Honeywell Bearings

Design Proposal for Bearing Characterization

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EXECUTIVE SUMMARY

For this project, the team is tasked with creating a device that will test the friction experienced by Honeywell's bearings of various sizes when subjected to thrust and radial loads they endure in service. To begin, the team developed black box, and functional models to create a visual representation of the necessary inputs and outputs to make the system perform as intended. The team researched any previous designs created to accomplish this task, however, the team found nothing publicly available. The client, Honeywell, provided no information regarding similar systems or their current system as the wanted to ensure our ideas were unique and did not succumb to design fixation. A quality function deployment (QFD) was then created to outline customer requirements and match them with engineering requirements. The team met with Honeywell to ensure the customer needs and engineering requirements were ranked and scored properly. The team then brainstormed and generated multiple concepts, most centered around using jacks to impart a force directly to the inner race of the bearing, and another to the outer race. The idea behind this was that reaction forces of the same magnitude experienced by the inner race would sufficiently simulate the forces it experiences in service. A design based on this idea was selected using a decision matrix. However, Honeywell determined that this concept does not meet their criteria. Therefore, the team selected and perfected a design meant to impart radial and thrust loads directly to the inner race through applying all forces to the shaft. A fully hydraulic system that utilized three hydraulic rams and a rotary actuator was initially envisioned, but budgetary constraints required a new paradigm to be explored. A system utilizing three electro-hydraulic car jacks and a stepper motor was selected instead. Pressure measurements are expected to be taken via pressure transducers and converted into force measurements via a computer program. A torque measurement is taken via a force sensor located at the end of the shaft, and another computer program to take the appropriate measurement. In this way all criteria are met.

ACKNOWLEDGEMENTS

Thanks to Honeywell, Dr. Trevas, and the NAU shop managers for dedicating time to help us with this project.

TABLE OF CONTENTS

Contents

1 BACKGROUND

1.1 Introduction

This project was established by Haley Flenner, an employee of Honeywell. The Honeywell Bearing project's main objective is to measure the torque needed to affect the movement on a shaft when there is a variable force on a bearing in the radial and thrust directions. The Honeywell team benefits from the project's completion by having a new machine to test their bearings on one of their popular products a butterfly shaft. As it will be explained later in this report, there isn't any current machines that test bearings in this specific way, although it was mentioned to the team that Honeywell currently has some ideas to test their bearings there very limited information about other machines that can measure the torque. The main issues that are accompanied to the project are keeping the system simple and safe. This involves safely meeting the force requirements which will be explained later and maintaining a safe shaft rotational speed. The team is attempting to keep the machine as simple as possible which would allow someone to use it without complicated calculations.

1.2 Project Description

The description of this project from Honeywell is:

"Design and manufacture a test fixture to test rolling element shaft bearings. The fixture will be used to record the torque required to rotate the inner race while simultaneously applying a radial and thrust load to the inner race. We want to quantify rotational friction as a function of radial and thrust loads".

Honeywell was adamant that the goal of the project was to quantify friction, not to test the bearings themselves. The machine must be capable of testing bearings of the following sizes; 38, R8, R12, and R14. The thrust and radial load imparted upon the bearings varies based on their size. The largest bearing (R14) is subject to forces up to 8000 lbs radially and 4000 lbs axially when in service, thus the fixture must be capable of imparting those loads. The smaller bearings are subjected to smaller loads. The inner race of the bearing must rotate 90° between 1 and 10°/s. Honeywell originally required the fixture to be capable of testing the bearings in temperatures between -65°F to 1000°F. However, both Honeywell and the faculty advisors agreed this requirement should be eliminated as it could not be achieved with the given budget. The test fixture must quantify and plot friction against the applied loads. Finally, Honeywell stated that the volume of the testing machine must be less than 9 ft^3.

2 REQUIREMENTS

Honeywell immediately specified what the test fixture must output, however, safety and power constraints were addressed later in meetings with them. Engineering requirements were then specified to establish quantifiable goals that indicate meeting the constraint.

2.1 Customer Requirements (CRs)

A list of requirements was compiled and matched with constraints. This is displayed within table 1.

Table 1: Customer Requirements and Constraints

Requirements	Constraints
Must meet loading requirements	Adjustable radial Load from 0 to 8,000 lbf Adjustable thrust load from 0 to 4,000 lbf Apply load to inner race
Must be safe	Injury cannot occur if something breaks
Must be portable	Easily moveable by two people
Must be compatible with different bearings	Fits R8, R12, R14, and 38 bearing diameters
Must output a torque vs. Load applied Graph	Torque vs. load applied plotted in real time
Must fit in an office space	Volume of system must be within $9ft^3$
Only source of power is electricity	120V electrical outlet
Must rotate the inner race	Bearing must rotate 90°
Affordable	Cost less than $1,500
Easy Maintenance	No lube cannot be fragile

The constraints listed represent boundaries the team cannot break. Regarding the loading requirements, the user must be able to vary the loads in the specified range.

2.2 Engineering Requirements (ERs)

Every customer constraint was matched with an engineering requirement which provides a quantifiable goal. These requirements are listed in table 2.

Table 2: List of engineering requirements

Engineering Requirements		
Property	Unit	Value
Radial Load	lbs.	8,000
Thrust Load	lbs.	4,000
Weight	lbs.	300
Shaft Rotation	°	90
Cost	USD	1,500
Voltage	V	120
Current	Amps	15

The radial and thrust load values represent what the hydraulic system must be capable of imparting upon the system. The value for weight was determined based on Honeywell's desire for the product to be mobile. The team determined that a weight of less than or equal to 300lbs would be acceptable. The value for cost represents the maximum amount the team can spend on the project, and the values for current and voltage represent what a common household electrical outlet can produce. In addition to the requirements listed in table 1, the team must ensure the frame and shaft are able to withstand the bending and shear stresses associated with the applied loads. The team must also produce separate shafts that will fit each bearing size to be tested. This will result in shaft diameters of 0.315in, 0.5in, 0.75in and 0.875in with a tolerance of +- 0.005in.

2.3 Testing Procedures (TPs)

To ensure that engineering requirements are met the team had to determine how to measure each one. The following list considers how the team tested each requirement.

1. Radial Loads (Up to 8000 lbs)

The radial loads are applied with two hydraulic jacks. The shaft needed to be capable of supporting that load. Solidworks finite element analysis (FEA) was used to simulate the shaft's behavior under the applied loads which ensured failure would not occur. The jacks needed to be tested to ensure they could attain the maximum loads. To do this they were disassembled and tapped so a pressure transducer could be installed. The jacks were then reassembled and mounted to the frame. The transducers were zeroed by placing a known weight upon one jack and then correcting the force displayed by the Arduino program. It was assumed that all jacks would exhibit the same force measurements, therefore, all jacks were corrected to the same degree. The jacks then loaded the shaft until the maximum radial load was displayed on the program.

2. Thrust Loads (Up to 4000 lbs)

The thrust load is applied using one hydraulic jack. The shaft had to be capable of handling a 4000lb compressive force, thus, this load was modeled in Solidworks FEA along with the radial loads to test for potential failure. The jack had to be capable of imparting the 4,000lb load axially. To test for this the jack was disassembled and tapped to mount a pressure transducer. The jack was then reassembled and mounted to the frame. The shaft was then loaded in the axial direction until the axial direction until the screen displayed the maximum thrust load.

After procedures 1 and 2 were conducted, the team tried applying all loads simultaneously to check if any errors were caused by doing this.

3. Weight (less than 300 lbs)

The test fixture was to weigh 300 lbs or less. Honeywell never gave a specific weight target but approved of this one. A scale large enough to weigh the fixture could not be found, therefore, the team demonstrated mobility by simply having two members lift it.

4. Volume (9 ft^3)

The test fixture had to fit within a 9ft^3 space. For this reason, the shafts had to be short and the supports needed to be made as short and/or thin as possible. Hydraulic jacks were used because they can of apply the required loads within a small space. The team measured the length, width, and height of the fixture to test for this.

5. Rotation (Up to 90°)

The shaft had to be rotated 90° at a speed of 10°/s or less. To test for this, the motor was hooked up to an Arduino board and was programmed to spin 90°. The was quantified by timing how long the shaft took to reach 90°.

6. Cost (Under $1500)

The team had a budget of $1500. It is the reason off-the-shelf car jacks were used to apply the forces, pressure transducers were used to measure the forces and an equation was used to measure torque measurements. A bill of materials was used to keep track of expenses.

7. Voltage (120V)

A standard wall outlet was all that was available to power the test fixture, meaning it had to work with a 120V power supply. A converter was installed on all components as all were designed to be powered by a 12V power supply. The components were plugged in and turned on to ensure they could be powered by a wall outlet.

8. Current (15A)

A standard wall outlet provides 15A of current. The team ensured the fixture could operate on this current by plugging it in and turning it on.

9. Shaft Size

Honeywell required the system to be capable of testing 4 different bearing sizes including R8, R12, R14, and 38. To do this, the team made a steel shaft for each bearing size. To test this requirement, the team press fit each bearing onto their respective shafts and ensured they could still spin which indicated the fit was not too tight.

2.4 House of Quality (HoQ)

To create the quality functional deployment shown in appendix B, 9.2.1, the team ranked each customer requirement on a scale of 1-4. The rankings were shown to Honeywell, and they were pleased with the results. The engineering requirements were then related to customer requirements. A rating of nine indicated a strong correlation to customer needs, three indicated a moderate one, and one indicated a weak correlation. For engineering requirement targets the team indicated the maximum values allowed. The maximum radial load is 8000 lbs. The maximum thrust load is 4000 lbf. The maximum weight the team felt two people could move is 300 lbf. Honeywell determined the maximum volume the device could occupy is $9ft^3$. The maximum speed the bearing needs to achieve is ten degrees a second. The estimated budget for our final design is $2700 which does not meet our preliminary budget requirement. The team will focus on cutting costs once a sound design is finished. The maximum voltage and amperage the device can operate on is 120V and 15A. The size of the shafts will fall between 0.315in to 1in. The maximum necessary yield strength of the material needs to be high enough to resist deformation during maximum loading.

3 EXISTING DESIGNS

A considerable amount of research was done to determine how to go about designing the test fixture. Inspiration was drawn from mechanisms designed for bearing work and from Honeywell's butterfly valves. Additionally, the functionality of the fixture was broken down to sub-system levels. Components that could meet those functional needs were then researched.

3.1 Design Research

An attempt was made to research fixtures that quantified friction in bearings, however, the team struggled to find these. This problem was brought up to Honeywell to which they responded by claiming most companies keep these systems secret and stating they wanted the team to have a fresh perspective the matter with no inspiration from similar products. Honeywell did provide a rough illustration as to how they wanted the test fixture to work this is shown in figure 1.

