinum and works on the basis that the electrical resistance of materials changes linearly with changing temperature. In terms of mode of operation, RTDs are very similar to thermocouples. Comparing the performance of a RTD to that of a thermocouple, a RTD has a larger operating temperature range, a bigger probe sheath, a slower response and a higher accuracy. However, RTDs are less rugged in high vibration environments and are more expensive than thermocouples. An average RTD costs approximately $50 to $90 each.

<u>Thermistor</u>

Thermistors are special solid temperature sensors that behave like temperature-sensitive electrical resistors. The negative temperature coefficient (NTC) types are used mostly in temperature sensing. Thermistors typically work over a relatively small temperature range and appear as embedded components in a larger application. However, most reasonably-priced thermistors($40 to $70) with reasonable level of sensitivity (±0.05°C) do not cover the entire temperature range our sponsor requires.

Radiation thermometer

Radiation thermometers are non-contact temperature sensors that measure temperature from the amount of thermal electromagnetic radiation received from a spot, line or area on the object of measurement. Infrared thermometers are the main type of radiation thermometers. These thermometers are generally bulkier than the other types of temperature sensors and may be difficult to implement in our entire assembly. Furthermore, radiation thermometers are more expensive, costing an average of $100 for a low-cost version.

Overall Evaluation

Based on the Pugh chart shown in Table 9, the thermocouple is the best option since it is cheap, has a reasonable level of accuracy and is easy to implement into our system. From the total scores, we observe that the RTD and thermistor are actually close substitutes for thermocouple. Therefore, in the final implementation, it is still possible to choose RTD or thermistor if they prove to be more compatible.

	Weight	Thermocouple	Resistance Temperature Detector	Thermistor	Radiation Thermometer
Cost	9	+	-	+	-
Size	9	+	+	+	-
Safety	9	-	0	-	+
Precision	9	+	+	+	+
Operation Ease	3	+	+	+	-
Maintenance	3	0	0	0	0
Noise	1	+	+	0	+
Environmental Impact	1	-	+	-	-
Total Score		21	14	20	3

Table 9: Pugh Chart of Temperature Measurement Sensors

Rotary Speed Measurement

The test specimen will be subjected to rotational motion about the vertical axis. The rotary speed needs to be the correct and constant amount for testing standards. The rotary motion is most likely applied by an AC motor, as explained in the rotary motion section previously, so it will be constant but to prove that it is correct, measurements need to be taken.

Contact Tachometer

Contact tachometer is a device that measures the rotary speed of a shaft by coming into contact with it. Some tachometers use a magnet and gear. The approaching and leaving of the teeth of a gear from the magnet creates an electric field and the electric field is measured and used to calculate the RPM. Some may use a switch and a small lever. As the shaft rotates, the small lever physically comes into contact with the switch and the number of “hits” is measured and used to calculate the rotary speed. It is not very expensive, but less accurate.

Laser Tachometer
Laser tachometer is similar to the contact tachometer but does not require physical contact with the shaft to measure rotary speed. It utilizes a laser or infrared light and a receiver. When the shaft rotates, light is reflected off the shaft and into the receiver. The rate the light is reflected is measured to find the rotary speed. It is very accurate, but very expensive.

Optical Encoder
Optical encoders consist of a disc and LEDs. The disc is usually transparent with opaque sections so that it is in a code pattern. The LED shines through the transparent sections of the disc and the photo resistors on the opposite side receive this information and translated into rotary speed. They are as accurate as the disc used and not very expensive. They are also easy to maintain because usually only the disc needs to be replaced.

Overall Evaluation
It is determined that optical encoders are the best option for rotary speed measurement based on the Pugh chart shown in Table 10. Optical encoders are usually less accurate than laser tachometers and as accurate as contact tachometers, but they still fulfill the accuracy requirements we need. They are also cheaper than the other two options. Hence, it is the best choice.

	Weight	Laser Tachometer	Optical Encoder	Contact Tachometer
Cost	9	--	++	-
Size	9	-	++	-
Safety	9	+	+	0
Precision	9	++	+	+
Operation Ease	3	+	+	-
Maintenance	3	-	+	0
Noise	1	+	+	-
Environmental Impact	1	0	0	0
Total Score		1	61	-13

Table 10: Pugh Chart of Motor Performance Measurements

Data Acquisition Devices (DAQ)
To obtain the test results, we require sensors that can measure the force, torque and temperature measurements. A DAQ serves the purpose of converting these measurements from analog to digital signals to be displayed and processed on a computer. Our sponsor requires a sampling rate of 20 kHz. It would be very helpful if the DAQ we use is able to interface with LabVIEW, since we have some experience in LabVIEW programming and the required software is readily available to us. This section serves to discuss the types of DAQs we considered.

Low Cost Data Acquisition Starter Kits
The starter kit is usually used as an educational tool to introduce students to the basics of DAQs. These DAQs are easy to install and use, and is very cost-effective. However, the maximum

sampling rate for the starter kits we found is 14.4kHz, which does not fulfill our sponsor requirements.

Low Cost Multifunction DAQ
Low cost multifunction DAQ usually have limited functions. For example, these DAQs may have a lower sampling rate, smaller number of channels for input and output or lack of grounding. The prices are approximately around $300.

Multifunction DAQ
These multifunction DAQs are able to support bridge-based sensors that will allow for more sensitive data acquisition. Simultaneous sampling is also more likely to be supported. Increased resolution is also common. However, the greatest drawback is its high price. Prices of such DAQs range from approximately one thousand dollars to a few thousand dollars.

Overall Evaluation
We chose low cost multifunction DAQ as the best option based on its price and ability to meet our sponsor's testing needs. The starter kit does not satisfy criteria such as sampling rate while the high end multifunction DAQs are a lot more expensive and have more functions which may be unnecessary. Therefore, the low cost multifunction DAQ strikes a good balance between function and cost. A sample low cost multifunction DAQ is shown in Appendix D. The Pugh chart in Table 11 shows how each type of DAQ fared against each other.

	Weight	Starter Kit	Low Cost Multifunction DAQ	Multifunction DAQ
Cost	9	+	0	--
Size	9	0	+	0
Safety	9	0	0	0
Precision	9	0	+	++
Operation Ease	3	+	+	+
Maintenance	3	+	+	+
Noise	1	0	0	0
Environmental Impact	1	0	0	0
Total Score		15	24	6

Table 11: Pugh Chart of Data Acquisition Device

Temperature Control
Temperature control of the lubricant is desired in the system, as lubricants exhibit different properties at different temperatures. Hence, lubricant tests may have to be conducted at different temperatures other than the ambient temperature. In addition, the temperature of the lubricant is likely to increase during the test, due to heat produced by friction. Because of these reasons, a temperature control device will have to be implemented into the system to actively control the temperature of the lubricant.

Peltier Device
The Peltier device is an electrical device which consists of a heating and cooling plate which can be controlled based on the input current. Because of the relative small size of the device, it can

be integrated easily into the system. Control of the temperature of the heating and cooling plates can also be done fairly easily by varying the input current. However, the heating and cooling plates are joint back-to-back, which means that the device cannot be inserted into an insulated control volume. Hence, the Peltier device must be exposed to a heat inlet or outlet reservoir to absorb or dispel heat into the surroundings.

Heat Sink

A heat sink is made up of a conductive metal, which is able to dispel heat to the surroundings. This is the most economical option, as there is little or no maintenance required for operation. However, it serves only to conduct heat, and to maintain the lubricant's temperature to be closer to the ambient temperature. This essentially leaves the user with little control over the lubricant temperature.

Large Reservoir

The large reservoir method attempts to address the unwanted frictional heating of the lubricant during the test. A large amount of lubricant will be required for this method. Increasing the volume and mass of the lubricant will in turn increase the total heat capacity of the lubricant. Hence, the temperature of the lubricants will be less sensitive to frictional heat inputs.

Refrigeration Cycle

The refrigeration cycle method involves implementing the 4-step process: evaporation, compression, condensation and throttling. This device will be able to provide separate heating and cooling elements, which could potentially achieve a large range of operating temperatures based on the power input and the selection of refrigerants. However, it may be too costly to implement, and may be too bulky given the small size of the actual test cup.

Resistive wire

The resistive wire is basically an electrical resistive heating element. Unlike the Peltier device, it can be inserted into an insulated environment to heat up the internal components. However, the resistive wire offers only heating capabilities, and is not able to cool the lubricant.

Overall Evaluation

The Peltier device is our best option, as it is able to provide for both heating and cooling, with a reasonable amount of control. It is also relatively cost-effective, and is small enough to be integrated into the system. Its requirement to be connected to the ambient environment can be easily accounted for during the design for its implementation.

	Weight	Peltier Heating/ Cooling	Heat Sink	Large Reservoir	Refrigeration Cycle	Resistive Wire
Cost	9	+	+	++	---	+
Size	9	++	++	-	--	++
Safety	9	+	++	++	-	-
Effectiveness	9	++	-	-	+++	+
Temperature Range	3	++	-	--	+++	0
Operation Ease	3	+	++	++	0	+
Maintenance	3	0	+	+	-	0
Noise	1	++	0	++	--	+
Environmental Impact	1	-	-	-	---	-
Total Score		65	50	23	-26	30

Table 12: Pugh Chart of Temperature Control

Temperature Uniformity

Temperature uniformity of the lubricant ensures proper mixing of the heated or cooled portions of the lubricant. This ensures that the lubricant that is in contact with the test ball (Four-Ball test) or the cylinder (Twist Compression test) is well circulated and is at the desired temperature.

Internal Stirring

Internal stirring involves submerging a small stirrer into the reservoir of lubricant, to internally mix the lubricants for temperature uniformity. The stirrer can come in the form of a “mini fan” or an egg beater which is electronically or mechanically powered. Mini blades can also be installed on the modular shaft, which can also act like a stirrer.

External Recirculation

External Recirculation involves installing a fluid pump on the system which drains the lubricant into a large external tank, where cooling and heating can take place. The cooled or heated lubricant is then pumped back into the test area. By controlling the temperature of the lubricant in the external tank, we can control the temperature of the lubricant in the system. The external recirculation system enables the lubricant to be easily tested and controlled, but is relatively harder to implement.

Overall Evaluation

After much consideration, it was decided that the temperature uniformity mechanism may not be required, as long as the surroundings of the test zone can be well controlled. Temperature of the lubricant in contact will be controlled by heat conduction through its surroundings. The “environment” around the test cup will be isolated with an external cup and a lid.

	Weight	Internal Stirring	External Recirculation
Cost	9	+	--
Size	9	+	-
Safety	9	+	++
Effectiveness	9	--	++
Operation Ease	3	0	-
Maintenance	3	--	++
Noise	1	--	++
Environmental Impact	1	+	0
Total Score		5	14

Table 13: Pugh Chart of Temperature Uniformity

Structure
The structure is the foundation upon which the entire structure will be build on, which means that it will have to be able to withstand the axial and torsional loads of the system. Safety is an important consideration, due to the large amount of forces within the system.

<u>Commercial Shop Press</u>
A commercial shop press can be purchased, with a hydraulic pump fitted into the structure. As the shop press is designed for only axial forces, modifications will have to be done for the structure to withstand the torsional load. In addition, as the hydraulic pump will have to be located at the bottom of the test zone, the hydraulic loading mechanism will have to be removed and refitted to suit our needs.

<u>Custom Built Structure</u>
A self-designed structure can be manufactured to suit our needs. Since we will have to modify the commercial shop press anyway, it may be easier to build the structure ourselves. This will allow us to choose the right dimensions with the appropriate structural strengths. However, fitting the hydraulic loading mechanism into the system may be very tedious, as the precision of the loading angle will be very important. In addition, purchasing the structural components and putting them together may cost more than the commercial shop press, which makes it not cost effective.

<u>Overall Evaluation</u>
Due to safety, cost and resources, we will proceed with the commercial shop press. Modifications to the structure, such as adding a base plate and additional diagonal supports, would add to the existing stability of the structure. The provision of the integration of the hydraulic loading mechanism is also an added advantage, and would enable the loading to be done at a more precise angle.

Vertical Alignment of the Shaft
The rotating axle and the test cup have to be aligned perfectly perpendicular to each other to allow for the normal forces and contact areas to be spread evenly through the test pieces. This is especially important for the Twist Compression test, where a slight angular misalignment would place the entire normal load on one edge of the annulus. Misalignments would then cause irregularities in the test results. To ensure that our design is able to accommodate possible

angular misalignments due to manufacturing, installation and operating procedures, we looked into several designs that would allow us to adjust the angular displacement of the test cup within a reasonable range of angles.

Based on our engineering specifications, we set a target of ±2.5° as the angular range from a vertical plane that the rotating axle will be expected to tilt during operations. In addition, we recognized that the design we use need to be able to withstand the large normal forces acting through the test cup. Hence, these factors became the the main motivations behind the design of our preliminary options.

Spring System

The spring system consists of 2 plates supported by 4 springs at each corner of the plate, as shown below in Figure 5: Spring System. This design allows the free angular movement of the top plate; in other words, each individual springs will be compressed depending on how the normal force is being applied on the plate. While this design is operationally easy to implement and use, our calculations show that each spring will have to withstand a minimum of 2,500N. This necessitates the use of custom manufactured springs that are larger than 3 inches in diameter. This would significantly raise the cost and size of the spring system. Spring systems are also very prone to vibration issues unless dampening is implemented.

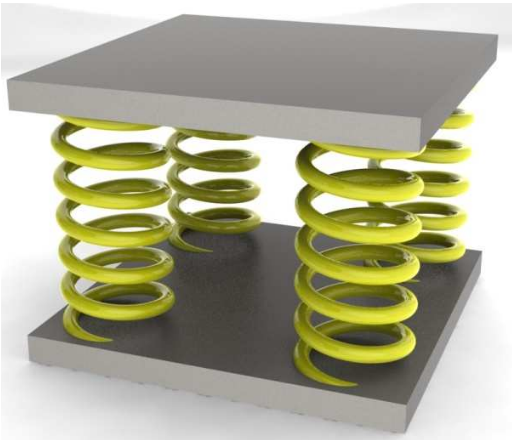

Figure 5: Spring System

Hinge System

The hinge system consists of 3 plates, which have a single cylindrical hinge joint placed between each plate, as shown in Figure 6. The hinge joints are placed perpendicular to each other, allowing for a wide range of angles in which the plate could tilt. This design consists of simple geometries, and would be relatively easy to manufacture. However, because the top plate is only able to rotate along 2 perpendicular axes, it will not able to cover the same vertical angular displacement for all directions. Therefore, to cover the target angular range, the system will have to be made bigger, such that the minimum angular tilt is sufficient to fulfill our requirements.

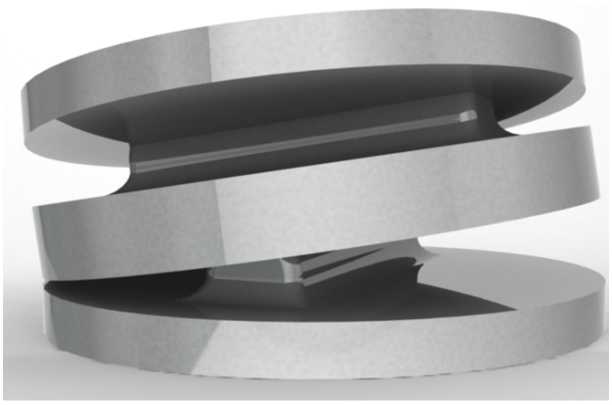

Figure 6: Hinge System

Fluid System

The fluid system consists of 2 plates, and a ‘balloon’ filled with a suitable fluid in between the plates. Fluids are easily displaced, and would allow for conformity to any forces being applied on the top plate. This system eliminates any complexity in manufacturing and installation, in addition to allowing for a wide range of angular movement similar to the spring system. However, the outer material of the balloon will need to be flexible while being strong enough to withstand the high normal loads applied on it. Finding such a material will be a problem.

Ball Joint System

The ball joint system consists of 2 plates, with a ball in between, as shown in Figure 7. This ball joint allows for a large degree of angular displacement in all directions. The design is also relatively simple to manufacture and install. However, the ball joint does not resist movement in the circumferential direction, and the test cup is able to spin about the ball during testing, so a design feature will be needed to stop this from happening.

Figure 7: Ball Joint System

Overall Evaluation

In order to decide on the vertical alignment mechanism we are to use, we created a Pugh chart to evaluate and compare each option based on a set of weighted criteria. This is shown below in Table 1Table 14.

	Weight	Spring	Hinge	Fluid	Ball Joint
Cost	9	-	+	0	++
Size	9	--	-	0	+
Safety	9	+	+	-	-
Effectiveness	9	+	-	0	+
Manufacturability	9	0	+	-	++
Operation Ease	3	0	+	0	+
Maintenance	3	0	+	--	+
Noise	1	0	0	0	0
Environmental Impact	1	0	0	-	0
Total Score		-9	15	-25	51

Table 14: Pugh chart on Vertical Alignment Mechanisms

As seen from the Pugh chart, we have determined the ball joint system to be the most suitable for our needs, followed by the hinge, spring and fluid systems.

Gripping Of Test Specimen

The gripping of the ball and annulus is crucial to the operation of our design, as the test standards dictate that the top ball and annulus must not spin relative to the rotating axle. Each grip design must allow for a large amount of friction between the axle and the test piece. Several options were considered and are evaluated below.

<u>Clamp System</u>

The clamp system makes use of a solid axle with a slit at the end to hold the test piece, as shown below in Figure 8. A screw brings the two sides of the axle together, clamping the test piece in place. However, machining such a shaft manually will be a challenge.

Figure 8: Clamp System

<u>Rubber System</u>

The rubber system consists of a single axle with a rubber lined groove as shown in Figure 9. The design uses the rubber to create a high friction surface between the axle and the test piece, allowing the grip to be increased under a high normal load. The geometries of this design are simple and easy to manufacture. However, the system will be more difficult to use, as it requires the operator to hold the test piece in place until a substantial normal load is applied to hold it in place. This could be a safety hazard for the operator due to the high operating loads that are being applied. In addition, this system does not employ an active gripping system. Should the test pieces slip, the system is not equipped to hold the piece back, and it would be propelled at high speeds into the surroundings. This will be a safety hazard, and safety precautions would have to be taken when using this design. Lastly, based on discussions with our sponsor, the lubricants could slip in and negate the frictional properties of the rubber. Some forms of rubber also react with lubricants, and could affect test results.

Figure 9: Rubber System

<u>Cap System</u>

The cap system consists of an axle and a threaded cap used to hold the ball in place, as seen below in Figure 10. The inner surface of the grooved axle will be roughened to increase the friction between the axle and the ball. Based on the high normal loads that will be placed on the test piece, the friction from this design will be sufficient to hold the ball in place. The cap also prevents the ball from leaving the system in the event that the forces do not act directly through the ball. However, this system is difficult to manufacture as it involves machining threads in small parts.

Figure 10: Cap System

<u>Groove System</u>

The groove system consists of an axle with a non-circular shaped rod end, and test pieces with the same grooves machined into them. This is shown in Figure 11. The design eliminates the need to use friction to maintain the grip. Instead, it makes use of geometrical constraints to move the ball. The design also ensures that the force would be translated through the test pieces and reduces the chance of any lateral movement of the test piece due to a misalignment of the high normal loads. However, this design is difficult to achieve and maintain, as each test piece will have to be individually machined before they can be used.

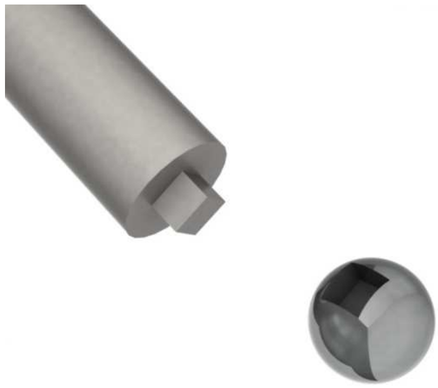

Figure 11: Groove System

Rod System

The cap design explained previously for Four-Ball test will not work for the Twist Compression annular cylinderAs a result, the next best alternative would be to use the rod system, which is similar to the rod system for the modular shaft. Instead of a rod through the shaft, there will be a rod through the cylinder. Similarly, the rod will be removed after positioning and before testing to prevent any potential hazards.

Overall Evaluation

In order to decide on the grip mechanism to be used, we created a Pugh chart to evaluate and compare each option based on a set of weighted criteria. This is shown below in Table 15.

	Weight	Clamp	Rubber	Cap	Groove	Rod
Cost	9	0	+	0	-	0
Size	9	0	0	0	0	0
Safety	9	+	-	++	+	+
Effectiveness	9	+	0	0	++	0
Manufacturability	9	-	+	-	--	-
Operation Ease	3	+	-	+	0	++
Maintenance	3	0	-	+	+	+
Noise	1	0	0	0	0	0
Environmental Impact	1	0	-	0	0	0
Total Score		12	2	15	3	9

Table 15: Pugh chart on grip mechanisms

As seen from the Pugh chart, we have determined that the cap system is the most suitable for our needs, followed by the clamp, groove and rubber systems.

Translational Alignment

Translational alignment is required to cater to positional errors that might occur, causing the axis of the test cup and the main axle to misalign. A misalignment of axes will generate a moment and given the high loadings that the machine is expected to run at, there will be a high probability of machine failure. After the position is adjusted, the position will be clamped down, before actual loading and testing occurs. This section discusses the various options to achieve this translational alignment. They are eventually scored against each other at the end of this section.

<u>Two-Layer Sliding Design</u>
This design involves two plates placed on top of each other, which have slots perpendicular to each other, shown in Figure 11. The top plate and bottom plate will slide along each other and would hence allow the test cup, which sits on the top, to move around. A bolt is then used to put through the slots of both plates and would be tightened to hold the plates down. This is an elegant design that is effective, simple and easy to manufacture.

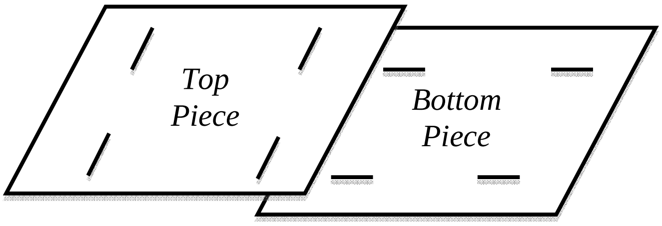

Figure 12: Two-Layer Sliding

<u>Cross Plate Design</u>
The cross plate design, shown in Figure 13, involves a cross-shaped plate placed on top of another plate driven by a hydraulic jack from the bottom. Before testing, the cross plate is free to slide on the bottom plate, hence allowing the test cup to be aligned. Upon alignment, a butterfly bolt will be tightened to hold the two plates together. In addition, C-shaped clamps will the bottom plate at the corners of the cross. This prevents the top plate from sliding during testing. This design is easy to manufacture due to the geometry of the components. Sheet metal and clamps can be purchased at low cost.

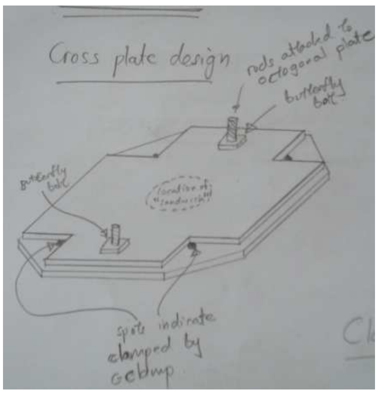

Figure 13: Cross Plate Design

<u>Clamp-only Design</u>
The clamp-only design allows the testing cup to be aligned with the main axle before it is clamped. The top plateholding the testing cup will be pushed upwards by another plate, which in turn is pushed by a hydraulic jack. This upward motion is guided by the perpendicular guides at the 4 corners as shown in Figure 14. Upon alignment, the top plate will be clamped to the bottom

plate. This design is a relatively low-cost design and is similar to the cross plate design. However, it is more bulky than the cross plate design due to the presence of more structures.

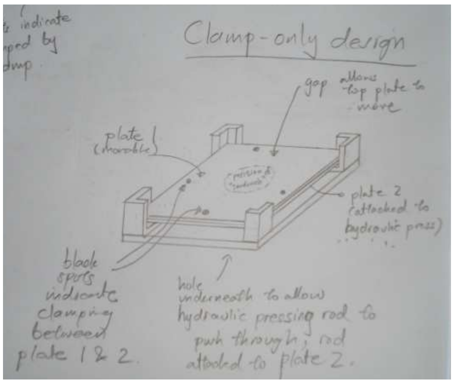

Figure 14: Clamp-Only Design

Free Bearing Design
The Free Bearing Design allows the test cup to realign during testing. Essentially, the test cup sits on a set of bearings that allows translational motion. However, there will be walls present to limit the translational motion, as shown in Figure 15. A disadvantage of this design is that it is potentially less safe compared to other designs. While theoretical calculations indicate that the motion is small, the actual operational situation is unknown.

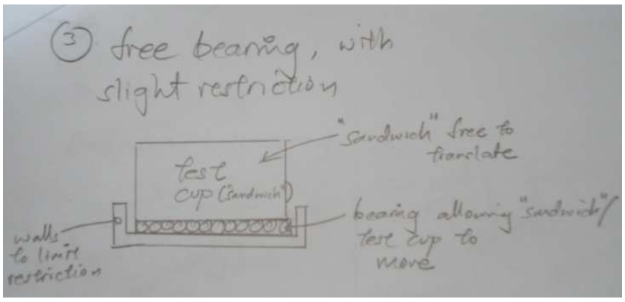

Figure 15: Free Bearing Design

Customized Bearing Design
The Customized Bearing Design allows the test cup to shift in alignment before testing. Upon testing, the bearing is compressed and an internal mechanism locks the position of the test cup. This is the most elegant design, however, it is also very fragile. Upon the high loads that the machine is expected to experience, there is a very high probability of mechanical failure due to the complexity of the design and degree of stress concentration. This design is shown in Figure 16.

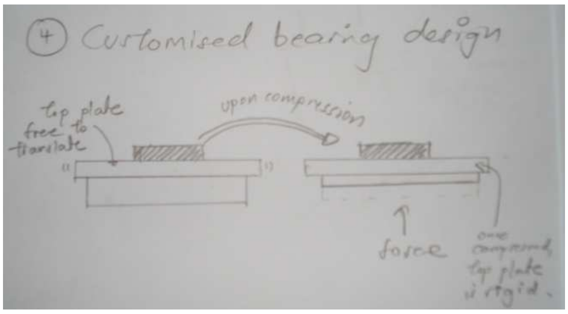

Figure 16: Customized Bearing Design

Overall Evaluation

The two-layer sliding design emerged the most suitable mainly due to its simplicity and cost-effectiveness.

	Weight	Two-Layer Sliding	Cross Plate	Clamp Only	Free Bearing	Customized Bearing
Cost	9	++	+	++	-	--
Size	9	+	+	+	++	++
Safety	9	++	++	-	-	0
Precision	9	+	+	++	++	++
Manufacturability	9	++	+	+	-	-
Operation Ease	3	+	+	++	++	++
Maintenance	3	+	+	+	+	+
Noise	1	+	+	0	0	+
Environmental Impact	1	0	0	0	0	0
Total Score		75	61	54	18	19

Table 16: Pugh chart on translational alignment design

Modular Test Piece

The modular design seeks to allow the testing piece (either the Four-Ball Test top ball, or the Twist Compression Test cylinder), to be attached to the main axle. The test piece is also required to be removable so that the machine can operate both types of tests. An important point to note is that in all designs, the main axle has a square hole and the modules have a corresponding square cross-section that will fit into the hole. This allows the main axle to transmit the torque to the module. For simplicity, the illustrations in this section have omitted this “square” attribute, and will be arbitrarily be represented by cylindrical shapes.

Rod Design

The rod design is a very simple design, involving only a rod to be placed after the module is put into the main axle. The rod would hold the module in place, and just before testing, the rod will be removed to ensure that there will be no stray spinning objects. This design requires the user to manually place and remove the rod, but it is a very simple, low cost and effective design. This is illustrated in Figure 17.

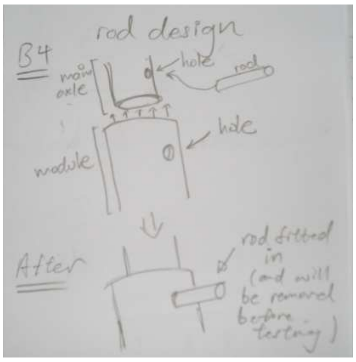

Figure 17: Rod Design

<u>Snap On (Ball-Spring Design)</u>
The ball-spring design involves a ball-spring mechanism located at the main axle, which allows the module to snap onto the main shaft. This design is safer, but involves a more complex design. The design is illustrated in Figure 18.

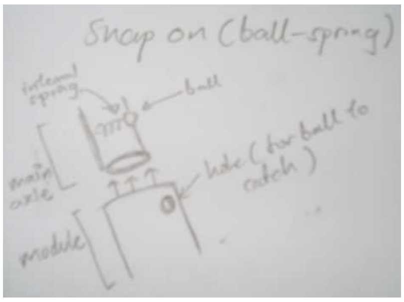

Figure 18: Snap On (Ball-Spring Design)

<u>Velcro Design</u>
The Velcro Design involves industrial strength Velcro to be placed in the module and the main axle. Once the module is placed into the main axle, the Velcro surfaces will interlock, hence holding the module in place. When required, a tug would release the module from the main axle. This design may provide a lot of convenience for the user, but it is heavily dependent on the Velcro which might be damaged under extreme pressures. The design is showed in Figure 19.

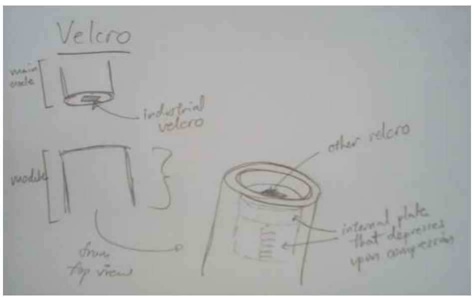

Figure 19: Velcro Design

Overall Evaluation

The rod design was determined to be the most apt due to the simplicity and ease of manufacturing. Although it convenience is compromised due to more manual work required, it is still the most effective option.

	Weight	Rod	Snap On	Velcro
Cost	9	++	0	0
Size	9	++	++	++
Safety	9	0	++	0
Precision	9	++	0	+
Operation Ease	3	-	+	+
Maintenance	3	+	+	-
Noise	1	0	0	0
Environmental Impact	1	0	0	0
Total Score		54	42	27

Table 17: Pugh Chart on Modular Test Piece

Vertical Motion of Plate

Inverted Cup Design

This design involves an inverted cup that will be driven by the force generator from the inside of the cup. As the inverted cup is pushed and moved upwards, it will slide along a support to ensure straight alignment. Essentially, the cup base has to be made of a strong material, while the cylinder can be made of a cheap and light material. Given the size of the hydraulics press piston and the alignment tolerance, it is determined that the operational forces will not point out of the piston to create a moment that will break the cup. It is expected that the force will be directed within the piston.

Four Damper Design

This design involves using four used car dampers to support the plate as it slides up and down due to the hydraulic press. The car dampers are expected to be sturdy, and may interfere with the angular alignment of the plate.

Slider Design

This design uses two poles to guide the plate vertically and is very similar to the four-damper design. However the slider design utilizes sliders that are less rigid and hence will more easily allow the plate to self-correct if it is out of line. It is also expected that the design might function as a stopper that prevent the cup from spinning due to the applied torque. Nonetheless, even though the sliders can allow the plate to realign, there is still a chance of the plate bending.

Overall Evaluation
It is determined that the inverted cup is the most suitable and this is due to the sturdiness of the design. It is also relatively cheap and easy to manufacture. It is also a safe design.

	Wt	Inverted Cup	Slider	Damper
Cost	9	+	+	+
Safety	9	+	0	-
Sturdiness	9	++	+	+
Size	3	0	+	+
Torque Control	3	0	0	++
Operation Ease	3	+	+	-
Maintenance	3	0	0	0
Noise	1	0	0	0
Environmental Impact	1	-	-	-
Total Score		38	23	14

Table 18: Pugh Chart of Vertical Motion of Plate

Holding Three Balls

Square Cut and Ball Groove
This design involves creating grooves that would allow the test balls to sit in. The groove is designed in a way that will prevent the balls and plate from rotating on the spot. The balls will be used for the Four-Ball test, while the square plate will be used for the Twist Compression test. This is easy to manufacture and allows for parts to be exchangeable should damage occuron the square plate or balls. The grooves are deep enough to keep the balls in place. To further prevent displacement of the balls, a top cap will be screwed down to hold the balls in place.

Ring Design and Replaceable Cup
In this design, the balls are held in a ring and hence are compressed against each other. The geometry of the ring reduces stress concentrations as compared to a more angular shape. Upon compression the ball will be fastened in place. As for the Twist Compression test, the cup, itself, is used as the flat surface and the whole cup will be replaced when worn out. Though this design is feasible, there is a chance that safety might be an issue and inconvenient as the test cup may have to be replaced repeatedly.

Alternate Shapes
In this design, the contact surfaces used in a regular test are achieved by creating test pieces of alternate geometry. For example, instead of using the balls, cylinders with ball ends could be considered. This is a very creative design, however the main issue involved would be the available of these test pieces in an alternate geometry. This indirectly leads to the cost of testing, as the plates, cylinder and ball surfaces will most likely have to be replaced for each new test.

Overall Evaluation

The square cut and ball groove was the most suitable design due to the low cost involved and the simplicity of the design. It is also interchangeable, which is a very cost effective method for testing, as compared to replacing full test cups or alternate shaped test pieces.

	Weight	Square Cut and Ball Groove	Ring Design and Replaceable Cup	Alternate Shapes
Cost	9	++	--	-
Size	9	+	0	+
Safety	9	+	-	+
Effectiveness	9	0	0	0
Manufacturability	9	+	-	--
Operation Ease	3	+	-	++
Maintenance	3	+	-	--
Noise	1	0	0	0
Environmental Impact	1	0	0	0
Total Score		51	-42	-9

Table 19: Pugh Chart of Holding Down of Bottom Test Surface

DESIGN CONCEPTS

Keeping in mind the options of each component presented earlier, we generated four design concepts. Design Concept #1 and Design Concept #2 are concepts that were evaluated to be feasible and well-balanced designs. Design Concept #3 is generated based on the ease of integrating the components together and Design Concept #4 is the concept that primarily addresses the engineering specifications without much consideration of cost or ease of integration.

Design Concept #1 (DC1): Feasible Design

The components of Design Concept #1 were selected based on the overall functionality, cost, safety and ease of integration, and are shown in Table 20 below. A rough illustration of Design Concept #1 is shown in Figure 20 below.

Category	Component	Selected Device
Drivers	Spinning Mechanism	Motor
	Loading Mechanism	Hydraulic
Sensors	Axial Load Sensor	Strain Gauge
	Torque Sensor	Strain Gauge
	Temperature Sensor	Thermocouple
Controls	Temperature Control	Peltier
	Tilt Control	Ball Balance
	Lateral Control	Bolted Plate
Modular Design	Modular Rod Securing Mechanism	Rod
	Four-Ball Modular Design	Cup
	Four-Ball Securing Mechanism	Cap Screw on
	Twist Compression Modular Design	Groove
Structure	Structure	Benchpress
	Shaft Straightening Design	Shaft Bearings
	Base Plate Alignment	Inverted Cup

Table 20: Components selection for Design Concept 1

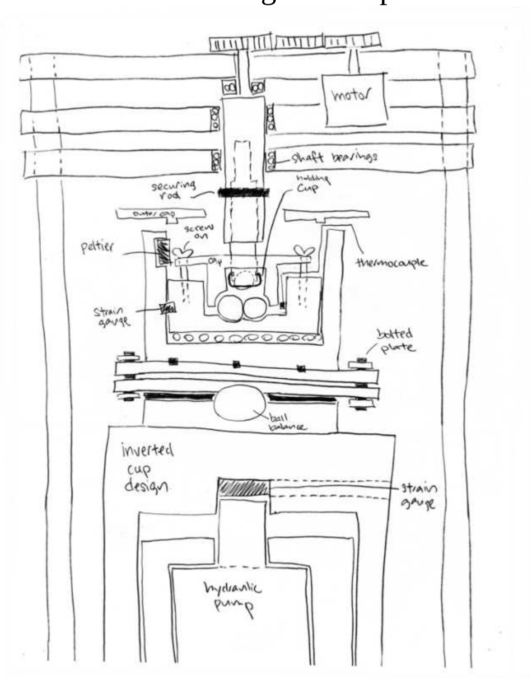

Figure 20: Initial sketch up of Design Concept #1

Components

Drivers: Drivers consist of both the spinning mechanism and loading mechanism. From the previous section, we concluded that the only feasible spinning mechanism will be an electric motor, and that the only feasible loading mechanism will be a hydraulic loading mechanism. The components of driving mechanisms will be similar for the other design concepts.

Sensors: Sensors consist of the axial load sensor, torque sensor, and temperature sensor. Strain gauges were decided to be the most cost-effective method to measure the exerted loads and torques. As torque sensors are generally too expensive, strain gauges will be placed on a dummy material, which will be aligned at the side of the cup in a position where it will experience the lateral strains of the cup. This will eventually enable us to calculate the torque transmitted through the shaft. Further elaboration on the design, positioning and calculations will be done in the analysis section. For temperature sensing, a thermocouple will be used, as it is also the most cost-effective option. Both the strain gauges and thermocouple will have detection ranges sufficient for our engineering specifications.

Controls: Controls consist of temperature control, tilt control and lateral control. For temperature control, an external cup and lid will be used to insulate the environment surrounding the test zone. This design aspect will be similar for the other design concepts. For design concept #1, a Peltier heater/cooler will be placed within the external cup to control the temperature of the system. This was chosen as it has the flexibility of both heating and cooling capabilities while being relatively cost-effective. For tilt control, the ball balance design was selected, as it is a simple design that is sufficient to automatically align itself to an angled load. For lateral control, we selected the bolted plate mechanism, as it is relatively quick and simple to manufacture, and is able to be easily integrated due to its simple nut and bolt components.

Modular Design: The modular design consists of the main modular rod securing mechanism, the Four-Ball modular components, and the Twist Compression modular component. For the main rod securing mechanism, we selected the through-rod design, as it is quick and simple to manufacture and operate, and yet able to accomplish its purpose of keeping it in place while the device is not in use. For the four-ball modular components, we selected the cup design to hold the top ball in place, and the cap screw on for the bottom three balls. The cap screw on mechanism for the bottom three balls will be similar for all the other design concepts. For Twist Compression, the groove concept is selected. The groove concept allows for flexibility of the tests as it allows for interchangeability while not compromising on test conditions.

Structure: The structural components consist of the overall structure, the shaft straightening design and the base plate alignment. For the overall structure, a shop press is used. The selection was primarily due to safety considerations, as the commercial shop press is commercially made to withstand the high loads. Modifications will have to be done to enable the shop press to handle torsional loads as well. For shaft straightening, shaft bearings will be used for vertical alignment. This method will be used for the other design concepts. For the base plate alignment, we chose the inverted cup design. It is the most feasible design due to safety and stability considerations, which justifies for the more material required.

Design Concept #2 (DC2): Alternative Feasible Design

Design Concept #2 puts together remaining feasible components that were not used in Design Concept #1. Hence, it should be also noted that the components in both Design Concept #1 and Design Concept #2 were preliminarily decided to be practical and well-balanced in terms of cost and functionality. The components are tabulated in Table 21, and the design is illustrated in Figure 21 below.

Category	Component	Selected Device
Drivers	Spinning Mechanism	Motor
	Loading Mechanism	Hydraulic
Sensors	Axial Load Sensor	Piezo Resistor
	Torque Sensor	Piezo Resistor
	Temperature Sensor	RTD
Controls	Temperature Control	Resistive Heating, Heat Sink
	Tilt Control	Ball Balance
	Lateral Control	Clamped Plate
Modular Design	Modular Rod Securing Mechanism	Snap on
	Four-Ball Modular Design	Clamp
	Four-Ball Securing Mechanism	Cap Screw on
	Twist Compression Modular Design	Clamp
Structure	Overall structure	Benchpress
	Shaft Straightening Design	Shaft Bearings
	Base Plate Alignment	Sliding Bars

Table 21: Components selection for Design Concept 2

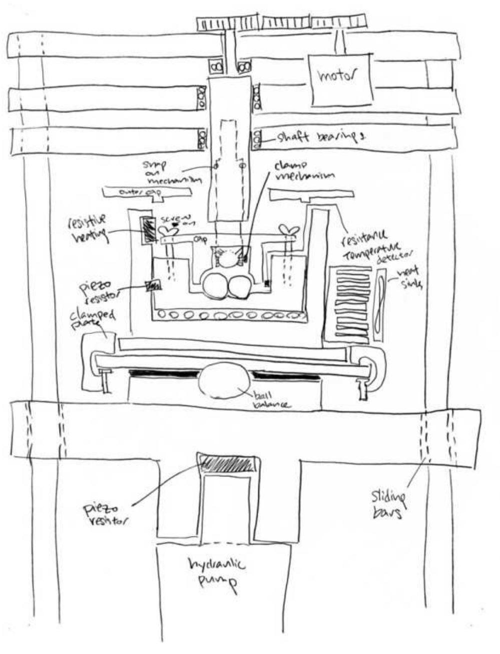

Figure 21: Initial sketch up of Design Concept #2

Components

As the selection criteria of the components were similar to DC1, only components which are different from DC1 will be elaborated in the sections below. In addition, since most of these components are close alternatives to those chosen in DC1, they will be frequently compared to each other in the sections below.