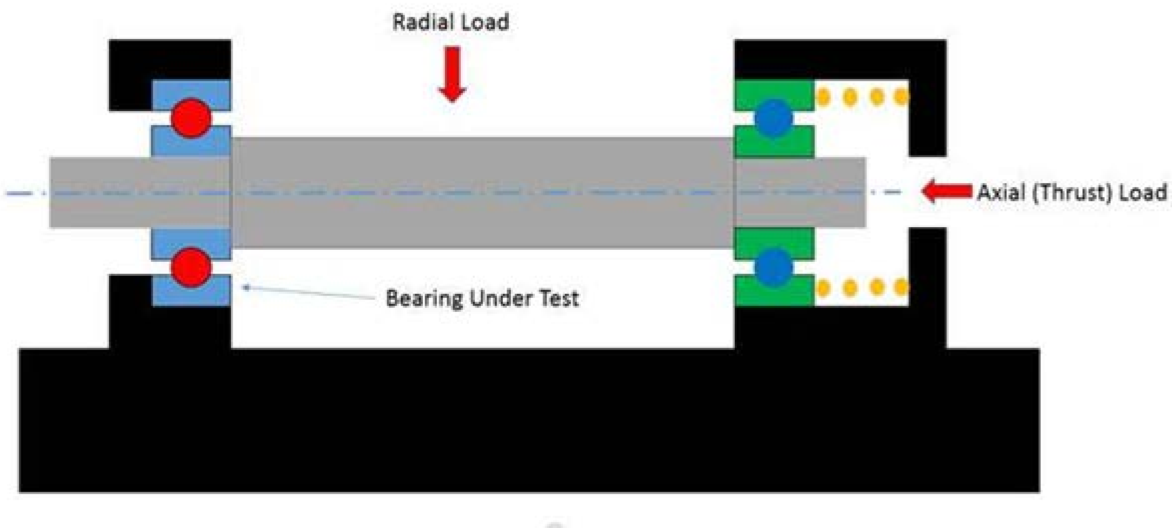

Figure 1: Load application

Honeywell wanted to simulate how their bearings are loaded on their butterfly valves, which meant all forces had to be imparted upon the shaft which would impart reaction forces upon the inner race of the bearing.

An opportunity to see Honeywell's existing system was eventually granted during a tour at their factory. It was unclear how the fixture worked and all engineers that knew how to work it retired. This fixture used a pneumatic ram to impart the thrust load. This required a large space to fit the tank and complete shield coverage to protect users in the event of a failure. A system of belts and pulleys imparted the radial loads though it is unclear how it could reach up to 4-tons of force. From this it was decided that pneumatic systems should be avoided and that it should be simple enough to use without the assistance of those who designed it.

3.2 System Level

An attempt was made to find a product that offered some similar capabilities to what the fixture needed. Hydraulic presses of various types offered these capabilities; thus, they were explored further.

3.2.1 Hydraulic Press

Hydraulic presses consist of a hydraulic jack, and a support. These systems are often used to impart a compressive force upon objects to make flat sheets or perform press fits. The team can draw inspiration from these systems by observing how the frame is designed to withstand the stresses imparted upon it by the jack when under operation. Additionally, the team can observe how the force is imparted as the jack either imparts the force directly or does so indirectly as shown in figure 2.

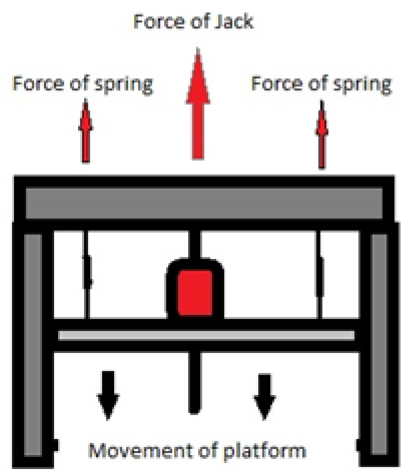

Figure 2: Hydraulic press design

The rod below the platform is interchangeable, which allows one to change the area of the surface being loaded. The area of the ram on off the shelf jacks cannot be changed without modification, the system shown in figure 2 will circumvent this problem.

3.3 Functional Decomposition

3.3.1 Black Box Model

The team created a black box model to help visualize the material, energy, and signal flows within the system. This is shown in figure 6.

Figure 3: Black Box model for Testing Bearing

The material inputs are the test bearing and the hand that is used to place the bearing on the shaft. The material outputs are the same as the inputs. The energy flows include the electrical energy used to turn the motor, the hydraulics used to apply the forces, and the human energy to start the system. The outputs are in the form of heat generated and kinetic energy in the rotating shaft. The signals include the instructions to operate and the switch to turn on and off the system. The signal output is the live test results which includes the torque required to spin the shaft when the forces is applied. By creating this model, the team was able to break down the basic requirements of the project.

3.3.2 Functional Model

The design of this fixture was broken down into 6 main components which is displayed in figure 4.

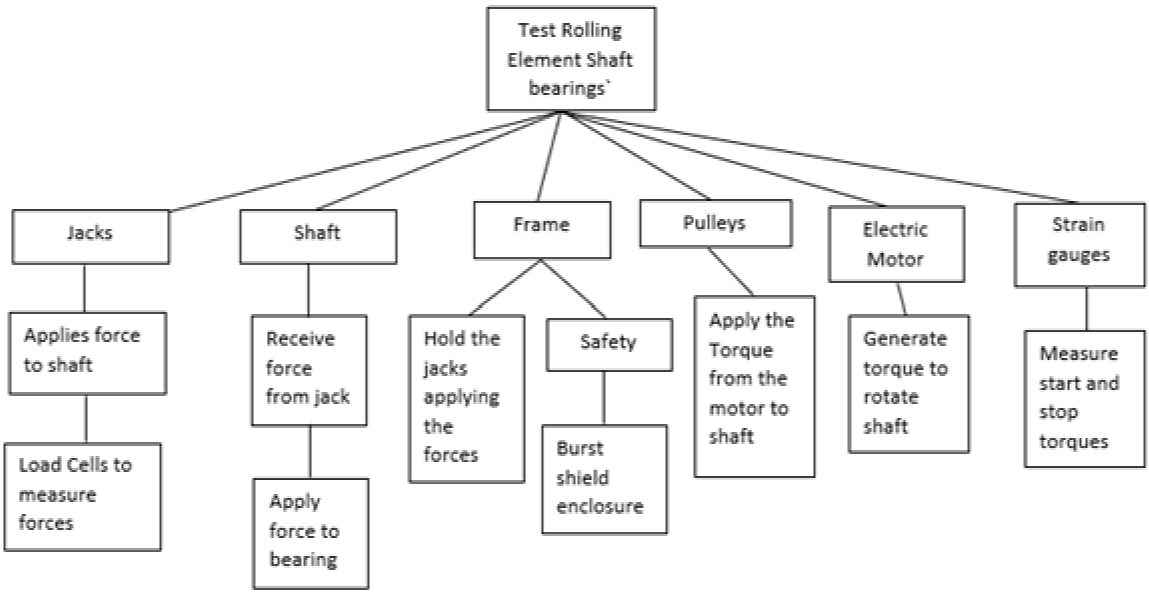

Figure 4: Decomposition of Testing Bearings

The first of these components are the jacks, which apply the forces to the shaft. Another component rotates the inner race of the bearing after loads are applied. The third is the frame which holds the jacks steady as forces are applied. The fourth and fifth are the pulleys and electric motor which apply and generate torque for the shaft respectively. The final component is the strain gage which measures the amount of torque needed to start and stop spinning the bearing. Figure 4 provided a basis for which components to research and develop a functional model with.

To visualize how the device would function given certain inputs the team made a functional model. This is shown in figure 5 below.

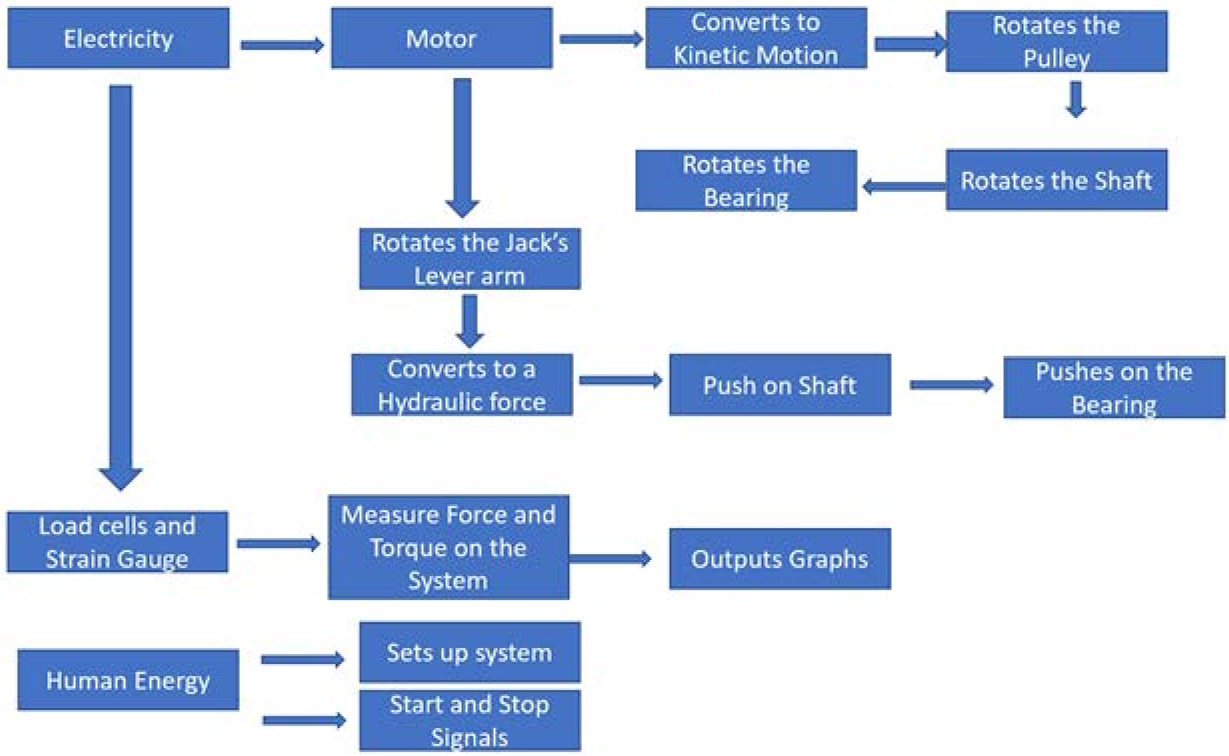

Figure 5: Functional Model

Figure 5 shows that the main energy input is electricity which will power most subsystems. Electricity will power the motor which will convert the electricity to kinetic energy. This kinetic energy will then spin the bearing. Someone will then actuate the jack which will cause a buildup of hydraulic forces. The forces will then be applied to shaft which will transfer the forces to inner race of the bearing giving us the radial and thrust loads. The last subsystem is also powered by electricity. This includes the force measuring devices which will measure the force and torque in the system. Once these measurements are acquired the data will be organized and displayed on the computer as graphs.

Human energy is also a major part of this design. It is needed to set up the test fixture, in which the correct shaft and bearing combination must be attached to the system. After completing the system set up, someone must initiate the test, and terminate it when completed.

3.4 Subsystem Level

This project dealt with multiple subsystems including the shafts, shaft retainers, the frame, shaft rotation mechanisms, power transmission systems, force application systems, and force measurement systems. Each of these components addressed a specific part of the design requirement.

3.4.1 Subsystem #1: Shafts

The shaft provided a place for the bearings to be mounted on. It needed to withstand the loads that will be imparted upon the system.

3.4.1.1 Step Shaft

The team considered using a step shaft first as it would allow us to mount every bearing simultaneously simplifying the design. Additionally, there would be no need to switch out shafts, so it could be permanently affixed to a retainer with a spline or press fit. This could reduce costs and enable the team to use a stronger mount. A drawback to this design is that a stress concentration exists at the shoulder of each step [1], therefore the total stress the shaft can withstand is reduced.

3.4.1.2 Uniform Shaft

The team also considered utilizing uniform shafts. This would eliminate stress concentrations on the shaft, increasing its strength. However, this also necessitates developing a way to quickly change out shafts when a new bearing must be tested. Resultantly, this increases the cost and complexity of the design as an adaptor must be designed, and/or a large chuck must be purchased.

3.4.2 Subsystem #2: Frame

The frame contains all components; therefore, it had to be strong enough to contain all loads. All forces needed to be absorbed by the frame to keep the system stationary. It was crucial that the frame stayed together and did not bend so results would be accurate.

3.4.2.1 Bearing Support

The bearing support system consists of a bearing support base that is secured to the frame. The first iteration featured 4 interchangeable supports for each bearing size that slid in and out of the base. The bearing supports needed to remain stationary and not deform or the bearing load data will be inaccurate. This design had been constructed in SOLIDWORKS and simulated at the max loads of 8000 lbs. radially and 4000lbs axially. It was able to support the loads and had a max deflection that was small enough to not affect the measurements. However, this design was replaced by a similar one that bolted in and out of the frame instead. It was lighter and passed the same tests.

3.4.3 Subsystem #3: Torque Application

3.4.3.1 Electric Motor

The team determined that the shaft would be spun by an electric motor, as it seems to be the only feasible power source available that will impart a torque strong enough to spin the shaft, while not taking up too much space. Very little power was necessary to spin the shaft; thus, any motor could have been used so long as it could be programmed or made to move 90° at 10°/s or less.

3.4.3.2 Hydraulic Rotary Actuator

The team considered using a hydraulic circuit to implement a hydraulic rotary actuator which would enable the team to spin the shaft 90° without the use of an electric motor. There were many different actuators the team considered including rack and pinion systems, scotch and yoke systems and enclosed piston crank actuators, all of which could have met the team's goal of turning the shaft. These systems would have been able to accommodate a shaft, pulley, or thrust bearing via a coupling or keyway.

3.4.4 Subsystem #4: Power Transmission

The team initially determined that a thrust force cannot be directly applied to the shaft, or the bearing when affixed to an electric motor. Therefore, the team considered systems that would transfer the torque from one position to another.

3.4.4.1 Belt and Pulley

A flat belt and pulley combination exhibited a high efficiency of about 98% [1], which is desirable as transmission losses would have had a minimal effect on test bearing friction measurements. Due to elastic creep, however, the belt may have moved at a speed slower than the pulley [1], therefore adjustments would have had to be made to ensure the shaft turns the full 90°. Another advantage considered was that a belt can be easily wrapped around a shaft or an adaptor, so torque could be transferred to the bearing with little modification. A small flat belt could have been had for around $22.00 [2], and an accompanying drive pulley would have cost around $13.00-$22.00 [3].

3.4.4.2 Chain and Sprocket

A chain and sprocket combination would have exhibited roughly the same efficiency as a flat belt and pulley [1], however there would have been no issues with slip. Additionally, there would have been no concern for wear due to the fact the motor will only turn 90°. However, extra modification would have been needed to enable it to spin the shaft, as another sprocket would have been needed to do so. The necessary sprockets and chain may have cost about $47.00 combined [4,5].

3.4.5 Subsystem #5: Force Measuring System

Force data needed to be collected so it could be used to calculate friction and illustrate its relationship to the applied loads.

3.4.5.1 Strain Gauge

A strain gage is a device containing a thin metal wire that stretches or compresses with a material given an applied load [6]. Based upon the resistance measured over that wire, one could tell how much a material has deformed, and back out a force measurement based on that. The team thought it could take advantage of this phenomenon to obtain the thrust and radial forces imparted upon the system. The strain gage kit may have cost about $332 [7].

3.4.5.2 Load Cell

The team also investigated compression load cells, which include four strain gages packed into a small steel casing [8]. This device therefore works on the same principle as strain gages do. When the steel casing compresses, the strain gages inside it either stretch or compress, changing resistance, and thereby giving a strain value that could have been used to back out force. These devices are only 1in in diameter and 1in tall and can withstand forces up to 10,000lbs [9], therefore the team could have directly mounted them to the force application system. This would have allowed the team to get a direct measurement of the force applied in real time. However, these devices were expensive as they cost around $350 each [9], and the team would have likely needed more than one since thrust loads act horizontally, and radial loads act vertically. Additionally, load cells cannot measure torque.

3.4.5.3 Pressure Transducer

The hydraulic jacks that apply the forces could be tapped with a pressure gauge. As the hydraulics apply a force, this pressure could be measured. A pressure transducer is used to read this pressure change and convert it to either a voltage or a current. Using an Arduino unit, this signal can be used to find the force applied by the jacks.

3.4.6 Subsystem #6: Force Application Systems

The team had to impart a maximum load of 8000 lbs. of force radially, and 4000 lbs. axially upon the inner race of the bearings. Honeywell provided nothing in terms of hydraulic, or pneumatic pressure. Therefore, the team only considered systems that would be able to impart the necessary forces on their own.

3.4.6.1 Bottle Jacks

Bottle jacks are typically used to lift cars in emergency situations. They utilize fluid actuated by a hand crank to impart forces strong enough to lift vehicles. Bottle jacks rated at 4-tons were available for about $20.00 [10]. Additionally, these jacks had a minimum height of 7.625in and a maximum height of 15.125in [10] which meant they fit well within the team's $9ft^3$ constraint, which allowed for design flexibility. A drawback with these systems was that they had to be hand cranked which would reduce the precision of the measurements, and they did not have a pressure readout therefore a sensor would have needed to be used to determine how much force they imparted. Tapping these could have given more precise readouts, however, this also had drawbacks. In the worst case, tapping the jack could break it rendering it unable to impart its maximum force.

3.4.6.2 Electric Jacks

Electric jacks utilize an electric motor to power a pump which pressurizes fluid providing a lifting force. These jacks were considerably shorter than hydraulic bottle jacks and are actuated with a button rather than a lever, which would make operating the system more straight-forward. However, they are also considerably more expensive, costing around $234 [11]. They were also subject to issues with directly measuring force, just as bottle jacks are.

3.4.7 Subsystem #7: Shaft Retainers

This was considered necessary for mounting the shaft to the system and allowing for a way for uniform shafts to be interchanged until it was determined the bearing housing could do this instead.

3.4.7.1 Lathe Chuck

A lathe chuck could have been used to very quickly change out shafts. It is made to be strong and would have had no problem withstanding the applied loads. Furthermore, a belt could have been wrapped around it creating an easy way for the torque to be transferred from the motor to the shaft. However, if heavy loads were applied the chuck could have dug into the shaft, compromising the integrity of the shaft. Additionally, some chucks may not have been able to fit the smallest shafts. Finally, lathe chucks tend to be expensive costing around $175.00 [12].

3.4.7.2 Thrust bearing

A thrust bearing is a specific type of ball bearing meant to withstand high thrust loads. This was considered ideal as it would likely hold up to the axial forces subjected to it through the shaft. It was also a cheap option costing around $51.00 [13]. However, given the inner diameter of this bearing could not be changed, an adaptor would have needed to be made to accommodate different shafts.

4 DESIGNS CONSIDERED

To begin the design process all members of the team developed concept sketches to illustrate understanding of the problem and establish a direction to take in terms of meeting the requirements. All concepts are displayed below.

4.1 Design #1: Funnel Design

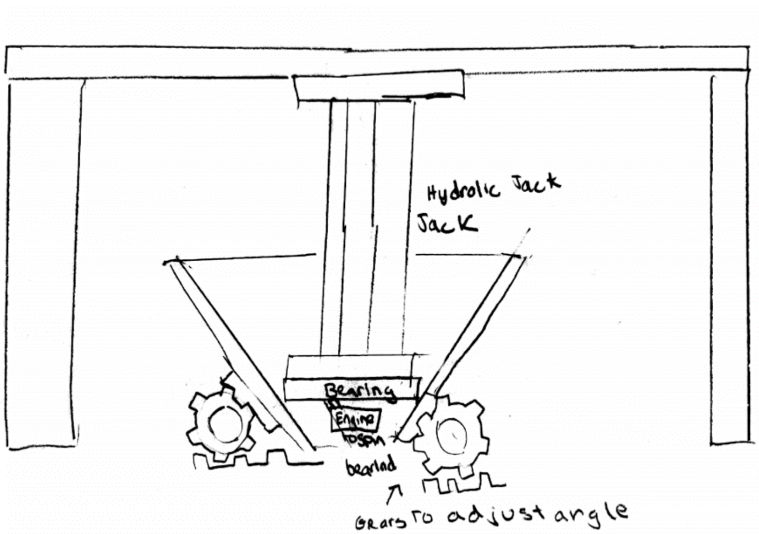

Figure 6: Funnel deign

This design would have placed the bearing in a funnel that could be adjusted with a rack-and-pinion system. The angle of the plates the bearing sits on would have been adjustable so that the force vectors in the radial and axial direction could be changed. Once the angle was determined and the bearing was put in place, the jack would apply a load to the face of the bearing. This was the cheapest concept, as it did not require the use of shafts. This design would have applied both loads to the outer race bearing, however, the client wanted the load imparted only on the inner race of the bearing, therefore, it was rejected.