Sensors: A different set of sensors were chosen for DC2, and they consist of the axial load sensor, torque sensor, and temperature sensor. For the axial load and torque sensors, the Piezo Resistive sensor was selected, due to its sensitivity to smaller temperature changes. However, it is comparatively more expensive than the strain gauge chosen in DC1. The Resistive Temperature Detector (RTD) is chosen for temperature sensing in DC2. Similarly, it is a more sensitive but costly alternative to the thermocouple chosen in DC1.

Controls: The selected temperature control and lateral control mechanism in DC2 is different from the ones selected in DC1. For temperature control, resistive heating and heat sink were considered, as they are easier to implement (as compared to the Peltier device which requires access to the external environment). However, it is unable to cool the lubricant to below the ambient temperature. A clamped plate was considered for lateral control. This design is slightly more elaborate as it consists of clamps and corner pieces, but could potentially withstand a higher torsional load.

Modular Design: All the modular components in DC2 are different as compared to DC1. The snap-on design was chosen for the modular rod securing mechanism, which is more user-friendly but has a more complicated manufacturing process than the through-rod design. For the Four-Ball modular design, the top ball will be clamped on to the shaft, instead of being simply supported by a cup beneath it. This will provide a more secure fit, although it is likely to be unnecessary due to a higher coefficient of friction at the contact surface of the top ball and the shaft, as compared to the coefficient of friction at the contact surface to the three balls. Similarly, a clamp design for the Twist Compression modular shaft provides a more secure fit, but may not be necessary.

Structure: The only structural change made in DC2 is the base plate alignment mechanism. Instead of the inverted cup design in DC1, the sliding bar concept is considered, which will allow the plate to move vertically with less materials and manufacturing. It can take a vertical loading, but poses a safety concern as it could potentially be misaligned during the loading process.

Design Concept #3 (DC3): Well-Integration of Components
DC3 was designed with components that will be easily integrated with each other with minimal compatibility conflicts. These components are tabulated in Table 22, and the design is illustrated in Figure 22 below.

Category	Component	Selected Device
Drivers	Spinning Mechanism	Motor
	Loading Mechanism	Hydraulic
Sensors	Axial Load Sensor	Load Cell
	Torque Sensor	Load Cell
	Temperature Sensor	Infrared Thermometer
Controls	Temperature Control	Peltier
	Tilt Control	Hinge
	Lateral Control	Four-Corner
Modular Design	Modular Rod Securing Mechanism	Manual Holding
	Four-Ball Modular Design	Rubber Grooves
	Four-Ball Securing Mechanism	Cap Screw on
	Twist Compression Modular Design	Permanent Fixture
Structure	Overall structure	Shop Press
	Shaft Straightening Design	Shaft Bearings
	Base Plate Alignment	Inverted Cup

Table 22: Components selection for Design Concept 3

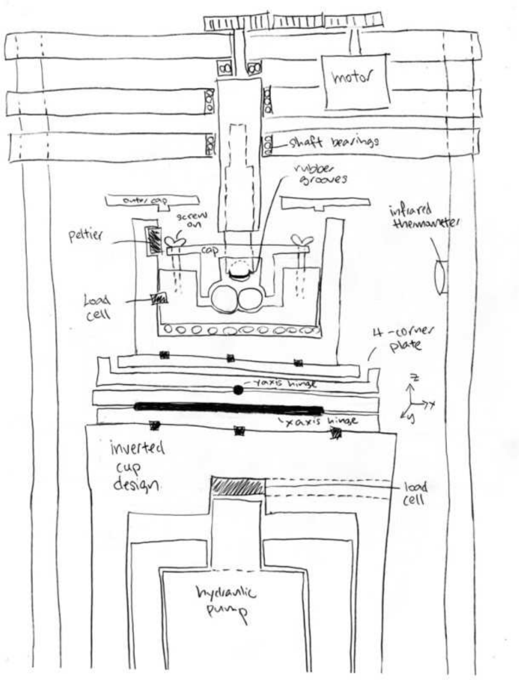

Figure 22: Initial sketch up of Design Concept #3

Components

As these components were selected based on how well they can be integrated together without much modifications, the sections below will focus primarily in this aspect. However, they may not be the best option in terms of functionality or cost, which would have been mentioned in the preceding sections that introduced these components.

Sensors: In DC3, three new sensors are selected. The load cell was selected to measure both the axial load and the torque of the system. Load cells are easy to integrate into the system, as the device comes as a single block that can be used without any modifications to it. An infrared thermometer was selected for temperature sensing, as it can be mounted away from the test zone, which eliminates the need for wiring into the system and modifying the cup design.

Controls: For temperature control, the Peltier device was selected, as it is relatively small and easy to integrate into the system. The hinge concept was selected for tilt control, as the individual tilt controls in the x and y direction makes the design simple to implement. Finally, the four-corner concept was selected for lateral control, as the design only involves putting metal at the corners of the base plate without affecting anywhere else on the surface of the plate.

Modular Design: The mechanisms for the modular designs were simplified to cut on unnecessary machining on these modular components, in order to reduce possible complications such as misalignment and precision issues. For the modular rod securing mechanism, the operator will have to manually hold the rod when the machine is not at its test position. For the Four-Ball modular design, rubber grooves will be used to hold the top ball in place when the device is not in use. The Twist Compression modular device will have its shaft and cylinder welded into a single piece as a permanent fixture.

Structure: The shop press and inverted structure were the components selected, as they are components that are independent of each other, unlike the sliding bars concept, which may be affected by slight torsional twisting of the shop press during tests.

Design Concept #4 (DC4):

DC4 consists of components which extend the testing capabilities of the machine. With that in mind, components with the best capabilities are selected, with less regard to cost and manufacturability. These components are tabulated in Table 23, and the design is illustrated in Figure 23 below.

Category	Component	Selected Device
Drivers	Spinning Mechanism	Motor
	Loading Mechanism	Hydraulic
Sensors	Axial Load Sensor	Piezo Resistor
	Torque Sensor	Piezo Resistor
	Temperature Sensor	RTD
Controls	Temperature Control	Refrigeration Cycle
	Tilt Control	6-axis Machine
	Lateral Control	6-axis Machine
Modular Design	Modular Rod Securing Mechanism	Snap on
	Four-Ball Modular Design	Clamp
	Four-Ball Securing Mechanism	Cap Screw on
	Twist Compression Modular Design	Clamp
Structure	Overall structure	Custom Design
	Shaft Straightening Design	Shaft Bearings
	Base Plate Alignment	Four-Slider

Table 23: Components selection for Design Concept 4

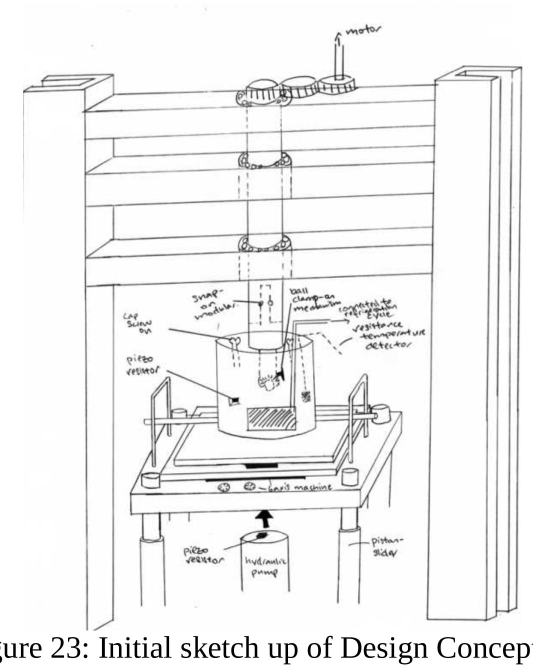

Figure 23: Initial sketch up of Design Concept #4

<u>Components</u>

Similar to DC3, the below sections will justify the components selection based on the capability of each device.

Sensors: Like DC2, the Piezo Resistor and the Resistive Temperature Detector (RTD) were selected for load sensing and temperature sensing respectively, due to its sensitivity to small changes.

Controls: More changes are seen in the controls of the system. For temperature control, a refrigeration cycle was chosen, as it is the option with the widest range of temperature control. For tilt and lateral control, an automated 6-axis machine was chosen, which can precisely move the test plate to its desired position.

Modular Design: For the modular rod securing mechanism, a snap-on mechanism was selected, as it will be the most user-friendly option. Both the Four-Ball and Twist Compression modular designs will have their top ball or test cylinder firmly mounted on using a clamp, as this provides additional support on the top ball/ test cylinder.

Structure: A customized design for the exterior structure will ensure that the physical dimensions and load capabilities of the structure will suit our needs. The four-slider base plate alignment mechanism will simplify the structure, and will ensure smooth vertical motion of the base plate.

ALPHA DESIGN SELECTION

With the four design concepts in mind, we generated a Pugh chart for the entire concept, shown in Table 24 below. The ratings of each design concept were given based on a collective evaluation of all the individual components of the concept.

	Weight	DC #1	DC #2	DC #3	DC #4
Fulfillment of Engineering Specifications	9	++	++	++	+++
Safety	9	++	+	++	0
Cost	9	-	-	---	---
Manufacturability	3	+	0	++	--
Compatibility	3	++	++	+++	0
Size	3	--	--	--	-
Ease of Operation	1	0	0	--	++
Maintenance	1	0	0	-	--
Noise	1	0	0	0	0
Environmental Impact	1	0	0	0	-
Total Weight		30	18	15	-10

Table 24: Pugh chart

Justifications for Ratings

<u>Fulfillment of Engineering Specifications</u>

The components that accounts for our engineering specifications are: Drivers, sensors, and control. DC1 is able to fulfill most of the required engineering specifications. The driving mechanisms, like all four design concepts, are able to fulfill the load and spinning requirements. Sensing and control capabilities are also within the required ranges. DC2 is able to fulfill most of the required engineering specifications. Although the sensors in DC2 are more sensitive than those in DC1, the added sensitivity adds little value to addressing the engineering specifications. In addition, the temperature control capabilities of DC2 does not fully satisfy the engineering specifications as the lubricant will not be able to cool to temperatures below the ambient temperature. DC3 is able to fulfill both the sensing and control requirements, as all its components are able to meet the engineering specifications. DC4 contains components which are both in DC1, DC2, DC3, with the exception of the Refrigeration cycle and the 6-axis machine. These two components contribute the increased rating of engineering specifications as the refrigeration cycle contains a higher temperature range, and a greater temperature range will enable a greater potential rate of heat transfer to the lubricant. The automated 6-axis machine will be able to achieve precise tilt and lateral control, which will better fulfill the engineering specifications, as minimal tilt precision will be beneficial to test conditions. Because of these two components, DC4 gets a higher rating than the other design concepts.

Safety

The safety-sensitive components are: Tilt and lateral control and structural components. These components affect the structural strength as well as the stability of the structure. For DC1, the tilt and lateral controls are theoretically stable and safe to manufacture. The structural components are also safe, as the shop press and the inverted cup concept are the most reliable options out of the generated concepts. DC2 essentially contains components with designs of similar reliability. However, the sliding bar mechanism which aligns the base plate has a remote possibility of being obstructed during the process of sliding. Hence, this presents a safety hazard which reduces the safety score of DC2. DC3 contains the same safety-sensitive components as DC1, with the exception of the hinge concept and the four-corner concept. As these two concepts can be said to not have obvious hazards, DC3 is a relatively safe design. The tilt and lateral controls of DC4 is done by a 6-axis commercial machine, which will be designed to take the test loads and is hence safer than the other design concepts. However, the structure of DC4 is custom designed, which poses the uncertainty of a self-built structure which is untested by time. As the structure poses a major safety concern, DC4 is not well rated for safety.

Cost

Most of the sensor and control components have cost as a major consideration during the selection process. DC1 contains the cheapest set of sensors. The costs of the control components are similar to all the other design concepts, with the exception of DC4 (which costs more). Hence, DC1 scored relatively well as compared to the other design concepts. The sensor components of DC2 are slightly more expensive than DC1, while the control components cost about the same. This gives DC2 a lower cost rating than DC1. The sensors selected in DC3 cost much more than those in DC1 and DC2, which gives it a lower cost rating. The control components in DC4 consist of the Refrigeration cycle and the 6-axis machine, which are the most expensive options in the controls category. This gives DC4 a low cost rating as well.

Overall Evaluation

Based on the ratings on the Pugh chart, DC1 appears to be the most viable design. Generally, DC1 outweighs DC2, as it contains components that satisfy the engineering specifications, yet not exceeding functionality requirements. For example, the selected sensors are able to cover the entire required range of engineering specification, hence being more cost effective than the sensors selected in DC2. In addition, it minimizes on unnecessary features such as the snap on modular rod and the clamp down lateral alignment.

PARAMETER ANALYSIS

After choosing an Alpha design, which was developed mostly qualitatively, parameter analysis was conducted to quantitatively justify and improve the Alpha design to develop the final design. As the machine has numerous components, a systematic and logical approach had to be used. As such, the analysis is done component by component. Within the discussion for each component, a description and the necessary quantitative reasoning are provided with the aid of, if applicable, engineering fundamentals, Solidworks, DFMEA, and DesignSafe.

In general, certain changes and modifications to the Alpha design were required due to numerous reasons. Some reasons include commercial constraints, where actual stock material and parts were not available in the geometry or dimension desired, or possible safety concerns that arose from the high speeds and loads that the machine is designed to operate at. Table 25 lists, in order, the components that are discussed.

Category	Component
Structure	Structure
Driving Mechanism	Motor and Motor Control
	Gear System
	Main Driving Mechanism Dimensions
Test Region	Four-Ball Test Module
	Test Contact Plate
	Test Cup Dimensions
	Test Cup Internal Temperature Control
Measurement Issues	Torque Measurement Rod
	Data Acquisition Device (DAQ)
	Normal Loading Force Measurement
	Torque Measurement
	Temperature Measurement
	Speed (RPM) Measurement
Safety	Protective Safety Shield

Table 25: Order of Discussed Components

These calculations and part selections were made based on the assumption that the Twist Compression cylinder inner diameter would be half the outer diameter. This increased the normal loads of the system to be 97000N. This assumption affected our motor, drive and material selections.

Structure

The shop press used in our main structure was analyzed for torque reactions. The Omega 60253 bench press is rated to hold up to 25000N in the vertical direction, which is higher than the required 10000N. However, it is not rated for any torque, making it necessary for us to carry out our own analysis to ensure that it can withstand at least 175 Nm as required.

The structure has a complicated geometry made up of U-channels and L-channels. In order to simplify the process of analysis, a Finite Element Analysis (FEA) was conducted using SolidWorks Simulation. The actual dimensions of the structure was created on Solidworks and analyzed for torque, as shown below in Figure 24. The structure was assumed to be fixed at the base area, and the torque applied to two mounting holes. These mounting holes were selected based on the overall position of the cantilever beam where the torque will be applied. The material of the structure was assumed to be A36 steel, which was found to be a common material for steel U-channels.

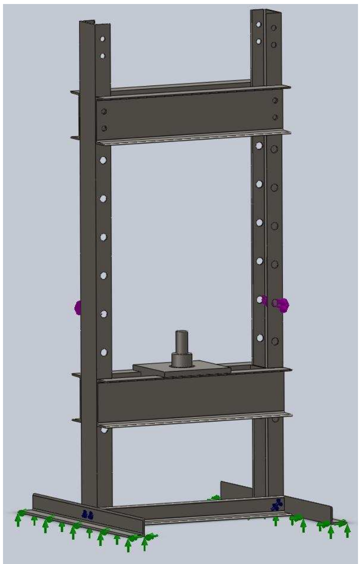

Figure 24: Constraints and Loads on Structure Stress Test

The results of the analysis are shown below in Figure 25. The factor of safety for the structure was determined to be 9.4. This eliminates the need for any modification of the structure to strengthen it.

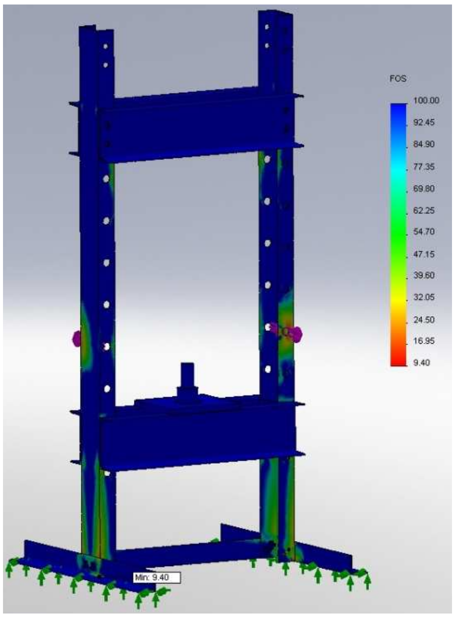

Figure 25: Factor of safety for structure under torque

Motor and Motor Control

The motor to be used in our system is the WEG 00518EP3E184T motor. This is a five horsepower motor that has a full load torque of 15.0 lb-ft and a full load speed of 1730 RPM. The motor drive that will be used to run the motor is a Yaskawa CIMR-VUBA0018FAA motor drive. This motor drive is rated to take a 240V single phase power supply and output a three phase power supply for a five horsepower motor.

Infrastructure Considerations

The five horsepower motor that was originally considered for our Alpha Design required a three phase power input. In order to supply this power, we need a motor controller that could convert a single phase power supply to a three phase power supply which is not available in the sponsor's lab due the high costs involved in its installation and the lack of infrastructure..From our correspondence with motor suppliers and motor drive manufacturers, we found that it was possible to obtain a motor controller that can convert the power phase from a single phase to three phase supply, at the same time, providing enough power to run a five horsepower motor.

Motor Selection

The final motor selection was dependent on various factors including the gearing requirements from the speed and torque of each test, as shown in the next section. We were also made aware of a three horsepower motor that is available in the university. However, our gearing requirements indicated that a three horsepower motor is insufficient to meet our power requirements, as horsepower ratings are usually higher than the actual power output from the motor. As such, a lower operating range would be available from a three horsepower motor. Given these circumstances we decided that a 5 horsepower motor at 1800RPM would be the best choice for our application, and decided upon the WEG 00518EP3E184T motor as the best choice due to cost and availability. Detailed specifications about the motor are available in Appendix K.

Motor Drive Selection

As mentioned in the motor selection, a motor drive or controller is required to start, stop and control the speeds of the motor. Based on recommendations by the building technician and motor drive suppliers, we have selected the Yaskawa CIMR-VUBA0018FAA as our motor drive. This motor drive is able to provide the full power for the motor, remotely start and stop the motor, and is programmable for various functions. Further details of the motor drive are shown in Appendix K.

Gear System

Our system will have a chain drive with 2 sets of sprockets with ratios of 3:1and 9:1. These ratios were calculated based on the motor specifications.

Gear Ratio selection for three horsepower motor

We did an analysis on the three horsepower motor to determine the feasibility of using it. From our discussions with our sponsor, we have determined that the system has to cover the full range of operating specifications for the Twist Compression test, while the Four-Ball test requires the system to be run at the maximum speed of 3600RPM, but with a reduced axial load due to the smaller motor. We were unable to obtain the full operating torque and speed for the motor, but chose a similar WEG 00318EP3E182T motor to estimate the required gear ratios. The motor torque and speed curves are shown below in Figure 26.

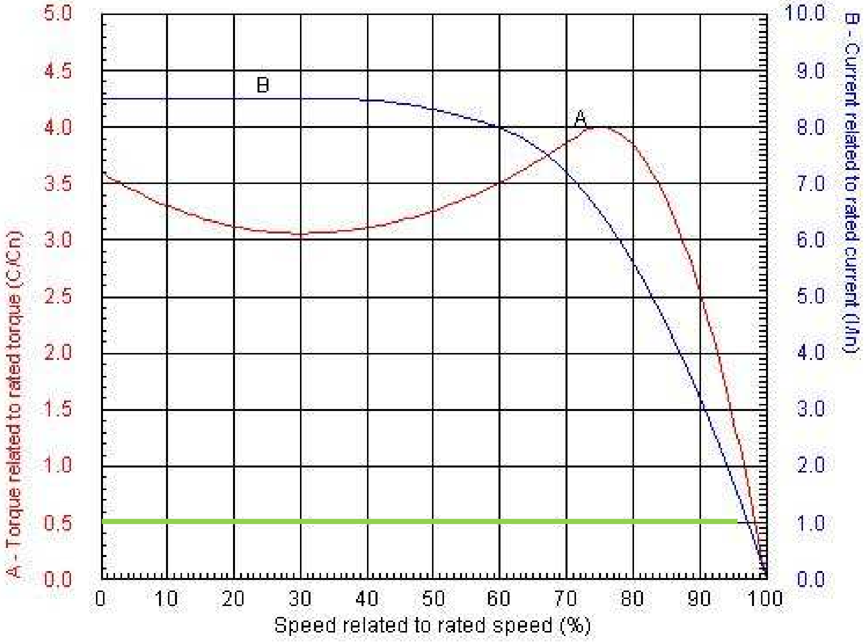

Figure 26: Torque vs Speed Curves for WEG 00318EP3E182T Motor

As shown in Figure 25, the motor is able to produce a torque above the maximum rated torque. However, this feature is meant to prevent the motor from failing, and the motor should not be run at a power above the green line to prevent a loss in the operating life of the motor. Thus, our gear ratio calculations were done using the full load torque and speeds of the motor, which is the area below the green line. These calculations are shown in Appendix C using the equations below.

$$\text{Power, } P = \text{Torque, } \tau \cdot \text{Speed, } \omega$$

$$\tau = F \cdot r \text{ where F is Force and r is radius of gear}$$

Force between gears are always equal,

$$\therefore \tau \propto r,\ \ \omega \propto \frac{1}{r}$$

Based on these calculations, we would require a max gear ratio of 15:1. However, sprockets are not easily available in these ratios. Choosing the three horsepower motor would increase the cost of power transmission, and reduce the operating capabilities of the system.

<u>Gear Ratio selection for five horsepower motor</u>

The gear ratios of the five horsepower motor were determined from the torque speed curves shown below in Figure 27. The detailed calculations are shown in Appendix C.

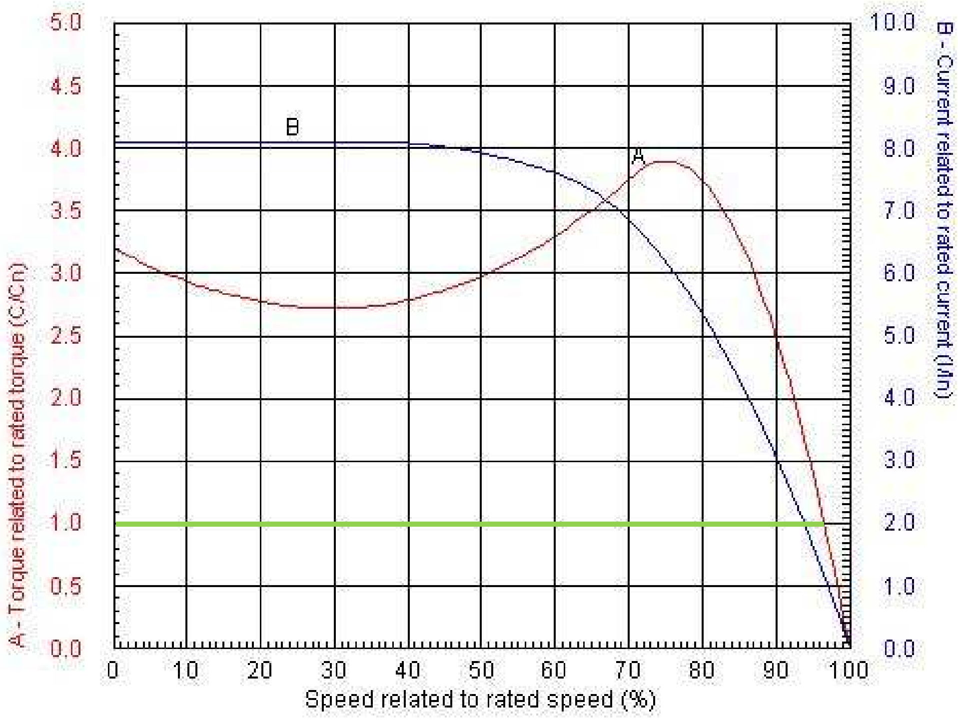

Figure 27: Torque vs Speed curves for WEG 00518EP3E184T motor

Based on these calculations, the maximum gear ratio required will be 8.6:1, and sprockets of this size are easily available.

Gear System Selection

Contrary to our Alpha Design, we decided to use a chain drive instead of a belt drive to transmit the power from the motor to the axles. This was due to the possibility of slip or wear in the timing belts. Sprockets and chains are made out of steel and will be more durable than the belt drive. To ensure that the sprocket teeth will be able to handle the loads from the motor, an FEA was conducted on the smaller sprocket. The maximum operating torque of 175 Nm was assumed to be exerted on only three spokes, and the inner bore of the sprocket was constrained to be fixed. As a conservative estimate, the material of the sprocket was assumed to be 1018 steel, although the actual sprocket is made of steel with hardened teeth. The constraints and results are shown below in Figure 28 and Figure 29.

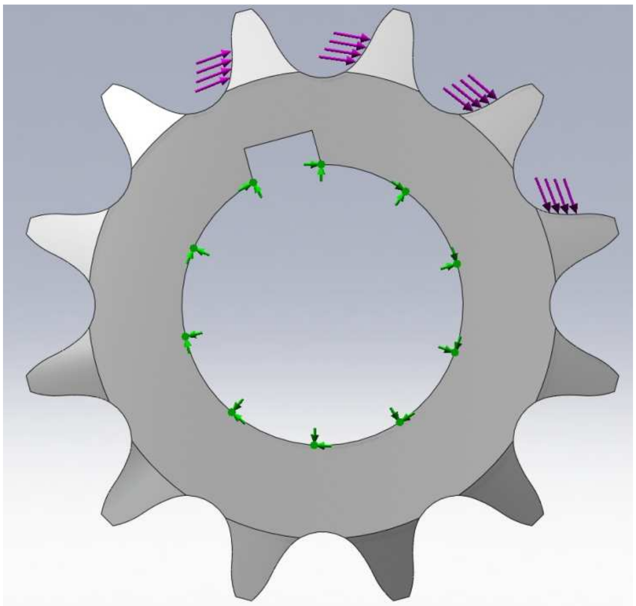

Figure 28: Load and constraints on sprocket

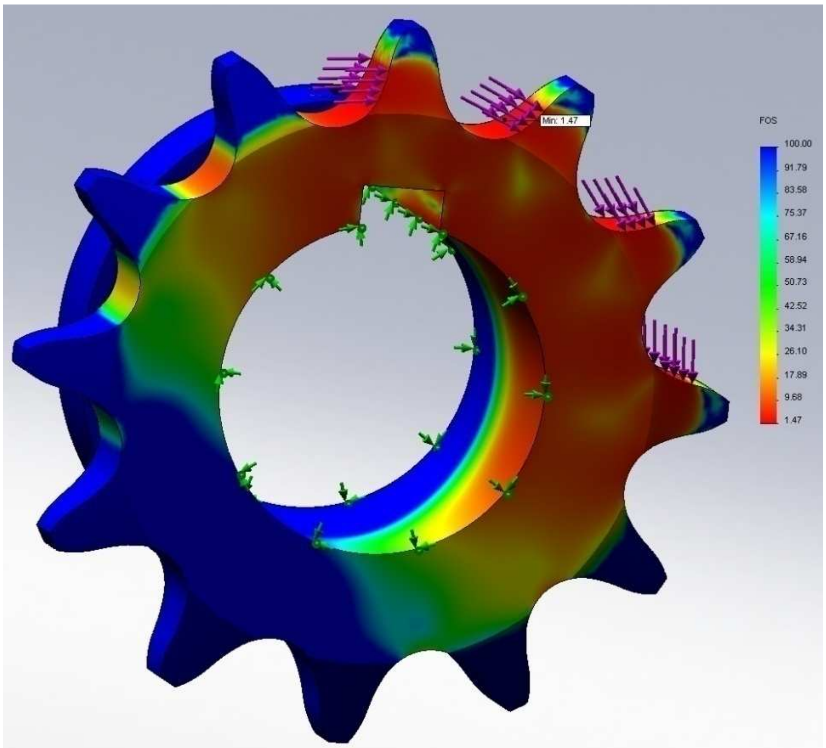

Figure 29: Factor of safety plot for sprocket stress test

Based on the maximum von Mises stress and yield strength of 1018 steel, the safety factor of the sprocket is 1.47.

Main Driving Mechanism Dimensions
The main driving mechanism refers to all axles involved in transmitting torque, from the gears to the test module. These axles have sizes that are dependent on the amount of stress that they will be subjected to, mainly from normal and torsion loads. Based on calculations, the maximum normal force and maximum torque are 120 kN and 175 N-m respectively, using the test conditions found in the commercial market. These loads come from the extreme testing range of the Twist Compression test. The equations for normal stress, σ, and shear stress, τ, are

$$\sigma = F/A \qquad \tau = T \times r/J,$$

where F = normal force, A = area of application, T = torque, r = radius, J = rotational moment of inertia.

After substituting in known variables, the normal and shear stress equations becomes,

$$\sigma = 120000/(\pi \times r^2) \qquad \tau = 350/(\pi \times r^3)$$

With the normal and shear stress, Mohr's circle can be applied to find the maximum stress, σ_{max}, in terms of the radius and height of the rod,

$$\sigma_{max} = \frac{\sigma}{2} + \sqrt{\left(\frac{\sigma}{2}\right)^2 + \tau^2}$$

The axles will most likely be made of different types of metal due to the differences in dimensions and market availability. However, when choosing the stock parts for these axles, special attention will given to ensure that the desired geometry and size can fit into the design and yet have a σ_{max} less than the yield strength of the material.

Four-Ball Module
The Four-Ball modular is inserted into the main driving shaft, so that the top ball for the Four-Ball test can be mounted on the shaft. This is illustrated in Figure 30 below.

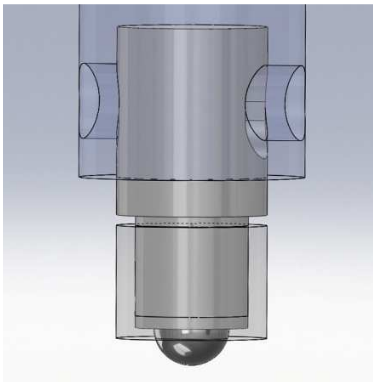

Figure 30: Four-Ball Module

The modular axle is made out of a solid rod, with a through hole through its diameter to hold it in its place. A securing pin will be inserted through this hole, but is not expected to experience high loads as most of the torque is expected to be transmitted though the contact area between the modular axle and the main shaft. A securing cap will be threaded below the modular axle to keep the top ball in place. The securing cap is not expected to take any loads. The outer diameter of the modular axle is designed to match the outer diameter (1in) of the Twist Compression annulus.

To verify the structural reliability, analysis was done using SolidWorks, illustrated in Figure 31 below.

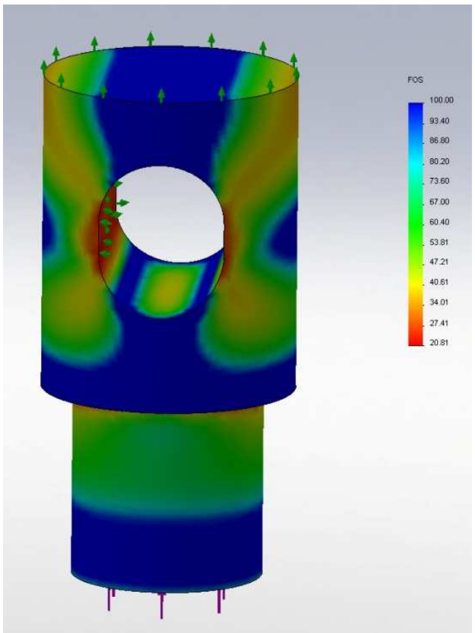

Figure 31: Factor of Safety of modular piece = 20

From our analysis, we have a safety factor of 20. This is not surprising due to the relatively low loads required in Four-Ball tests.

Test Contact Plate

The test cylinder and test balls used for testing are made of relatively high yield-strength AISI E 52100 steel and as such, methods have to be employed to ensure Aluminum in the test region does not yield during high loads. Figure 32 shows a schematic of the test region. In direct contact with the test cylinder or balls is a square plate made of AISI E 52100 steel, sitting in a square groove of the test cup. The square plate distributes the load to reduce the pressure on the test cup. The test cup is made of 6061-Aluminum T6 which has a yield strength of about 145 MPa. The steel square plate will be 1.5” (length) by 1.5” (width) 0.5” (thickness), resulting in a pressure of 62.5 MPa when operating at the maximum load of 100 kN which is a safety factor above 2.

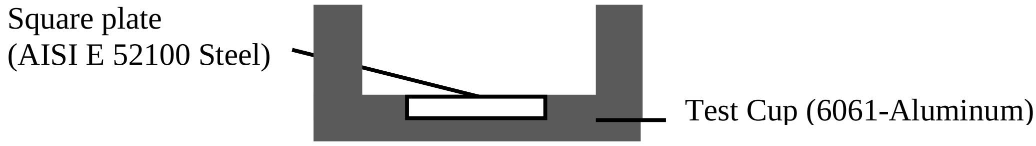

Figure 32: Schematic of Test Region

Test Cup Dimensions

The size of the Four-Ball test cup was determined by the dimensions of other components: the cup cap and the driving shaft and pin. This is illustrated in Figure 33 below.

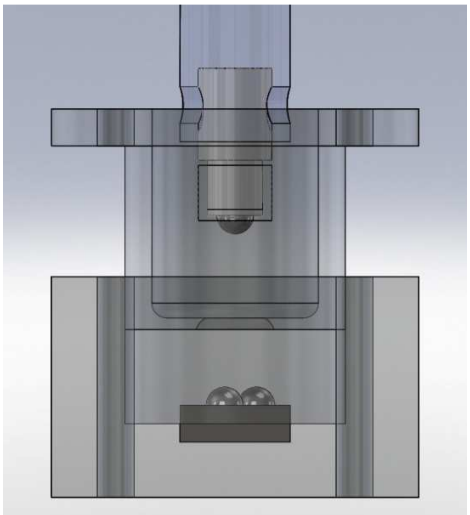

Figure 33: The dimensions of the test cup are determined by fitting requirements

The cup cap will have to fit into the cup, which in turn has to have an inner diameter wide enough to contain the main driving shaft and its securing pin (1.75”). Hence, for sufficient clearance, the cup cap is designed to have an inner diameter of 2.25”. Taking into account the thickness of the cup cap, the outer diameter of the cup cap as well as the inner diameter of the Four-Ball test cup was designed to be 3”.

The exterior of the cup was chosen to be a cube, so that heating elements such as Peltier devices can be easily mounted on the flat surface. There would be 4 sides of 5” by 3”.

For ease of integration, the Twist Compression test cup is designed to have similar dimensions as the Four-Ball test cup.

Test Cup Internal Temperature Control

Due to surface friction during testing, heat will be generated and hence, Peltier coolers are required to cool the test cup to maintain the desired operating temperature. Using the surface heat generation formula by Kaviany, Figure 34 shows the rate of heat generation based on the operating conditions. As it was calculated that the Twist Compression test will generate the most heat, only operating conditions of the Twist Compression test is shown.

$$Q = \mu \times F \times v,$$

where: Q is the rate of heat generation
μ is the coefficient of friction
F is the normal force and
v is the rubbing velocity

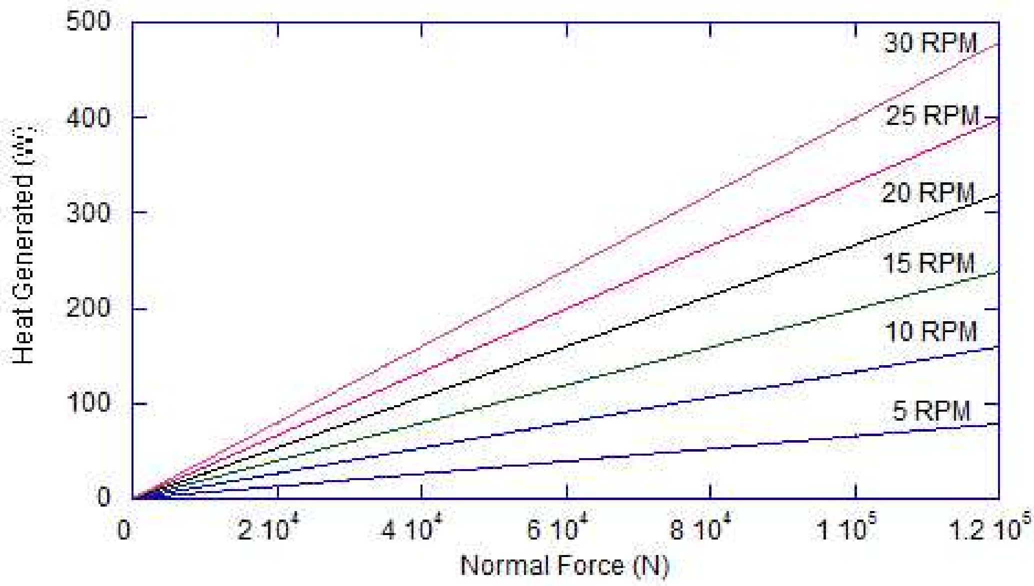

Figure 34: Rate of heat generation at different operating conditions

The proposed Peltier coolers are rated at 400W. Figure 35 is a typical graph for a 400W Peltier Cooler. Note that the original graph shows the performance at $T_h = 27^oC$ and hence, superimposed is a line that estimates the decrease in performance. The line represents the heat extraction at a temperature difference of 40^oC with $T_h = 40^oC$, with the use of the heat sink. Operating at maximum power, the rate of heat removal can reach 100W. To handle the maximum rate of heat generation of 500W, 5 Peltier coolers will be required.

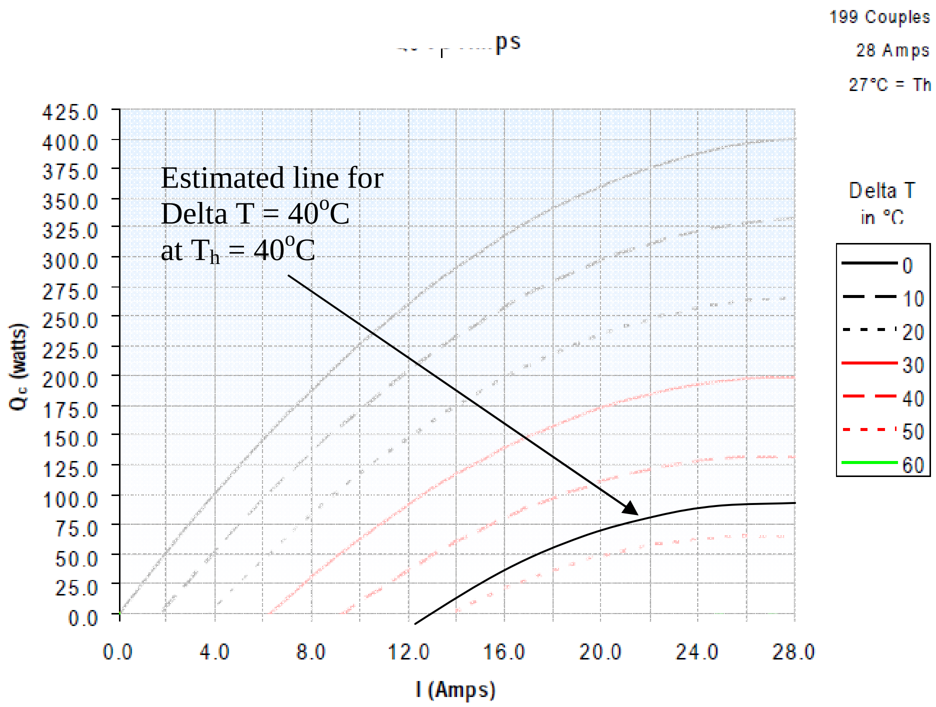

Figure 35: Performance curve of Peltier cooler.

Torque Measurement Rod

The torque rod is used to convert the torque experienced by the test specimen during the test into a force exerted on the cantilever beam that is mounted onto the H-bar. It has a secondary function of ensuring safety by preventing the test cup from spinning out of control during testing. This is a huge safety concern since the test specimen will be spinning at speeds of up to 3600RPM. Two torque rods will be installed, one on either side of the test cup. The torque rod has screw threads that are used to screw into the side of the test cup. The screw threads are ½-13. The diameter of the torque rod is 0.75 in. The length of each torque rod that extends out of the cup is 6 in. The material that the torque rod is made of is 4130 Alloy Steel. In designing the torque rod, we have to ensure that the rod is thick enough to withstand the shear forces that it will experience during testing. However, we cannot make the torque rod too big that it we will be difficult to install on the test cup. The CAD model of the torque rod is shown in Figure 36.