4.2 Design #2: Shaft Step

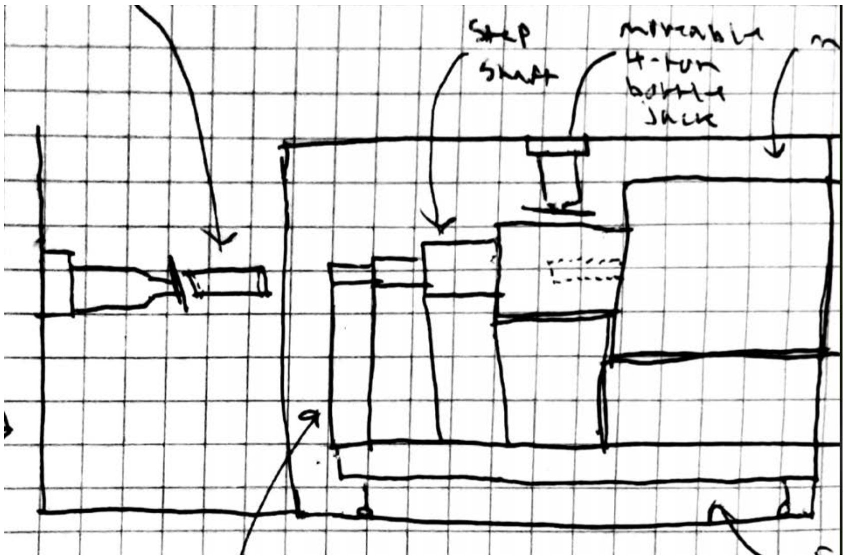

Figure 7: Stepped Bearings

This design utilized a step shaft directly mounted to an electric motor to mount the bearings without the need to switch out shafts. A bracket would have supported the bearings so when a force was imparted upon them a moment would not be experienced by the shaft. The forces would be imparted by two bottle jacks. The first jack would impart a force directly to the outer race of a bearing, as it was expected that the inner race would experience a reaction force of the same magnitude. This jack would have been moveable so one could select which bearing to test. The second jack would have been mounted horizontally and would have pushed a hollow tube directly into the inner race of the bearing to impart the thrust load. It was proposed that industrial scales be used to measure the forces, however this would not have worked as these scales could not handle forces the team needed to impart. Therefore, load cells would have been mounted to each jack instead. This design allowed for each bearing to be quickly tested as there was no need to switch out shaft assemblies. Additionally, calculations may have been easier as the forces were applied directly to the bearings, and no transmission losses would have to be considered given the shaft was directly attached to the motor. Because of this, however, the motor would have experienced the full thrust force which could break it. Honeywell ultimately determined that this method of testing the bearings was too great a departure from simulating the loads experienced by these bearings when attached to a butterfly valve, and therefore rejected it outright.

4.3 Design #3: Inner Race Design

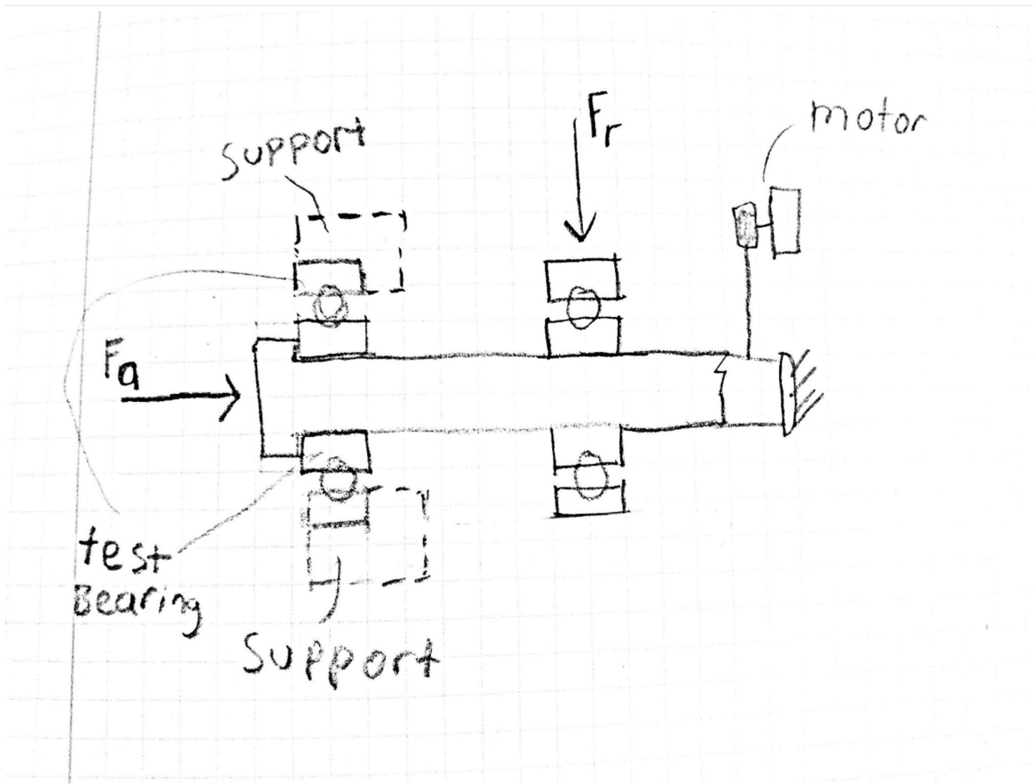

Figure 8: Inner Race Design

The rationale behind this design was that loads were applied directly to the shaft to ensure that the forces act on the inner race of the test bearing. The thrust load would have been applied to the shaft and the radial load was to be applied to another bearing which deflects the shaft ensuring the radial load is applied to the inner race of the test bearing. The design would have been supported by the frame which connected the bearings. An oversight on this iteration of the design was that the system was supported where the radial load was applied. To ensure that the radial load was applied to the shaft, the shaft had to be supported elsewhere. However, since this design did not apply the loads directly to the test bearing, it is the design that the Honeywell engineers preferred.

4.4 Design #4: Reaction Load Analysis

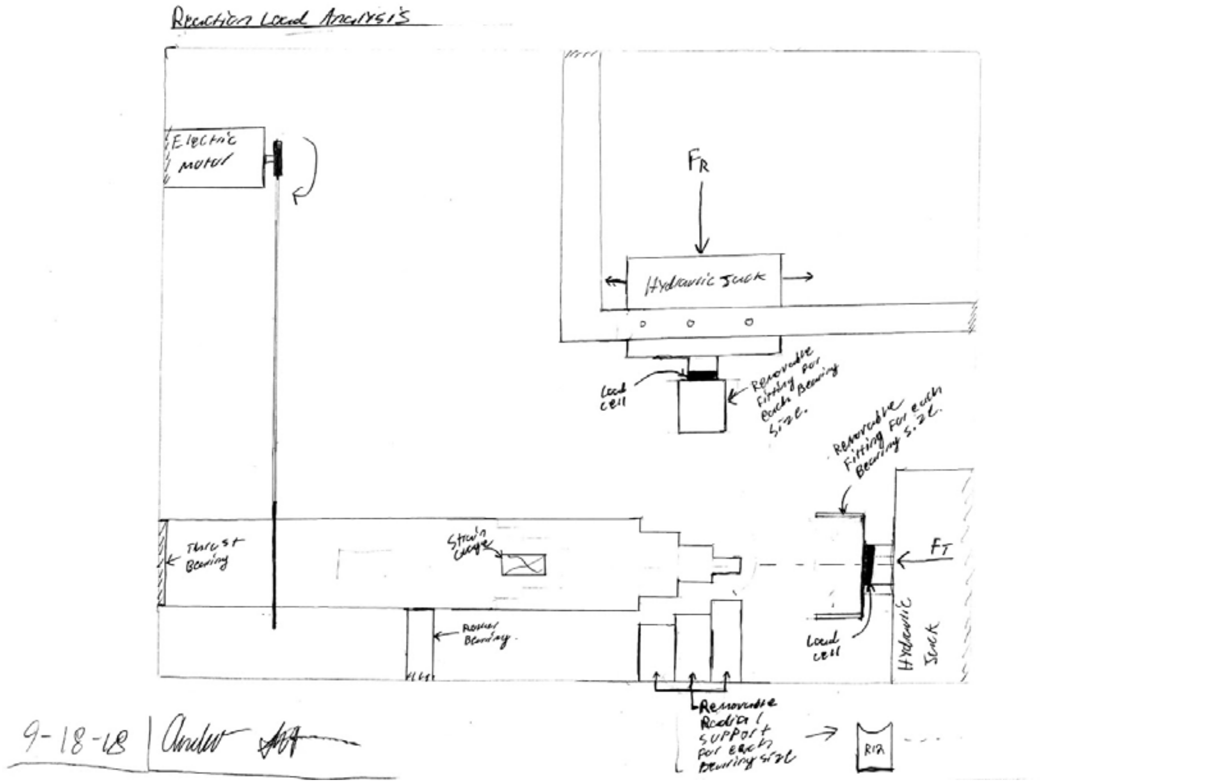

Figure 9: Reaction Load Analysis

This design focused on applying the radial and axial loads to the outer race of the test bearing. A step shaft would have been used to make the test apparatus easy to use as a new shaft did not have to be installed to test a new bearing. An electric motor connected to a pulley would have provided the rotational motion needed to test the break force and kinetic friction within the bearing. There would have been a strain gauge on the shaft to measure strain which was necessary for calculating the rotational torque. There would have been removable radial supports for each bearing size to support the bearing while it was loaded radially. These supports would have lowered the bending moment on the shaft and ensure the radial load was dispersed equally. The jack that would have supplied the radial load had removable fittings at the end that are unique to each bearing size. They would have been semicircular, so the load could be equally distributed throughout the bearing. The jack would have been placed on a track, so it could move to each bearing. The thrust load jack would have featured removable fittings that are unique for each bearing size. Both jacks would have applied the radial and thrust load to the outer race of the bearing. The reaction forces on the inner race would have been measured. Load cells would have been placed under the fittings of the jacks to output a live reading of the loads on the bearing. This design may have been simple and easy to maintain with minimal moving parts.

4.5 Design #5: Lever Arm

This design, shown in Appendix A, 10.1.1, would have utilized a special shaft that would be forced apart three ways to apply the radial load. The thrust load would have been applied by a jack directly to the inner race. This design did not require the use of a shaft and applied the load directly to the inner race as required by Honeywell. The major problem with this design was its high potential cost and low feasibility.

The small rods spreading apart would need a very high yield strength making them prohibitively expensive, it was therefore not considered by the team.

4.6 Design #6: Bearing-go-round

This design was meant to directly connect several uniform shafts to the bearing. These shafts, with the bearings attached, would have been mounted to a carousel that would allow the user to quickly select a bearing to test, and connect it to the motor via a spline, or keyway. Everything would have been connected to a slide, so the user could center each bearing and prepare it for a test. A bottle jack mounted to a slide on top of the frame would have imparted the radial load. The bearing would have been placed on a bracket to avoid imparting a moment on the shaft. Another jack mounted horizontally would have pushed a hollow pipe into the inner race imparting the thrust load. The problems with this design were numerous. For one, every sliding component adds complexity to the design as it introduces stress concentrations to the frame which is problematic as the frame must withstand all forces being imparted simultaneously. Additionally, a way to lock everything in place needed to be devised. Another problem was the input shaft on the electric motor may have been too large to fit some of the smaller shafts. Finally, the electric motor would have had to absorb the entire thrust force, which could have led to failure. The only advantage this design exhibited was that fact that uniform shafts were used which increases the stress they can take before failure.

4.7 Design #7: Four Points

This design, shown in Appendix A, 10.1.2, applied the thrust load to the test bearing using a prong system attached to a jack. It would have also applied the radial load directly to a test bearing. However, Honeywell wanted both loads applied to the inner race of the bearings, therefore, both methods of application did not satisfy the requirements. The design would have been able to apply loads to the bearing, but it did not meet Honeywell’s requirements, so the idea was not considered.

4.8 Design #8: Gear Turner

This design shown in Appendix A, 10.1.3, was very similar to other designs in which two jacks were used to apply the forces to the shaft which were then reacted onto the inner race of the bearing. Gears would have been attached to the motor input shaft and the testing shaft to apply the correct amount of torque needed. The main problem with this design was that the torque imparted upon the shaft would have been large and could have potentially damaged the gears. It was also too complicated for the team to design because determining the correct gear ratios may have taken too much testing and extra gears may have been needed in case of failure.

4.9 Design #9: Air Compress

This design, shown in Appendix A, 10.1.4, focused on using air compressors to apply enough pressure to the shaft to match the required forces for a given area. The main problem with this design was that the forces needed cannot be imparted by air compressors. The remaining portions of the design were very similar to the other designs in which strain gauges are used to measure the torque.

4.10 Design #10: Gravity Assist

This design shown in Appendix A, 10.1.5, utilized a bolt and washer that would have pulled down on the inner race of the bearing to apply the thrust loads on the shaft and use jack to apply radial loads to outer bearing. Weights would be attached to the washer and bolt. This design was one of the less expensive designs and used a unique thrust load delivery system. However, due to volume constraints and safety concerns this design was deemed impractical.

5 Design Selection

To start the selection process the team put all ten designs into the Pugh chart shown in table 3.

Table 3: Pugh Chart

	Concepts									
Criteria	Reaction Load Analysis	Inner race Design	Funnel	Lever Arms	Merry go bearing	Stepped Bearing	Four Points	Gravity Assist	Gear Turner	Air compress
Meets Loading Requirements	D	+		+				-		-
Is Safe			+	-	-	+	-		+	+
Is portable	A		-			+		+	-	
Is compatable with the 4 Bearing sizes		-	+	+	+		+			
Outputs torque vs applied load graphs	T	+								
Outputs friction vs applied load graph		+			-					
Fits within an office Space						+	-	-	-	-
Electric power	U			-						
Rotates the bearing				-	+					
Easy to maintain		+		-	-		-	-	-	-
Affordable	M		+				-			
sum +		4	3	2	2	3	1	1	1	1
sum -		1	1	3	3	0	4	3	3	3
sum		3	2	-1	-1	3	-3	-2	-2	-2

The reaction load analysis design was used as the datum and was compared to all other designs. The team eliminated any design that received 3 or more negative signs in the Pugh chart. This was done so only designs deemed competitive against the datum would be presented to Honeywell. The Datum, and the three design that passed this criterion were passed on to Honeywell and were considered within the team’s decision matrix shown in table 4.

Table 4: Decision Matrix

	Ease of Assembly	Cost	Size	Ease of Maintaince	Ease of Use	Precision	
	# of moving parts	# of parts that are expensive	How big	# wear parts	# of steps to operate	# of sources of error	
Weight	4	6	6	5	4	7	Total (Lowest Score wins)
Reaction Load Anaylisis	3	5	3	2	5	3	111
Inner race Design	3	5	3	2	3	3	103
Step Shaft Design	3	5	4	2	5	4	124
Funnel	6	4	2	4	6	7	153

The team then used the decision matrix to score the top four designs. The designs were scored based on six criteria; ease of assembly, cost, size, ease of maintenance, ease of use, and precision. The team assigned weighted values to each of these criteria. To quantify each design's score, a value was assigned based on the following parameters; number of moving parts, number of expensive parts, how big, number of wear parts, number of steps to operate, and sources of error. Do note, a higher score meant a design had more negative attributes, therefore, the design with the lowest score won. Ultimately, the Inner race design had the best score. The team presented this to Honeywell, and they agreed with the team's decision.

5.1 Design Description

The design selected by this process is shown in figure 8 as it was the only one that did not impart a load upon the outer race of the bearings. The team improved upon this design and pitched a system that utilized two jacks to impart a force radially by contacting the shaft at opposite ends, and one that loaded the shaft axially. Honeywell liked this idea better, therefore, the team designed the fully hydraulic system shown in figure 10.

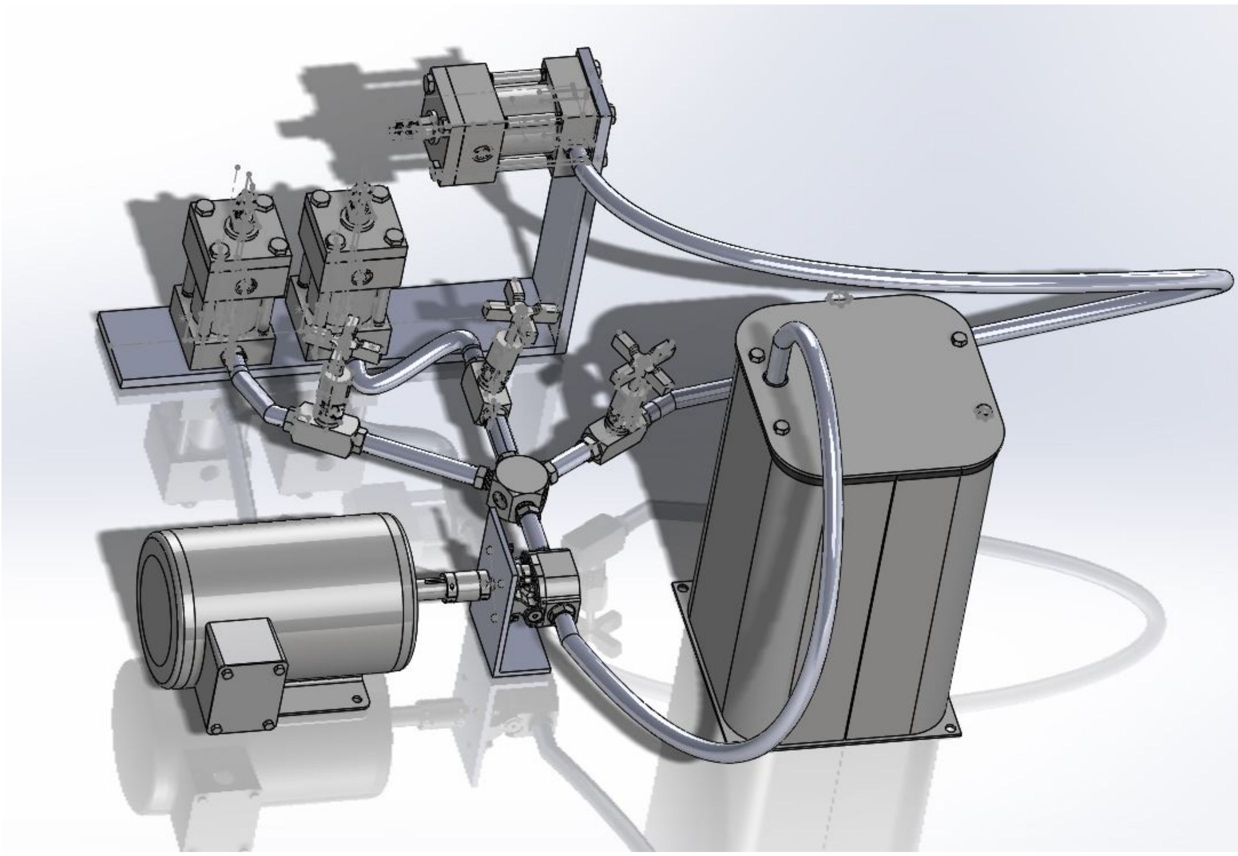

Figure 10: Solidworks Model of Design

This 3D model shows how the major components of this design came together. Those components included the shaft, supports, hydraulic system, and power source. An electric motor was to drive a hydraulic pump that pressurized fluid from the reservoir and flowed through flow control valves which could regulate the pressure of the fluid and the force of the rams as a result. The loads were to be applied to the bearings through the forces imparted upon the shaft via the hydraulic rams. The rams were to act on both sides of the shaft to keep bending moments from being imparted upon the bearing. The horizontally oriented ram was to apply the thrust load directly to the shaft. All force measurements were to be taken via manometers of pressure transducers leading to the rams. A rotary hydraulic actuator was to be used to spin the shaft instead of the stepper motor. A manometer or pressure transducer was to be connected to a line leading to the actuator, so a force could be measured and multiplied by a known lever arm to determine the torque needed to spin the shaft. All bearings would have been press fit to a shaft and placed within the bearing housing shown in figure 11.