We did an analysis for the maximum stress that torque rod will be subjected to. The maximum stress is at the point of contact between the torque rod and test cup. The yield strength of 4130 Alloy Steel annealed is 46.4 ksi. The maximum yield strength that the torque rod will experienced is 1.98 ksi, which gives a safety factor of about 23.

$$\sigma = \frac{My}{I},\ I\text{: Moment of inertia } (\mathrm{m}^4)$$

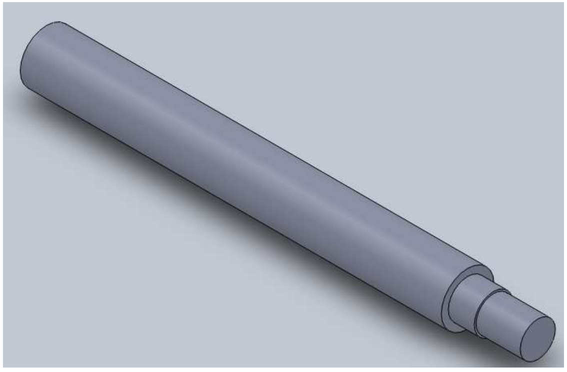

Figure 36: Torque rod

Data Acquisition Device (DAQ)

We will be using a NI USB-9237 DAQ to obtain the readings from the strain gages. The DAQ has a sampling rate of 50 kHz per channel, which exceeds the 20 kHz sampling rate required by our sponsor. The DAQ also comes with channel-to-channel isolation to ensure that readings from the DAQ device are not affected by difference in ground potentials or common-mode voltages. Signal conditioning is built-in so that the signals from the strain gages are amplified and the significance of noise can be negated. For obtaining temperature readings from the thermocouple, we will be using a USB-9211A. This DAQ features integrated signal conditioning, 250Vrms channel-to-earth ground isolation for safety, noise immunity, and high common-mode voltage range. In addition, there is cold-junction compensation (CJC) for the thermocouple to ensure accurate readings of temperature. These DAQs are bus-powered and have built-in excitation for the connected sensors.

For the optical encoder, a NI USB 6501 portable digital I/O device is used for reliable data acquisition and control of the digital signals at a low price.These DAQs are chosen to be connected to the computer by USB because of the accessibility and ease of use of USB ports. Furthermore, these DAQs are relatively portable and can be easily disconnected and used for other applications.

Normal Loading Force Measurement

We will measure strains on a load cell placed above the hydraulic jack as shown in Figure 37, and calculate the corresponding normal loading force. Equation 3 is derived using Hooke's Law and stress equation. We assumed that the material is homogeneous and isotropic; the force is exerted uniformly in the axial direction and stress and strain in the material occurs equally in all directions within the linear-elastic region.

The strain gages are installed in a Wheatstone bridge circuit with a half-bridge configuration, as shown in Figure 38. When the load cell material is strained, the resistance in the strain gage changes and causes a change in the bridge output voltage. Using the bridge output voltage, the known lead-wire resistance, nominal strain gage resistance, gage factor and Poisson's ratio of the load cell material, we are able to measure the strain in the load cell material.

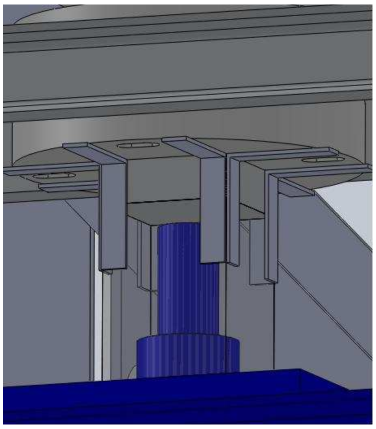

Figure 37: Illustration of Load Cell

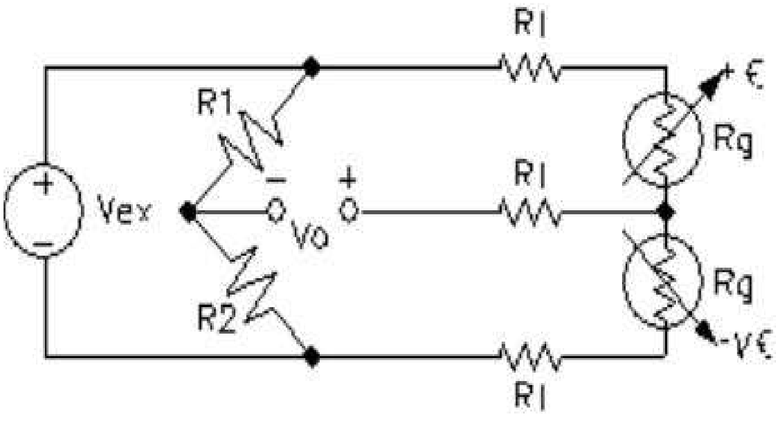

Figure 38: Wheatstone bridge circuit with half-bridge configuration for measuring uniaxial strain

$$\varepsilon = -\frac{4\left(\frac{V_{o,strained} - V_{o,unstrained}}{V_{ex}}\right)\left(1 + \frac{R_l}{R_g}\right)}{GF(1+\nu) - 2\left(\frac{V_{o,strained} - V_{o,unstrained}}{V_{ex}}\right)(\nu - 1)}$$

$$F = \varepsilon E A$$

σ: Average uniaxial normal stress at any point on the cross-sectional area (MPa)

E: Young's modulus (MPa)

ε: Average normal strain (m/m)

F: Normal loading force (N)

A: Cross-sectional area of the cantilever beam (m^2)

$V_{o,strained}$: Bridge output voltage when cantilever beam is loaded
$V_{o,unstrained}$: Bridge output voltage when cantilever beam is unload, or initial bridge offset
V_{ex} : Bridge excitation voltage
R_l : Lead-wire resistance
R_g : Nominal strain gage resistance
GF: Gage factor (sensitivity to strain)
ν: Poisson's ratio

<u>Data Processing of Normal Force Measurement</u>
Using the LabVIEW interface, we are able to obtain measurements of strain. With the measured strain, the known Young's modulus of the load cell material, the exposed surface area that the force is exerted on, we are able to calculate the normal loading force.

Torque Measurement
We will be measuring strains on a cantilever beam placed on the middle U-channels, and calculating the corresponding bending force exerted on the end of the beam by the torque rod. We assume that the material is homogeneous and isotropic, the force is exerted at a point on the cantilever beam and stress and strain in the material occurs within the linear-elastic region.
The strain gages are mounted on the load cell as shown in Figure 39 and Figure 40. The strain gages are installed in a Wheatstone bridge circuit, with a half-bridge configuration, as shown in Figure 41. When the load cell material is strained, the resistance in the strain gage changes and causes a change in the bridge output voltage. Using the bridge output voltage, the known lead-wire resistance, nominal strain gage resistance and gage factor, we are able to measure the strain in the load cell material.

The strain gages are mounted on the cantilever beam. The strain is measured using a Wheatstone bridge circuit, with a half-bridge configuration, as shown in Figure 41.

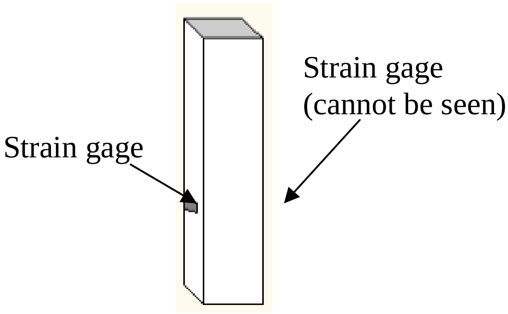

Figure 39: Initial State of Cantilever Beam

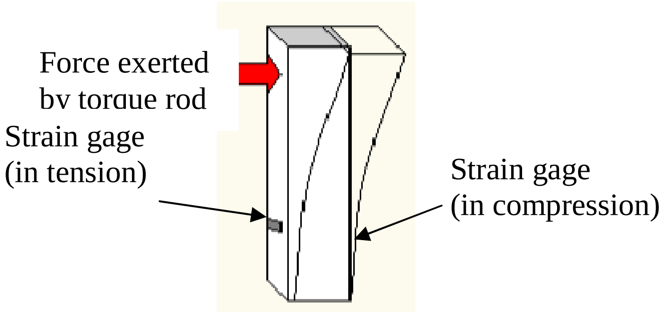

Figure 40: Cantilever beam with force exerted

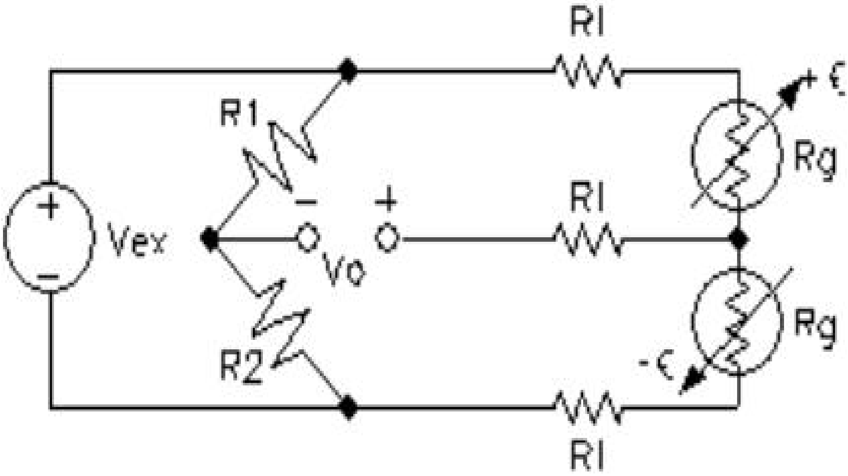

Figure 41: Wheatstone Bridge Circuit with Half Bridge Configuration for Measuring Bending Strain

Data Processing of Torque Measurement

Using the LabVIEW interface, we are able to obtain measurements of strain. With the measured strain, the known Young's modulus of the cantilever beam material, the width and thickness of the cantilever beam and the distance of the point of contact to the base of the cantilever beam, we are able to calculate the normal loading force.

ε: Strain (m/m)

P: Force exerted by torque rod on the cantilever beam (N)

D: Distance of the point of contact to the base of the cantilever beam (m)

E: Young's modulus (MPa)

w: Width of the cantilever beam (m)

t: Thickness of the cantilever beam (m)

$V_{o,strained}$: Bridge output voltage when cantilever beam is loaded

$V_{o,unstrained}$: Bridge output voltage when cantilever beam is unload, or initial bridge offset

V_{ex} : Bridge excitation voltage

R_l : Lead-wire resistance

R_g : Nominal strain gage resistance

GF: Gage factor (sensitivity to strain)

Test Cup Internal Temperature Measurement

We will be measuring the temperature inside the test cup by placing a thermocouple probe inside the test cup as shown in Figure 42. We will use a hollow tube thermocouple probe, which is a PFA insulated lead wire epoxy potted into a stainless steel sheath. It is a T calibration type which allows us to measure within a temperature range of -250°C to 350°C, which is well within the temperature range of 0°C to 100°C specified in our engineering specifications. The thermocouple has a 6 inch grounded probe, 1/8 inch diameter and 36 inches of PFA insulated 24AWG stranded wire. The length of the probe is sufficient for us to slot into our test cup during testing. The diameter is also small enough such that it does not impede the rotational motion of the modular test piece.

Figure 42: Thermocouple

Speed (RPM) Measurement

We will be measuring the transmitted speed by mounting a hollow shaft optical encoder on the drive shaft. The drawing of the optical encoder is shown in Figure 43. The optical encoder has a frequency response of 100 kHz, an operational temperature range of 0°C to 70°C and requires a voltage input of 5VDC. The encoder output is digital signals, and will be measured using the NI-USB 6501. In choosing the optical encoder, we have to ensure that the diameter of the shaft collar was big enough so that it can be mounted on our drive shaft. We also have to ensure that the optical encoder is heavy duty so that it can withstand the high torques transmitted through our drive shaft.

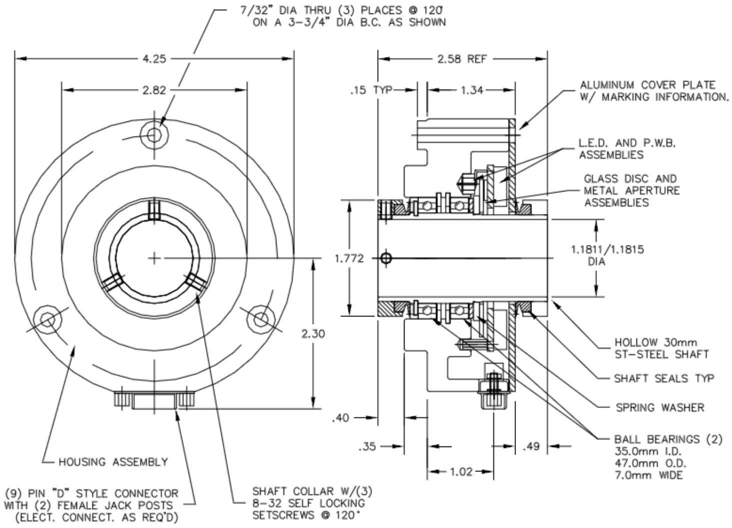

Figure 43: Drawing of the Incremental Optical Encoder

Protective Safety Shield

The safety shield is needed to mainly prevent users from coming into contact with moving parts and potential pinch points during operation. However, due to the unpredictability of nature, secondary use of the safety shield is to also act as a container to prevent any parts of the machine from flying outwards towards the user. As the machine will be tested thoroughly within the confines of a safety test cell during the safety validation (which is discussed later), the machine is not expected to fail and as such, the safety shield is a precautionary measure. The most probable material of choice is Tuffak A© Polycarbonate, which has high impact strength. It is also transparent and this allows the user to keep track of the machine's operation visually. As the polycarbonate is also a rubber-based material, it is not brittle and will not shatter.

A small box will be assembled around the cup area, shown below.

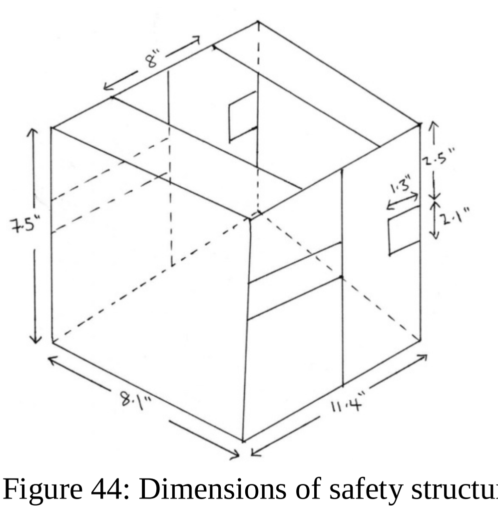

Figure 44: Dimensions of safety structure

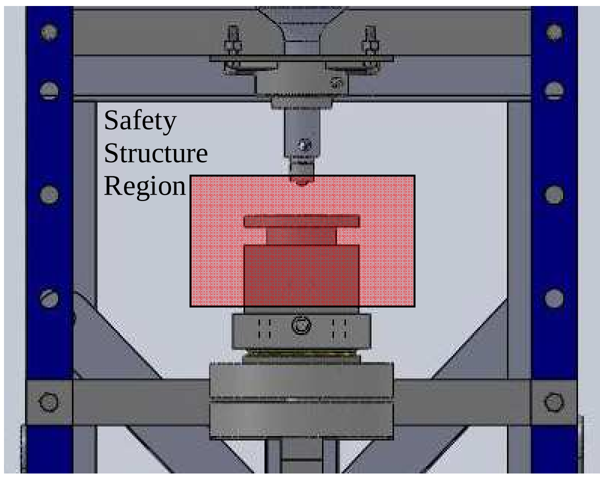

Figure 45: Area enclosed by safety structure

The safety structure will be made out of polycarbonate sheets of thickness 0.25", and will be joined together using brackets.

Motor Structure
Our tentative selection for the motor structure is a Little Giant 1500-lb capacity machine table. The motor weight is small compared to the rated capacity of the table, and the table should be able to take its full weight and loads during operation. To account for vibrational issues, we will have connection beams between the motor structure and main structure. In addition, neoprene pads will be placed under and at the mounting feet of the motor to absorb any vibrations. The motor structure has dimensions of 48” (width), 24” (depth) and 42” (height) is shown below in Figure 46.

Figure 46: Motor Structure

FINAL DESIGN DESCRIPTION
As explained in the introduction paragraph of the Parameter Analysis section, our prototype will be the full size model of our actual design. This section describes the operation of the machine. As the complete final design is lengthy in description, to aid the flow of information, the description will be in order of contact, starting from the motor. Figure 47 renders the machine while Figure 48 provides a schematic of the order of description.

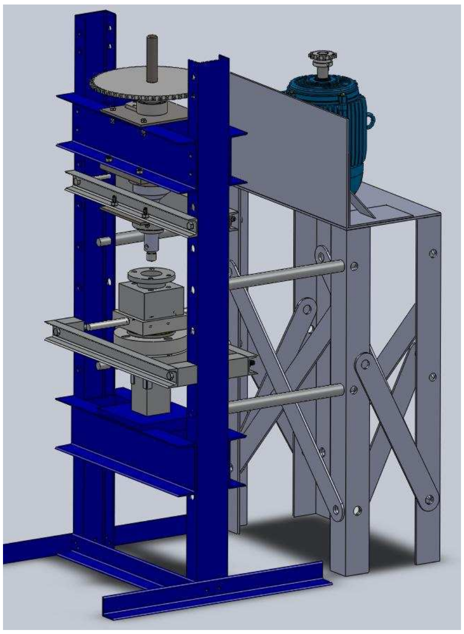

Figure 47: Illustration of Overall Machine

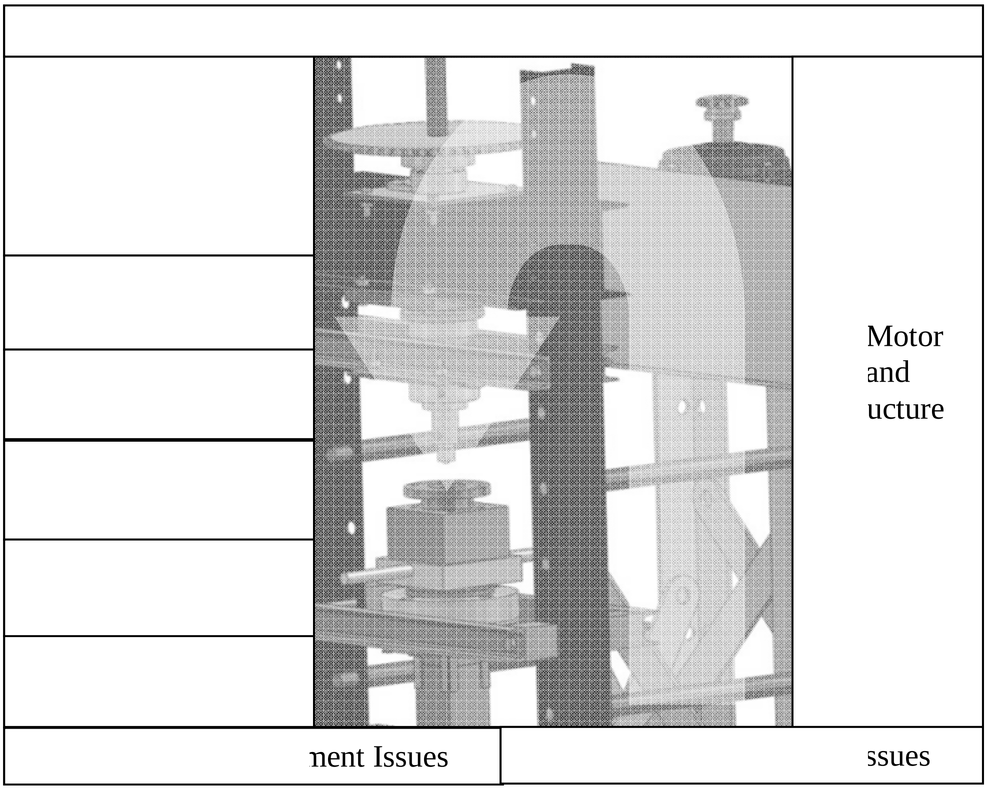

Figure 48: Schematic of Description Order

Motor and Structure

The motor will be mounted onto an industrial shelf which we will purchase. The motor will be bolted onto the industrial shelf, with its axle pointing vertically upwards. The upward direction was chosen to have the motor axle as close to the top of the main structure as possible to reduce the complications in power transmission, and reduce the potential number of pinch points compared to a system with the power being transmitted from the ground level. Because the sprockets have a bore size of 1 inch, a axle reduction piece will be installed on the motor axle to accommodate the sprockets. The motor operation will then be controlled by a motor drive. Figure 49 shows the motor and a representation of the structure.

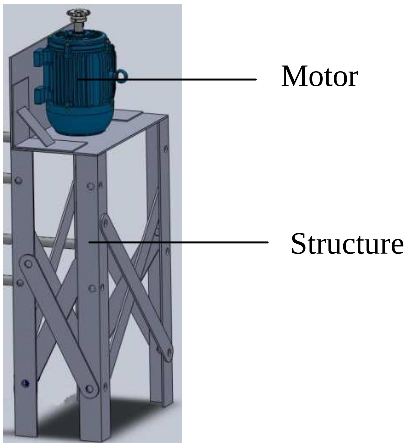

Figure 49: Illustration of Motor and Structure

Gear System

The gear system involves two sprockets, one connected directly to the motor, while another connected to the driving axle of the machine. The two sprockets are linked by a roller chain. Figure 50 illustrates how the gear system looks like.

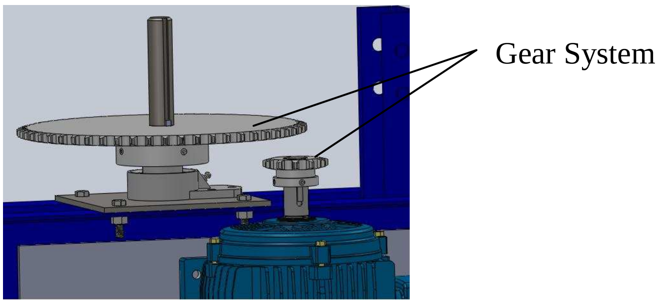

Figure 50: Illustration of Gear System

Rotational System

One of the sprockets in the gear system connects to the driving axle through a keyed gear shaft. The driving axle then transmits the rotational power to the test pieces. To measure the speed of rotation, a hollow shaft incremental optical encoder will be used and set screwed onto the gear shaft. As the machine will be loaded with a force from the bottom, a thrust bearing is used to connect the axle body to a top bar mounted on to the main structure. The thrust bearing enables the axle to spin, while providing a normal contact force to hold the axle stable. To stabilize and align the lower axle body, it is fitted through a flange-mounted roller bearing placed at the centre of a horizontal bar mounted onto the main structure. This reduces the amount of bending faced by the axle body during high speed rotations. Figure 51 provides an illustration of the rotational system.

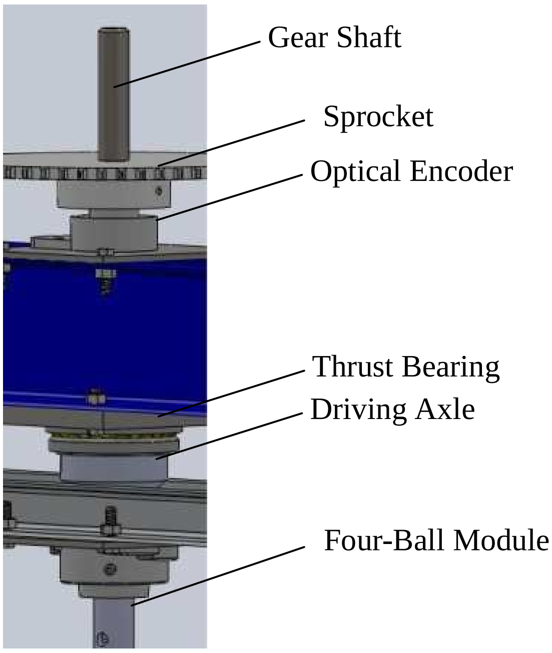

Figure 51: Illustration of Main Driving Axle.

Upper Test Region

The upper test region comprises of the modular design that allows both the Twist Compression test and Four-Ball test to be conducted on the same machine. Essentially, either the Twist Compression Cylinder or the Four-Ball upper ball will be connected directly to the axle body. In the case of the Twist Compression test, the Twist Compression annulus will be placed inside a groove located at the base of the main driving axle body. This annulus is then attached to the axle by a bolt. The holes located at the axle will be elongated in the vertical direction. This is so that during actual testing, the compressive forces are not transmitted to the bolt, which is unable to withstand high loadings. However, it is important to note that the bolt serves as the main turning mechanism for the Twist Compression annulus. Figure 52 gives an illustration of the Twist Compression setup.

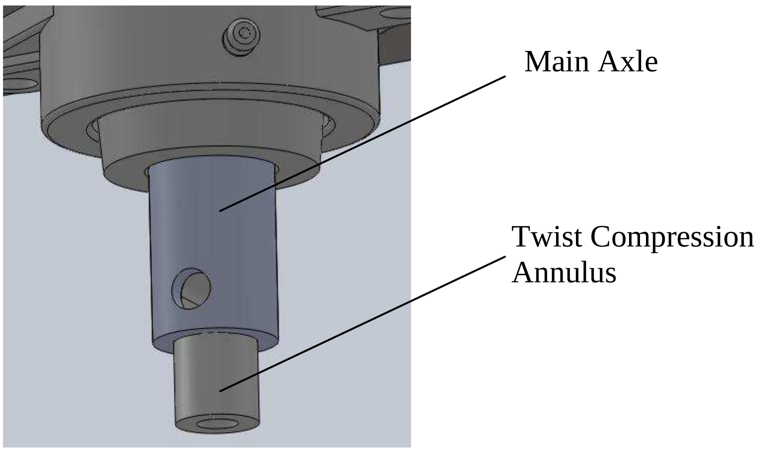

Figure 52: Illustration of Twist Compression Module.

In the case of the Four-Ball test, the upper ball is attached to a modular piece through a screw cap. This modular piece connects to the driving axle through a bolt, similar to the Twist Compression annulus mounting method. The steel cylinder was chosen mainly for its high coefficient of friction with the test balls. For the Four-Ball test, the turning mechanism for the upper ball is solely through friction. As this point is always dry, the friction here is greater than the friction within the test cup which is filled with lubricant, hence allowing the upper ball to spin. Figure 53 provides an illustration of the Four-Ball configuration.

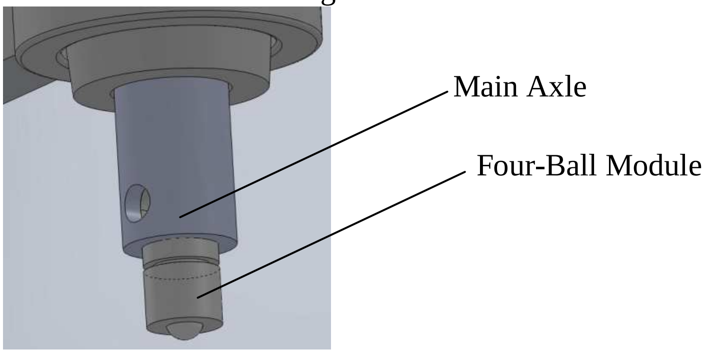

Figure 53: Illustration of the Four-Ball Module.

Test Cup

The test cup is used to hold the surfaces on which the test pieces will spin against. Within the test cup is a square groove that will house the square test plate. There are two kinds of square plates. For the Twist Compression test, it will just be flat plate and for the Four-Ball test, 3 grooves will be cut into the plate to sit the bottom balls. The dimension of the square plate distributes the load, so as to prevent the test cup from yielding. For the Twist Compression test, the design is straightforward as all that is required is for the upper Twist Compression annulus to be pressed against the flat square plate. However in the Four-Ball case, the bottom 3 balls have to be fastened down by a cap that is bolted onto the test cup. The test cup will also be fitted with 5 Peltier coolers on the external walls. These thermoelectric coolers will enable temperature within the test cup to be controlled. Heat sinks will be placed on the hot side of the thermoelectric coolers, to reduce the temperature difference and hence increase the rate of heat extracted. Also, a thermocouple will be placed within the test cup so that the temperature may be monitored. Figure 54 shows the lower test region assembly.

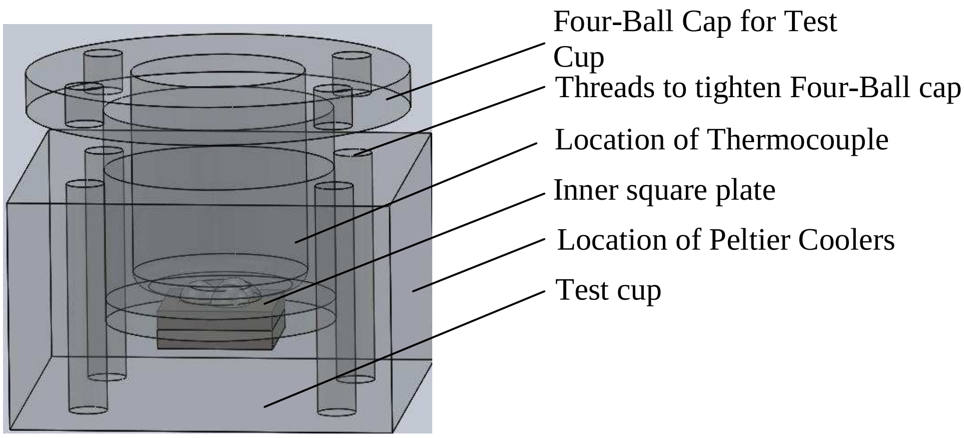

Figure 54: Lower Test Region Assembly

Torque Measurement

The torque measurement region involves rods extending from opposites sides of the cup plate, which is where the test cup sits on. Beneath the base plate is a thrust bearing that is placed on an alignment plate assembly, which will be discussed later. This thrust bearing only needs to reduce friction effects and allow the base plate and test cup to rotate slightly during testing as a result of the torque generated by the test. The extruded rods will then turn along with the base plate, until it presses against a cantilever beam. The cantilever beam is located away from the cup and has strain gauges attached on opposite sides of the beam. The force exerted by the rod onto the cantilever beam, causes the beam to strain in tension on one side and compression on the other. The strain gages measure the strain and allow the force and thus, torque to be measured. Figure 55 illustrates more clearly how the torque measurement region appears to be.

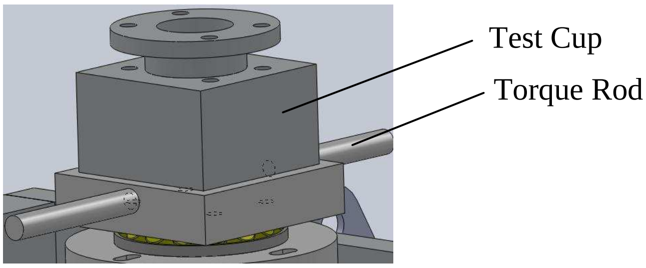

Figure 55: Torque Measurement Region

Alignment System

The alignment system serves two functions – translational alignment during the Four-Ball test and angular alignment during the Twist Compression test. The alignment plate comprises of two plates with spherical grooves located at the centre between the plates. During the Four-Ball test, the alignment plates sit flat on top of each other, allowing horizontal translation to ensure that the four balls in the Four-Ball test are all in contact during testing. The two alignment plates have perpendicular slots, which allow bolts to be tightened once an optimum position is found through translation in the horizontal plane. Figure 56 illustrates the alignment plate for the case of the Four-Ball test.

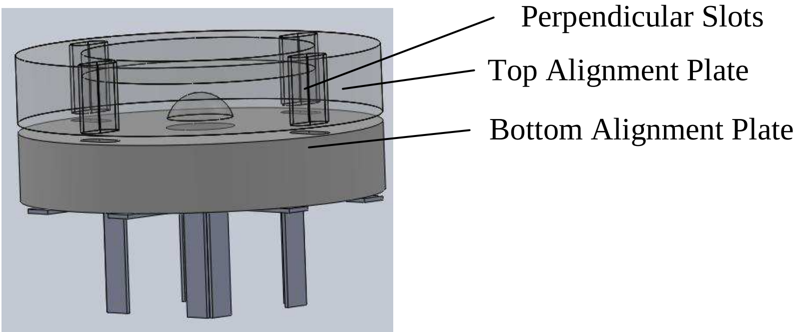

Figure 56: Alignment Plate During Four-Ball Test

During the Twist Compression test, the alignment plate will be fitted with a ball in the center, between the plates. The ball now allows the top plate to tilt at an angle, so that the Twist Compression cylinder may press perpendicularly against the flat square plate described previously. The slots used in the case of the Four-Ball test will not be used here. Figure 57 shows the alignment plate in the Twist Compression test scenario.

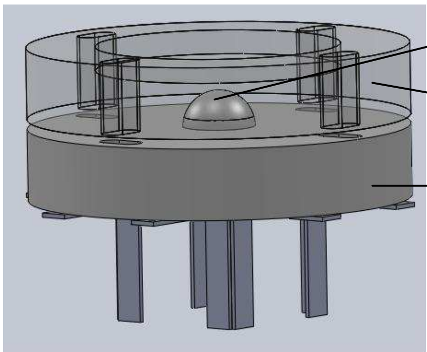

Figure 57: Alignment Plate During Twist Compression Test

Hydraulic Press Region

The alignment system discussed previously sits on the hydraulic press, augmented with a mechanical guiding system and force measurement system. Eight angled brackets are attached to the bottom alignment plate to guide the square head modification that sits on the hydraulic press. The brackets form the four corners around the square modification piece that is fitted around the hydraulic press. The brackets act as a guiding mechanism to prevent the tip of the hydraulic press from pushing the force sensor and alignment plate at a misalignment. In between the alignment plate and the square head modification is a load cell we plan to use to measure the axial load. The load cell comprises of a aluminum block with strain gauges mounted to its sides. During operation, the hydraulic tip will push against square head modification, onto the force sensor, onto the alignment plate. Figure 58 shows the hydraulic press region.

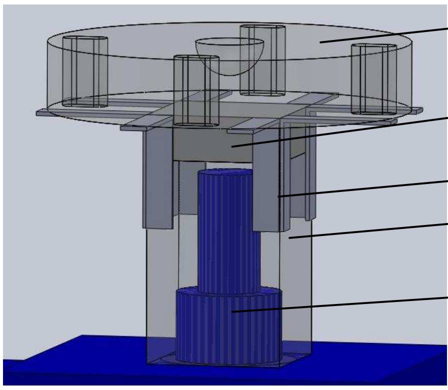

Figure 58: Illustration of Hydraulic Press Region

Auxiliary Attachment Issues

This section covers how certain parts of the machine described above are attached to the machine. Figure 59 illustrates the other parts of the machine and describes the other attachments that are required in the machine.

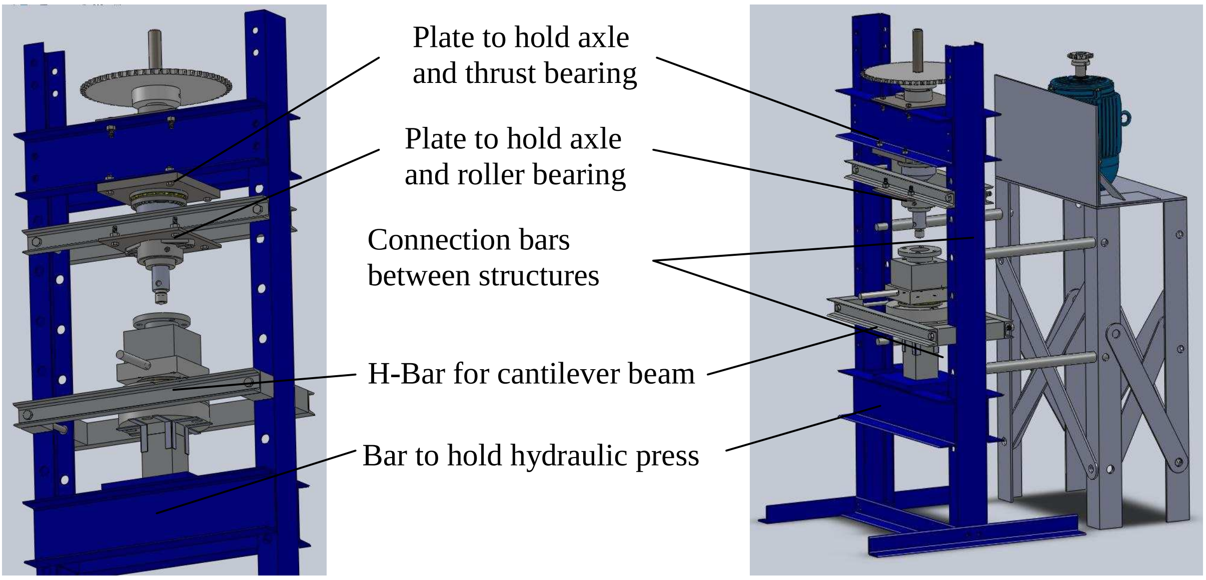

Figure 59: Illustration of Auxiliary Attachments

Power and Electrical Issues

The section covers the power and electrical requirements of the machine. The areas that will be discussed are a) motor power requirements, b) power supply to the Peltier Coolers and c) how the thermocouple and strain gauges will be attached to an external computer for measurement.

<u>Motor Power Requirements</u>

The motor (WEG – 00518EP3E184T) requires a three phase AC power input, 230V, 60 Hz and up till 13.6A. Currently this is limited by logistical and infrastructural issues as the current available power supply is single phase. As such, the motor power is still subjected to changes.

<u>Power Supply to Peltier Coolers</u>

The power rating of the Peltier coolers chosen is about 336W and hence a DC power supply of 12V, 1680W will be required to operate 5 of these coolers needed to cool the test cup to achieve 0 to 100^{o}C test temperature range. Tentatively one possible solution is to use the DS2000-3 Distributed Power System that is able to provide an output of 2000W, 12V DC power. However this is subjected to further approval and changes.

<u>Connecting the Optical Encoder, Thermocouple and Strain gauges</u>

The optical encoder (Sensor Systems, Model VOH42) will be connected to a DAQ (NI USB-6501) while the thermocouple (T-calibration Hollow Shaft Probe) and strain gauges (Foil gauges in Wheatstone Bridge Configuration) will be connected to another DAQ (NI USB-9237) via wires. The wires will then be screwed and fastened to the DAQ. The DAQ is in turn connected to a LabVIEW© enabled computer.

Cost

The total cost of our system was determined to be $7234.04. This includes all components. However, this value may change pending the finalization of our parts. In addition, we may be sending some parts for external fabrication, and this may increase the costs further. The detailed cost breakdown is in Appendix O.

FINAL MODIFICATIONS TO DESIGN

Several modifications were made to our final design. These included the motor and motor drive, sprockets, the trust bearing and material of the load cell.

Motor and Motor Drive

The motor selection has been changed from a 5 Hp WEG 00518EP3E184T motor to a 3 Hp A.O. Smith E265M motor. This reduction in power is due to the reduction in the area of the twist compression cylinder that thus requires a lower gear ratio for the twist compression test.

The original 5Hp motor was selected, as the original loads required a gear ratio that would be too large to be used with a 3Hp motor without complications to the design. This smaller gear ratio makes it possible to find a set of gears that will generate the right torques and speeds that are required by the test. Although the limiting factor for the power required is from the four-ball test, the sponsor is willing to run the system at a lower load but at the maximum speed of 3600 RPM. Calculations of the new loads are presented in Appendix Q-1.

The motor drive was also changed accordingly to suit our new motor. As such, an ABB ACS-150-01U-09A8-2 was selected to run the motor. This drive runs on a single phase 208-230V power supply, and is able to supply a three phase constant torque 3Hp power supply to the motor.

Sprockets

As shown in Appendix Q-2 the sprocket ratios would be changed to accommodate for the changes in the load requirements and motor capability. The gear ratios for the Twist Compression test are now 9.16:1, and the rear ratios for the Four-Ball Test are now 2.5:1. The actual sprockets chosen now have a gear ratio of 93.33:1 and 2.33:1.

Thrust Bearing

The lower thrust bearing was changed to a thrust bearing with a lower load rating since it does not require the same speeds as the bearing on the driving axle. This also helped to reduce costs by about $300.

Material of Load Cell and Cantilever Beam

Due to the lower loads that will be exerted on our system, the original choice of aluminum was not suitable as it could not produce the right magnitude of strain that could be detected by our electronic system. Thus a glass filled polycarbonate was chosen as a replacement material.

INITIAL FABRICATION PLAN

Our manufacturing process involves many components and processes. The breakdown of components and manufacturing operations are briefly summarized in Table 26 below. The Drawings, Materials, Machining operations and conditions of each of the 22 parts (excluding the U Channels and the shop press structure) to manufacture are shown in Appendix M.

Part	**#**	**Machines**
Plates	3	Drill, Lathe
Shaft	4	Mill, Lathe, Broaching Tool, Tap
Discs	3	Mill, Lathe, Tap
Blocks	12	Mill, Lathe, Tap, Drill, Band Saw
U Channels (Al)	4	Drill
Structure (Steel)	7	Drill

Table 26: Summary of Components

Machine	**# of operations**
Drill	66
Lathe	16
Mill	86
Tap	22
Broaching Tool	2
Band Saw	8

Table 27: Summary of Manufacturing Operations

Determination of Machining Process

<u>Drill</u>

A drill will be used to drill holes that do not require a high precision for its hole location. This is selected so that less machining time will be required at the mill, which we expect to have limited access to. As the main structure is bulky and hence hard to mount onto the mill, the drill would be used to drill the additional required bolt holes. Holes on the main structure, U-channels, and upper mounting plates will be created using the drill.

<u>Lathe</u>

The lathe is used when high precision is required for smooth, circular dimensions. Hence, it is used to manufacture the shape of the main shaft, indents on the plates for the thrust bearings and the cavity of the Four-Ball and Twist Compression test cups. Large discs, plates and blocks will first have to be mounted on a face plate, which would in turn be mounted onto the lathe. The lathe will also be used to manufacture external screw threads.

Mill
The mill is used to machine simple dimensions of the components, as well as to drill holes which have to be precise. The mill will also be used to flatten critical surfaces for angular precision of the components.

Tap
A tap will be used to create internal screw threads.

Band Saw
A band saw will be used to cut the stock materials to its required dimensions. This is mainly used for the 4 spacers as well as the 4 cantilever beam holders.

Determination of Tooling and Cutting Speeds
Different materials are selected for our components due to the different amount of loading and stresses each component is expected to experience. As a result, different tools will be required to manufacture each component to account for the differing hardness and manufacturability of the stock materials. The general choice of tools is shown below in Table 28.

Material	**Tool**
Al 6061, Steel 1018, 1020	High Speed Carbon
Steel O1, 52100, 4140	Carbide

Table 28: Tools Material Selection

The operational machining RPM speeds are calculated based on the recommended cutting speeds of the material, shown in Equation 1 below.

$$RPM = \frac{Cutting\ Speed}{\pi(\max Diameter)}$$

The cutting speeds used for the calculations are tabulated in Table 29.

Machine	Material	Tool	Cutting Speed (FPM)
Lathe	Al 6061	HSS	500
	Steel 1018, 1020	HSS	125
	Steel O1, 52100	Carbide	590
	Steel 4140	Carbide	430
Mill	Al 6061	HSS	165
	Steel 1018, 1020	HSS	65
	Steel O1, 52100	Carbide	50
	Steel 4140	Carbide	35

Table 29: Material Cutting Speeds

ASSEMBLY METHODS

Different assembly and fastening methods are employed in our design. The specific chosen method of assembling each component together is easier presented in the CAD drawings. The rationale behind the selection of the different assembly methods is elaborated below.

Press Fit

The press fit methods are used for components that have to be sturdy and will not have to be removed. This applies to the bottom inverted cup that fits into the punch of the hydraulic press, as well as the upper plate that holds the thrust bearing.

Snug Fit

Snug fits are chosen for components that have to be removed frequently but does not bear any safety consequences for not being fastened. The top alignment plate and the cup plate are designed to fit snugly into the thrust bearing without being overly tight. This is done because these plates have to be frequently removed by the user to change between the Four-Ball and Twist Compression tests, and to change operating conditions.

Bolts in Through-Holes

Bolts in through-holes are used when the tightness of the bolted plates are important for the safety and operation of the machine. This is chosen as a wrench and nut tightening system would be easier to tighten as compared to a screw system. In addition, the strength of the bolts and nuts will not be dependent on screw threads which are susceptible to wear.

Screw in Screw Threads

Screw and screw threads are chosen when the tightness is not a crucial consideration. It is also used when the user does not have access to the opposite end of the screw, which makes it harder to implement a bolt-tightening system.

VALIDATION

There are 2 types of validation tests that will be conducted, namely 1) the Safety Validation and 2) the Results Validation. The Safety Validation validates that that machine can be operated safely, while the Results Validation validates that the engineering specifications were achieved.

Safety Validation

This section covers the procedure that will be taken to test the safety of the operation of the machine. The test of operation will not test the correctness of the machine, but rather that it works and is safe under prolonged periods of operation. The main approach is to run the machine starting from the motor only, and thereafter add in more components test after test, until the whole machine is assembled. Thereafter, the limits of the loads will tested. Table 30 summarizes the testing procedure.

Team Members	Type of Test	Speed	Load	Duration of Test
A & B	Motor on Structure only	3600 RPM	NA	1 Hr
B & C	Add Test Machine (Top rotating assembly only)	60, 500, 1000, 2000 RPM	NA	20 Min
B, C & D	Add Test Machine (Top rotating assembly only)	3600 RPM	NA	1 Hr
D & E	Add bottom test region assembly (Twist Compression Configuration)	30 RPM	1kN, 10kN	1 Hr
E & A	Add bottom test region assembly (Twist Compression Configuration)	30 RPM	100kN	1 Hr
A & B	Add bottom test region assembly (Four-Ball Configuration)	3600 RPM	1kN, 10kN	1 Hr
B & C	Add bottom test region assembly (Four-Ball Configuration)	3600 RPM	100kN	1 Hr

Table 30: Summary of Test Schedule

<u>Test Logistics</u>

The venue of testing will be test cells located at the FXB and Auto Lab. These test labs are used to test engines and are certified safe. As each test is expected to last for at least 1 hour, team members will take turns to supervise the test proceedings. As the team has 5 members, each test will be conducted by 2 team members and after every 2 hours, one of the team members will be replaced by another. This ensures that at any time, there will be one team member who was present in the previous test to help initiate the next test.

<u>Test 1: Motor on structure only</u>

The first test will involve mounting the motor onto the structure, with the motor running at 3600 RPM for 1 hour. The 1 hour limit is taken with reference to ASTM Test D2266 and D4172 which requires the tests to run for 60 ± 1 minutes. The rationale of the test is to check on the stability of the structure as the motor will be vibrating at an elevated height while being bolted to the structure, which is in turn bolted to the ground. The foreseen weakest link of the structure is the joint between the ground and the motor structure.

<u>Test 2: Add test machine, but with top rotating assembly only.</u>

This test will entail the top rotating assembly being driven by the motor at 60, 500, 1000 and 2000 RPM, each for 5 minutes, before running at 3600 RPM for 1 hour. No vertical force will be applied. The rotating assembly will essentially be connected to the motor via the top gear sprockets and chain drive. The gradual increment in speed is to allow identification of problems with damaging the machine. Problems might arise in how the gear grips to the chain, possibly causing a skip in a gear tooth. Finally, after asserting that the chain drive is able to run smoothly, the 3600 RPM test identifies any weak points in the machine that might fail.

Test 3: Add bottom test region assembly (Twist Compression Configuration)
This test will involve the Twist Compression configuration to be used. Loads of 1kN, 10kN will be applied and run for 30 minutes each, before applying a load of 100kN for 1 hour. A speed of 30 RPM will be used. The 1kN and 10kN test preliminarily determines the safety of the machine, as they allow major points of failure to manifest, in the absence of the maximum load that could damage the machine. The last 1 hour, 100kN test is to ensure that the machine will be safe under the maximum load and speed for the Twist Compression test.

Test 4: Add bottom test region assembly (Four-Ball Configuration)
This test involves the Four-Ball configuration to be used. Loads of 1kN, 10kN will be applied and run for 1 hour each, before applying a load of 100kN for 1 hour. A speed of 3600 RPM will be used. Similar to Test 3, the 1kN and 10kN tests allows failure, if any, to occur in the absence of high loads. However, due to the higher speed of the test (3600 RPM), the 1 hour duration is chosen to better ensure the safety of the machine. The final cornerstone test, will involve the machine operating at 3600 RPM at maximum load of 100kN. This final test tests the limits that the machine was built to achieve and will conclude the safety test.

Results Validation
The target engineering specifications are listed in Table 31. Table 31 also summarizes the validation method for each specification. With the exception of dimensional issues, which are straightforward, the other issues will be discussed below.

<table>
<tr><th colspan="3">Loading Issues</th></tr>
<tr><td>Loading Range</td><td>Up to 91,700N</td><td rowspan="2">To use ME 395 Load Cells</td></tr>
<tr><td>Precision</td><td>± 1%</td></tr>
<tr><th colspan="3">Temperature Issues</th></tr>
<tr><td>Temperature Range</td><td>0^oC to 100^oC</td><td rowspan="3">Self-experimentation with liquid-in-glass thermometer</td></tr>
<tr><td>Precision</td><td>± 3%</td></tr>
<tr><td>Fluctuation</td><td>± 5^oC</td></tr>
<tr><th colspan="3">Dimension Issues</th></tr>
<tr><td>Size of machine</td><td>3 ft by 3 ft by 4 ft</td><td rowspan="4">Typical Measurements</td></tr>
<tr><td>Outer diameter of cylinder</td><td>1 in</td></tr>
<tr><td>Cylinder wall thickness</td><td>0.05 in</td></tr>
<tr><td>Diameter of balls</td><td>0.5 in</td></tr>
<tr><th colspan="3">Speed Issues</th></tr>
<tr><td>RPM Range</td><td>Up to 3600 RPM</td><td rowspan="2">Strobe light/ Tachometer</td></tr>
<tr><td>RPM Measurement Precision</td><td>± 2%</td></tr>
<tr><th colspan="3">Alignment Issues</th></tr>
<tr><td>Angular alignment range</td><td>± 0.25^o</td><td rowspan="2">Digital angle gauge</td></tr>
<tr><td>Translational alignment range</td><td>± 1 cm</td></tr>
</table>

Table 31: Summary of Verification Method of Specifications

Loading Issues

The only way to verify that the strain gages have been installed and calibrated correctly is to check with another load cell. As such, load cells used in ME 395 classes will be used to verify the strain gages. With the load cell, the measurement metal will be compressed. Small load differences will also be tested to test the sensitivity of the strain gages. The range of forces will also range from 0 to 100 kN.

Temperature Issues

The thermocouple will be tested by self-experimentation. A simple water bath and liquid-in-glass thermometer will be used. From the boiling temperature of water (100^{o}C), the water will be allowed to cool to room temperature. During the cooling process, the liquid-in-glass thermometer will be used to record the temperature. The results will then be compared to the digital data collected using the thermocouple via LabVIEW. Ice will also be used to test the temperature from 0^{o}C to room temperature.

Speed Issues

There are two ways to test the RPM speed. One way is to use a strobe light of variable frequency. After marking a spot on the rotating axle and running the machine, the strobe light will be activated and its frequency adjusted until the spot appears to be stationary. An out of phase frequency will result in the spot to appear like it is moving. Should a strobe light be unavailable, a light emitting device connected to a variable power supply, which are abundant within the Electrical Engineering Department, could also be used. Alternatively, a non-contact tachometer could be purchased or borrowed.

Alignment Issues

To test the angular alignment, a digital angle gauge could be used. A digital angle gauge is fairly inexpensive and could also be borrowed from within the Mechanical Engineering Department.

LOGISTICAL AND SPECIAL CHALLENGES

There will be certain obstacles and challenges as our project progresses, most of which associated with the manufacturing phase. This section serves to describe the challenges we foresee and how we plan to overcome them.

Challenges in Motor and Power Transmission

Motor Selection

There is a possibility that the motor may not perform ideally due to unexpected and unforeseen frictional losses. However, even in the event that this happens, the only consequence is that the machine will only be able to achieve a fraction of its desired operating range. A check with the sponsor will have to be conducted to assess the necessity to make improvements as the sponsor might still be satisfied with a small decrease in operating range. Should improvements be necessary, an additional gearing system can be added to the current design to allow for the increase in operation range.

Motor Vibration

There is a risk of the motor vibration being excessive which may cause the machine to vibrate excessively. A solution is to increase the mass of the structure which the motor is attached to.

Options include attaching weights to the structure simply to increase its mass and hence, reduce the amplitude of vibration.

Manufacturing Challenges

The magnitude and complexity of the amount of machining that is required of this project is tremendous. While some parts are expected to be sent out for external manufacturers to manufacture, there is still a possibility that the amount required to self-machine may not be completed in time. In the event that this happens, the most likely solution will be to send parts out for express manufacturing. Care needs to be taken to determine what parts need to be sent out as such outsourcing gets exponentially more expensive when it is due more urgently. A consistent careful monitoring of the parts to be manufactured is required, coupled with fair amount of judgment, to achieve the balance between cost and amount of outsourcing. Elaborated below are specific manufacturing challenges that we foresee.

<u>Tooling</u>

Two of the components of our design have geometries that require specialized tools for manufacture. A broaching tool is required to machine the keyway in the main shaft. Alternatively, an EDM machine can be used to manufacture the key way accurately, but would cost significantly more. A large 1.5” ball mill is also required to manufacture the cavity for the ball bearing for the alignment plates. These tools are uncommon and unavailable in our machine shop and will have to be purchased in order for us to manufacture the parts. As these tools have to be made out of cobalt or carbide which tends to be more costly, it may be more cost effective to have these parts sent out to an external machinist to have them manufactured.

<u>Precision</u>

Precision of the dimensions and the location of most machined geometries are crucial for the success of our test device. This is because, a slight misalignment of machined geometries, such as the drilled holes of the U-Channels, would cause the main driving shaft to be slanted. These geometrical imperfections would greatly affect the reliability of the test device which is sensitive to even the slightest deviations. Hence, many of the crucial geometries will be precisely milled if possible.

Assembly Challenges

There are potential challenges we may face while assembling our components together and ensuring compatibility. As major compatibility concerns will have already been resolved before any parts were purchased, we only foresee minor issues that may arise from our manufacturing processes.

<u>Fitting of Components</u>

The dimensions of the cup cap, modular axles and thrust bearing plates were designed to be a perfect fit with other components of the structure (i.e. cup, main shaft and thrust bearings respectively). This is done as the precise fitting of these parts are crucial to ensure the prefect alignment of the shaft. However, it is also likely that the tight fit would interfere with the smooth insertion and integration of these components. Hence, to solve this problem, we will sand the side and corners of the contact areas.

Assembling Electrical Components

In the final phase of the project, we will be assembling the entire operational prototype, which consists of incorporating the mechanical components with the electrical components of the design. We recognize that our knowledge of electrical components as Mechanical Engineers may be lacking. Therefore, we foresee that we may have to spend a significant amount of time reading up and studying about assembling the electrical system. Managing the electrical system is also a significant safety issue that we do not intend to treat lightly since the consequences of poor assembling could be as serious as death. Mounting strain gages may be a problem because they are very small and the alignment has to be very precise.

Potential Operational Issues

Care and consideration has been taken during our design phase of the project to account for all possible operational issues. Despite these measures, it is still possible for these problems to occur, which is only detectable after manufacturing and assembling is done.

Misalignment of Shaft

Low tolerance levels and poorly aligned parts may cause the shaft to be misaligned or unsteady during operation. The main possible cause of this problem would be due to holes drill on the main structure using the drill press, which has a low accuracy of the location of the drilled holes. To prevent this, special attention would be given to the precision of the drilled holes. If the final assembled structure has an angular misalignment of less than 1°, it would still be acceptable as this can be accounted for by the tilt alignment plates. However, if the misalignment exceeds 1°, we would have to disassemble the structure and re-drill the holes, with possibly the application of welding to keep secure the joint.

Reliability of Tilt Alignment Plate

Tilt alignment plates may not be able to tilt freely on the ball due to wear or abrasion. To prevent this, extra effort will be made to ensure that the contact surface of the ball and plates would be manufactured to be as smooth as possible. If the problem still occurs, we will look into adding lubricants on the contact surface to reduce the coefficient of friction.

Poor Sensing Capabilities

The low sensitivity and repeatability of the measurements is highly dependent on the magnitude of the changes in strain experienced by the strain gauge or temperature difference "visible" to the thermocouple. In the event that small changes cannot be distinguished, the solution to the problem is a simple one, though there might be insufficient time. To allow small load measurements, the strain will have to be increased either by increasing the stresses (reducing the area of contact) or by reducing the Young's Modulus (choosing another material). Some possible materials that have lower Young's Modulus than T6 6061 Aluminum include metal like Magnesium, Neodymium, Selenium, Lead or Gallium. However special care is required when dealing with these metals as they can be potentially toxic, flammable and/or hazardous. Nonetheless, the required increase in strain can only be determined after the actual machine is functional.

Logistical Challenges
<u>Transporting the Test Device</u>
As our test device will be bulky and heavy, more work and consideration will be needed to transport it. It is likely that the motor drive and the motor structure will have to be disassembled from the main structure before transportation. In addition, due to the weight of the shop press, it would have to be loaded on a pallet, and transported using a pallet lift.

ELECTRONICS VALIDATION
Due to time constraints and unexpected problems, the electronics of the design were not fully functional. It is hence the aim of this section to provide a record of what happened during the design to faciliate future trouble shooting. Figure 60 illustrates the problems the occurred.

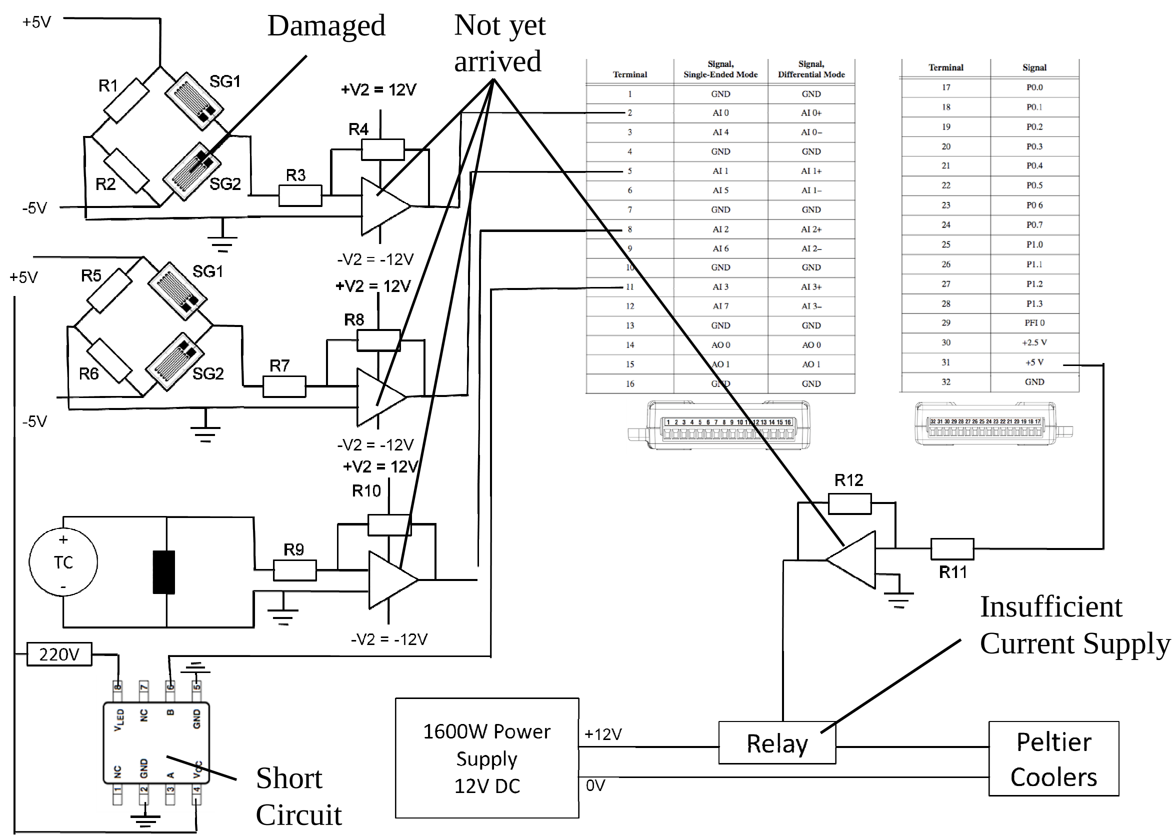

Figure 60: Schematic of Circuit Diagrams and Problems Associated

Strain Gauge
The strain gauge is incorporated in a Wheatstone bridge circuit and is expected to change in resistance with respect to strain. Unfortunately, one of the strain gauges was damaged, as proven by using a multimeter to measure the resistance of it. Without a resistance reading, it essentially meant the circuit could have been open where the strain gage is placed. However, other than the damaged strain gauge, the other three were operational. These were tested by applying dead

weights to the load cells and measuring the change in resistance. For the torque load cell, we exerted force to the end of the beam and recorded a slight change in resistance.

To improvise the validation test, a dummy resistance was used to replace the damaged strain gauge. We measured voltage changes while applying dead weights to the load cell. However, when that was done, LabVIEW did not detect a clear voltage increase. This could be due to the voltages being too small. The operational amplifiers that were supposed to be incorporated into the design did not arrive in time.

The proposed solution is to have the damaged strain gauge replaced and implement the operational amplifier when it arrives. Based on our previous theoretical calculations, the amplification of 255 times should be sufficient for LabVIEW to record the changes in strain.

Thermocouple
When tested with a multimeter, the thermocouple did produce a voltage across its junctions when a temperature difference occurred across the thermocouple probe. However, the voltage measured was in the order of millivolts, which was detectable by the multimeter, but not the DAQ in LabVIEW. As previously mentioned, the delayed delivery of the operational amplifiers resulted in the failure to amplify the thermocouple voltage. The algorithm in LabVIEW was nonetheless created for the thermocouple reading. When tested with a simulation signal, the output did produce the correct temperatures.

It is proposed that the operational amplifiers be incorporated into the circuit, with an amplification of 20 times. It is strongly believed that the temperature reading will be accurate.

Optical Encoder
The optical encoder, during the soldering process, had a short circuit occurring between the voltage in (V_{cc}) and a ground. The LED voltage in (V_{LED}) when connected to the ground with a 5V supply, was operational as it lit up. However, due to the short circuit, the channel outputs could not output any readable signal. It is proposed that a new optical encoder be purchased, or at the very least, the current optical encoder be investigated with regards to this short-circuit issue.

Temperature Control
The temperature control is controlled by a relay that requires at least 0.9A to turn on. This translates to a voltage between 7 to 12V. However, an unforeseen problem occurred when the DAQ (which had a voltage output), could not achieve the 0.9A that the relay requires. The output was a few milliamperes at maximum. This is a relatively small problem as the remedy requires a current amplifier. With a current amplifier, the relay can then be turned on and off by controlling the DAQ output voltage. The voltage output is in turn activated by the voltage recorded by the thermocouple. This essentially behaves as a temperature controller.

Peltier Coolers
Due to delay in shipping, the peltier coolers could not arrive in time. The temperature aspect of the project could not be tested then. However, the heat sink, fan and necessary adhesive did arrive in time and it is proposed that the peltier coolers be attached with the heat sink and fan,

when the coolers arrive. The coolers will then be required to be wired to the relay, which is in turn wired to the power supply. The wires for the peltier coolers were already prepared.

STRENGTHS AND WEAKNESSES

As Team 22 looks back and reviews on their design and design process, strengths and weaknesses can be determined for future reference. Strengths and achievements include in-depth exploration of a wide range of methods to achieve subsystem abilities and completion of the overall structure and large number of parts within a certain precision error. Weaknesses include the absence of the validation and failure of electronics. The team feels these weaknesses can be better addressed with more time.

WHAT COULD HAVE BEEN DONE DIFFERENTLY

Based on our evaluated strengths and weaknesses, our team has thought about some things that we would have done differently. Analysis and evaluations done up to this point in the project does not show any fundamental flaws in the operational concept in our test device. Because of this, we would not have changed any of our concept generation process nor design considerations. On hindsight, however, we should have hastened these design processes to allocate more time for assembly, validation and troubleshooting of the final assembled product. This would have left us with more reaction time to make any necessary minor modifications for our final product to work.

Specifically, we should have completed our concept generation phase earlier, which would enable us to finalize and order our stock parts earlier. This would ultimately enable us to complete our manufacturing and assembling earlier. Similarly, the electronic sensing components should have been purchased earlier once the engineering specifications had been set. This would enable allow us to build and test these electronic components early.

RECOMMENDATIONS AND FUTURE MODIFICATIONS

Several modifications and improvements need to be made to ensure that the system will be able to be run safely within the expected operating conditions.

Mechanical Systems

A shaft collar should be installed on the keyed shaft. This will act as an additional support for the sprockets to sit on, and prevent the sprockets from slipping down the shaft. Care should be taken while attaching the main structure to the motor table such that the structures should be bolted in a way that both structures are level to the ground. Should the structures be bolted too tightly, they may tilt towards each other, and the sprockets will be operating at an angle.

Electronic Systems

A new strain gauge needs to be installed on the load cell to replace the broken strain gauge. Parts that have not arrived, such as the operational amplifiers, need to be installed before testing can be conducted. Advice should be sought from professors and other experts in the field on how to properly set up the electronic systems and troubleshoot the devices. More time and work will be necessary to ensure that the electronic system will work with our device.

Safety
The safety shield for the sprockets will have to be redesigned and manufactured. This shield should be made of a suitable material that can withstand high loads in the event that the sprocket and chains fail. Our current system only covers the system for the sole purpose of eliminating pinch points, but may not be adequate for any mechanical failures.

Validation
Due to the short amount of time available to us, we were unable to complete our validation plan. This will need to be fully completed before the system can be used for normal operations. There is no indication that the validation plan that we laid out is insufficient.

CONCLUSION

It is unfortunate that Team 22 could not accomplish the project thoroughly. Given the time and logistical constrains, the team was faced with a very challenging task of completing the project. As there are currently many outstanding issues left before this project can be completed, a new project team will have to be take over from where we left off. To aid in this handing over process, our team has documented these problems. It is also not concluded if the machine is able to accomplish the engineering specifications set by our sponsor, as it has not been deterministically tested. Although it is expected that the machine will be able to meet the specifications, work is required to ensure the smooth operation of the machine.

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APPENDIX A: SPECIFICATION SHEETS FROM RESEARCH

Appendix A-1: Specification Sheet from Koehler Instrument

KK93100 Four Ball Wear and EP Tester
Four Ball Tester performs both Wear Preventative (WP) and Extreme Pressure (EP) analyses for measuring the wear and frictional properties of lubricants under sliding-on-steel test conditions. Tests are performed in accordance to the latest ASTM and IP published methods. Normal load on the ball assembly and frictional torque are measured through load cells. Wear scars on the steel balls are measured with a graduated-scale microscope and can be recorded with an optional CCD camera. Data is processed and stored utilizing TriboDATA, and advanced data acquisition and processing software package. Test results can be plotted and compared, as well as exported to other programs.

Specifications
Conforms to: ASTM D2266, D2596, D2783, D4172, D5183*; IP 239

Electrical Requirements:
220V, 60Hz, 3 phase
440V, 50Hz, 3 phase

Drive Motor Power: 1.5 kW

Test Speeds: 1200, 1440, 1760 rpm

Optional Test Speeds (min/max): 1000/3000, 300/3000 rpm

Maximum Axial Load: 10000 N at 3000 rpm or 12000 N at 1800 rpm

Test Duration (min/max): 1/9999 min

Test Ball diameter: 12.7mm

Included Accessories
Set of Weights
Ball Chucks
Ball Pot
Ball Chuck Remover
Ball Rack
Ball Clamp Ring
Ball Holder Base Disc
Set of Hand Tools
Torque Wrench
Graduated-Scale Microscope
Electronic Controller
Connecting Cables
TriboDATA Software
Calibration and Test Reports

Shipping Information
Shipping Weight: 1360 lbs (620 kg)
Dimensions: 45 Cu. Ft.

Appendix A-2: Specification Sheet from Institute of Sustainable Technologies (IST)

T-02 FOUR-BALL TESTING MACHINE

FOR DETERMINATION
OF LUBRICATING PROPERTIES
OF OILS AND GREASES

INSTITUTE FOR SUSTAINABLE TECHNOLOGIES
NATIONAL RESEARCH INSTITUTE RADOM

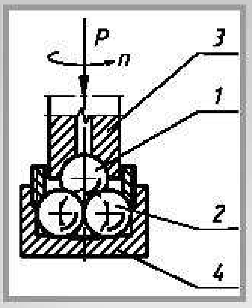
ul. K. Pulaskiego 6/10, 26-600 Radom, POLAND

fax: (+48-48) 3644765 www.itee.radom.pl www. tribologia.org trib-dep@itee.radom.pl

CHARACTERISTIC OF T-02 TESTING MACHINE

T-02 Four-Ball Testing Machine is intended for determination of extreme pressure (EP) and antiwear (AW) properties of lubricants like oils and greases.

T-02 Machine makes it possible to carry out experiments in accordance with the following standards: **ASTM D 2783, ASTM D 2596, ASTM D 4172, ASTM D 2266, IP 239, DIN 51350, Fiat 50500, PN-76/C-04147**. After using an optional kit the realisation of surface fatigue tests according to **IP 300** is also possible.



The tribosystem consists of the three stationary balls (2) fixed in the ball pot (4) and pressed at the required load P against the top ball (1). The top ball is fixed in the ball chuck (3) and rotates at the defined speed n.

T-02 Four-Ball Testing Machine is equipped with a control-measuring system which consists of:

- a set of measuring transducers,
- digital measuring amplifier,
- PC and special software for measurements and data acquisition.

During the tests the following quantities are measured:

- friction torque,
- applied load,
- lubricant temperature,
- rotational speed,
- time.

The measured values are displayed on the monitor screen and saved on the computer disk. The motor of the tribotester is automatically stopped when the preset time elapses or when the preset friction torque is reached. After test completion one can print a report presenting curves of changes in the particular quantities versus time.

A unique feature of T-02 Machine is a possibility of automatic, continuous increasing of the load during the run, which is necessary to carry out research under conditions of scuffing, according to a test method developed at the Institute for Sustainable Technologies in Radom. What is more, before the run the load can be applied without any effort - it is enough just to press the button and the weight will slide along the loading lever increasing the load in this way. This prevents the operator from carrying heavy weights.

TECHNICAL SPECIFICATIONS

– type of movement	sliding
– contact geometry	non-conformal (point)
– nominal ball diameter	12.7 mm (0.5 in.)
– rotational speed	up to 1800 rpm
– applied load	up to 7848 N
– speed of load increase	409 N/s
– tribotester dimensions (W × H × D)	1700 × 1700 × 620 mm
– tribotester weight	210 kg
– power supply	380 V / 50 Hz
– max. power consumption	2.1 kW

TÜV CERT EN ISO 9001 RWTÜV

Appendix A-3: Specification Sheet from PTI

PTI

TRIBOLOGY TESTING EQUIPMENT

For the 21st Century

P.O. Box 489 • Hinckley, IL 60520

Phone: (630) 556-4622 • Fax: (630) 556-4624 • E-mail: PTIINCTEST@aol.com

MULTI - FOUR BALL TEST MACHINE

FEATURES AND SPECIFICATIONS

SPECIMENS: Three-point contact. Rotates a ½ " diameter test ball against three ½" diameter balls. Other specimen configurations and geometry's can be adapted.

LOADS: Computer controlled load system applies load to 1000.0 kilograms load.

SPEEDS: Computer controlled test ball speed is 3 to 3,600 rpm

WEAR: Digital micrometer type system allows rate of wear measurement during test and total wear. The system measures wear to 0.0001 ins.

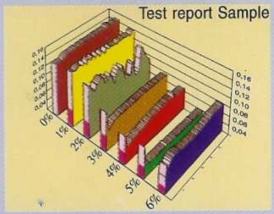
FRICTION: Digital Friction force Indicator (0.00 to 50.00 kilograms.)

ENVIRONMENT: Liquid or grease. Computer controlled heater with control to 300 degrees F. This system is designed for the evaluation of varies fluids and greases, etc.

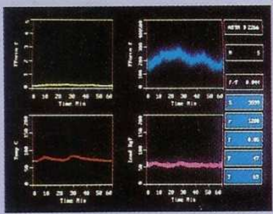
METHODS AND SPECIFICATIONS:

D 2596	IP-239
D 2266	FTM-6503
D 2783	FTM-6520
D 4172	

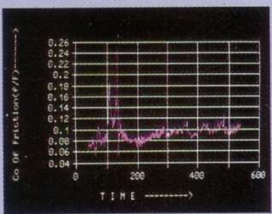
Appendix A-4: Specification Sheet from Tribotesters



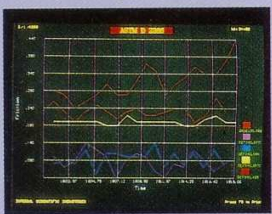
Coefficient of friction Vs additive quantity



Run/Replay test screen enables the user to monitor:
- Digital Display readings, traces
- Co-eff. of friction Value

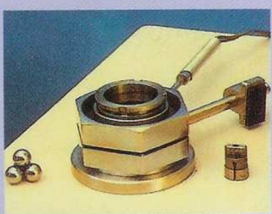


Facility to zoom graphs (initial seizures)



Individual colour specifying a particular test item

Plotting graph Options



4-Ball E.P./wear cup, chuck, welded balls

IMPERIAL FOUR BALL MACHINE IS DESIGNED TO PERFORM TESTS AS PER THE FOLLOWING INTERNATIONAL TEST STANDARD METHODS

ASTM D2596 - **Extreme-Pressure Properties of**
ASTM D2783 **Lubricating Fluids & Greases**
ASTM D2266 - **Wear Preventive characteristics of**
ASTM D4172 **Oils & Greases**
FTMS 791-6503
IP 239 - **Extreme-Pressure Properties, Friction and wear tests for lubricants**
GOST 9490-77

A rotating ball is pressed onto three stationary balls of same size and quality. Wear scars on surface, welding of balls, torque and size of seizure delay is measured with continuously varialble speed.

COMPRISES OF

2HP variable speed drive, Pneumatic loading system, (manual / programmed) load cells for Friction Force and load measurements, digital display for temperature, speed, time, load, friction force.

LOAD

Pneumatically activated, continuous loading max range 1000 kg.

SPEED

Variable speed drive, continuously variable 2HP/22OV Max. Range 60-3000 rpm *(specify voltage while ordering).*

TEMPERATURE

Thermostatically controlled. Temp. of specimen holder range ambient to 250'C

FRICTION FORCE

High accuracy load cell for measurement of Friction Force between specimens. Range 0-5 Kg.

TEST TIME

Digital timer/for automatic start or shut off.

COMES WITH

- Basic Machine.
- Built in Metal Stand.
- The Control Panel : Microprocessor based Variable Drive AC.
- Digital Display System : to monitor Load, Friction force, RPM & Temp.
- Steel Test Balls, Material EN31, with Selected Hardness & Tolerance.
- Test Cups to Run Cyclic Fatigue Test and Shear Stability Test.
- Accessories & Spares to conduct all the above tests.
- Spare Chucks.
- Spare Balls (Pack of 2000 Balls).

MICRO PROCESSOR CONTROLLED SYSTEM FOR DATA ACQUSITION & REAL TIME (ON/OFF) ANALYSIS

Pentium II, 256, MB RAM, 8.4 GB HDD, CD ROM, 17" Colour Monitor, Inkjet printer PLC , Dedicated user friendly software and Digital camera (for detailed scar dia. analysis).

REAL TIME ANALYSIS CAN BE DONE ON THE FOLLOWING GRAPHS:

Temp	Vs	Time
Load	Vs	Time
Coeff. of friction (f/F)	Vs	Time
Coeff. of friction (f/F)	Vs	Temperature
Coeff. of friction (f/F)	Vs	Load
Coeff. of friction (f/F)	Vs	Scar dia
Coeff. of friction (f/F)	Vs	Additive quantity

IMPERIAL SCIENTIFIC INDUSTRIES
P.O. Box - 4827, Sarojni Nagar, New Delhi-110023, India
Phone : +91-11-616 7133 • Fax: +91-11-616 7133 • e-mail: sales@ tribotek.com
website: www.tribotek.com

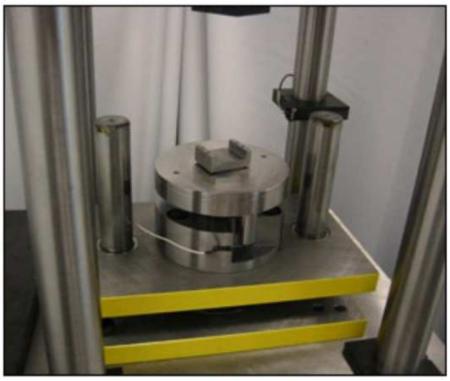
Appendix A-5: Specification Sheet from Tribsys

TRIBSYS
31 Second Ave.
Coniston, ON, Canada P0M 1M0
Tel: (705) 694-9605
e-mail: gdalton@tribsys.com

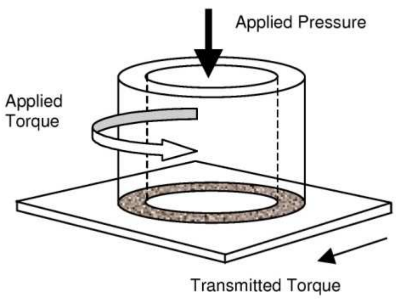
99 West 550 North
Valparaiso, IN, USA 46385
Tel: (219) 531-2583
e-mail: tmcclure@tribsys.com

The TribSys Twist Compression Test (TCT)

The Twist Compression Test (TCT) is used to evaluate lubricants and die materials for application in metalforming processes. This test measures the transmitted torque between a rotating annular cylinder and a lubricated flat sheet specimen. The 25mm (1") diameter annular cylinder is driven by a hydraulic motor for smooth delivery of the applied torque at speeds up to 30 RPM (90 inches/minute). The pressure may be adjusted up to 35,000 psi to best duplicate the tribological conditions of the metalforming process being studied.



Data is collected electronically and the coefficient of friction is calculated from the ratio of transmitted torque to applied pressure.
The TCT is best used as a comparative rather than an absolute test. The simplicity of the TCT and good laboratory practice minimizes variations.



However, it is advisable to include a reference for each series of evaluations.

LUBRICANT PERFORMANCE INDICATORS IN THE TWIST COMPRESSION TEST

Friction – Indicates the effectiveness of the lubricant at reducing the interfacial shear stress. Interfacial shear prevents movement of sheet material into a die or distribution of material over a punch.

Time to Breakdown – Indicates ability of the lubricant to prevent adhesion between the tool and the workpiece. This is a function of the lubricant film strength and additive efficacy including extreme pressure (E.P.) lubricants.

Pickup/Galling – The nature of metal transfer in the TCT has been found to be an excellent indicator of tool wear.

CRITICAL FEATURES

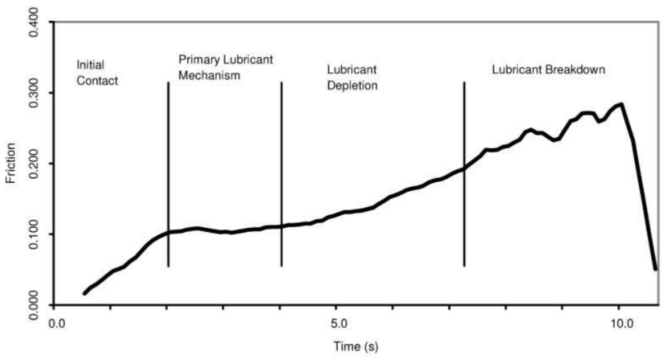
The TribSys Twist Compression Test has several features that are critical for the successful evaluation of lubricants or die materials.

- **Self centering tooling to ensure concentricity of annular tool and sheet specimen holder**
- **Self aligning sheet holder on a high capacity spherical bearing maintaining parallelism of tool and workpiece**
- **High capacity ultra-low friction bearing for low losses**
- **Engineered for low noise and maximum signal to noise ratio**

Understanding TCT Results

The frictional force transmitted by the tool to the workpiece changes as the lubricated interface changes with tool rotation under the applied load. In a typical test, the following stages can be identified.

Typical Twist Compression Test Output



Stage 1. Initial Contact – The rotating tool contacts the lubricated sheet. The transmitted torque increases rapidly as pressure at the interface builds and the lubricant is displaced. This stage is governed by the following parameters:

- sheet and tool characteristics
- lubricant viscosity
- contact pressure
- speed of rotation

If determining a static coefficient of friction, the load is applied before beginning rotation.

Stage 2. Primary Lubricant Mechanism – In most cases, the interface will reach a period of stability and exhibit a stable frictional force. This period of stability may be brief or continue for several revolutions depending on test conditions.

- Hydrodynamic lubrication – may occur briefly in TCT results, usually involving viscous lubricants at low pressure.
- Boundary lubrication – common in TCT results, the full load of the tool is carried by the points of contact with the sheet. Viscosity will have little effect on boundary lubrication.
- Mixed lubrication – also common in TCT results, pockets of lubricant trapped in the sheet surface are pressured. The pressurized pockets replenish the lubricant at the sheet/tool interface.
- Solid film lubrication – the sheet and tool are separated by a solid film.