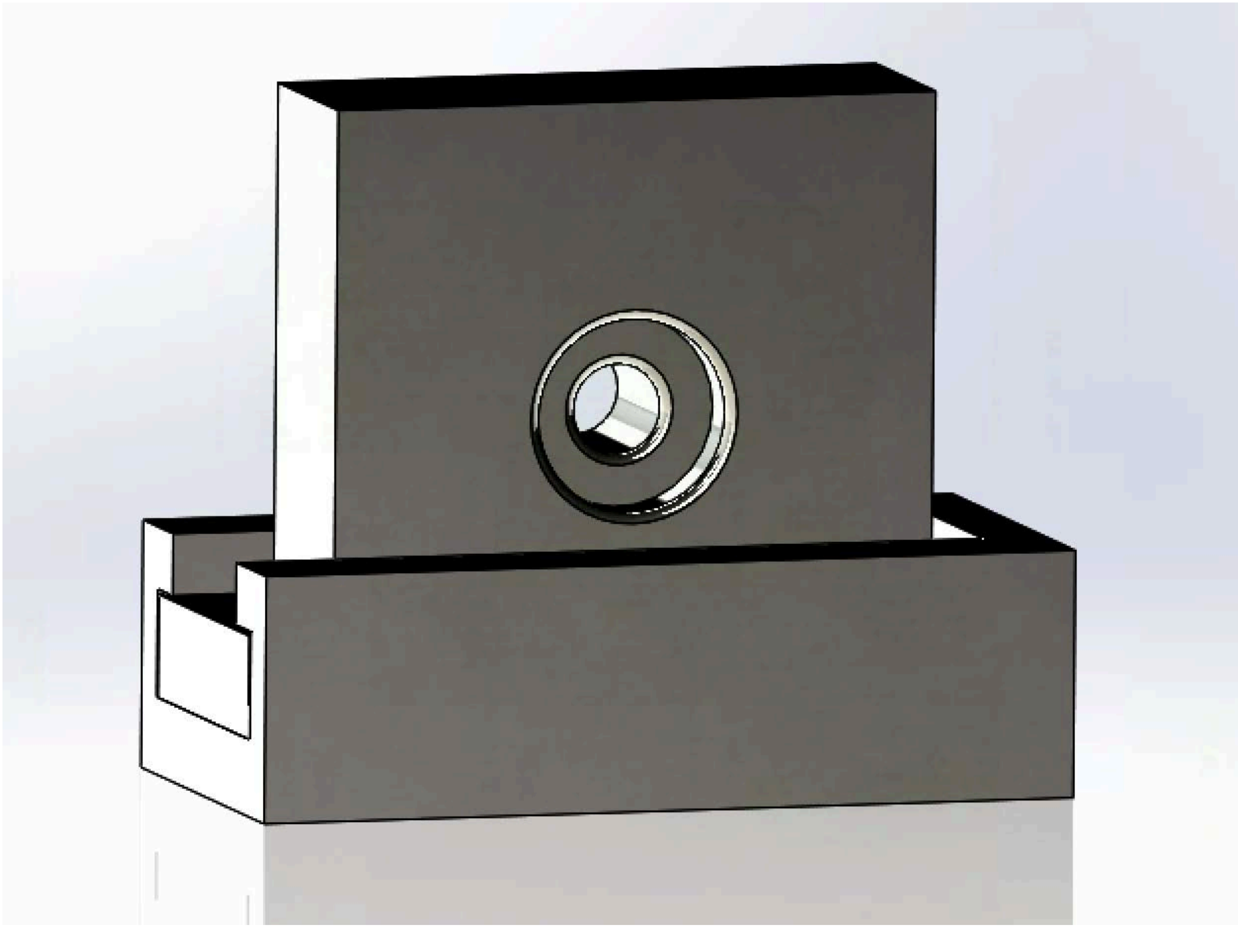
Figure 11: First Bearing Housing Iteration

This housing would have slid in and out of the frame enabling quick test bearing changes.

An exploded view of this design is depicted within figure 12.

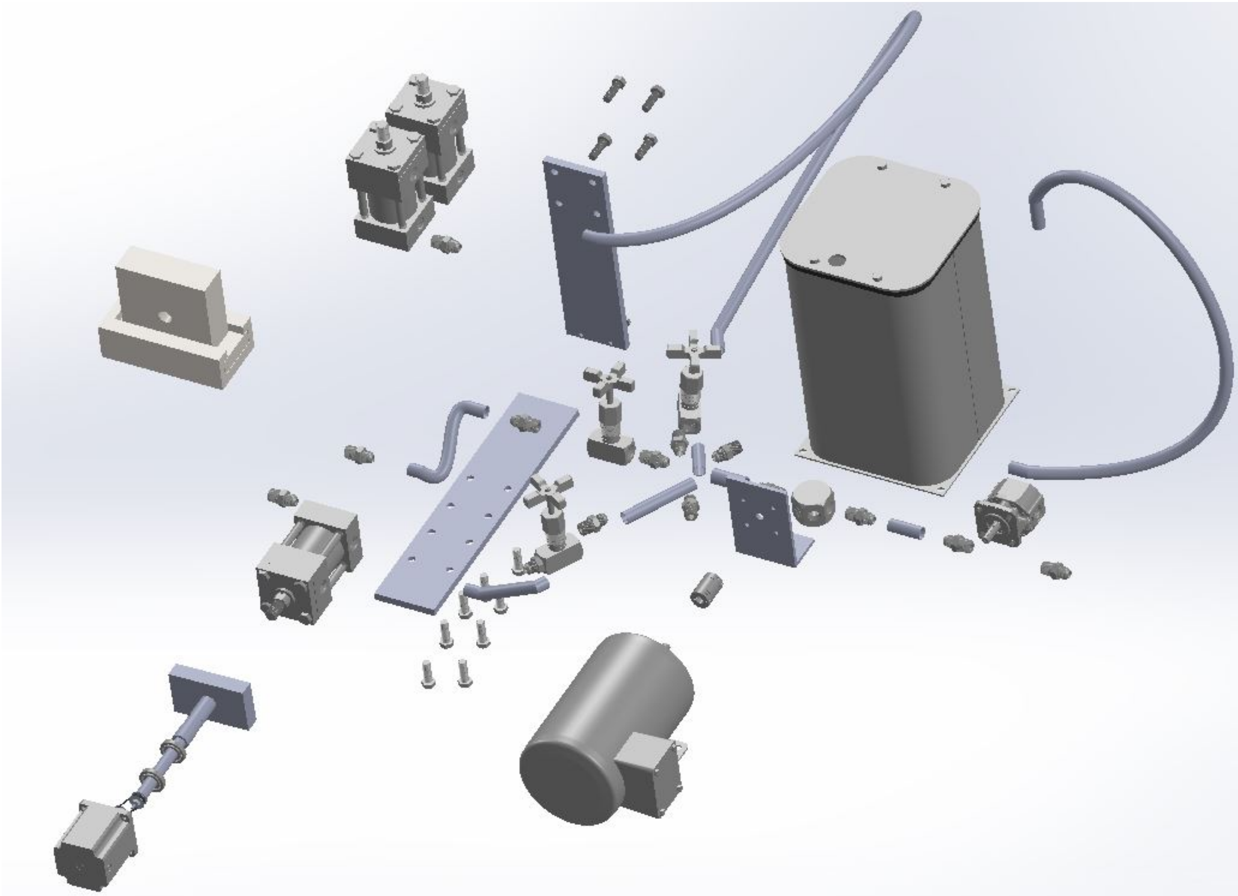

Figure. 12 Exploded view of CAD model

This exploded view shows vast number of parts that would have been needed to make this system work, it was determined that it would be better to outsource production of this system. This was also deemed safer than attempting to build it from scratch. Thus, the cost of all hydraulic components shown in table 5 were based off quotes from the company, Hydraulic Controls Incorporated.

Table 5: Hydraulics Budget

Part	Price per unit		Quantity	Total
Pump	580		1	580
Hose	20		5	100
Pump Motor	586.2		1	586.2
Piston	30		3	90
Metal	0.69	per in^3	200	138
Bearing	7		5	35
Shaft Coupling	41.73		1	41.73
Pressure Relief Valve	52.07		1	52.07
Flow contrl. valves x3	94.91		3	284.73
Hose fittings x15	7.42		15	37.5
Steel tubes (6ft)	22.38		1	22.38
Check Valve	15.24		1	15.24
1/2"-20 screws (pack of 10)	5.67		1	5.67
3/8"-16 screws (pack of 50)	9.18		1	9.18
Jack Manifold	163.63		1	163.63
Shaft Motor	300		1	300
Total				2461.33
Tax				246.133
Total Plus tax				2707.463

This design exceeded the team's max budget by about $1,200 dollars. As a result, the team had to investigate cheaper alternatives, specifically within the force application system. It was discovered that hydraulic car jacks work on a similar principle to the purely hydraulic system considered. They too utilized an electric motor and pump to pressurize fluid and force it through a ram to impart a force. These jacks could be had for far cheaper, thus a design using these was proposed to Honeywell.

6 Proposed Design

The design selected for this project is shown in figure 12.

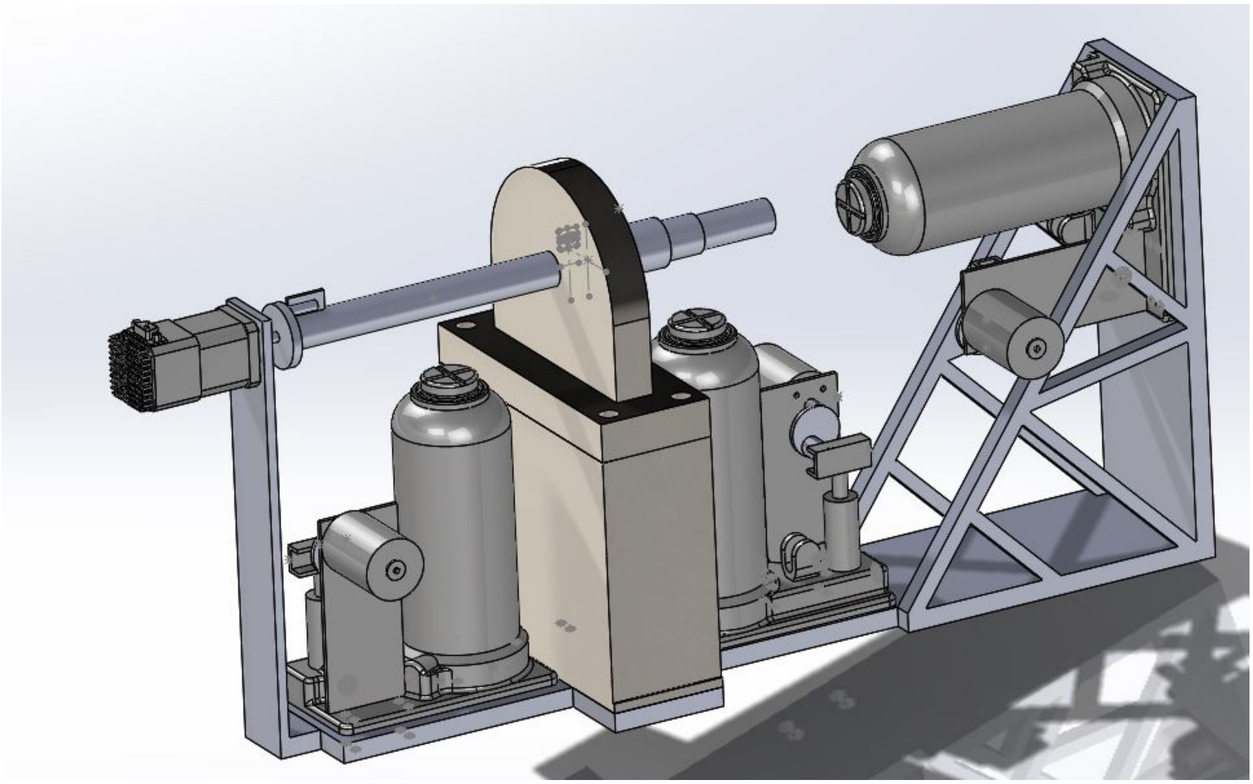

Figure 12: First Hydraulic-Jack Iteration

Both the team and Honeywell determined this would be the correct approach to take. It was a design capable of meeting all engineering requirements while staying within budget. It was also deemed easier to use given the jacks and stepper motor could be controlled through an Arduino system. The steps for implementation the team sought to use are shown in table 6.