Stage 3. Lubricant Depletion – With continued sliding contact the lubricant is depleted and the above mechanisms may fail. In the presence of EP additives, the heat generated at the contact points may be sufficient to cause a reaction between the additive and the metal surfaces. In such cases low shear strength reaction products will form on the surface(s) preventing or delaying breakdown.

Stage 4. Lubricant Breakdown – When lubricant mechanism failure occurs friction rises dramatically and becomes unstable as pickup and galling form. The test is usually stopped at this point to preserve the tool.

Test Results

A number of responses can be measured or calculated. The most common are:

- initial peak friction (end of Stage 1)
- average friction (Stage 2)
- time to breakdown (time from test start to beginning of Stage 4)

Pickup and galling can be evaluated from the tool and sheet specimens. Tribochemical residues may be analyzed.
Contact area can be measure from the sheet specimens using optical microscopy and image analysis.

www.tribsys.com

TRIBSYS INC.

Quotation

Date	Estimate #
9/23/2009	T9022 - B01

valid for 60 days

P.O. Box 720
31 Second Ave.
Coniston, ON
P0M 1M0

Name / Address
University of Michigan
Ann Arbor, MI 48109 USA
1 734 764-1817 .
Attn: Kok-Shing Chu [kokshing@umich.edu]

Warranty	Payment Terms
1 year	**25% down - Balance on shipping**

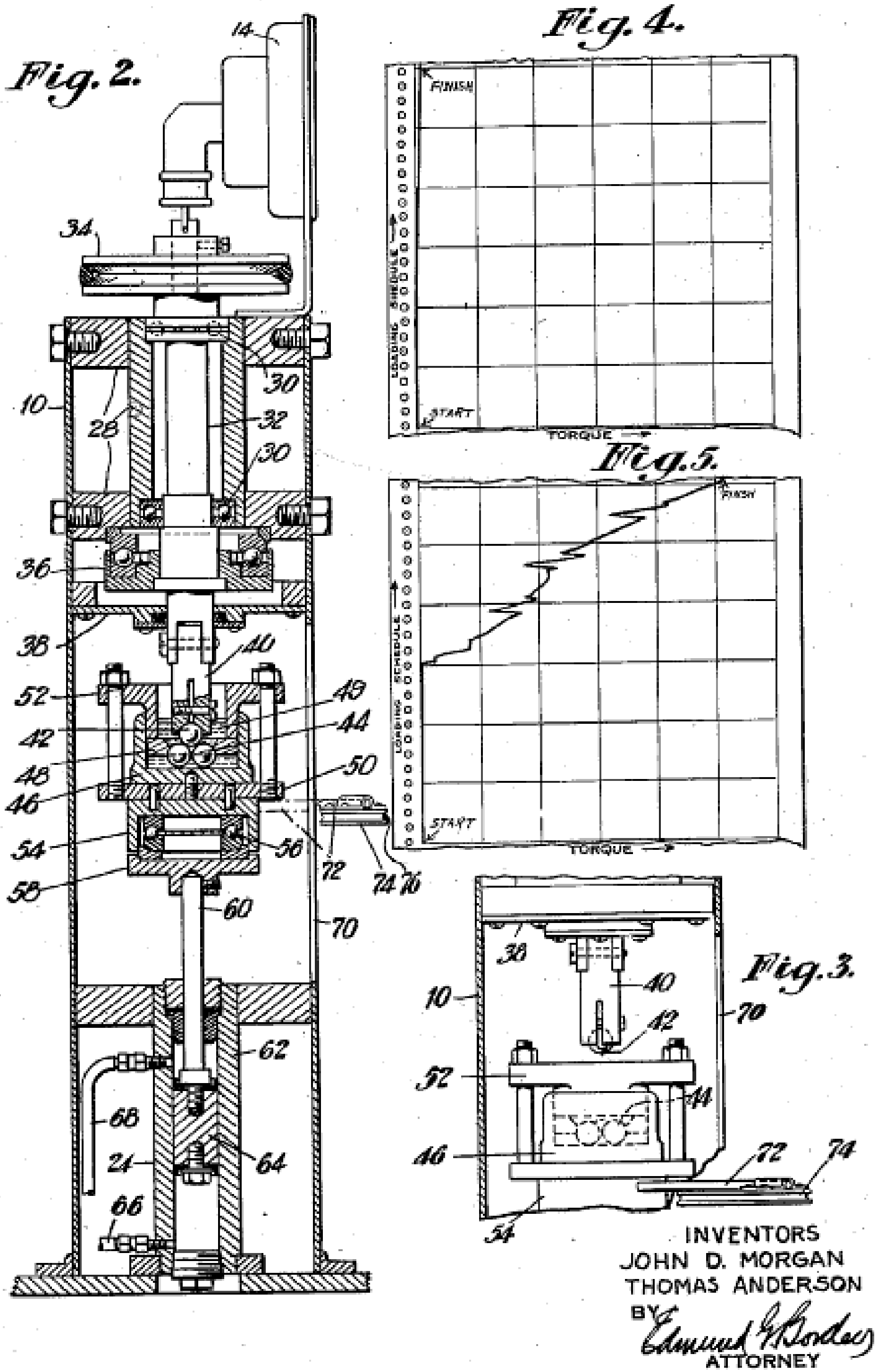
Item	Description	Unit	Rate	Total
TCT Unit	**Twist Compression Test - Base Unit**	1	41,176.47	41,176.47
Data A&A	Computer (monitor not included), printer, data acquisition	1	9,500.00	9,500.00
	system, TribSys proprietary software		120.00	0.00
HP Unit	Hydraulic Power supply for press frame	1	8,823.53	8,823.53
Press Frame	Hydraulic Press with actuators and valving	1	32,281.00	32,281.00
Training	2 days training and comissioning included	1	0.00	0.00
Travel	Transportation, accomodation, meals (estimated)	1	3,000.00	3,000.00
TCT Tools	100 annular tools included	1	0.00	0.00
	Upgrades:			
TCT Unit	50k psi interface pressure testing capability	1	1,404.30	1,404.30
TCT Tools	100 annular tools	1	2,000.00	2,000.00
TCT dual range capability	Dual range hydraulic actuator and controls	1	2,315.70	2,315.70
Press Upgrade	Load frame upgrade for 50K psi testing	1	15,438.00	15,438.00
	Total GST Business Number: 86690 698			0.00

Confirmation of Order - Please remit payment in either US$ or Cdn $ - US$ rate quoted on request (valid for 30 days)	**Total**	Cdn. Dollars	**$115,939.00**

APPENDIX B: PATENTS FROM RESEARCH

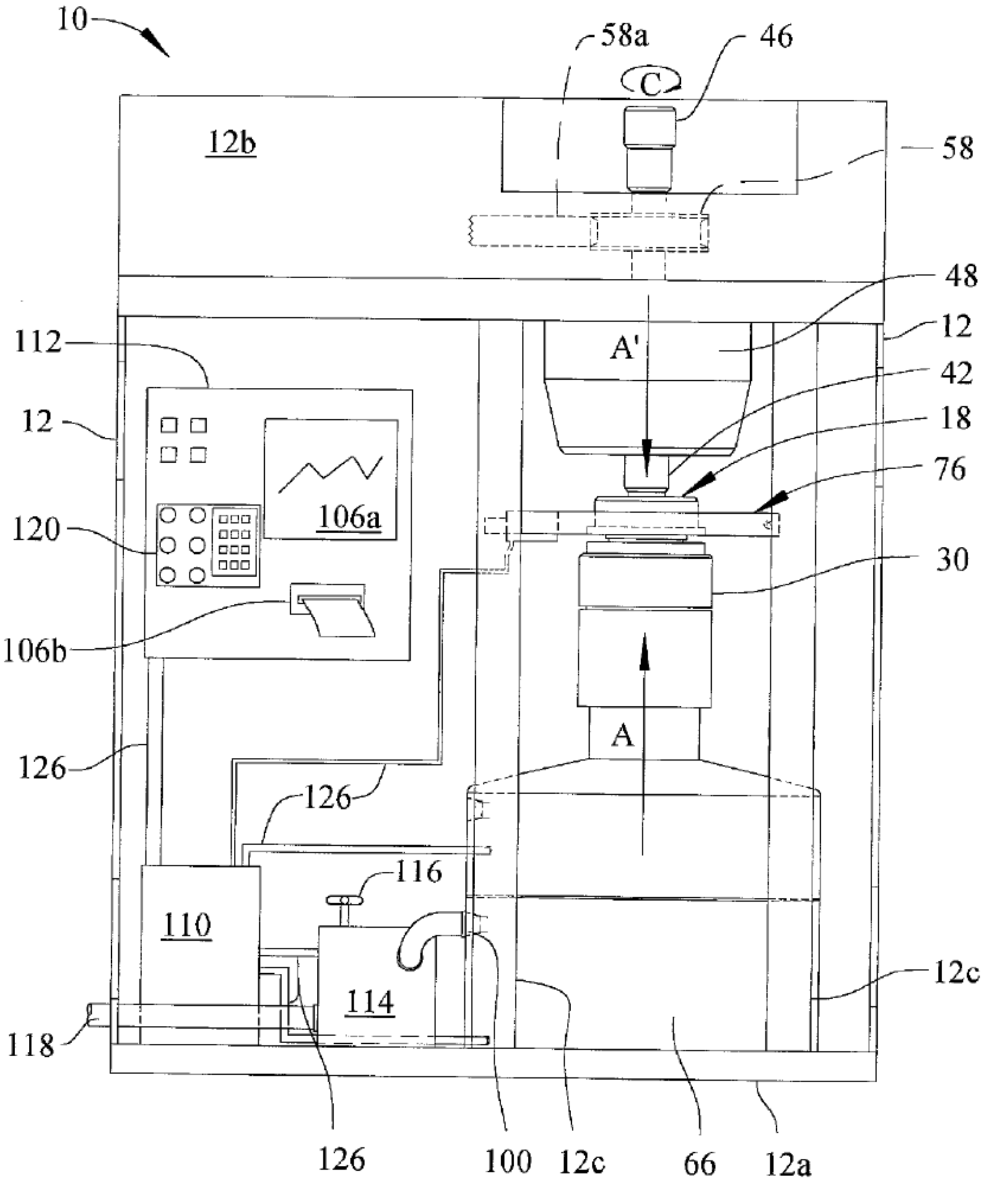
Appendix B-1: Patent 2,370,606

Apparatus for Testing Lubricants



Appendix B-2: Patent NO. US 2003/0101793

Machine For Testing Wear, Wear-Preventative and Frition Properties of Lubricants and Other Materials



APPENDIX C: HAND CALCULATIONS

Appendix C-1: Twist Compression Test Calculations

Twist Compression:

ASTM Standards: Max Pressure applied is 35 ksi, Max Speed is 30 RPM, and Cylinder Outer Diameter is 1 inch
$P = 35\ ksi = 241.3\ MPa$ $V = 30\ RPM = 0.0399\ m/s$ $OD = 1\ in = 0.0254\ mm$

Assume Max Frictional Coefficient $\mu = 0.15$

Max Cylinder Thickness $= t = \frac{1}{4} \times\ OD = 0.00635\ m$
Inner Diameter $= ID = OD - 2 \times t = 0.0123\ m$
Area $= A = \pi\left(\left(\frac{OD}{2}\right)^2 - \left(\frac{ID}{2}\right)^2\right) = 0.0003879\ m^2$

Normal Force $= F_N = P \times A = 93.6\ kN$

Friction Force$= F_F = F_N \times \mu = 14.040\ kN$

Power $= P_W = F_F \times V = 560.2\ W = 1.0185\ Hp$
$1\ W = 0.00134\ Hp$
Torque $= T = F_F \times \frac{OD}{2} = 178.3\ N - m = 131.41\ lb/ft$

Appendix C-2: Four-Ball Test Calculations

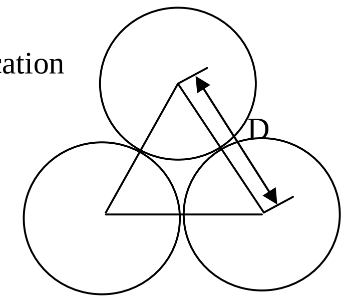
Four-Ball:

ASTM Standards: Max Force applied is 10 kN, Max Speed is 3600 RPM, and Ball Diameter is 0.5 inch

$F = 10\ kN$ $\qquad\qquad\qquad$ $V = 3600\ RPM$

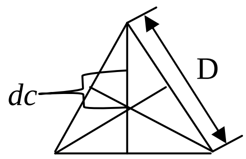
Assume Max Frictional Coefficient $\mu = 0.15$

Assume distance from contact points to center of load application $= dc$



$$dc = \frac{1}{3}\sqrt{D^2 + (0.5D)^2}$$

$$dc = 0.28868\ D$$



$$dc = 0.00367\ m$$

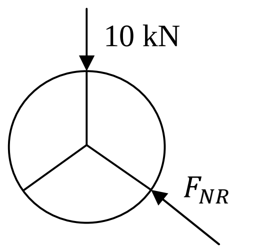
Normal Reaction Force= $F_{NR} = 10.46\ kN$

Friction Force= $F_F = F_{NR} \times \mu = 1.57\ kN$

Power= $P_W = F_F \times V = 2172.2\ W = 2.905\ Hp$

$1\ W = 0.00134\ Hp$

Torque = $T = F_F \times dc = 7.972\ N - m = 5.88\ lb/ft$



APPENDIX D: SPECIFICATION SHEET FOR DAQ (NI USB-6009)

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™

Every M Series data acquisition device also includes a copy of LabVIEW SignalExpress LE data-logging software, so you can quickly acquire, analyze, and present data without programming. The NI-DAQmx Base driver software is provided for use with Linux, Mac OS X, Windows Mobile, and Windows CE operating systems.

Recommended Accessories

The USB-6008 and USB-6009 have removable screw terminals for easy signal connectivity. For extra flexibility when handling multiple wiring configurations, NI offers the USB-600x Connectivity Kit, which includes two extra sets of screw terminals, extra labels, and a screwdriver.

In addition, the USB-600x Prototyping Kit provides space for adding more circuitry to the inputs of the USB-6008 or USB-6009.

NI USB DAQ for OEMs

Shorten your time to market by integrating world-class National Instruments OEM measurement products into your embedded system design. Board-only versions of NI USB DAQ devices are available for OEM applications, with competitive quantity pricing and available software customization. The NI OEM Elite Program offers free 30-day trial kits for qualified customers. Visit **ni.com/oem** for more information.

Information for Student Ownership

To supplement simulation, measurement, and automation theory courses with practical experiments, NI has developed the USB-6008 and USB-6009 student kits, which include the LabVIEW Student Edition and a ready-to-run data logger application. These kits are exclusively for students, giving them a powerful, low-cost, hands-on learning tool. Visit **ni.com/academic** for more details.

Information for OEM Customers

For information on special configurations and pricing, call (800) 813 3693 (U.S. only) or visit **ni.com/oem**. Go to the Ordering Information section for part numbers.

Ordering Information

NI USB-6008[1]	779051-01
NI USB-6009[1]	779026-01
NI USB-6008 OEM	193132-02
NI USB-6009 OEM	193132-01
NI USB-6008 Student Kit[1,2]	779320-22
NI USB-6009 Student Kit[1,2]	779321-22
NI USB-600x Connectivity Kit	779371-01
NI USB-600x Prototyping Kit	779511-01

[1] Includes NI-DAQmx software, LabVIEW SignalExpress LE, and a USB cable.
[2] Includes LabVIEW Student Edition.

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.

APPENDIX E: SPECIFICATION SHEET FOR OPTICAL ENCODER SYSTEM

Appendix E-1 : Optical Encoder Module

US DIGITAL HEDR Reflective Optical Encoder Modules

Description

The HEDR encoder module uses reflective technology to sense rotary or linear position. This sensor consists of an LED light source and a photodetector IC in a single surface mount package. When used with a reflective codewheel or codestrip, this device can sense rotary or linear position.

The reflective surface mount optical encoders provide two square wave outputs in quadrature for count and direction information.These TTL compatible outputs correspond to the alternating reflective/non-reflective pattern of the codewheel or codestrip.

The HEDR reflective optical encoder modules are available in three resolutions of 75, 150, and 180 LPI (lines per inch).

See the online HEDR-8000/HEDR-8100 Datasheet for complete information.

Features

- Reflective technology
- Surface Mount SO-8 package
- Two-channel quadrature outputs for direction sensing
- Available in 3 resolutions
- Small size
- TTL compatible
- Single 5V supply

US DIGITAL
1400 NE 136th Avenue
Vancouver, Washington 98684, USA
info@usdigital.com
www.usdigital.com
Local: 360.260.2468
Toll-free: 800.736.0194

Appendix E-2 : Codewheel

Description

US Digital offers a wide variety of standard hub / disk assemblies (optical encoder disks attached to an aluminum hub) to aid with mounting on a shaft. Encoder disks may also be ordered as a stand alone item (see the Disks page). These rotary encoder disks are made from mylar polyester film. This material allows for a wide temperature range and is virtually unbreakable.

HUBDISK-2 consists of a precision machined aluminum hub fastened to a 2" diameter optical encoder disk (DISK-2). One or two set screws (depending on the hub's ID) are used to fasten the hub to a shaft. The codewheel is available with bore sizes ranging from 2 mm to 1 inch. All hub bores are held to a very tight tolerance of -0.0000 to +0.0005 inches.

The index track option is not available with 512 CPR. The following resolutions are only available with the index option: 64, 1800 and 2500 CPR. Other resolutions are available with or without an index track.

Mechanical alignment drawings for optical encoder rotary disks and optical encoder modules are available (see the EM1 and HEDS pages).

Features

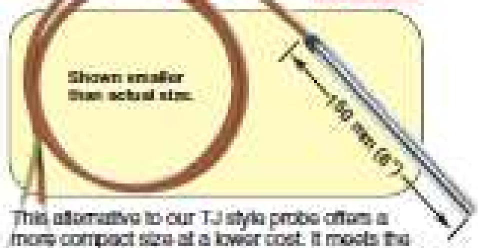
- 2" outside diameter
- .920" optical radius
- 64 to 2500 cycles per revolution (CPR)

APPENDIX F: SPECIFICATION SHEET FOR THERMOCOUPLE (HTTC36-T-18G-6)

Specialty Thermocouple Probes

Low-Cost Hollow-Tube Thermocouple Probe

- 1 m (40") PFA-Insulated Lead Wire Epoxy Potted into a 304 SS Sheath
- Probe Rated Up to 450°F
- Probe Diameters of 1.5, 3.0, 4.5, and 6.0 mm (1/16", 1/8", 3/16", and 1/4")
- J, K, T, E, and N Calibrations
- Glass-Insulated Leads Available for Higher Temperatures (Up to 900°F)
- Made with Special Limits of Error Wire
- Grounded Junction



This alternative to our TJ style probe offers a more compact size at a lower cost. It meets the requirements of limited-space applications, without the metal transition fitting or strain relief spring.

MOST POPULAR MODEL HIGHLIGHTED!

To Order *(Specify Model Number)*

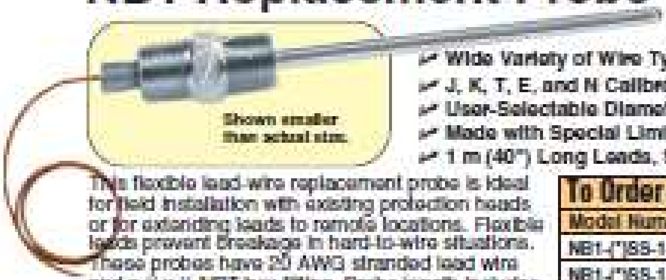
Model Number	Probe Dia.	Lead AWG	Price
HTTC36-(*)-116G-(**)	1.5 mm (1/16")	30 Solid	$22
HTTC36-(*)-18G-(**)	3.0 mm (1/8")	24 Stranded	19
HTTC36-(*)-316G-(**)	4.5 mm (3/16")	20 Stranded	21
HTTC36-(*)-14G-(**)	6.0 mm (1/4")	20 Stranded	22

** Specify calibration type: J, K, T, E, or N. ** Specify probe length in inches: 1½ to 6" (38 to 150 mm). Over 150 mm (6") add $1/in. 32 mm (1.25") minimum; 1.2 m (48") maximum. Over 1.2 m (48") consult Custom Engineering Department. Over 1 m (40") leads, add $1/ft, and modify model number. Add suffix "GG" for fiberglass insulated lead wire. [Does not include 1.5 mm (1/16") dia.]*

Ordering Example: *HTTC36-K-18G-6-GG, hollow tube thermocouple Type K calibration, 3 mm (1/8") dia., 150 mm (6") probe length, 1 m (40") glass insulated leads, $19.*

Options: *Modify model number to meet your specific needs. For connector terminations add suffix "-OSTW-M" or "-SMPW-M" and add $5 to the price. For ungrounded probe change G in part number to U and add $6 to price.*

NB1 Replacement Probe With Lead Wire



- Wide Variety of Wire Types Available
- J, K, T, E, and N Calibration Types
- User-Selectable Diameter and Length
- Made with Special Limits of Error Wire
- 1 m (40") Long Leads, Standard; Other Lengths Available

This flexible lead-wire replacement probe is ideal for field installation with existing protection heads or for extending leads to remote locations. Flexible leads prevent breakage in hard-to-wire situations. These probes have 20 AWG stranded lead wire and a ⅛ x ½ NPT hex fitting. Probe length includes 13 mm (½") of hex fitting threads.

To Order *(Specify Model Number)*

Model Number	Description	Price
NB1-(*)SS-14G-(**)-RP-†	Up to 300 mm (12") probe length	$45
NB1-(*)SS-14U-(**)-RP-†	Up to 300 mm (12") probe length	47

** Specify calibration type: CA, IC, CP, CX, NN. ** Probe length in inches. † Wire insulation type: TT: PFA, GG: fiberglass and length in inches.*

Over 1 m (40") leads, add $1/ft, and modify model number. TT-SS: PFA with SS overbraiding, GG-SB: fiberglass with SS overbraiding and length in inches, add $3 to the price. Over 1 m (40") leads, add $2.25/ft, and modify model number.

Ordering Example: *NB1-CASS-14G-12RP-TT36, Type K "replacement probe", 300 mm (12") long, 6 mm (1/4") Dia., grounded junction with 1 m (40") PFA leads, $45 (for other diameters and configurations, contact Sales).*

Metal Handle Probe

- 1.5, 3.0, 4.5, and 6.0 mm (1/16, 1/8, 3/16, and 1/4") Diameters
- 300 mm (1') of Retractable Cable Expands to 1.5 m (5')
- Probe Leads Terminate in a Male SMP Miniature Connector
- Heavy-Duty Aluminum Handle
- Bendable Probe
- Variations Available—Contact the Quotations Department

The MHP handle probe picks up where the HPS handle probes leave off, offering unparalleled durability and no probe length limitations. A compression fitting rigidly secures the probe to the cast-aluminum handle and provides the rugged construction and stability that long probes and heavy-duty applications require.

To Order *(Specify Model Number)*

Model Number	Price
MHP-(*)SS-14G-(**)-SMP	$67

** Specify calibration type: CA, IC, CP, CX, NN for K, J, T, E and N. ** Specify probe length in inches.*

Ordering Example: MHP-CASS-14G-12-SMP, metal handle probe, Type K calibration, 6 mm (1/4") diameter, 300 mm (12") length, SMP connector, $67 (for other diameters and configurations, contact Sales).

APPENDIX G: SPECIFICATION SHEET FOR STRAIN GAGE (EA-06-125AD-120)

Stress Analysis Gages

The Strain Gage Designation System described below applies to Vishay Micro-Measurements General-Purpose Strain Gages.

Self-Temperature-Compensation

Foil Alloy

Carrier Matrix (Backing)

Active Gage Length in Mils (0.001 in [0.0254 mm])

Grid and Tab Geometry

Resistance in Ohms

Optional Feature

Example: E A - 06 - 250 BG - 120 Option LE

E: Open-faced general-purpose gage with tough, flexible cast polyimide backing. Various options are available, including lead connection features and protective encapsulation.

CE: Flexible gages with a cast polyimide backing and encapsulation featuring large, rugged, copper-coated solder tabs. This construction provides optimum capability for direct leadwire attachment.

N2: The 'N2' matrix provides an open faced gage on a thin, high-performance laminated polyimide film backing.

S2: Gage grid and solder tabs fully encapsulated in a thin, flexible, laminated polyimide film. Provided with large (0.030 in [0.75mm]) solder pads for ease of leadwire attachment.

W: Provides a gage fully encapsulated in glass-fiber-reinforced epoxy-phenolic resin. High-endurance leadwires.

S: Full encapsulation identical to the W matrix, but with solder dot connections instead of leadwires.

A: Constantan alloy in self temperature-compensated form.

P: Annealed constantan. Used for high-elongation or post-yield gages.

D: Isoelastic alloy. High gage factor and high fatigue life.

K: Nickel-chromium alloy (similar to Karma) used for high-performance self-temperature-compensated gages.

The S-T-C number is the approximate thermal expansion coefficient in ppm/°F of the structural material on which the gage is to be used. The following S-T-C numbers are available:

A alloy: 00, 03, 05, 06, 09, 13, 15, 18

P alloy: 08

K alloy: 00, 03, 05, 06, 09, 13, 15

For S-T-C's of 30, 40, or 50, refer to high S-T-C patterns.

The D alloy is not available in self-temperature-compensated form. 'DY' is used instead.

STANDARD OPTIONAL FEATURES

Most of the following options apply to the EA- or ED-Series gages as indicated in the General-Purpose Listings:

- **W:** Integral printed circuit terminal, polyimide encapsulation
- **E:** Polyimide encapsulation, leaving a portion of solder tab exposed
- **SE:** Solder dots plus polyimide encapsulation
- **L:** Preattached, soft, formable copper leads
- **LE:** Leads plus polyimide encapsulation
- **P:** Preattached leadwire cables and encapsulation
- **P2:** Preattached leadwire cables for CEA-Series gages

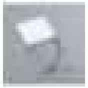
Please refer to the Optional Features data sheet for technical details on these and other available optional features.

Standard Strain Gage Series Selection Chart

GAGE SERIES	DESCRIPTION AND PRIMARY APPLICATION	TEMPERATURE RANGE	STRAIN RANGE	FATIGUE LIFE	
				Strain level in με	Number of Cycles
EA	Constantan foil in combination with a tough, flexible, polyimide backing. Wide range of options available. Primarily intended for general-purpose static and dynamic stress analysis. Not recommended for highest accuracy transducers.	Normal: –100° to +350°F [–75° to +175°C] Special or Short-Term: –320° to +400°F [–195° to +205°C]	±3% for gage lengths under 1/8 in [3.2 mm] ±5% for 1/8 in and over	±1800 ±1500 ±1200	10^5 10^6 10^8
CEA	Universal general-purpose strain gages. Constantan grid completely encapsulated in polyimide, with large, rugged copper-coated tabs. Primarily used for general-purpose static and dynamic stress analysis. 'C'-Feature gages are specially highlighted throughout the gage listings.	Normal: –100° to +350°F [–75° to +175°C] Stacked rosettes limited to +150°F [+65°C]	±3% for gage lengths under 1/8 in [3.2 mm] ±5% for 1/8 in and over	±1500 ±1500 *Fatigue life improved using low-modulus solder.	10^5 10^6*
N2A	Open-faced constantan foil gages with a thin, laminated, polyimide-film backing. Primarily recommended for use in precision transducers, the N2A Series is characterized by low and repeatable creep performance. Also recommended for stress analysis applications employing large gage patterns, where the especially flat matrix eases gage installation.	Normal Static Transducer Service: –100° to +200°F [–75° to +95°C]	±3%	±1700 ±1500	10^6 10^7
WA	Fully encapsulated constantan gages with high-endurance leadwires. Useful over wider temperature ranges and in more extreme environments than EA Series. Option W available on some patterns, but restricts fatigue life to some extent.	Normal: –100° to +400°F [–75° to +205°C] Special or Short-Term: –320° to +500°F [–195° to +260°C]	±2%	±2000 ±1800 ±1500	10^5 10^6 10^7
SA	Fully encapsulated constantan gages with solder dots. Same matrix as WA Series. Same uses as WA Series but derated somewhat in maximum temperature and operating environment because of solder dots.	Normal: –100° to +400°F [–75° to +205°C] Special or Short-Term: –320° to +450°F [–195° to +230°C]	±2%	±1800 ±1500	10^6 10^7
EP	Specially annealed constantan foil with tough, high-elongation polyimide backing. Used primarily for measurements of large post-yield strains. Available with Options E, L, and LE (may restrict elongation capability).	–100° to +400°F [–75° to +205°C]	±10% for gage lengths under 1/8 in [3.2 mm] ±20% for 1/8 in and over	±1000 EP gages show zero shift under high-cyclic strains.	10^4
ED	Isoelastic foil in combination with tough, flexible polyimide film. High gage factor and extended fatigue life excellent for dynamic measurements. Not normally used in static measurements due to very high thermal-output characteristics.	Dynamic: –320° to +400°F [–195° to +205°C]	±2% Nonlinear at strain levels over ±0.5%	±2500 ±2200	10^6 10^7
WD	Fully encapsulated isoelastic gages with high-endurance leadwires. Used in wide-range dynamic strain measurement applications in severe environments.	Dynamic: –320° to +500°F [–195° to +260°C]	±1.5% — Nonlinear at strain levels over ±0.5%	±3000 ±2500 ±2200	10^6 10^7 10^8
SD	Equivalent to WD Series, but with solder dots instead of leadwires.	Dynamic: –320° to +400°F [–195° to +205°C]	±1.5% See above note	±2500 ±2200	10^6 10^7
EK	K-alloy foil in combination with a tough, flexible polyimide backing. Primarily used where a combination of higher grid resistances, stability at elevated temperature, and greatest backing flexibility are required. Supplied with Option DP.	Normal: –320° to +350°F [–195° to +175°C] Special or Short-Term: –452° to +400°F [–269° to +205°C]	±1.5%	±1800	10^7
WK	Fully encapsulated K-alloy gages with high-endurance leadwires. Widest temperature range and most extreme environmental capability of any general-purpose gage when self-temperature compensation is required. Option W available on some patterns, but restricts both fatigue life and maximum operating temperature.	Normal: –452° to +550°F [–269° to +290°C] Special or Short-Term: –452° to +750°F [–269° to +400°C]	±1.5%	±2200 ±2000	10^6 10^7
SK	Fully encapsulated K-alloy gages with solder dots. Same uses as WK Series, but derated in maximum temperature and operating environment because of solder dots.	Normal: –452° to +450°F [–269° to +230°C] Special or Short-Term: –452° to +500°F [–269° to +260°C]	±1.5%	±2200 ±2000	10^6 10^7
S2K	K-alloy foil laminated to 0.001 in (0.025 mm) thick, high-performance polyimide backing, with a laminated polyimide overlay fully encapsulating the grid and solder tabs. Provided with large solder pads for ease of leadwire attachment.	Normal: –100° to +250°F [–75° to +120°C] Special or Short-Term: –300° to +300°F [–185° to +150°C]	±1.5%	±1800 ±1500	10^6 10^7

The performance data given here are nominal, and apply primarily to gages of 0.125-in [3-mm] gage length or larger. Refer to Gage Series/Optional Feature data sheet for more detailed description and performance specifications.

APPENDIX H: SPECIFICATION SHEET FOR PELTIER COOLER ()



400W 12V Thermoelectric Cooler Peltier Plate

Product SKU : 001540-017

Your Price: $14.99

Add to cart

Airmail $4.99

Payment Methods | Shipping Methods

This thermoelectric cooler [TEC], otherwise known as a Peltier plate freezes to icy cold in just a matter of minutes, or alternatively, reverse the polarity and heat it to boiling point.

What Can I Use It In?

- CPU's
- Picnic cooler/ice boxes
- Drinks cooler/ice boxes
- Laser diodes
- CCD cameras

How Does It Work?

Utilising electricity, a TEC has an electrified metal plate that generates a heat pump which works when it is slid in between the CPU [for example] and the heat sink to keep the CPU side cool while the heat sink side stays warm.

Benefits:

- Capable of generating electricity when one side is kept cool and heat is applied to the other
- No moving internal parts to damage when in transit
- Makes absolutely no noise and does not vibrate
- Long life
- Slim and compact
- Excellent quality
- Brand new and unused

Specifications:

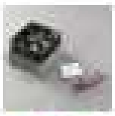
- Type: TEC1-12728
- 400W
- 12V
- Couples: 127
- Imax [A]: 28
- Vmax [V]: 15.4
- QCmax [W] 177.8
- Tmax °C: 68
- Dimensions: 50 x 50 x 3.65mm [2 x 2 x 0.15"]
- Wire length: 101mm [4"]
- Fully sealed for protection against moisture

V1.1 w.

$7.99

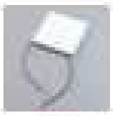
$14.99

$25.99



$24.99

$53.99



$34.99

APPENDIX I: SPECIFICATION SHEET FOR POWER SUPPLY ()

ULTRA

X3 ATX Modular Power Supply

All New X3® Modular Power Supplies

Ultra pulls out all of the stops for their latest power supply. By combining our "EE" (Energy Efficient) platform, which converts as much as 85% of AC power into DC power during typical loads, with a quiet 135MM ball bearing fan, our patented modular interface and our Limited Lifetime Warranty, Ultra clearly dominates the market with the new X3 power supply!

The X3 is super efficient, using less electricity than other power supplies when providing power to your computer and it generates less heat. Cooling is substantial yet silent by utilizing a thermostatically controlled 135MM ball bearing fan and a honeycomb cut exhaust grill. The X3 also has full range active PFC, which improves apparent power used by the computer. The modular interface allows you to only plug in the cables you actually require for your particular PC. The X3 is ready for even the most high end gaming rigs by providing up to four 6-pin PCI-e connectors, and the 800W and 1000W models provide the newest 8-pin PCI-e connector for the latest 300W graphics cards.

If the you're looking for an efficient, quiet, powerful, modular power supply with a Lifetime Warranty, there's no reason to look any farther than the Ultra X3!

Features

- Supports AMD and Intel Motherboards
- Low Acoustic Noise
- 135mm Fan
- Meets ATX Version 2.2, v1.3, and ATX 12V Version 2.2 Specifications
- Short Circuit Protection
- In-Rush Current Protection
- Thermal Overload Cutoff Protection
- MTBF > 100,000 Hours at Full Load, 110VAC and 25°C Ambient Condition
- FCC and UL Recognized

Modular Cabling Technology

FLEXFORCE

What is FlexForce?

- **Manageable** - FlexForce Cables are designed to be easily routed and hidden within your case.
- **Increased Airflow** - Special design promotes easy cable management and 30% better airflow than ordinary cables by easily routing cable under the motherboard, behind drive bays, etc.
- **Design** - Cables are dressed in a sleek looking titanium like finish.

OUTPUT **1600 Watt**

	+5V	+3.3V	+12V	-12V	+5VSB
max output current	28A	24A	117A	0.5A	5A
	180 W		1404 W	6 W	30 W
max combined wattage	1584 W			30 W	
	1600 W				

Total Output Power:
1600 W (Full Load, Nominal Input Voltage)
80% Efficiency at Typical Load

Input:
AC Input Voltage: 115V/230V
AC Input Frequency: 50-60Hz

AC Input Currents:
- 13A (RMS) for 115VAC input
- 8A (RMS) for 230VAC input

1000 Watt Energy Efficient ATX Power Supply
Part #: ULT40070
UPC#: 0227694000709
Weight: 7 lbs. Dimensions: 9.50X5.75x3.25"

ULTRA
1-800-830-0051
www.ultraproducts.com

ULTRA

APPENDIX J: SPECIFICATION SHEET FOR OPERATIONAL AMPLIFIER ()

Refer to attached file

APPENDIX K: MOTORS

Appendix K-1: Maximum Horsepower Ratings for 110V AC Motor

Motor Supplier	**Galco**	**Galco**	**Applied**	**Grangier**
Brand	WEG	WEG	Baldor	Leeson
Model	00336OS1BG56	182TCDR7001	L1408T-50	113905.00
Horsepower (Hp)	3	3	3	2
Voltage (V)	115	115	110	110
RPM	3600	3600	1500	2850
Phase	1	1	1	1
Service Factor	1.15	1.15	1.15	1.15
Weight (lb)	53	57	89	
Cost ($)	243.44	354.87	551.54	315

Appendix K-2: Motor Comparison between single and three-phase motors

Brand	**WEG**	**WEG**	**Baldor**	**Baldor**
Phase	**1**	**3**	**1**	**3**
Model	00536OS1D184T	00536OP3E182T	L1409T	VM3212
Horsepower (Hp)	5	5	5	5
Voltage (V)	230	208-230/460	208-230	208-230/460
RPM	3490	3480	3450	3450
Rated Torque (lb-ft)	4.45	7.44	7.6	7.71
Service Factor	1.15	1.15	1.15	1.15
Weight (lb)	42	57	76	56
Cost ($)	441.78	354.87	572.17	346.09

Appendix K-3: Motor Option 1 – WEG 00536OP3E182T
5Hp, 3450RPM

No.

Date 10/23/2009

DATA SHEET

Three-phase induction motor - Squirrel cage rotor

Customer	:	
Product line	:	W21 Severe Duty and General Purpose - High Efficiency - Three-Phase . ODP (IP21/IP23)
Frame	:	182T
Output	:	5 HP
Frequency	:	60 Hz
Poles	:	2
Full load speed	:	3480
Slip	:	3.33 %
Voltage	:	208-230/460 V
Full load current	:	13.5-12.2/6.10 A
Locked rotor current	:	86.6/43.3 A
Locked rotor current (Il/In)	:	7.1
No-load current	:	4.60/2.30 A
Full load torque	:	7.44 lb.ft
Locked rotor torque	:	200 %
Breakdown torque	:	290 %
Design	:	B
Insulation class	:	F
Temperature rise	:	80 K
Locked rotor time	:	12 s (hot)
Service factor	:	1.15
Duty cycle	:	S1
Ambient temperature	:	-20°C - +40°C
Altitude	:	1000 m
Degree of Protection	:	IP21
Approximate weight	:	67 lb
Moment of inertia	:	0.1279 sq.ft.lb
Noise level	:	---

	D.E.	N.D.E.	Load	Power factor	Efficiency (%)
Bearings	6206 ZZ	6205 ZZ	100%	0.88	86.5
Regreasing interval	---	---	75%	0.82	86.5
Grease amount	---	---	50%	0.73	85.5

Notes:

Performed by:

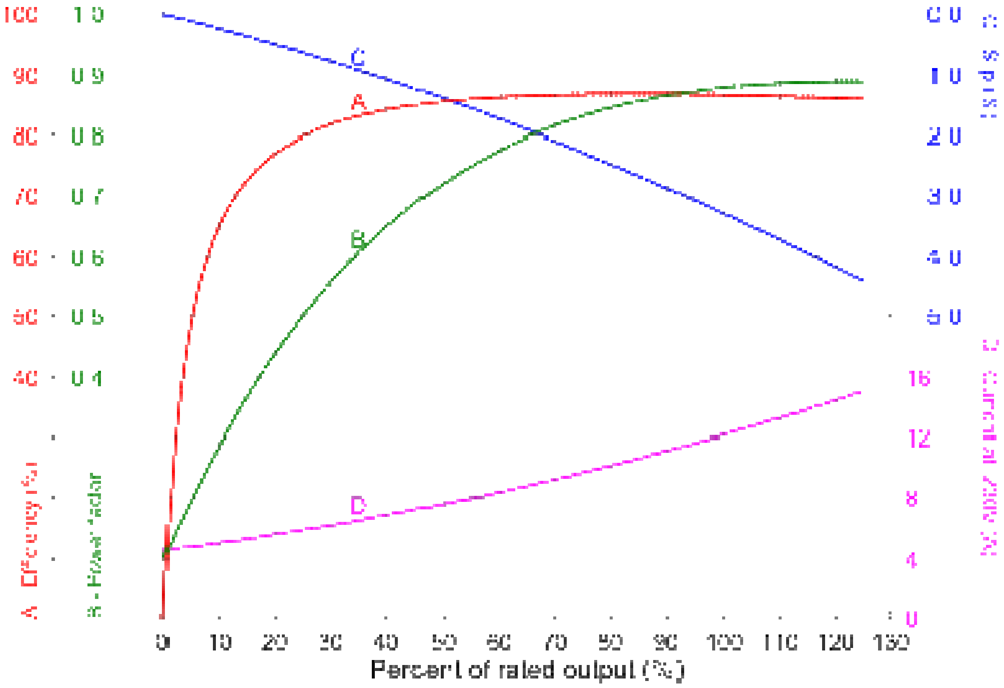
Checked

No.

Date 10/23/2009

PERFORMANCE CURVES RELATED TO RATED OUTPUT

Three-phase induction motor - Squirrel cage rotor



Customer	:
Product line	: W21 Severe Duty and General Purpose - High Efficiency - Three-Phase - ODP (IP21/IP23)

Output	: 5 HP	Locked rotor current (Il/In)	: 7.1
Frame	: 182T	Duty cycle	: S1
Full load speed	: 3480	Service factor	: 1.15
Frequency	: 60 Hz	Design	: B
Voltage	: 208-230/460 V	Locked rotor torque	: 200 %
Insulation class	: F	Breakdown torque	: 290 %
Full load current	: 13.5-12.2/6.10 A		

Notes:

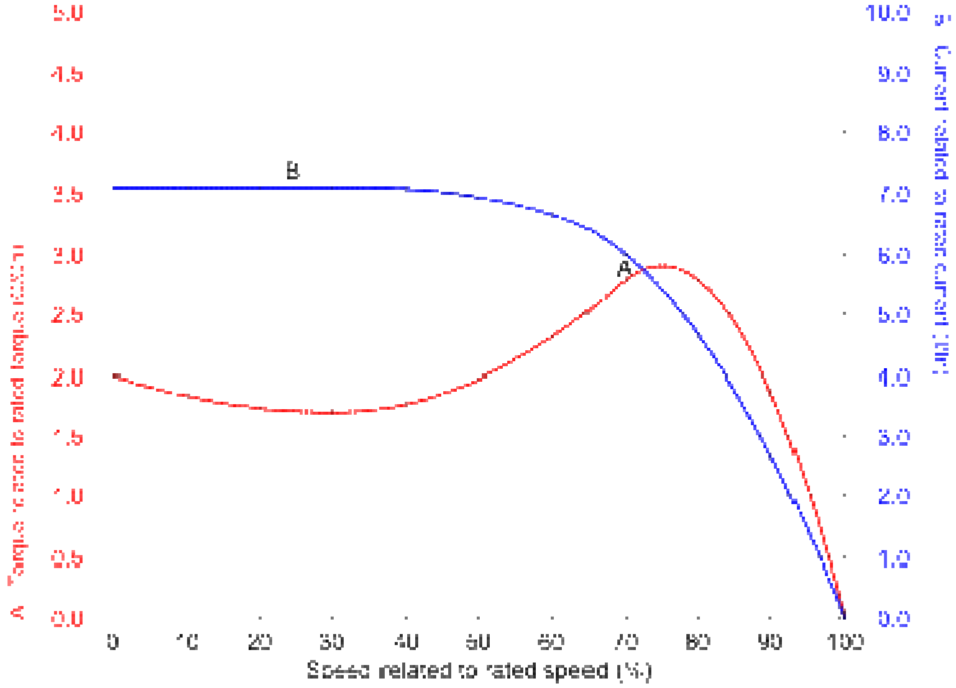
Performed by: Checked:

No.

Date : 10/23/2009

CHARACTERISTIC CURVES RELATED TO SPEED

Three-phase induction motor - Squirrel cage rotor



Customer	:				
Product line	:	W21 Severe Duty and General Purpose - High Efficiency - Three-Phase - ODP (IP21/IP23)			
Output	:	5 HP	Locked rotor current (Il/In)	:	7.1
Frame	:	182T	Duty cycle	:	S1
Full load speed	:	3480	Service factor	:	1.15
Frequency	:	60 Hz	Design	:	B
Voltage	:	208-230/460 V	Locked rotor torque	:	200 %
Insulation class	:	F	Breakdown torque	:	290 %
Full load current	:	13.5-12.2/6.10 A			

Notes:

Performed by: Checked:

Appendix K-4: Motor Option 2 – WEG 00518OP3E184T
5Hp, 1755RPM

No.

Date 10/23/2009

DATA SHEET

Three-phase induction motor - Squirrel cage rotor

Customer	:
Product line	: W21 Severe Duty and General Purpose - High Efficiency - Three-Phase . ODP (IP21/IP23)
Frame	: 184T
Output	: 5 HP
Frequency	: 60 Hz
Poles	: 4
Full load speed	: 1755
Slip	: 2.50 %
Voltage	: 208-230/460 V
Full load current	: 14.1-12.8/6.39 A
Locked rotor current	: 90.7/45.4 A
Locked rotor current (Il/In)	: 7.1
No-load current	: 6.00/3.00 A
Full load torque	: 14.8 lb.ft
Locked rotor torque	: 210 %
Breakdown torque	: 300 %
Design	: B
Insulation class	: F
Temperature rise	: 80 K
Locked rotor time	: 12 s (hot)
Service factor	: 1.15
Duty cycle	: S1
Ambient temperature	: -20°C - +40°C
Altitude	: 1000 m
Degree of Protection	: IP21
Approximate weight	: 69 lb
Moment of inertia	: 0.31774 sq.ft.lb
Noise level	: ---

	D.E.	N.D.E.	Load	Power factor	Efficiency (%)
Bearings	6206 ZZ	6205 ZZ	100%	0.83	87.5
Regreasing interval	---	---	75%	0.77	86.5
Grease amount	---	---	50%	0.66	84.0

Notes:

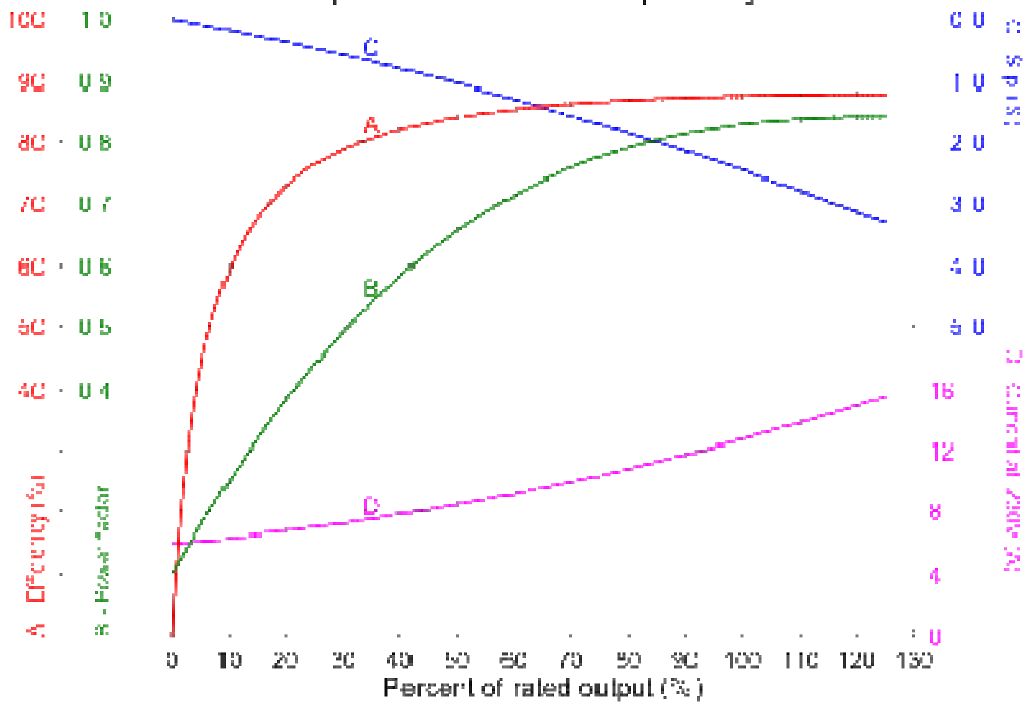
Performed by: Checked

No.

Date 10/23/2009

PERFORMANCE CURVES RELATED TO RATED OUTPUT

Three-phase induction motor - Squirrel cage rotor



Customer	:			
Product line	:	W21 Severe Duty and General Purpose - High Efficiency - Three-Phase - ODP (IP21/IP23)		
Output	:	5 HP	Locked rotor current (Il/In)	: 7.1
Frame	:	184T	Duty cycle	: S1
Full load speed	:	1755	Service factor	: 1.15
Frequency	:	60 Hz	Design	: B
Voltage	:	208-230/460 V	Locked rotor torque	: 210 %
Insulation class	:	F	Breakdown torque	: 300 %
Full load current	:	14.1-12.8/6.39 A		

Notes:

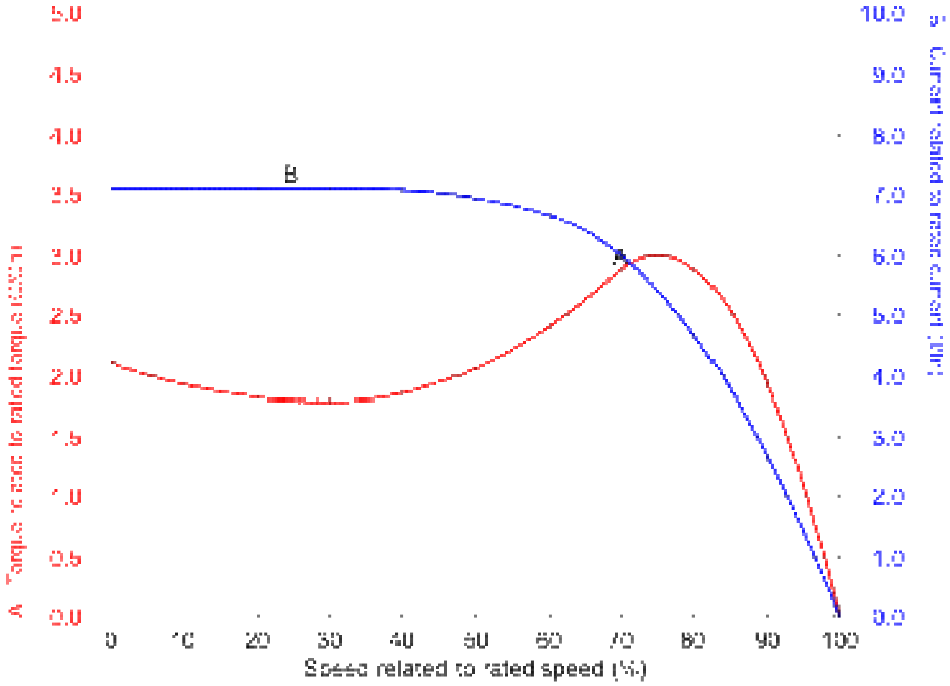
Performed by: Checked

No.

Date 10/23/2009

CHARACTERISTIC CURVES RELATED TO SPEED

Three-phase induction motor - Squirrel cage rotor



Customer	:				
Product line	:	W21 Severe Duty and General Purpose - High Efficiency - Three-Phase - ODP (IP21/IP23)			
Output	:	5 HP	Locked rotor current (Il/In)	:	7.1
Frame	:	184T	Duty cycle	:	S1
Full load speed	:	1755	Service factor	:	1.15
Frequency	:	60 Hz	Design	:	B
Voltage	:	208-230/460 V	Locked rotor torque	:	210 %
Insulation class	:	F	Breakdown torque	:	300 %
Full load current	:	14.1-12.8/6.39 A			

Notes:

Performed by: Checked:

Appendix K-5: Motor Option 3 – Baldor M2504T
5Hp, 850RPM

BALDOR • RELIANCE   Product Information Packet: M2504T - 5HP,850RPM,3PH,60HZ,254T,3734M,OPSB,F1

Product Detail							
Revision:	N	**Status:**	PRD/A	**Change #:**		**Proprietary:**	No
Type:	AC	**Prod Type:**	3734M	**Elec. Spec:**	37WGX971	**CD Diagram:**	CD0005
Enclosure	OPSB	**Mfg Plant:**		**Mech Spec:**	37F599	**Layout:**	37LYF599
Frame:	254T	**Mounting:**	F1	**Poles:**	08	**Created Date:**	
Base:	RG	**Rotation:**	R	**Insulation:**	B	**Eff. Date:**	11-20-2007
Leads:	9#14	**Literature:**		**Elec. Diagram:**		**Replaced By:**	

Nameplate NP1256L							
CAT.NO.	M2504T						
SPEC.	37F599X971						
HP	5						
VOLTS	230/460						
AMP	19.6/9.8						
RPM	850						
FRAME	254T	**HZ**	60	**PH**	3		
SER.F.	1.15	**CODE**	J	**DES**	B	**CLASS**	B
NEMA-NOM-EFF	77	**PF**	62				
RATING	40C AMB-CONT						
CC		**USABLE AT 208V**	N/A				
DE	6309	**ODE**	6206				
ENCL	OPSB	**SN**					

Performance Data at 460V, 60Hz, 5.0HP (Typical performance - Not guaranteed values)			
General Characteristics			
Full Load Torque:	30.9 LB-FT	**Start Configuration:**	DOL
No-Load Current:	7.1 Amps	**Break-Down Torque:**	114.0 LB-FT
Line-line Res. @ 25°C.:	2.39 Ohms A Ph / 0.0 Ohms B Ph	**Pull-Up Torque:**	71.0 LB-FT
Temp. Rise @ Rated Load:	80°C	**Locked-Rotor Torque:**	80.0 LB-FT
Temp. Rise @ S.F. Load:	98°C	**Starting Current:**	47.0 Amps

Load Characteristics							
% of Rated Load:	**25**	**50**	**75**	**100**	**125**	**150**	**S.F.**
Power Factor:	26.0	41.0	53.0	62.0	68.0	72.0	66.0
Efficiency:	63.7	74.8	77.5	77.2	75.3	71.9	76.1
Speed:	888.0	876.0	863.0	847.0	830.0	806.0	837.0
Line Amperes:	7.2	7.7	8.6	9.8	11.4	13.5	10.8

Appendix K-6: Motor Option 4 – Baldor CM3212T
5Hp, 3450RPM

BALDOR · RELIANCE   Product Information Packet: CM3212T - 5HP,3450RPM,3PH,60HZ,182TC,3535M,OPSB,F1

Product Detail							
Revision:	S	**Status:**	PRD/A	**Change #:**		**Proprietary:**	No
Type:	AC	**Prod Type:**	3535M	**Elec. Spec:**	35WGT481	**CD Diagram:**	CD0005
Enclosure	OPSB	**Mfg Plant:**		**Mech Spec:**	35N884	**Layout:**	35LYN884
Frame:	182TC	**Mounting:**	F1	**Poles:**	02	**Created Date:**	06-22-2007
Base:	RG	**Rotation:**	R	**Insulation:**	B	**Eff. Date:**	01-12-2009
Leads:	9#18	**Literature:**		**Elec. Diagram:**		**Replaced By:**	

Nameplate NP1256L							
CAT.NO.	CM3212T						
SPEC.	35N884T481H1						
HP	5						
VOLTS	208-230/460						
AMP	13-12.2/6.1						
RPM	3450						
FRAME	182TC	**HZ**	60	**PH**	3		
SER.F.	1.15	**CODE**	J	**DES**	B	**CLASS**	B
NEMA-NOM-EFF	85.5	**PF**	89				
RATING	40C AMB-CONT						
CC	010A	**USABLE AT 208V**	13				
DE	6206	**ODE**	6203				
ENCL	OPSB	**SN**					

Performance Data at 460V, 60Hz, 5.0HP (Typical performance - Not guaranteed values)							
General Characteristics							
Full Load Torque:		7.71 LB-FT		**Start Configuration:**		DOL	
No-Load Current:		2.47 Amps		**Break-Down Torque:**		31.7 LB-FT	
Line-line Res. @ 25°C.:		2.95 Ohms A Ph / 0.0 Ohms B Ph		**Pull-Up Torque:**		23.7 LB-FT	
Temp. Rise @ Rated Load:		55°C		**Locked-Rotor Torque:**		27.8 LB-FT	
Temp. Rise @ S.F. Load:		68°C		**Starting Current:**		53.1 Amps	
Load Characteristics							
% of Rated Load:	**25**	**50**	**75**	**100**	**125**	**150**	**S.F.**
Power Factor:	55.0	77.0	85.0	91.0	93.0	93.0	92.0
Efficiency:	80.4	86.0	86.7	86.1	84.6	82.6	85.2
Speed:	3561.0	3526.0	3488.0	3448.0	3401.0	3352.0	3420.0
Line Amperes:	2.86	3.73	4.93	6.15	7.62	9.24	7.03

Appendix K-7: Sample Gear Ratio Calculations

Using Motor Option 1 Performance Curves:

$$\text{Lowest torque occurs at 30\% of max RPM} = 1.75 \cdot \text{Rated Torque}$$
$$= 1.75 \cdot 14.8 \text{ lb-in}$$
$$= 25.9 \text{ lb-in}$$

RPM range for lowest torque: 0 to 90% of max RPM

For Four Ball Test:

$$\text{Max RPM needed} = 3600$$
$$\text{Max Torque Needed} = 4.25 \text{ lb-in}$$
$$\text{To achieve max RPM needed, gear ratio} = \frac{3600}{0.9 \cdot 1755}$$
$$= 1:2.279$$
$$\approx 1:3$$
$$\text{Minimum Torque available at } 1:3 \text{ gear ratio} = 25.9 \div 3$$
$$= 8.633 \text{ lb-in} \rightarrow \quad > 4.25 \text{ lb-in}$$

For Twist Compression Test:

$$\text{Max RPM Needed} = 30$$
$$\text{Max Torque needed} = 129.07 \text{ lb-in}$$
$$\text{To achieve max torque needed, gear ratio} = \frac{129.07}{25.9}$$
$$= 4.983:1$$
$$\approx 5:1$$
$$\text{Maximum RPM available at } 5:1 \text{ Gear Ratio} = \frac{0.9 \cdot 1755}{5}$$
$$= 315.9 \rightarrow \quad > 30$$

Appendix K-8: Motor Data Sheet

	No.:
	Date: 11/20/2009

DATA SHEET
Three-phase induction motor - Squirrel cage rotor

Customer	:
Product line	: W21 Severe Duty and General Purpose - High Efficiency - Three-Phase : W21 TEFC (IP55) - Ball Bearings

Frame	: W184T
Output	: 5 HP
Frequency	: 60 Hz
Poles	: 4
Full load speed	: 1730
Slip	: 3.89 %
Voltage	: 208-230/460 V
Full load current	: 15.0-13.6/6.80 A
Locked rotor current	: 110/55.1 A
Locked rotor current (Il/In)	: 8.1
No-load current	: 7.40/3.70 A
Full load torque	: 15.0 lb.ft
Locked rotor torque	: 320 %
Breakdown torque	: 390 %
Design	: B
Insulation class	: F
Temperature rise	: 80 K
Locked rotor time	: 8 s (hot)
Service factor	: 1.25
Duty cycle	: S1
Ambient temperature	: -20°C - +40°C
Altitude	: 1000 m
Degree of Protection	: IP55
Approximate weight	: 87 lb
Moment of inertia	: 0.26340 sq.ft.lb
Noise level	: 54 dB(A)

	D.E.	N.D.E.
Bearings	6206 ZZ	6205 ZZ
Regreasing interval	---	---
Grease amount	---	---

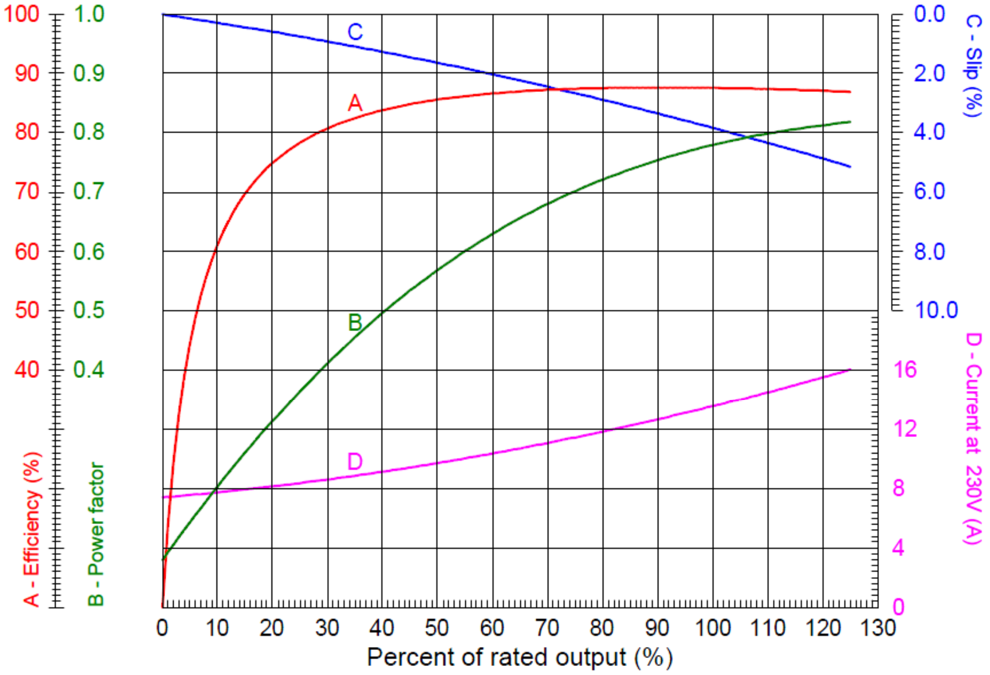
Load	Power factor	Efficiency (%)
100%	0.78	87.5
75%	0.70	87.5
50%	0.57	85.5

Appendix K-9: Performance Curves Related to Rated Output

No.:	
Date: 11/20/2009	

PERFORMANCE CURVES RELATED TO RATED OUTPUT

Three-phase induction motor - Squirrel cage rotor



Customer	:	
Product line	:	W21 Severe Duty and General Purpose - High Efficiency - Three-Phase : W21 TEFC (IP55) - Ball Bearings

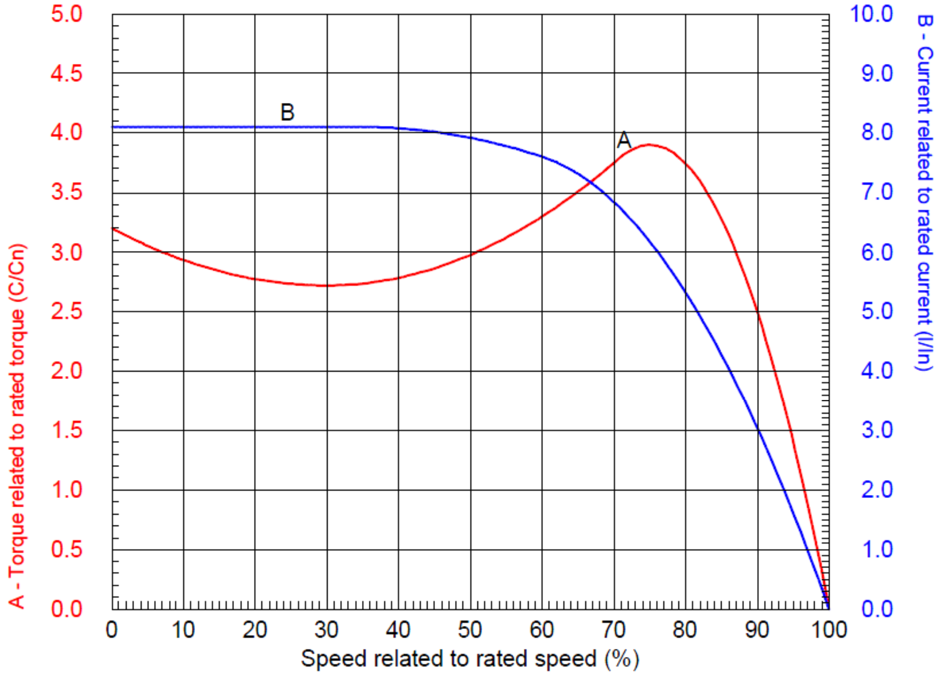
Output	:	5 HP	Locked rotor current (Il/In)	:	8.1
Frame	:	W184T	Duty cycle	:	S1
Full load speed	:	1730	Service factor	:	1.25
Frequency	:	60 Hz	Design	:	B
Voltage	:	208-230/460 V	Locked rotor torque	:	320 %
Insulation class	:	F	Breakdown torque	:	390 %
Full load current	:	15.0-13.6/6.80 A			

Appendix K-10: Characteristic curves related to speed

No.:	
Date: 11/20/2009	

CHARACTERISTIC CURVES RELATED TO SPEED

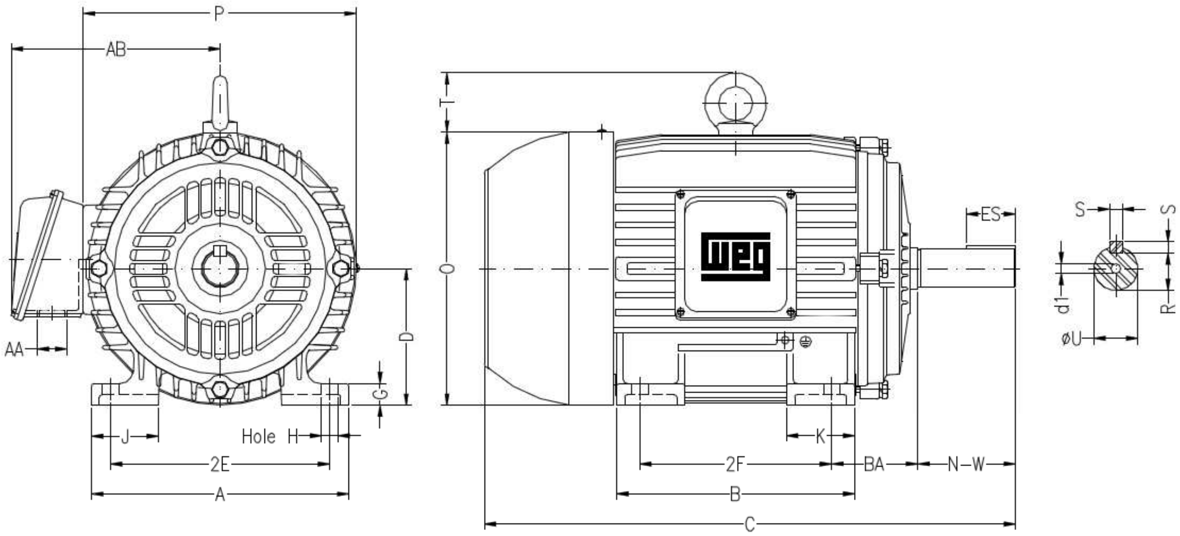
Three-phase induction motor - Squirrel cage rotor



Customer	:
Product line	: W21 Severe Duty and General Purpose - High Efficiency - Three-Phase : W21 TEFC (IP55) - Ball Bearings

Output	: 5 HP	Locked rotor current (Il/In)	: 8.1
Frame	: W184T	Duty cycle	: S1
Full load speed	: 1730	Service factor	: 1.25
Frequency	: 60 Hz	Design	: B
Voltage	: 208-230/460 V	Locked rotor torque	: 320 %
Insulation class	: F	Breakdown torque	: 390 %
Full load current	: 15.0-13.6/6.80 A		

Appendix K-11: Motor Dimensions



2E 7.500	2F 4.500/5.500	H 0.406	BA 2.750	A 8.661
B 6.969	C 15.630	D 4.500	G 0.720	J 1.890
K 2.441	O 8.909	P 7.795	T 1.772	ES 1.969
S 0.250	N-W 2.750	U 1.125	R 0.984	AB 6.614
AA NPT 1	d1 A4			

Notes:

Performed by:

Checked:

Customer:

W21 Severe Duty and General Purpose - High Efficiency - Three-Phase : W21 TEFC (IP55) - Ball Bearings

Three-phase induction motor | 11/20/2009

Frame W184T - IP55

Version

Appendix K-12: Motor drive Specifications

Item				Specification						
Three-Phase: CIMR-V☐2A				0001	0002	0004	0006	0010	0012	0020
Single-Phase: CIMR-V☐BA <1>				0001	0002	(0003)	0006	0010	0012	0018 <2>
Maximum Motor Size Allowed (HP) <3>			ND Rating	0.13	0.25	0.5/0.75	1.0/1.5	2.0/3.0	3.0	5.5 <2>
			HD Rating	0.13	0.25	0.5/0.75	0.75/1.0	1.5/2.0	3.0	5.0
Input	Input Current (A) <4>	Three Phase	ND Rating	1.1	1.9	3.9	7.3	10.8	13.9	24.0
			HD Rating	0.7	1.5	2.9	5.8	7.5	11.0	18.9
		Single Phase	ND Rating	2.0	3.6	7.3	13.8	20.2	24.0	–
			HD Rating	1.4	2.8	5.5	11.0	14.1	20.6	35.0
Output	Rated Output Capacity (kVA) <5>		ND Rating	0.5	0.7	1.3	2.3	3.7	4.6	7.5
			HD Rating	0.3	0.6	1.1	1.9	3.0	4.2	6.7
	Output Current (A)		ND Rating <6>	1.2	1.9	3.5 (3.3)	6.0	9.6	12.0	19.6
			HD Rating	0.8 <7>	1.6 <7>	3.0 <7>	5.0 <7>	8.0 <8>	11.0 <8>	17.5 <8>
	Overload Tolerance			ND Rating: 120% of rated output current for 1 minute HD Rating: 150% of rated output current for 1 minute (Derating may be required for applications that start and stop frequently)						
	Carrier Frequency			2 kHz (user-set, 2 to 15 kHz)						
	Max Output Voltage (V)			Three-phase power: Three-phase 200 to 240 V Single-phase power: Three-phase 200 to 240 V (both proportional to input voltage)						
	Max Output Frequency (Hz)			400 Hz (user-adjustable)						
Power Supply	Rated Voltage Rated Frequency			Three-phase power: Three-phase 200 to 240 V 50/60 Hz Single-phase power: 200 to 240 V 50/60 Hz						
	Allowable Voltage Fluctuation			-15 to 10%						
	Allowable Frequency Fluctuation			±5%						
Harmonic Corrective Actions	DC Reactor			Optional						

<1> Drives with single-phase power supply input will output three-phase power and cannot run a single-phase motor.
<2> CIMR-V☐BA0020 only. CIMR-V☐BA0018 is available with a Heavy Duty rating only.
<3> The motor capacity (HP) refers to a NEC rated 4-pole motor. The rated output current of the drive output amps should be equal to or greater than the motor rated current.
<4> Input current rating varies depending on the power supply transformer, input reactor, wiring connections, and power supply impedance.
<5> Rated motor capacity is calculated with a rated output voltage of 230 V.
<6> Carrier frequency is set to Swing PWM. Current derating is required in order to raise the carrier frequency.
<7> Carrier frequency is set to 10 kHz. Current derating is required in order to raise the carrier frequency.
<8> Carrier frequency is set to 8 kHz. Current derating is required in order to raise the carrier frequency.

Appendix K-13: Gear Ratio Calculations for Three Horsepower Motor

Equation Section (Next)

$$\text{Power, } P = \text{Torque, } \tau \cdot \text{Speed, } \omega \quad (1)$$

$$\tau = F \cdot r \text{ where F is Force and r is radius of gear} \quad (2)$$

$$\text{Force between gears are always equal,}$$
$$\therefore \tau \propto r,\ \omega \propto \frac{1}{r} \quad (3)$$

$$\text{Full Load Torque of Motor} = 15.0 \text{ lb-ft} \quad (4)$$

$$\text{Full Load Speed of Motor} = 1730 \text{ RPM} \quad (5)$$

$$\begin{aligned}\text{From graph, max RPM at max torque} &= 0.95 \cdot 1730 \\ &= 1643.5 \text{ RPM}\end{aligned} \quad (6)$$

<u>Four-Ball Test</u>

$$\text{Max RPM for Four Ball Test} = 3600 \text{ RPM} \quad (7)$$

$$\text{Max Torque for Four Ball Test} = 4.25 \text{ lb-ft} \quad (8)$$

$$\begin{aligned}\text{To achieve max RPM, gear ratio } &= \frac{3600}{1653} \\ &= 2.17 \\ &\approx 2.5:1\end{aligned} \quad (9)$$

$$\begin{aligned}\text{RPM available for 2.5:1 gear ratio} &= 2.5 \cdot 1653 \\ &= 4132.5\end{aligned} \quad (10)$$

$$\begin{aligned}\text{Torque available for 2.5:1 gear ratio} &= 8.93/2.5 \\ &= 3.572 \text{ lb-ft}\end{aligned} \quad (11)$$

$$\text{Max Normal Load available for Four Ball Test} = 8797.42N \quad (12)$$

<u>Twist Compression Test</u>

$$\text{Max RPM for Twist Compression Test} = 30 \text{ RPM} \quad (13)$$

$$\text{Max Torque for Twist Compression Test} = 129.07 \text{ lb-ft} \quad (14)$$

$$\begin{aligned}\text{To achieve max torque, gear ratio} &= 129.07/8.93 \\ &= 14.45 \\ &\approx 15:1\end{aligned} \quad (15)$$

$$\text{Max Gear Ratio available} = 6:1 \quad (16)$$

$$\text{RPM available for 6:1 gear ratio} = 275.5 \text{ RPM} \quad (17)$$

$$\text{Torque available for 6:1 gear ratio} = 53.58 \quad (18)$$

Appendix K-14: Gear Ratio Calculations for Five Horsepower Motor

Equation Section (Next)

$$\text{Power, } P = \text{Torque, } \tau \cdot \text{Speed, } \omega \tag{1}$$

$$\tau = F \cdot r \text{ where F is Force and r is radius of gear} \tag{2}$$

Force between gears are always equal,

$$\therefore \tau \propto r,\ \omega \propto \frac{1}{r} \tag{3}$$

$$\text{Full Load Torque of Motor} = 8.93 \text{ lb-ft} \tag{4}$$

$$\text{Full Load Speed of Motor} = 1740 \text{ RPM} \tag{5}$$

$$\begin{aligned}\text{From graph, max RPM at max torque} &= 0.95 \cdot 1730 \\ &= 1643.5 \text{ RPM}\end{aligned} \tag{6}$$

Four-Ball Test

$$\text{Max RPM for Four Ball Test} = 3600 \text{ RPM} \tag{7}$$

$$\text{Max Torque for Four Ball Test} = 4.25 \text{ lb-ft} \tag{8}$$

$$\begin{aligned}\text{To achieve max RPM, gear ratio } &= \frac{3600}{1643.5} \\ &= 2.19 \\ &\approx 3:1\end{aligned} \tag{9}$$

$$\begin{aligned}\text{RPM available for 3:1 gear ratio} &= 3 \cdot 1643.5 \\ &= 4930.5 > 3600\end{aligned} \tag{10}$$

$$\begin{aligned}\text{Torque available for 2.5:1 gear ratio} &= 15.0/3 \\ &= 5.0 \text{ lb-ft} > 4.25 \text{ lb-ft}\end{aligned} \tag{11}$$

Twist Compression Test

$$\text{Max RPM for Twist Compression Test} = 30 \text{ RPM} \tag{12}$$

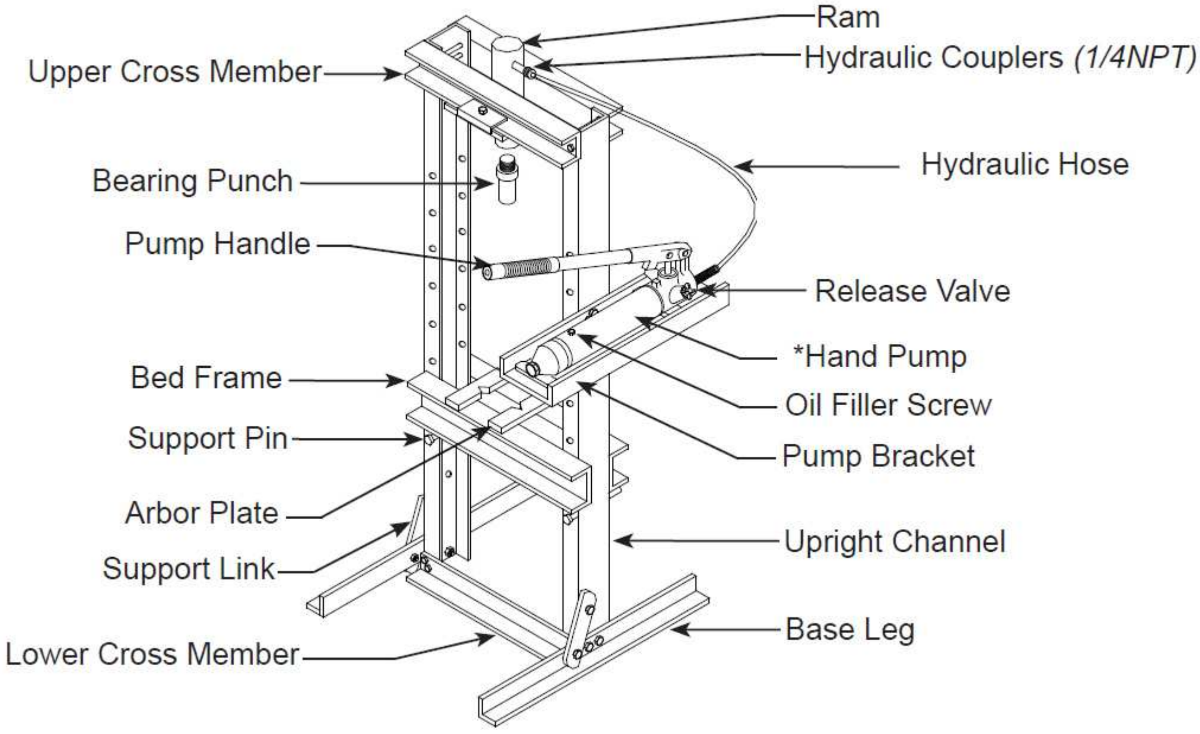
$$\text{Max Torque for Twist Compression Test} = 129.07 \text{ lb-ft} \tag{13}$$

$$\begin{aligned}\text{To achieve max torque, gear ratio} &= 129.07/15.0 \\ &= 8.60 \\ &\approx 9:1\end{aligned} \tag{14}$$

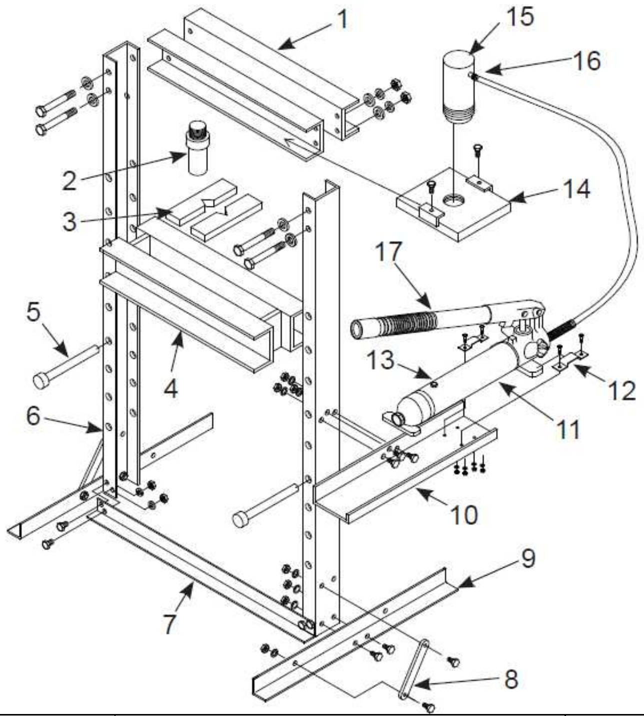
$$\text{RPM available for 9:1 gear ratio} = 182.61 \text{ RPM} > 30 \tag{15}$$

$$\begin{aligned}\text{Torque available for 9:1 gear ratio} &= 9 \cdot 15.0 \\ &= 135 \text{ lb-ft} > 129.07 \text{ lb-ft}\end{aligned} \tag{16}$$

APPENDIX L: SHOP PRESS COMPONENTS



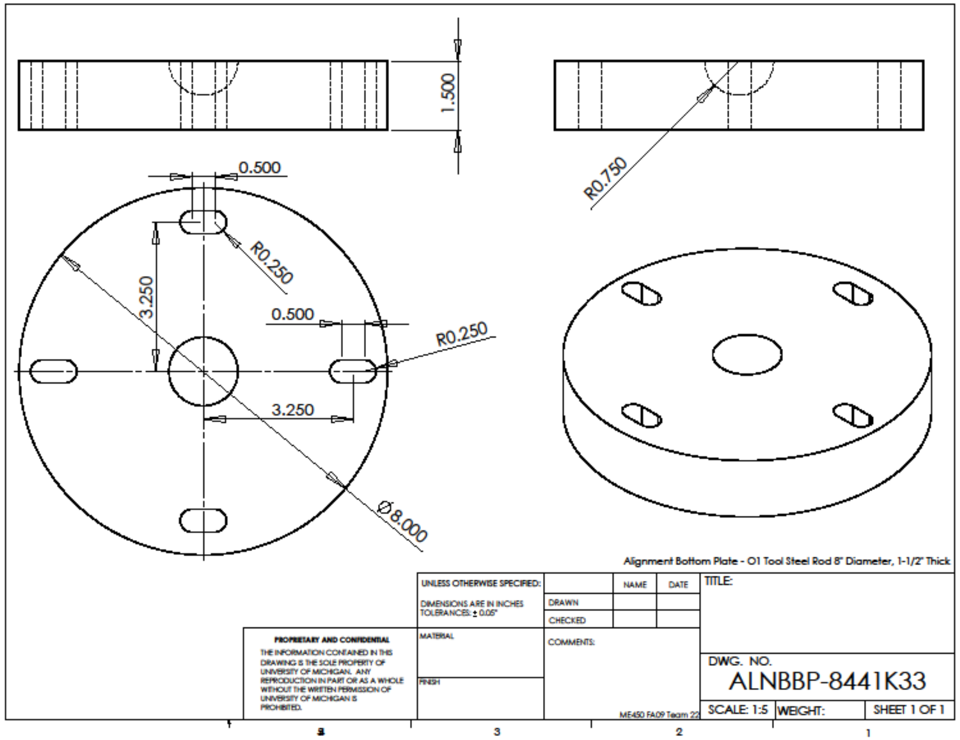
Model	Capacity	Dimensions (W x D x H)	Min. Working Space	Max. Working Space	Bed Positions	Hydraulic Stroke
60123	12 Ton	28" x 28" x 59"	4 5/8"	36 3/8"	8	6"