Table 6: Initial Schedule

Build the Shaft	January 17 to February 10
Build the support	January 25 to March 10
Order the Hydraulic Parts	February 12 to March 10
Combine parts and Test	March 11 to April 15

This schedule was considered ideal and did not reflect what truly occurred as design changes were later implemented

A complete bill of materials is shown in table 7.

Table 7: Bill of Materials

Bill of Materials							
		Team					
Part #	Part Name	Qty	Description	Material	Dimensions	Cost per unit	Total Cost
1	Electric Hydraulic Jack	3	The hydraulic system that will make radial and thrust loads	Steel		84	252
2	Arduino Uno	1	Electronics used to run the Stepper motor	Electronics		100	100
3	Stepper Motor	1	Rotates the shaft			75	75
4	Metal	N/a	Shaft, support, etc	Steel		300	300
5	Pressure Gage	3	Measures the pressure change in the jacks calculate the force	Stainless steel	200in^3	100	300
6	Bearings	7	Test bearings, while also applying the force onto the shaft in radial and thrust	Steel		35	245
7	Torque Load cell	1	measures the torque	Steel		50	50
8	optical rotary encoder		Measures rotation and speed			40	1322

The cost of this design came in under budget and meets all design requirements. A list of parts needed and what requirements they meet are listed with table 7.

7 Implementation

This section will cover design changes implemented after the proposal and the process of manufacturing the fixture.

7.1 Manufacturing

The materials and products required to manufacture the test fixture are displayed in the bill of materials (BOM) shown in table 8.

Table 8: Bill of Materials

Bill of Materials								
Part #	**Part Name**	**Qty**	**Description**	**Material**	**Dimensions**	**Cost per unit ($)**	**Total Cost ($)**	**Source**
1	Electric Hydraulic Jack	3	The hydraulic system that will make radial and thrust loads	Steel	-	84	252	[Amazon]()
2	Arduino Uno	1	Electronics used to run the Stepper motor	Electronics	-	35	35	Amazon
3	Stepper Motor	1	Rotates the shaft		-	75	75	[www.robotshop.com]()
4	Plate	1	Used as Mounting point for all components	1018 Low carbon steel	32inx8inx1.5in	300	300	Pheonix Industrial Metal Supply
5	Plate	1	Used for bearing housing base, front and back of bearing riser and the motor mount	1018 Low carbon steel	20inx5inx0.5in			
6	Plate	1	Purpose is the same as other 0.5in thick plate	1018 Low carbon steel	22inx5inx0.5in			
7	Plate	1	Used for the thrust frame and right and left side fo bearing riser	1018 Low carbon steel	41inx3.5inx1in			
8	Square stock	1	Used for trusses	1018 Low carbon steel	32inx1.5in			
9	Cylindrical stock	2	Used to make all shafts	1018 Low carbon steel	47inx0.8750in diam			
10	Pressure Transducers	3	Measures the pressure change in the jacks, calculates the force	Stainless steel	-	74.33	222.99	[Amazon]()
11	Bearings	3	Placed on the shaft at points the jack contacts	Steel	-	51.67	155.01	Grainger
12	Crank	1	Presses onto motor, rotates the shaft	PLA Plastic	-	0	0	3D Printed
13	Electrical AC to DC converters	3	Enables the jacks to be plugged into a standard wall outlet	Electronics	-	54	162	Amazon
14	Arduino Driver	1	Needed to run the stepper motor	Electronics	-	15	15	Amazon
15	Screws	8	Needed to mount the Radial jacks	Grade 3 Steel	M6x60mm	0 (donated)	0	Copperstate Nut and Bolt
16	Screws	4	Needed to mount thrust jack	Grade 3 Steel	M6x50mm	1	4	Copperstate Nut and Bolt
17	Screws	4	Needed to mount motor	Steel	M5x40mm	0.5	2	Home Depot
18	Screws	4	Needed to mount bearing housing	Grade 8 Steel	1/4in long 1/2in diam.	1.5	6	Copperstate Nut and Bolt
	Total						1229	

This BOM is different from the one shown in table 7 as it is complete and showcases all parts used to build the test fixture. All fabrication work is completed within Northern Arizona University's (NAU's) machine shop in building 98c. Most milling and welding operations require work orders for the shop managers to complete as most team members lack expertise using these machines. However, team members can conduct drilling, reaming, and turning operations without issue.

The first part to fabricate is the base plate. The 1.5in thick plate is cut to about 20 ¾ inches in length. 6mm holes are drilled into it to provide a mounting point for the radial jacks. A 1in hole is drilled in between 6mm holes to provide clearance for the pressure transducers. Next, the 1in thick plate is cut to about 14 ¾ inches in length to create the base of the thrust frame. 6mm holes are drilled into this plate to provide the mounting point for the thrust jack shown in figure 13.

Figure 13: Thrust Jack Mount

A 1in hole is drilled in between the 6mm holes to accommodate the pressure transducers. The next part prepared was the motor mount shown in figure 14.

Figure 14: Motor Mount

This part is cut from a ½ inch thick plate. It measures about 9.9in long. Four 5mm holes and one 12mm hole is drilled into the plate to mount the motor. The completed motor mount is shown in figure 15.

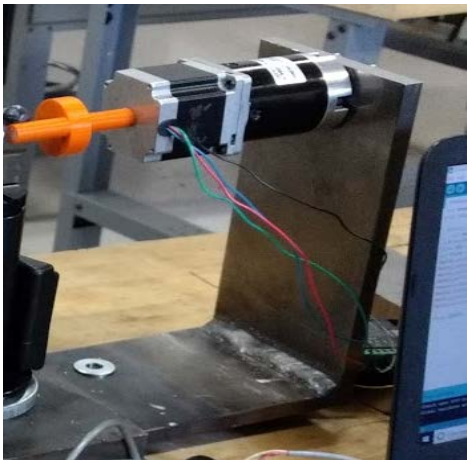

Figure 15: Completed Motor Mount

Another 1/2in plate is cut to about 6.65 inches in length to serve a spacer that provides clearance for the motor and crank.

The first part to weld is the bearing riser shown in figure 16.

Figure 16: Bearing Riser

The front and back plates are cut from a ½ in thick plate of steel, whereas the left and right plates were cut from a 1in thick plate, so threaded ½ in holes could fit on them. These plates are then welded together and welded to the base plate. Production of the bearings housings and shafts is conducted next. All housings are cut from a 1in thick plate and are 5in long. A mill is used to create the holes the bearings slide into. The shafts are all turned on a lathe. The results of these operations are shown in figure 17.

Figure 17: Shafts and Bearing Housings

Shown are the housing and shafts for the R14 bearing (1), R12 bearing (2), R8 bearing (3) and 38 bearing (4). The housings screw into the bearing riser via 1/2 in-20 screws. The bearings are press fit to their respective shafts. All shafts feature a threaded end to which a cap threads onto. This is shown in figure 18.


Figure 18: Shaft Cap

This cap enables radial load bearings of the same size to be used on both sides of the shaft and features a flap that enables the shaft to be spun by the crank and motor combination. Pressure transducers are connected to each jack, so pressure readings can be converted into electrical signals, uploaded to a computer, and converted into force readings. To do this, the base plate of each jack was tapped using ¼ NPT pipe threading, so the transducer could be placed in the bottom. The jacks were tapped directly where the pressurized fluid meets the piston. This is shown in figure 19.

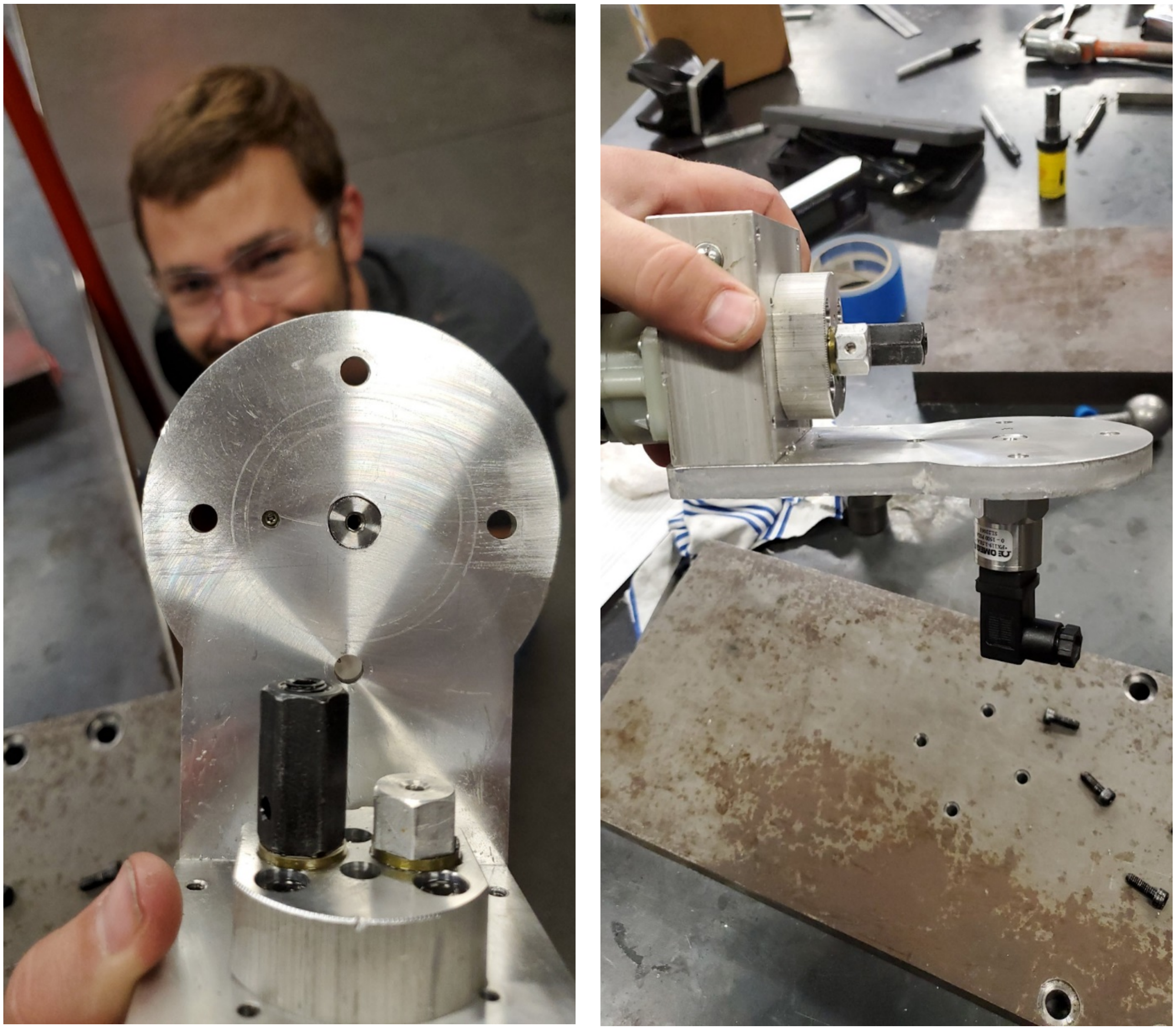

Figure 19: Pressure Transducer Mount

After tapping the jacks, they are reassembled, attached to the frame and filled with hydraulic fluid. This concludes the manufacturing work. After the motor and crank is installed and electrical equipment is hooked up the fixture should appear as shown in figure 20.