Item	Part#	Description	Qty
1	T184-00001-000	Upper Cross Member	2
2	T184-00008-000	Punch	1
3	T184-00007-000	Arbor Plates *(pair)*	1
4	T184-02000-000	Bed Frame	1
5	T184-01000-000	Support Pin	2
6	T184-00002-000	Upright Channel	2
7	T184-00006-000	Lower Cross Member	1
8	T184-00004-000	Support Link	2
9	T184-00005-000	Base Leg	2
10	T184-00003-000	Pump Bracket	1
11	F100-90004-K01	Hand Pump	1
12	T125-00008-000	Fixed Bracket	2
13	F040-90107-K02	Oil Filler Screw	1
14	T184-03000-000	Head Plate	1
15	F100-30000-000	Ram	1
16	F040-90009-K04	Coupler, Female 1/4NPT	1
	F040-90009-K05	Coupler, Male 1/4NPT	1
17	F100-90009-K01	Pump Handle	1

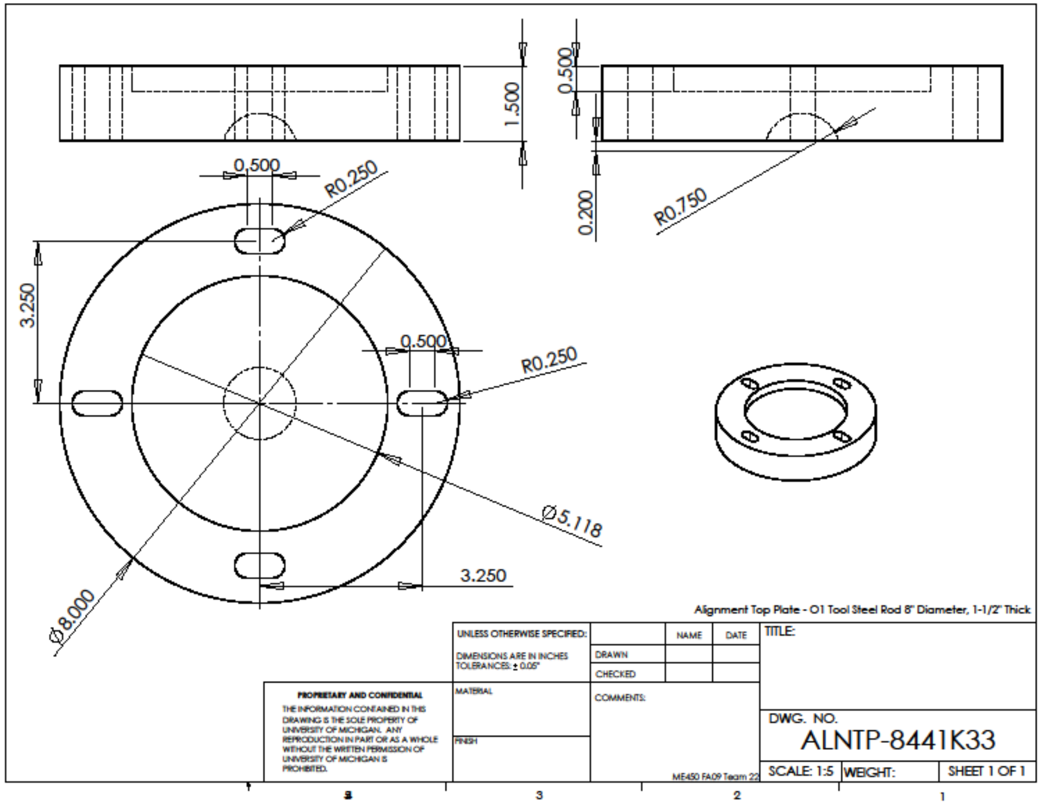
APPENDIX M: ENGINEERING DRAWINGS

Appendix M-1: Bottom Alignment Plate Drawing



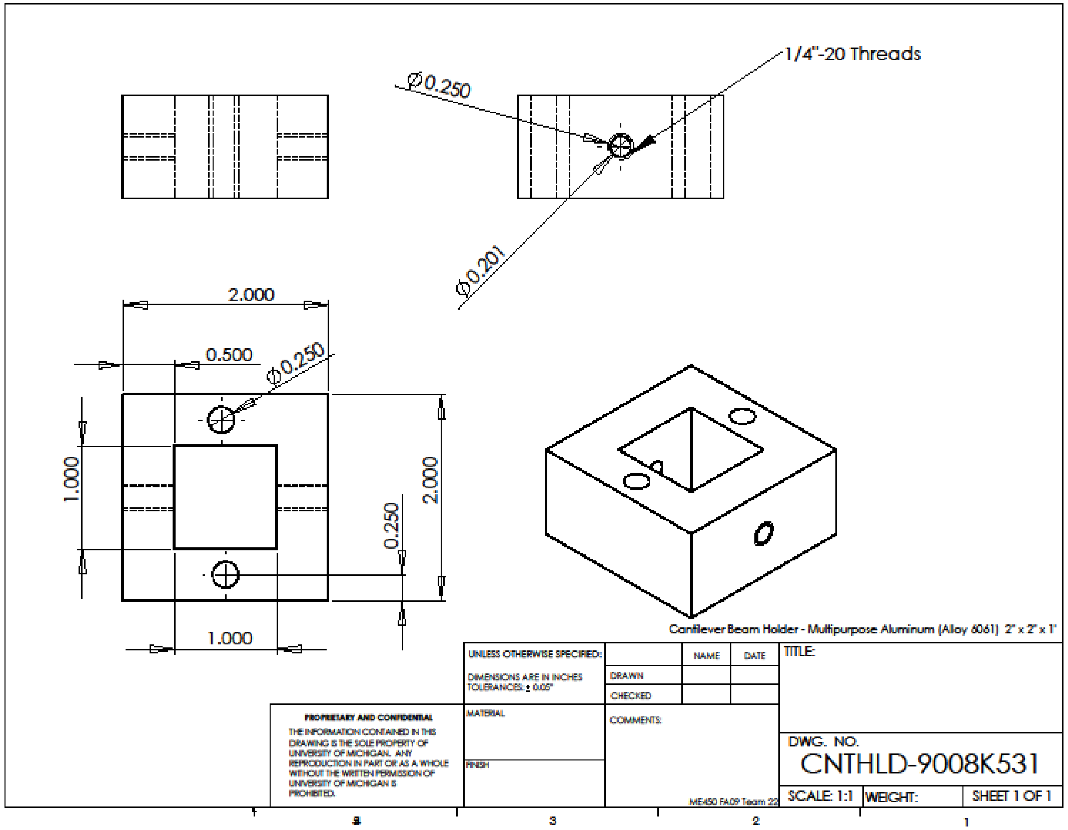
S/N	Purpose	Machine	Tool Type	Spindle Speed
1	Flatten Surfaces	Mill	HSS, 1in, End Mill	248
2	Screw Holes	Mill	HSS, 1/4"	993
3	Ball Groove	Mill	HSS, 1.5", Ball (purchased)	166
4	Bracket Screw Holes	Mill	HSS,	1799

Appendix M-2: Top Alignment Plate Drawing



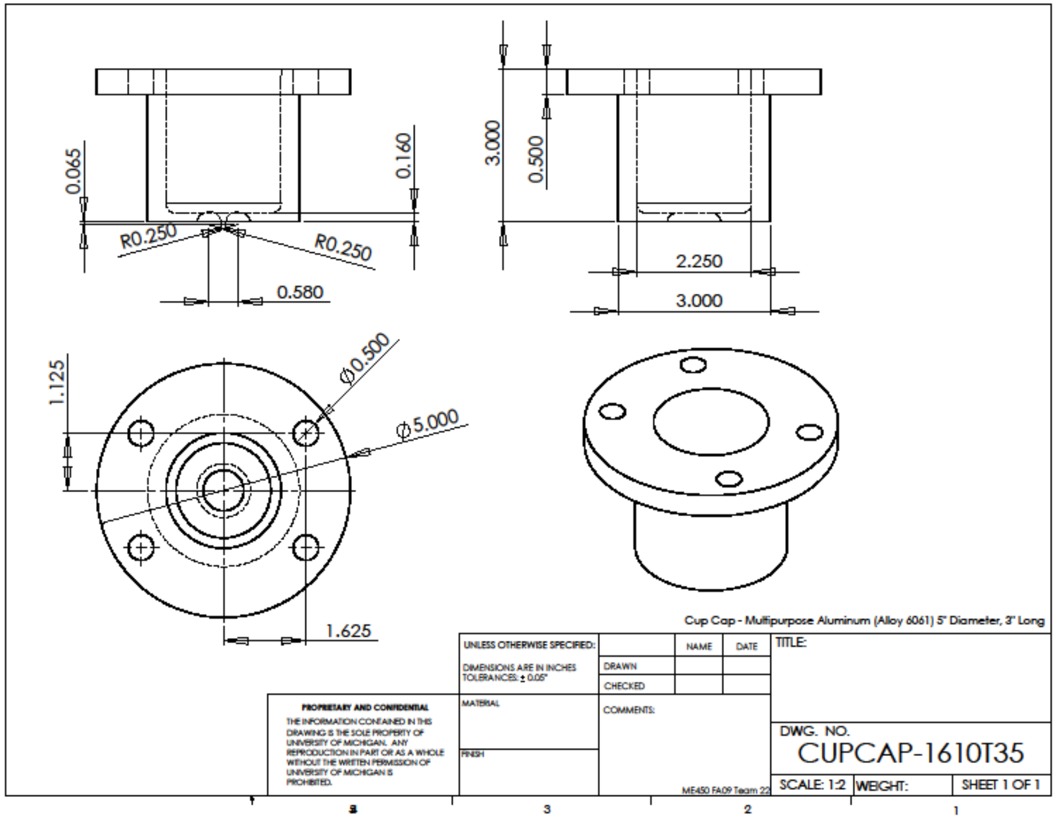
S/N	Purpose	Machine	Tool Type	Spindle Speed
1	Flatten Surfaces	Mill	HSS, 1in, End Mill	248
2	Thrust Groove	Lathe	HSS, Boring tool	97
3	Screw Holes	Mill	HSS, 1/4"	993
4	Ball Groove	Mill	HSS, 1.5", Ball (purchased)	166

Appendix M-3: Cantilever Beam Mount Drawing



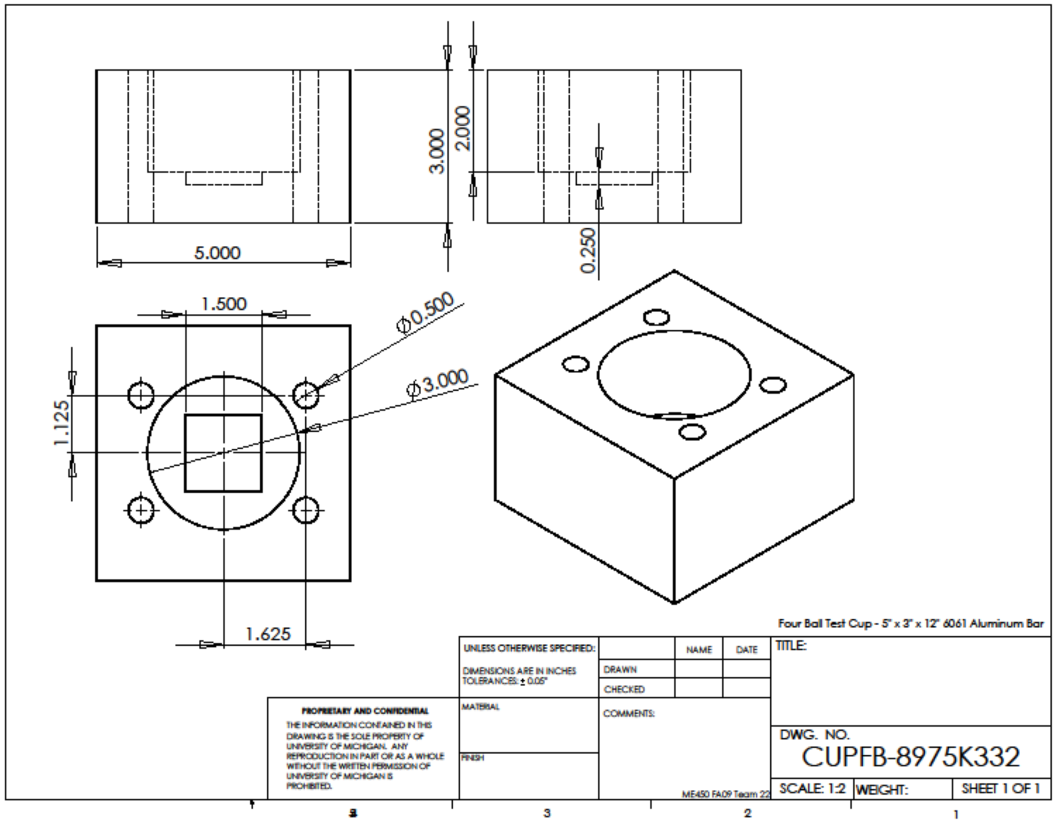
S/N	Purpose	Machine	Tool Type	Spindle Speed
1	Cut to Shape	Band Saw	-	-
2	Flatten Surfaces	Mill	HSS, 1"	630
3	Rod Holes	Drill	HSS, Drill Bit, 1/4"	630
4	Set Screw Holes	Drill	HSS, Drill Bit, 1/5"	1261
5	Set Screw threads	Tap	Threading: 1/4"-20	-

Appendix M-4: Top Alignment Plate Drawing



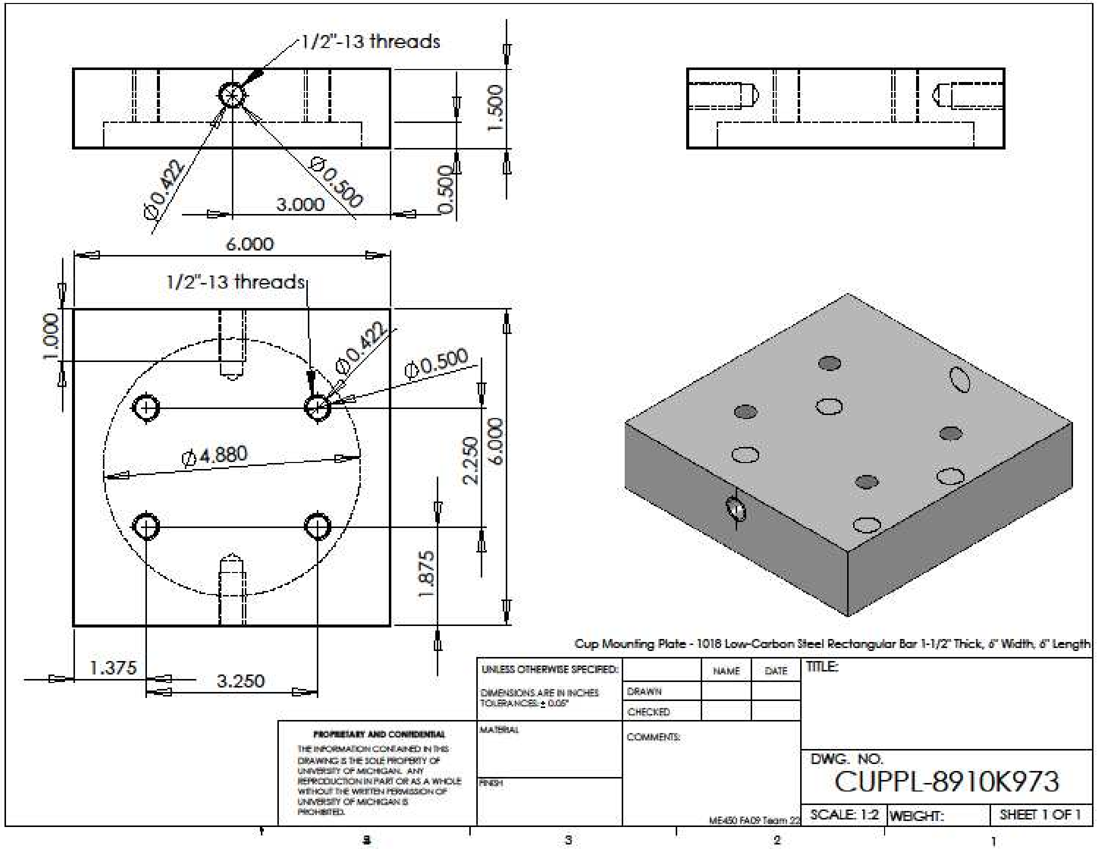
S/N	Purpose	Machine	Tool Type	Spindle Speed
1	Exterior Narrowing	Lathe	HSS, Square Nose Tool	382
2	Interior Hollow	Lathe	HSS, Boring Tool	841
3	Ball Groove	Lathe	HSS, Round Nose Tool (1/2")	761
4	Holes	Mill	HSS, (1/2")	1898

Appendix M-5: Test Cup Drawing



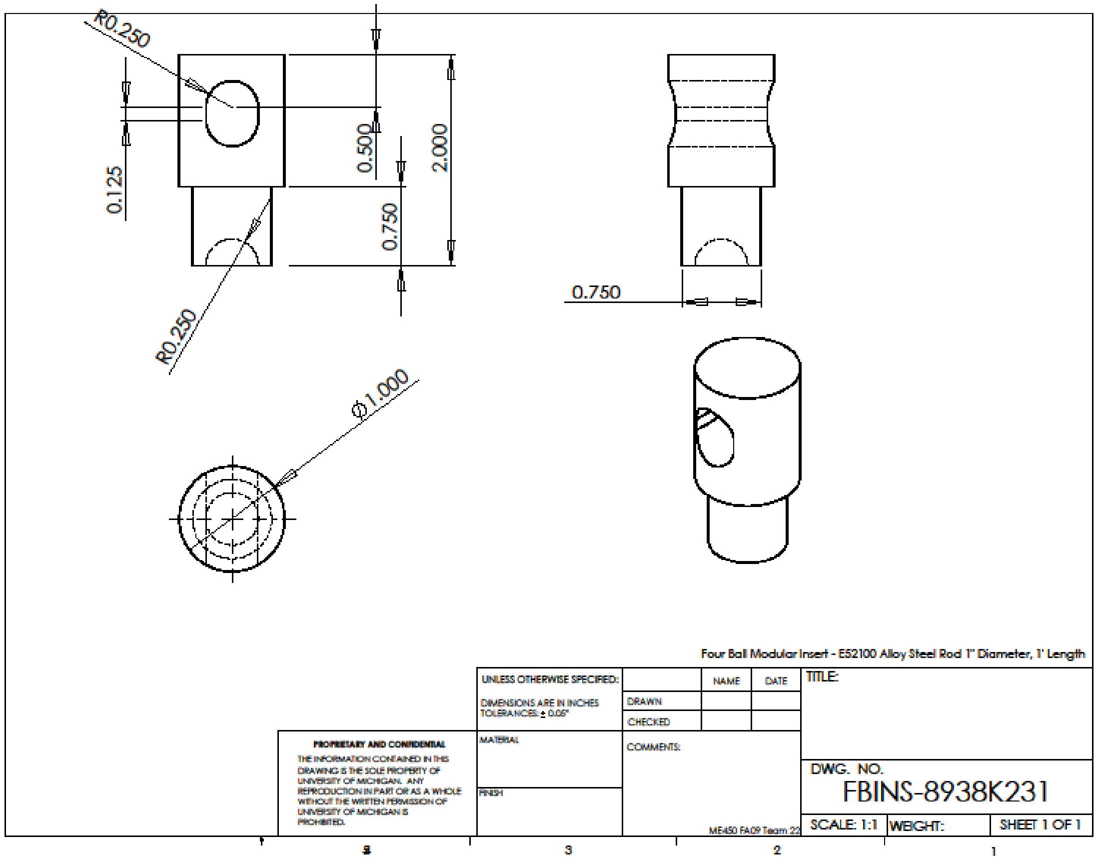
S/N	Purpose	Machine	Tool Type	Spindle Speed
1	Flatten Surfaces	Mill	HSS, 1"	630
2	Main Hole	Lathe	HSS, boring tool	637
3	Screw Holes	Mill	HSS, (1/2")	1898
4	Plate Indent	Mill	HSS, 1/8", End Mill	5000

Appendix M-6: Cup Plate Drawing



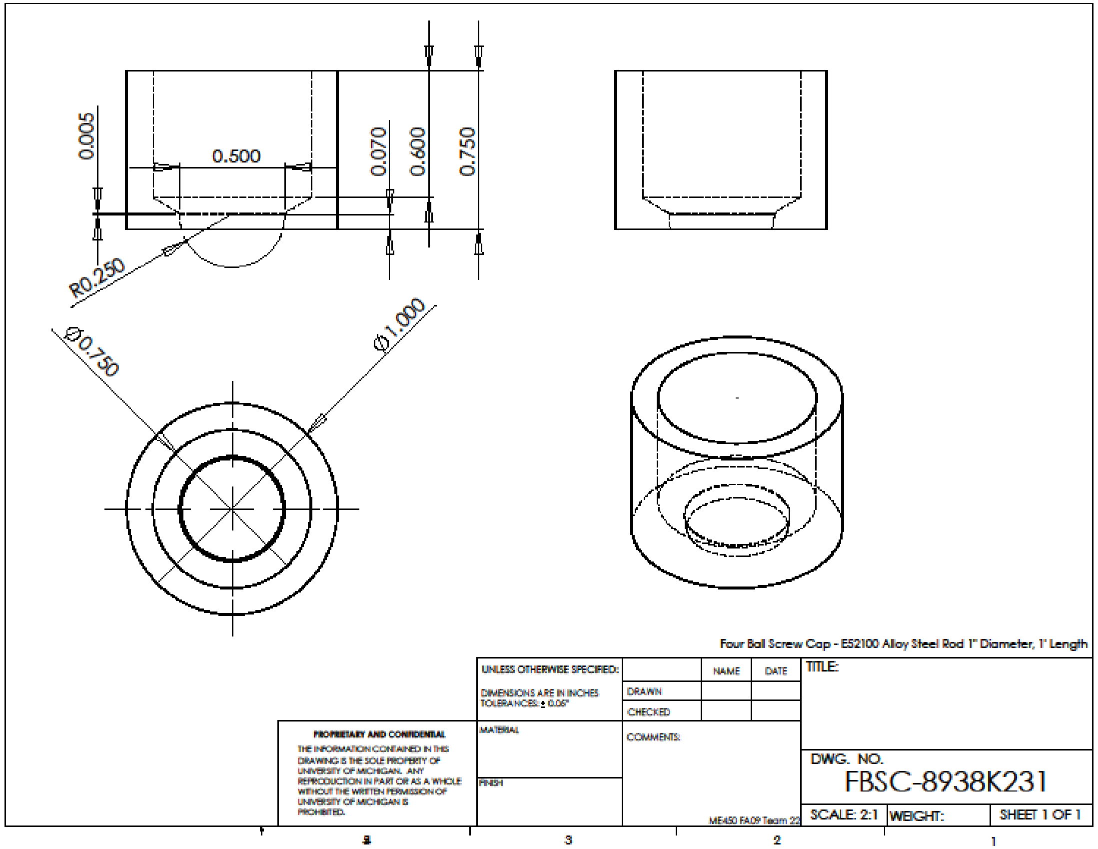
S/N	Purpose	Machine	Tool Type	Spindle Speed
1	Flatten Surfaces	Mill	HSS, 1in, End Mill	248
2	Thrust Groove	Lathe	HSS, Boring Tool	97
3	Torsion Rod Hole	Mill	HSS, 0.422"	567
4	Torsion Rod Hole	Tap	Threading: 1/2"-13	-
5	Screw Holes	Mill	HSS, 0.422"	722
6	Screw Threads	Tap	Threading: 1/2"-13	-

Appendix M-7: Four-Ball Modular Piece Drawing



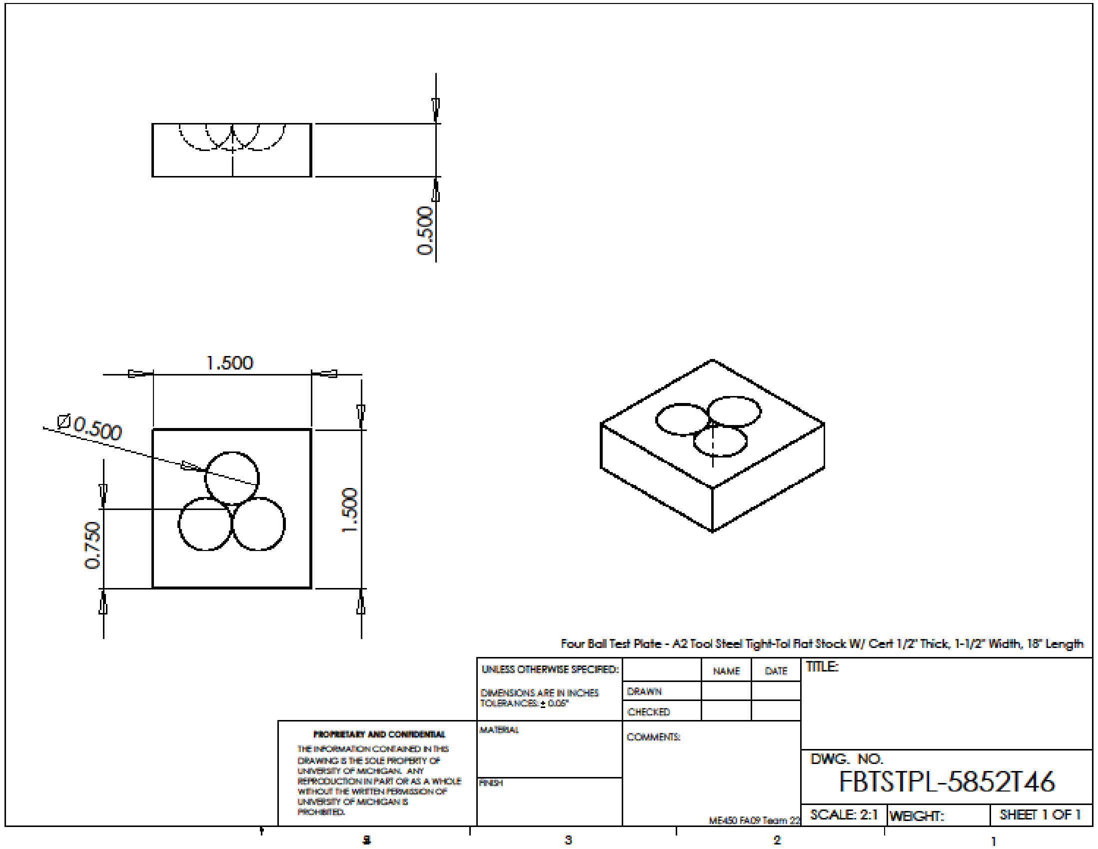
S/N	Purpose	Machine	Tool Type	Spindle Speed
1	Reduce OD for cap	Lathe	Carbide, Round Nose Tool Bit	2254
2	Threads for cap	Lathe	Carbide, Threading Tool Bit (3/4"-16)	60
3	Ball Groove	Lathe	Carbide, Drill Bit (1/2")	4507
4	Securing Slot	Mill	Carbide, 1/2"	382

Appendix M-8: Screw Cap Drawing



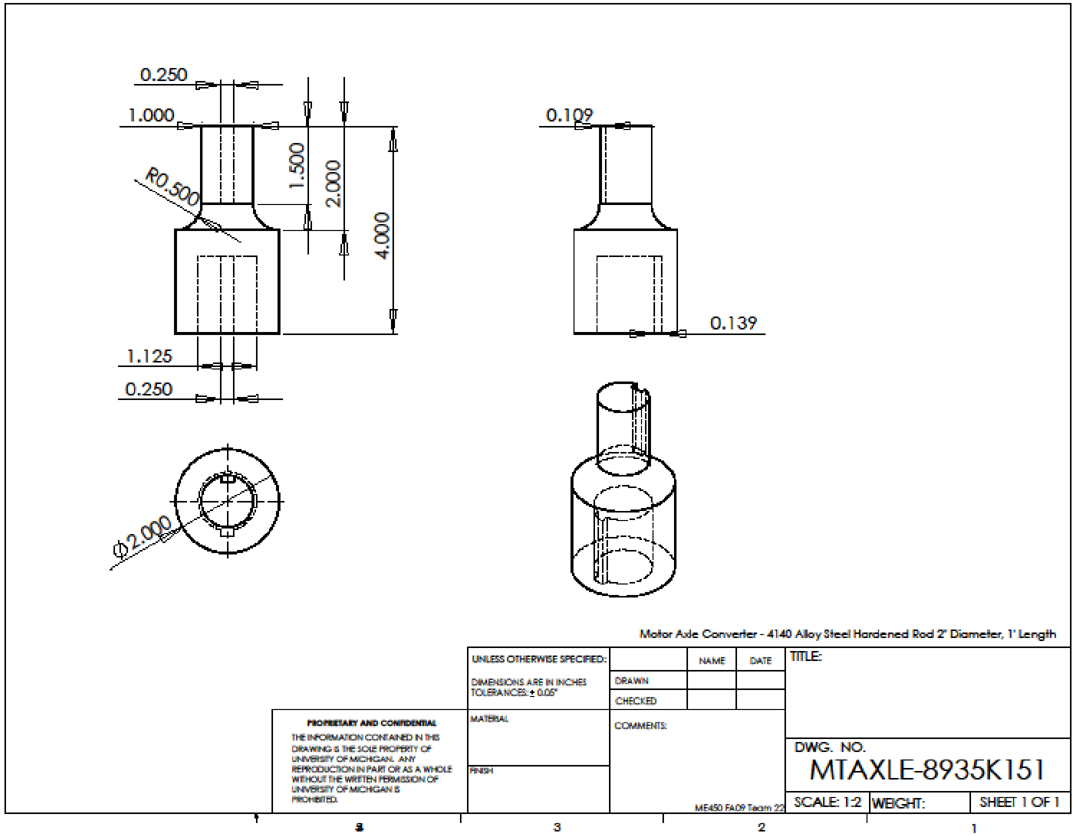
S/N	Purpose	Machine	Tool Type	Spindle Speed
1	Inner Hole	Lathe	Carbide, Boring Tool	3005
2	Inner Thread	Lathe	Carbide, Threading Tool Bit (3/4"-16)	60
3	Through Hole	Lathe	Carbide, 1/2"	4507

Appendix M-9: Four-Ball Test Plate Drawing



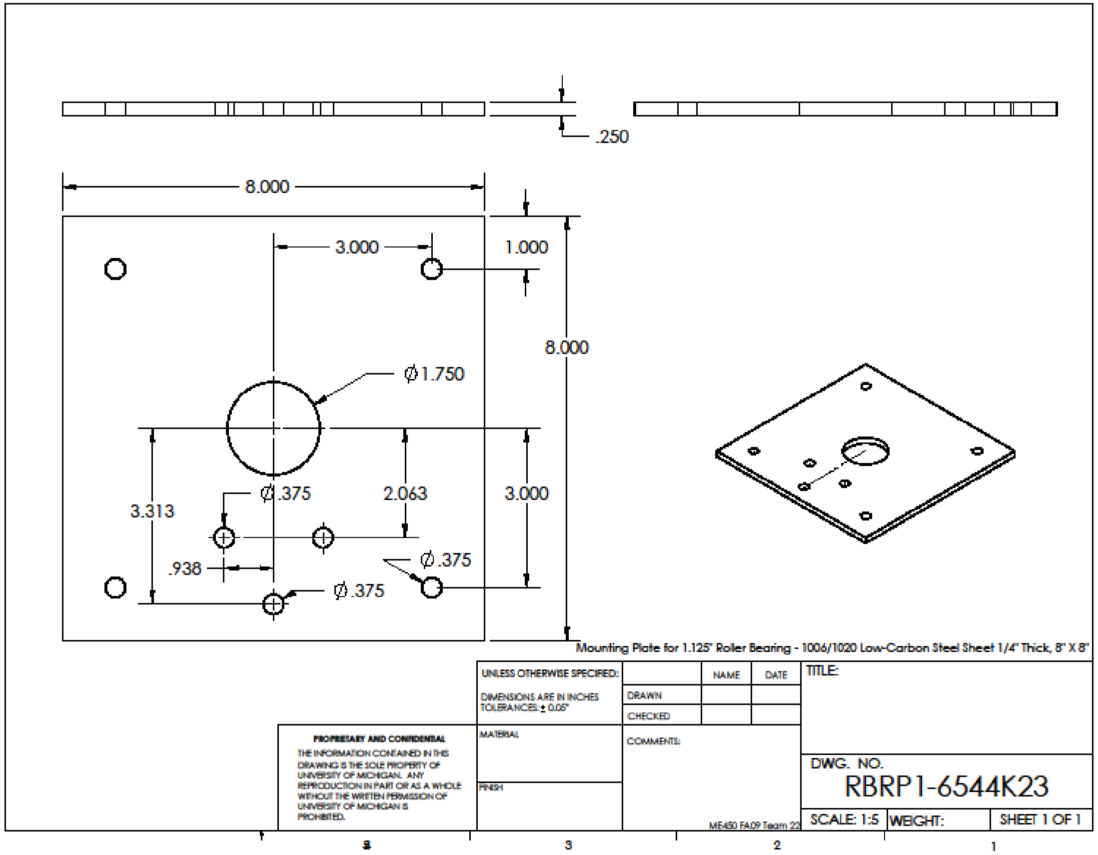
S/N	Purpose	Machine	Tool Type	Spindle Speed
1	Flatten Surfaces	Mill	Carbide, 1in, End Mill	248
2	Cut Grooves	Mill	Carbide, Ball mill (0.5")	382

Appendix M-10: Motor Axle Modular Piece Drawing



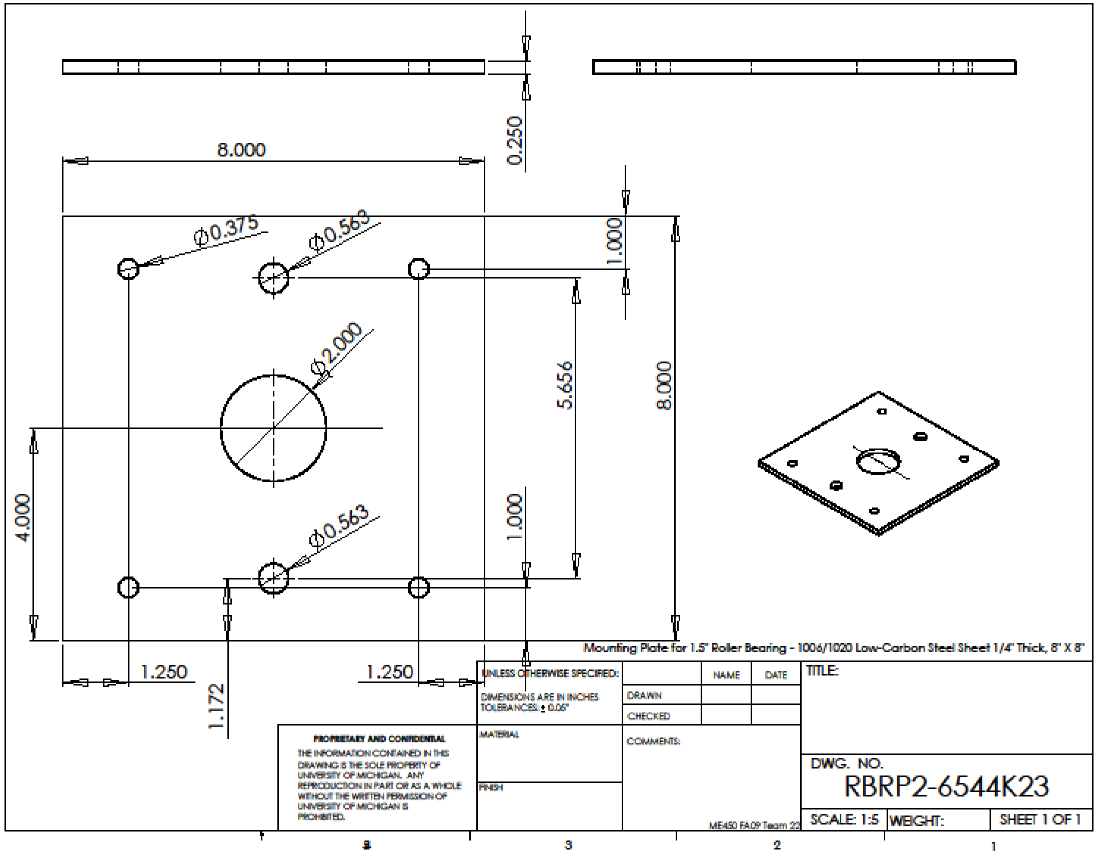
S/N	Purpose	Machine	Tool Type	Spindle Speed
1	Reduce Length to 4"	Lathe	Carbide, Parting Tool	821
2	Create General Shape	Lathe	Carbide, Round Nose Tool Bit	821
3	Create External Key way	Mill	Carbide, Flat Mill 1/4"	535
4	Create Shaft Hole	Lathe	Carbide, Drill Bit 1"	1642
5	Create Internal Key Way	Reciprocating Head	Broaching Tool	-

Appendix M-11: Top Mounting Plate Drawing



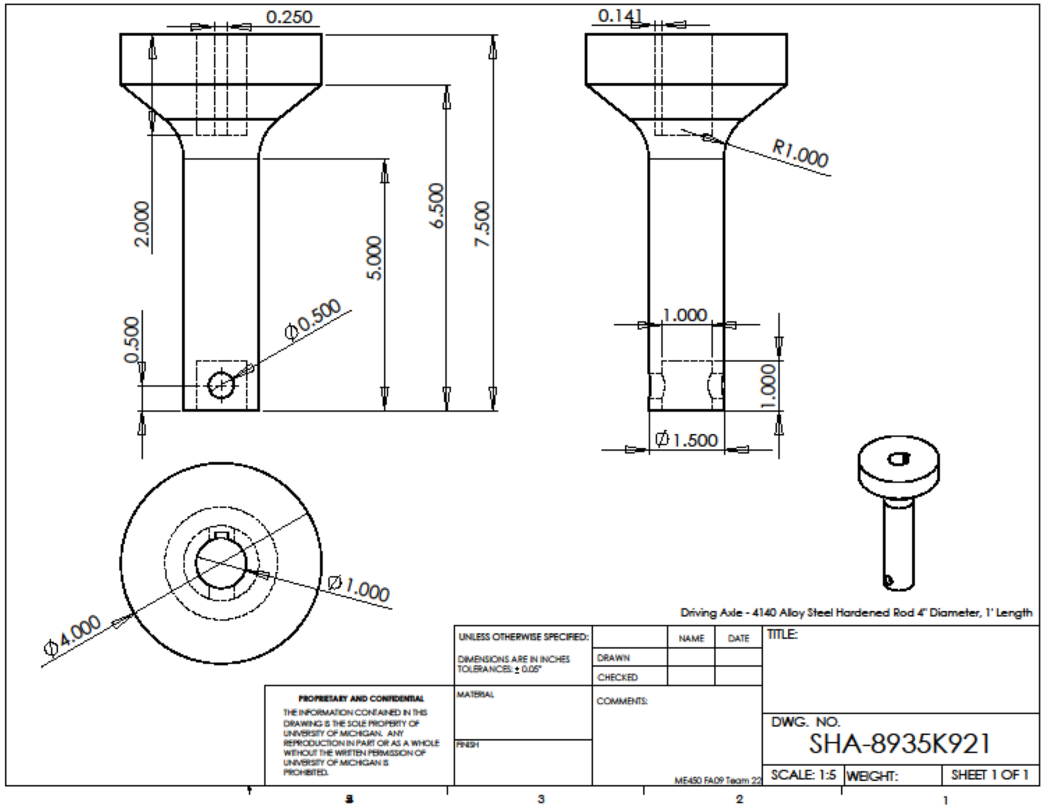
S/N	Purpose	Machine	Tool Type	Spindle Speed
1	Flatten Surfaces	Mill	HSS, 1in, End Mill	248
2	Large Center Hole	Mill	HSS, 1.5in	166
3	Thread Holes	Mill	HSS, 3/8"	662
4	Securing Thread Holes	Mill	HSS, 3/8"	662

Appendix M-12: Bottom Mounting Plate Drawing



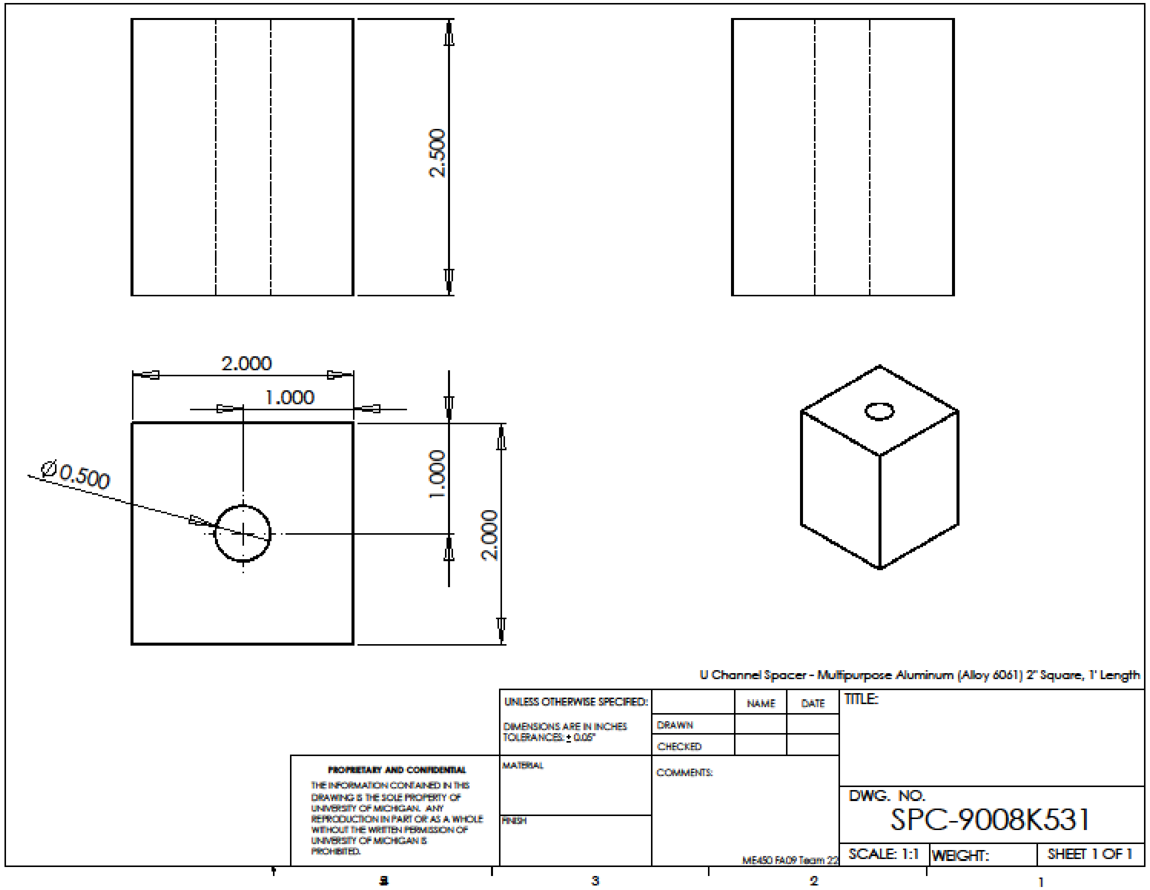
S/N	Purpose	Machine	Tool Type	Spindle Speed
1	Flatten Surfaces	Mill	HSS, 1in, End Mill	248
2	Thrust Groove	Lathe	HSS, Boring Tool	97
3	Securing Thread Holes	Mill	HSS, 3/8"	662

Appendix M-13: Driving Axle Drawing



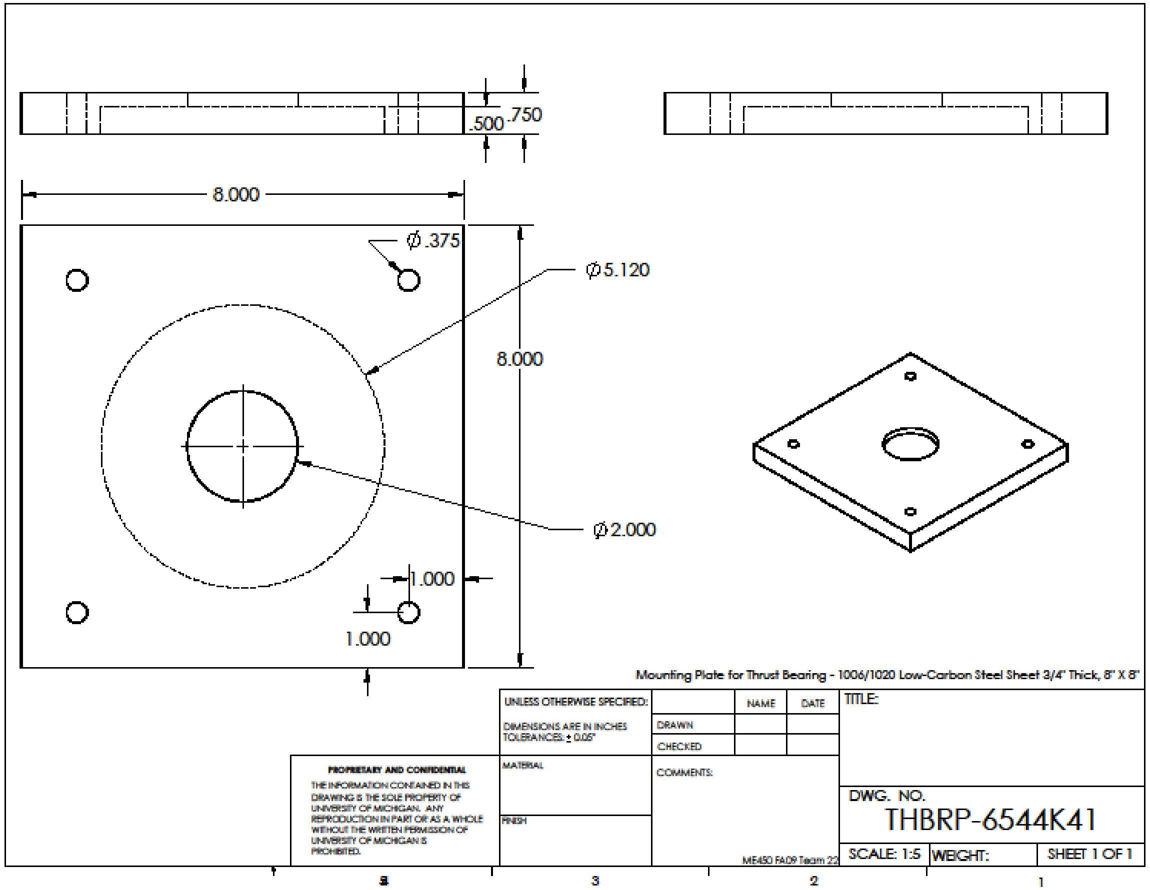
S/N	Purpose	Machine	Tool Type	Spindle Speed
1	Reduce Length to 7.5"	Lathe	Carbide, Parting Tool	410
2	Create General Shape	Lathe	Carbide, Round Nose Tool Bit	410
3	Top Hole	Lathe	Carbide, Drill Bit 9/16"	2915
4	Bottom Hole for Annulus	Lathe	Carbide, Drill Bit 1"	1643
5	Annulus Securing Hole	Drill Press	Carbide, Drill Bit 0.5"	240
6	Key Way	Reciprocating Head	Broaching Tool	-

Appendix M-14: Spacer Drawing



S/N	Purpose	Machine	Tool Type	Spindle Speed
1	Cut to Shape	Band Saw	-	-
2	Flatten Surfaces	Mill	HSS, 1"	630
3	Rod Holes	Drill	HSS, Drill Bit, 0.5"	1261

Appendix M-15: Thrust Bearing Plate Drawing



S/N	Purpose	Machine	Tool Type	Spindle Speed
1	Flatten Surfaces	Mill	HSS, 1in, End Mill	248
2	Thrust Groove	Lathe	HSS, Boring Tool	97
3	Securing Thread Holes	Mill	HSS, 3/8"	662

APPENDIX N: FAILURE MODE AND EFFECTS ANALYSIS (FMEA)
Appendix N-1: FMEA Table of Motor

Potential
Failure Mode and Effects Analysis Worksheet

FMEA Team: ME 450 Team 22	Test machine **System**	**FMEA Number:** 1A
Team Leader: Adolphus Lim	Power Transmission **Subsystem**	**FMEA Date: (Original)** 11/12/2009
	Motor **Component**	**(Revised)** -

Design FMEA									
Item and Function / Requirements	**Potential Failure Mode**	**Potential Effect(s) of Failure**	**Severity**	**Potential Cause(s)/ Mechanism(s) of Failure**	**Occurrence**	**Current Controls - Prevention**	**Current Controls - Detection**	**Detection**	**RPN**
Convert electrical energy input into mechanical rotational force	Motor does not work	Loss of use of machine	7	3-phase power supply unavailable	3	Ensure that the lab has 3-phase power supply	Go down to the lab to check	1	21
				Motor is not plugged into power supply or connection is loose	2	Ensure that power supply is plugged in before turning on the motor	Check before turning on the motor	1	14
				Motor is faulty	1	Test the motor before purchasing it	Check that motor is working	1	7
	Motor does not run at required speed	Machine does not function as desired	5	Motor is faulty	1	Test the motor before purchasing it	Check that motor is working	1	5

Appendix N-2: FMEA Table of Chain Drive

Potential
Failure Mode and Effects Analysis Worksheet

FMEA Team: ME 450 Team 22	Test machine **System**	**FMEA Number:** 1B	
Team Leader: Adolphus Lim	Power Transmission **Subsystem**	**FMEA Date: (Original)** 11/12/2009	
	Chain Drive **Component**	**(Revised)** -	

Design FMEA

Item and Function / Requirements	Potential Failure Mode	Potential Effect(s) of Failure	Severity	Potential Cause(s)/ Mechanism(s) of Failure	Occurrence	Current Controls - Prevention	Current Controls - Detection	Detection	RPN
Transmits torque from motor to drive shaft	Torque is not transmitted	Loss of use of machine	7	Chain does not grip onto gears	2	Use proper materials	Materials testing	3	42
						Specify tolerances to design standards	Analysis of critical measurements	1	14
				Chain breaks	2	Use proper materials	Materials testing	1	14
	Wrong torque is transmitted	Machine does not function as desired	5	Gear ratio is incorrect	1	Gears with correct gear ratios are mounted onto the motor axle and drive shaft	Build to design	3	15
				Chain is slipping	1	Chain and sprockets are dimensioned with appropriate tolerances correctly	Analysis of critical measurements	2	10

Appendix N-3: FMEA Table of Gear Shaft/Driving Shaft

Potential
Failure Mode and Effects Analysis Worksheet

	Test machine **System**	**FMEA Number:** 1C
FMEA Team: ME 450 Team 22	Power Transmission **Subsystem**	**FMEA Date: (Original)** 11/12/2009
Team Leader: Adolphus Lim	Gear Shaft/Driving Shaft **Component**	**(Revised)** -

Design FMEA									
Item and Function / Requirements	**Potential Failure Mode**	**Potential Effect(s) of Failure**	**Severity**	**Potential Cause(s)/ Mechanism(s) of Failure**	**Occurrence**	**Current Controls - Prevention**	**Current Controls - Detection**	**Detection**	**RPN**
Transmits torque from gear shaft to drive shaft	Torque is not transmitted	Loss of use of machine	7	Key is not securely fitted into shaft	1	Specify tolerances to design standards	Analysis of critical measurements	1	7
				Key breaks in shaft	2	Use proper materials	Materials testing	1	14
				Shaft is not vertically aligned and is prevented from rotating about its axis	2	Specify tolerances to design standards	Analysis of critical measurements	2	28
				Thrust bearings are not installed properly	2	Specify tolerances to design standards	Analysis of critical measurements	1	14
Transmits torque from drive shaft to modular test piece	Torque is not transmitted	Loss of use of machine	7	Key is not securely fitted into shaft	1	Specify tolerances to design standards	Analysis of critical measurements	1	7
				Key breaks in shaft	2	Use proper materials	Materials testing	1	14
				Shaft is not vertically aligned and is prevented from rotating about its axis	2	Specify tolerances to design standards	Analysis of critical measurements	2	28
				Thrust bearings are not installed properly	2	Specify tolerances to design standards	Analysis of critical measurements	1	14

Appendix N-4: FMEA Table of Hudraulic Jack

Potential
Failure Mode and Effects Analysis Worksheet

FMEA Team: ME 450 Team 22	Test machine **System**	**FMEA Number:** 2A	
Team Leader: Adolphus Lim	Normal Loading **Subsystem**	**FMEA Date: (Original)** 11/12/2009	
	Hydraulic Jack **Component**	**(Revised)** -	

Design FMEA									
Item and Function / Requirements	**Potential Failure Mode**	**Potential Effect(s) of Failure**	**Severity**	**Potential Cause(s)/ Mechanism(s) of Failure**	**Occurrence**	**Current Controls - Prevention**	**Current Controls - Detection**	**Detection**	**RPN**
Transfer of mechanical energy from human input to normal loading in the test	Mechanical energy is not transferred efficiently	Machine does not function as desired	5	Normal loading is not aligned perfectly vertically upwards	2	Specify tolerances to design standards	Analysis of critical measurements	2	20
				Leak of hydraulic fluid	2	Ensure that power supply is plugged in before turning on the motor	Check the hydraulic jack before using it	1	10
				Hydraulic bottle rupture	1	Build shielding so that flying debris is kept in	Check the hydraulic jack for signs of failure before using it	1	5

Appendix N-5: FMEA TABLE OF SHOP PRESS

Potential
Failure Mode and Effects Analysis Worksheet

FMEA Team: ME 450 Team 22
Team Leader: Adolphus Lim

Test machine **System**
Structure **Subsystem**
Shop Press **Component**

FMEA Number: 3A
FMEA Date: (Original) 11/12/2009
(Revised) -

Design FMEA									
Item and Function / Requirements	**Potential Failure Mode**	**Potential Effect(s) of Failure**	**Severity**	**Potential Cause(s)/ Mechanism(s) of Failure**	**Occurrence**	**Current Controls - Prevention**	**Current Controls - Detection**	**Detection**	**RPN**
Hold the test machine together	Subsystems cannot be attached to structure	Loss of use of machine	7	Parts to be assembled are not dimensioned correctly	2	Specify tolerances to design standards	Analysis of critical measurements	1	14
				Modifications to the structure cannot be made	2	Plan to attach parts by other methods that does not require modifying structure	Analysis of critical measurements	1	14
Withstand axial and torsional loads on the system	Structure unable to withstand axial loads	Structure breaks down and flies apart	10	Structure is not strong enough to withstand the axial loads	1	Purchase a commercial shop press that is rated for a greater axial load than required.	Testing after assembly	1	10
	Structure unable to withstand torsional loads	Structure breaks down and flies apart	10	Structure is not strong enough to withstand the torsional loads	1	Bolt the structure to a base plate and connect 2 stiff bars to the motor structure	Testing after assembly	1	10

Appendix N-6: FMEA Table of Motor Structure

Potential
Failure Mode and Effects Analysis Worksheet

	Test machine **System**	**FMEA Number:**	3B
FMEA Team: ME 450 Team 22	Structure **Subsystem**	**FMEA Date: (Original)**	11/12/2009
Team Leader: Adolphus Lim	Motor Structure **Component**	**(Revised)**	-

Design FMEA

Item and Function / Requirements	Potential Failure Mode	Potential Effect(s) of Failure	Severity	Potential Cause(s)/ Mechanism(s) of Failure	Occurrence	Current Controls - Prevention	Current Controls - Detection	Detection	RPN
Support the motor	Structure unable to support the weight of the motor	Injury and loss of use of machine	10	Structure is not strong enough to support the weight of the motor	1	Use proper materials	Materials testing	1	10
	Structure unable to withstand the vibrations caused by the motor	Injury and loss of use of machine	10	Structure is not strong enough to withstand the weight of the motor	1	Use proper materials. Bolt the structure to a base plate with a layer of neoprene underneath to damp the vibrations	Materials testing	1	10
Elevate the motor	Motor is elevated too much or too little	Torque not transmitted properly to the chain drive	7	Parts of motor structure are not dimensioned correctly	1	Specify tolerances to design standards	Analysis of critical measurements	1	7

APPENDIX N-7: FMEA Table of Normal Loading Mearsurement

Potential
Failure Mode and Effects Analysis Worksheet

Test machine **System**
Data Acquisition **Subsystem**
Normal Loading Measurement **Component**

FMEA Team: ME 450 Team 22
Team Leader: Adolphus Lim

FMEA Number: 4A
FMEA Date: (Original) 11/12/2009
(Revised) -

Design FMEA									
Item and Function / Requirements	**Potential Failure Mode**	**Potential Effect(s) of Failure**	**Severity**	**Potential Cause(s)/ Mechanism(s) of Failure**	**Occurrence**	**Current Controls - Prevention**	**Current Controls - Detection**	**Detection**	**RPN**
Measure normal force by measuring strain exerted on the dummy material	Strain measurement is not accurate	Test results are inaccurate	5	Strain caused by thermal expansion in the specimen material is not accounted for	3	Strain gages arranged in a Wheatstone bride circuit, with a half-bridge configuration	Measure temperature and use a correction curve to correct the data	1	15
				Lead wire resistance vary with temperature and affect strain gage readings	3	Strain gages arranged in a Wheatstone bride circuit, with a half-bridge configuration	Measure temperature and use a correction curve to correct the data	1	15
				Sensitivity to noise distorts the strain gage readings	3	Increase excitation voltage and thus, improve the signal-to-noise-ratio	Examine the zero point of the channel when no load is applied and progressively increase the excitation until instability is observed, then lower excitation until stability returns	1	15

Appendix N-8: FMEA Table of Torque Measurement

Potential
Failure Mode and Effects Analysis Worksheet

FMEA Team: ME 450 Team 22	Test machine **System**	**FMEA Number:** 4B	
Team Leader: Adolphus Lim	Data Acquisition **Subsystem**	**FMEA Date: (Original)** 11/12/2009	
	Torque Measurement **Component**	**(Revised)** -	

Design FMEA									
Item and Function / Requirements	**Potential Failure Mode**	**Potential Effect(s) of Failure**	**Severity**	**Potential Cause(s)/ Mechanism(s) of Failure**	**Occurrence**	**Current Controls - Prevention**	**Current Controls - Detection**	**Detection**	**RPN**
Measure torque by measuring strain exerted on the cantilever beam	Strain measurement is not accurate	Test results are inaccurate	5	Strain caused by thermal expansion in the specimen material is not accounted for	3	Strain gages arranged in a Wheatstone bride circuit, with a half-bridge configuration	Measure temperature and use a correction curve to correct the data	1	15
				Lead wire resistance vary with temperature and affect strain gage readings	3	Strain gages arranged in a Wheatstone bride circuit, with a half-bridge configuration	Measure temperature and use a correction curve to correct the data	1	15
				Sensitivity to noise distorts the strain gage readings	3	Increase excitation voltage and thus, improve the signal-to-noise-ratio	Examine the zero point of the channel when no load is applied and progressively increase the excitation until instability is observed, then lower excitation until stability returns	1	15

Appendix N-9: FMEA Table of Temperature Measurement

Potential
Failure Mode and Effects Analysis Worksheet

FMEA Team: ME 450 Team 22
Team Leader: Adolphus Lim

Test machine **System**
Data Acquisition **Subsystem**
Temperature Measurement **Component**

FMEA Number: 4C
FMEA Date: (Original) 11/12/2009
(Revised) -

Design FMEA									
Item and Function / Requirements	**Potential Failure Mode**	**Potential Effect(s) of Failure**	**Severity**	**Potential Cause(s)/ Mechanism(s) of Failure**	**Occurrence**	**Current Controls - Prevention**	**Current Controls - Detection**	**Detection**	**RPN**
Measure temperature by producing a voltage related to a temperature difference	Temperature measured is not accurate	Test results are inaccurate	5	1 thermocouple junction is not kept at a constant temperature	3	Cold junction compensation (CJC) provided by the DAQ	Use a lab thermometer to verify that data from thermocouple is accurate	1	15
				Unintentional thermocouple junctions due to connection of leads of different thermocouple material	2	Ensure that any connectors used must be made of the same thermocouple material and at the correct polarity	Check the thermocouple probes before use	1	10
				Temperature readings are sensitive to noise	3	Use thicker thermocouple leads	Measure resistance of the thermocouple before use	1	15
				Thermal shunting	2	Use thicker thermocouple extension wires connected to the thin thermocouple leads	Measure resistance of the thermocouple before use	1	10

Appendix N-10: FMEA Table of RPM Measurement

Potential
Failure Mode and Effects Analysis Worksheet

FMEA Team: ME 450 Team 22	Test machine **System**	**FMEA Number:** 4D
Team Leader: Adolphus Lim	Data Acquisition **Subsystem**	**FMEA Date: (Original)** 11/12/2009
	RPM **Component**	**(Revised)** -

Design FMEA									
Item and Function / Requirements	**Potential Failure Mode**	**Potential Effect(s) of Failure**	**Severity**	**Potential Cause(s)/ Mechanism(s) of Failure**	**Occurrence**	**Current Controls - Prevention**	**Current Controls - Detection**	**Detection**	**RPN**
Measure RPM of drive shaft	RPM measured is not accurate	Test results are inaccurate	5	Improper encoder alignment	2	Use mechanical method of centering	Check that optical encoder is mounted correctly before use	2	20
				Oil, dirt or water get inside the encoder due to seal failure	2	Ensure that seals are intact	Check the seals before use	2	20
				Bearings fail due to stresses	3	Ensure that the optical encoder is rated for the RPM we expected to test at	Check that bearings are still operational	1	15
				Optical disk may shatter during vibration or impact	2	Protect the optical disk with shielding and material to damp the vibrations	Check that optical disk is not shattered	1	10
				Sensitivity to noise distorts the RPM readings	3	Use encoder cable to protect the digital signals from ground-loop and interference problems	Check that optical disk is not shattered	1	15

APPENDIX O: COST BREAKDOWN

1st Purchase Order

Quantity	Unit	Item #	Item Description	Unit Price	Total
1	ea	8975K564	Multipurpose Aluminum (Alloy 6061) 3" Thick X 3" Width X 6" Length	33.24	33.24
8	ea	1556A44	Steel Corner Bracket Galvanized, 2-1/2" Length of Sides, 19/32" Width	0.85	6.8
1	ea	9528K64	E52100 Alloy Steel Ball 1-1/2" Diameter, Grade 50	5.34	5.34
2	ea	8441K33	O1 Tool Steel Rod 8" Diameter, 1-1/2" Thick	77.31	154.62
1	pkg	91257A726	Grade 8 Alloy Steel Hex Head Cap Screw Zinc Yellow-Plated, 1/2"-13 Thread, 3-1/2" Length	11.14	11.14
1	ea	8910K973	Low-Carbon Steel Rectangular Bar 1-1/2" Thick, 6" Width, 6" Length	56.25	56.25
1	ea	8935K921	Driving Axle - 4140 Alloy Steel Hardened Rod 4" Diameter, 1' Length	122.84	122.84
2	ea	1630T471	Multipurpose Aluminum (Alloy 6061) U-Channel, 2" Base X 1-1/4" Legs, 5' Length	25.89	51.78
1	ea	9008K531	Multipurpose Aluminum (Alloy 6061) 2" Square, 1' Length	28.72	28.72
2	ea	9157A163	Grade 8 Alloy Steel Hex Head Cap Screw Zinc Yellow-Plated, 1/2"-13 Thread, 10-1/2" Length	7.94	15.88
1	pkg	91247A734	Grade 5 Zinc-Plated Steel Hex Head Cap Screw 1/2"-13 Thread, 5-1/2" Length	6.37	6.37
2	pkg	91475A033	300 Series SS MS35338 Split Lock Washer 1/2" Screw Size, Dash #143, 0.87" OD	6.03	12.06
2	pkg	91849A635	18-8 Stainless Steel Heavy Hex Nut 1/2"-13 Thread Size, 7/8" Width, 31/64" Height	7.15	14.3
4	ea	91025A732	Black-Oxide Steel Spacing Stud 1/2"-13 Sz, 5" L O'all, 1-3/4" & 3/4" Thrd Lengths	1.75	7
2	ea	6673T251	Torque Shaft - 4130 Alloy Steel Aircraft-Grade Rod 3/4" Diameter, 1' Length	7.03	14.06
1	ea	8975K332	Multipurpose Aluminum (Alloy 6061) 3" Thick X 5" Width X 1' Length	78.89	78.89
1	ea	5852T46	A2 Tool Steel Tight-Tol Flat Stock W/ Cert 1/2" Thick, 1-1/2" Width, 18" Length	68.71	68.71
1	pkg	9528K24	E52100 Alloy Steel Ball 1/2" Diameter, Grade 25	14.01	14.01
1	ea	1610T35	Multipurpose Aluminum (Alloy 6061) 5" Diameter, 3" Long	35.64	35.64
2	ea	8938K231	E52100 Alloy Steel Rod 1" Diameter, 1' Length	12.57	25.14
1	ea	6544k41	Low-Carbon Steel Sheet 3/4" Thick, 8" X 8"	50.33	50.33
2	ea	6544k23	Low-Carbon Steel Sheet 1/4" Thick, 8" X 8"	26.18	52.36
1	pkg	92865A628	Grade 5 Zinc-Plated Steel Hex Head Cap Screw 3/8"-16 Thread, 1-1/2" Long, Fully Threaded	12.83	12.83
1	pkg	91475A031	300 Series SS MS35338 Split Lock Washer 3/8" Screw Size, Dash #141, 0.68" OD	5.2	5.2
1	pkg	91849A625	18-8 Stainless Steel Heavy Hex Nut 3/8"-16 Thread Size, 11/16" Width, 23/64" Height	7.26	7.26

Subtotal	890.77

2nd Purchase Order

1	ea	HB6M	Hollow Bore Optical Encoder, shaft speed 6000 RPM, bore 3/4"	215.25	215.25
1	ea	5968K790	Cast Iron Flange-Mounted Steel Ball Bearing for 1-1/2" Shaft Diameter, 6-3/4" Base Length	59.4	59.4
1	ea	29412 E	SKF Spherical Roller Thrust Bearing for 60mm Shaft Diameter, 130mm Outer Diameter	480.6	480.6
1	ea	52408	SKF Double Direction Thrust Ball Bearing for 30mm Shaft Diameter, 90mm Outer Diameter	85.41	85.41
1	ea	1497K961	Fully Keyed 1045 Steel Drive Shaft 1" OD, 1/4" Keyway Width, 18" Length	34.45	34.45
2	ea	98870A430	Plain Steel Machine Key Square Ends, Oversized, 1/4" Square, 2" Length	5.72	11.44
1	ea	00518EP3E184T-WEG	Motor, AC, 5 HP, 1800 RPM, 208-230/460	270.09	270.09
1	ea	3602300	LITTLE GIANT 1500-Lb. Capacity Machine Tables 48" width 24" depth 42" height	162	162
1	ea	CIMR-VUBA0018FAA-YASK	Motor Drive 230 VAC 1PH 5HP/17 5A/VT, 5HP/17 5A/CT 400HZ NEMA 1	606.72	606.72
1	ea	2500T445	Steel Hardened-Teeth Finished-Bore Sprocket for #40 Chain, 1/2" Pitch, 12 Teeth, 1" Bore	14.89	14.89
1	ea	2737T861	Steel Finished-Bore Roller Chain Sprocket for #40 Chain, 1/2" Pitch, 112 Teeth, 1" Bore	98.96	98.96
1	ea	6236K188	Steel Finished-Bore Roller Chain Sprocket for #40 Chain, 1/2" Pitch, 36 Teeth, 1" Bore	40.99	40.99
1	ea	8935K151	4140 Alloy Steel Hardened Rod 2" Diameter, 1' Length	38.17	38.17
1	ea	5976K108	Ultra Premium Carbon Steel ANSI Roller Chain #40, Carbon Steel, 1/2" Pitch, 8' L	141.68	141.68
1	ea	9008K143	Multipurpose Aluminum (Alloy 6061) 1" Square, 3' Length	24.45	24.45
1	ea	9008K531	Multipurpose Aluminum (Alloy 6061) 2" Square, 1' Length	28.72	28.72
1	pkg	93190A547	Type 316 SS Fully Threaded Hex Head Cap Screw 1/4"-20 Thread, 1-3/4" Length	8.23	8.23
1	pkg	90670A029	Aluminum Hex Nut 1/4"-20 Thread Size, 7/16" Width, 7/32" Height	5.96	5.96
2	ea	SS-080-050-500-PBB-S1	Semiconductor "Bar" Gage SS-080-050-500-PBB-S1	49.41	98.82
1	ea	HTTC36-T-18G-6	T Type, 6 Inch Grounded Probe 1/8 Inch Diameter with 36 Inches of PFA Insulated 24 AWG Stranded Wire	19	19
1	ea	780137-01	NI USB-9237 4 ch USB High Speed Bridge/Strain	1214	1214

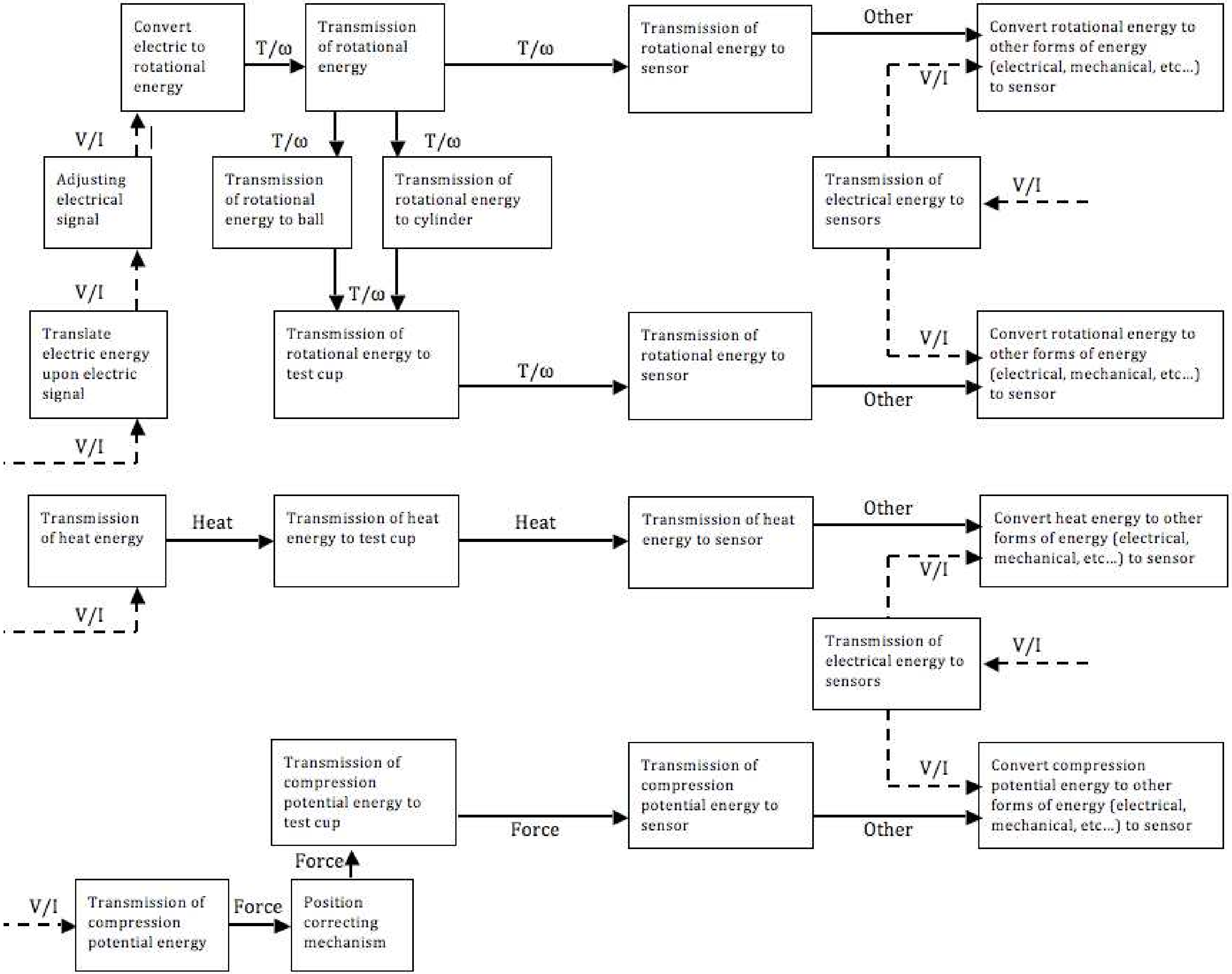
1	ea	779436-01	USB-6501, 24-Channel Digital I/O, programmable 5 V TTL or 3.3 V, 8.5 mA and NI-DAQ Drivers	476.1	476.1
1	ea	779205-01	USB-9211A 4ch, 24-Bit Thermocouple Input Module for Windows	89.1	89.1
5	ea	001540-017	400W 12V Thermoelectric Cooler Peltier Plate	14.99	74.95
1	ea	DS2000-3	Distributed Power Bulk Front-End Total Output Power: 2000 Watts	600	600
				Subtotal	4899.38

4	ea	LT1167CN8#PBF-ND	IC Prec Intrstrment Amp Prog 8-DIP	6.38	25.52
1	ea	ASTA- 7G	Arctic Silver (2 PC Set)	12.99	12.99
4	ea	5C12B3	Masscool 50mm Ball CPU Cooler-Retail	3.99	15.96
4	ea	001540-017	400W 12V Thermoelectric Cooler Peltier Plate	14.99	59.96
1	ea	85645K13	Glass-Filled Black Polycarbonate Sheet 1" Thick, 6" Width X 6" Length	63.54	63.54
1	ea	779026-01	USB-6009 Low-Cost Multifunction I/O and NI-DAQmx	251.10	251.10
1	ea	HTTC36-T-18G-6	T Type, 6 Inch Grounded Probe 1/8 Inch Diameter with 36 Inches of PFA Insulated 24 AWG Stranded Wire	19.00	19.00
1	ea	HEDR-8000-K	HEDR Reflective Optical Encoder Modules	14.07	14.07
1	ea	HUBDISK-2-400-1000-N	HUBDISK-2 2" Transmissive Rotary Codewheel	30.07	30.07
1	ea	802-124-910	General Purpose / Industrial Relays SPDT 12VDC	46.71	46.71
1	ea	ULT40070	Ultra X3 ULT40070 1600-Watt Power Supply - ATX, SATA-Ready, PCI-E Ready, Energy Efficient, Modular, Lifetime Warranty	299.99	299.99
				Subtotal	838.91

4	ea	90281A860	Black-Oxide Steel Spacing Stud 3/4"-10 Sz, 6-1/2" L O'all, 2" & 2" Thread Lengths	3.33	13.32
1	pkg	95479A128	Black Oxide Grade 5 Steel Hex Nut 3/4"-10 Thread Size, 1-1/8" Width, 41/64" Height	9.71	9.71
1	pkg	91247A330	Grade 5 Zinc-Plated Steel Hex Head Cap Screw 7/16"-20 Thread, 4" Length	6.58	6.58

1	pkg	93827A236	7/16"-20 Thread Hex Nuts	8.8	8.80
4	ea	98957A639	ASTM A193 Grade B7 Alloy Steel Threaded Rod Plain Finish, 7/8"-9 Thread, 3' Length	10.84	43.36
4	pkg	90648A240	Grade 2 Steel Nylon-Insert Heavy Hex Locknut Zinc-Plated, 3/4"-10 Thread Sz, 1-1/4" W, 1-1/64" H	5.5	22.00
1	ea	7786T62	Low Carbon Steel Rod 5" Diameter, 1/2" Length	11.17	11.17
1	ea	7786T52	Low Carbon Steel Rod 4" Diameter, 1/2" Length	8.34	8.34
1	ea	2500T434	Steel Hardened-Teeth Finished-Bore Sprocket for #40 Chain, 1/2" Pitch, 11 Teeth, 7/8" Bore	13.44	13.44
1	ea	6236K157	Steel Finished-Bore Roller Chain Sprocket for #40 Chain, 1/2" Pitch, 28 Teeth, 7/8" Bore	34.63	34.63
1	ea	2500T445	Steel Hardened-Teeth Finished-Bore Sprocket for #40 Chain, 1/2" Pitch, 12 Teeth, 1" Bore	14.89	14.89
1	ea	2737T861	Steel Finished-Bore Roller Chain Sprocket for #40 Chain, 1/2" Pitch, 112 Teeth, 1" Bore	98.96	98.96
1	ea	46715T24	Medium Duty Steel Machine Table 24" Width X 18" Depth, 42" Height	121.39	121.39
2	ea	1845A38	Heavy Duty Galvanized Steel Bracket Corner W/Brace, 8-1/4" L of Sides, 5-1/4" W, .187" Thk	23.99	47.98
1	ea	9455K158	Medium-Strength Neoprene Rubber Plain Back, 1/2" Thick, 12" X 24", 30A Durometer	34.3	34.30
1	ea	89015K32	Multipurpose Aluminum (Alloy 6061) .190" Thick, 12" X 24"	52.91	52.91
1	ea	9783T4	Step-Down Clamp-on Shaft Adapter 1-1/8" Bore, 7/8" Shaft Outside Diameter	63.2	63.20
				Subtotal	604.98
				Total	7234.04

APPENDIX P: FUNCTIONAL DECOMPOSITION



APPENDIX Q: UPDATED MOTOR CALCULATIONS AND GEAR RATIOS

Appendix Q-1: Updated Twist Compression Test Calculations

Standards: Max Pressure applied is 35 ksi, Max Speed is 30 RPM, and Cylinder Outer Diameter is 1 inch

$P = 35\ ksi = 241.3\ MPa \quad V = 30\ RPM = 0.0399\ m/s \quad OD = 1\ in = 0.0254\ m$

Assume Max Frictional Coefficient $\mu = 0.15$

Mean Cylinder radius = 11mm = 0.011m

Max Cylinder Thickness $= t = OD - 0.022 = 0.0034\ m$

Inner Diameter $= ID = OD - 2 \times t = 0.0186\ m$

Area $= A = \pi\left(\left(\frac{OD}{2}\right)^2 - \left(\frac{ID}{2}\right)^2\right) = 2.35 \times 10^{-3} m^2$

Normal Force $= F_N = P \times A = 56.70\ kN$

Friction Force$= F_F = F_N \times \mu = 8.51\ kN$

Power $= P_W = F_F \times V = 339.37\ W = 0.455\ Hp$

$1\ W = 0.00134\ Hp$

Torque $= T = F_F \times \frac{OD}{2} = 108.02\ N - m = 79.67\ lb - ft$

Appendix Q-2: Updated Gear Ratio Calculations

Motor Specifications
A.O. Smith Century E226M 3HP 1800RPM Motor

Full Load RPM = 1765, Assume available RPM = 1588.5

Full Load Torque = 8.7 lb-ft (Assuming 2.95 HP output)

Gear Ratio Calculations for Three Horsepower Motor

$$\text{Power, } P = \text{Torque, } \tau \cdot \text{Speed, } \omega \quad (17)$$

$$\tau = F \cdot r \text{ where F is Force and r is radius of gear} \quad (18)$$

Force between gears are always equal,

$$\therefore \tau \propto r,\ \omega \propto \frac{1}{r} \quad (19)$$

Four Ball Test

$$\text{Max RPM for Four Ball Test} = 3600 \text{ RPM} \quad (20)$$

$$\text{Max Torque for Four Ball Test} = 4.25 \text{ lb-ft} \quad (21)$$

$$\begin{aligned} \text{To achieve max RPM, gear ratio } &= \frac{3600}{1588.5} \\ &= 2.266 \quad (22) \\ &\approx 2.5:1 \end{aligned}$$

$$\begin{aligned} \text{RPM available for 2.5:1 gear ratio} &= 2.5 \cdot 1588.5 \\ &= 3971.25 \end{aligned} \quad (23)$$

$$\begin{aligned} \text{Torque available for 2.5:1 gear ratio} &= 8.7/2.5 \\ &= 3.48 \text{ lb-ft} \end{aligned} \quad (24)$$

$$\text{Max Normal Load available for Four Ball Test} = 6321.5N \quad (25)$$

Twist Compression Test

$$\text{Max RPM for Twist Compression Test} = 30 \text{ RPM} \quad (26)$$

$$\text{Max Torque for Twist Compression Test} = 79.67 \text{ lb-ft} \quad (27)$$

$$\begin{aligned} \text{To achieve max torque, gear ratio} &= 79.67/8.7 \\ &= 9.16 \quad (28) \\ &\approx 9.33:1 \end{aligned}$$