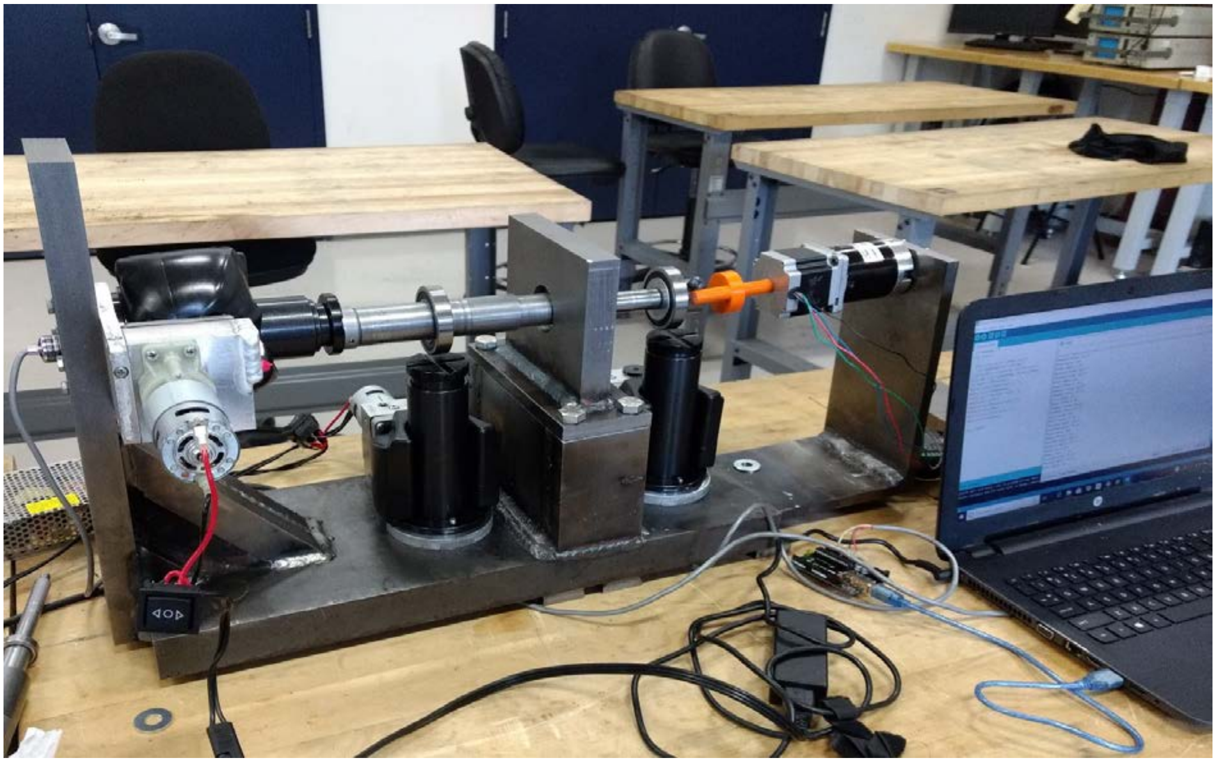

Figure 20: Final Assembly

This assembly does not appear like the design shown in figure 12 as several design changes were implemented based on weight reduction, parts availability, and ease of machining. The timeframe needed to build this fixture is depicted in table 9.

Table 9: Revised Schedule

Task	Date Completed
Buy raw materials	2/28/19
Make the bearing support	3/14/19
Make bearing support riser	3/15/19
Buy prefabricated parts	3/16/19
Machine shaft	3/20/19
Machine Stepper Motor Holder	3/25/19
Machine Jack Holder	3/26/19
Assemble everything	4/22/19

This schedule represents the timeframe the team needed to complete all work.

7.2 Design Changes

There have been several changes to many of the systems within the overall design. Most of these design changes were a result of budgetary or machinability issues.

7.2.1 Bearing Support System

The bearing support system is a crucial part of the design and needs to be able to hold the test bearing firmly in place. The previous bearing support design consisted of a bearing support base and four bearing supports, one for each test bearing size. The goal of the design was to be easily interchangeable. This design can be seen below in figure 7.2.1.1.

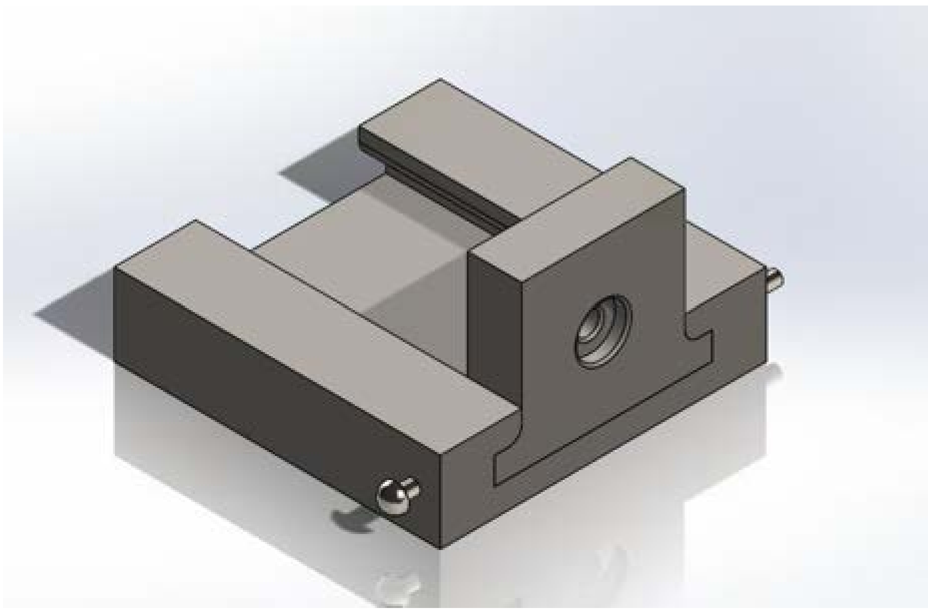

Figure 21: Sliding Frame and Bearing Housing

The issue with this design was the amount of material needed to make the bearing support base and the time required to machine it. To resolve this problem the bottom of each bearing support was widened to allow for four holes to be drilled out. These holes are for bolts that will hold the bearing support down to the frame. This makes it slightly more time consuming to exchange the bearing supports, but this change reduced the amount of material needed as well as the overall weight of the machine. Figure 22 below shows the current design.

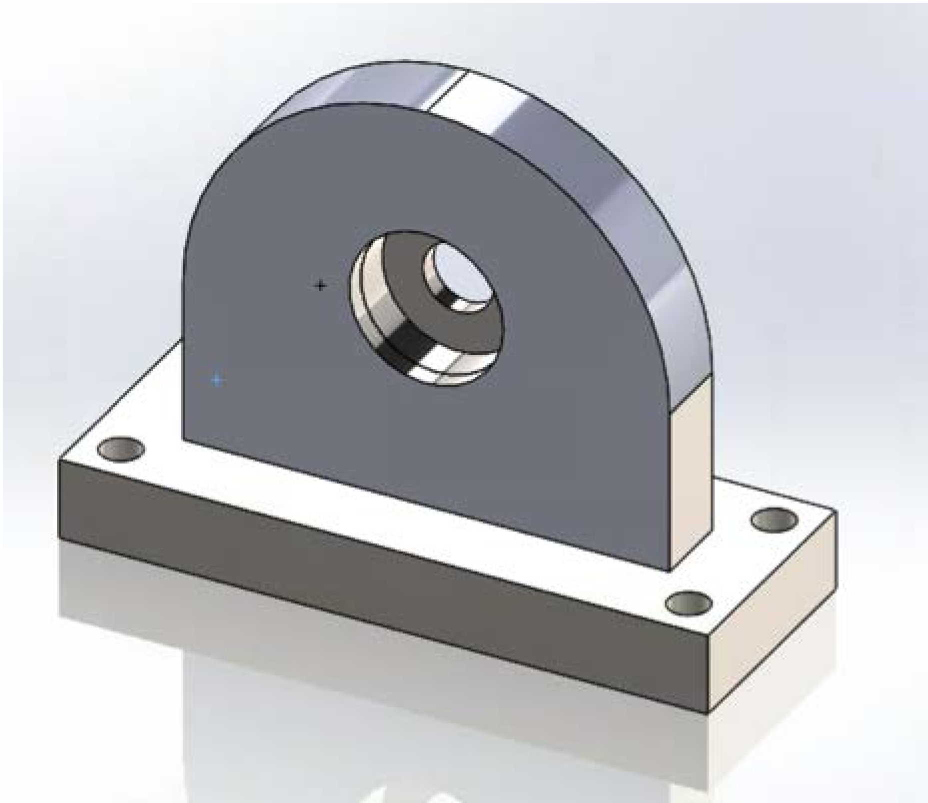

Figure 22: Bolt-on Bearing Housing

SolidWorks simulations were run on each R14, R12, R8, and 38 bearing support. The bearing supports passed the simulations with a minimum factor of safety of 2. Simulations were also run on the grade 8 ½” -20 bolts and they passed with a factor of safety of 4. Overall, the redesign lowered the amount spent on this system by over $100.00USD and cut over 4 hours off the machine time. The team determined later that the filet was not necessary and would simply add to the time it took to machine it. Therefore, the edges were left square.

7.2.2 Force Application System

The force application system has gone through three major redesigns due to budgetary constraints and new design considerations. It was initially determined that a fully hydraulic system would best meet Honeywell’s needs as it allows for the most precise load application. Direction control valves could be used to shut off flow to the ram as soon as the correct pressure had been achieved. It was soon discovered that the cost of this system would be prohibitive as a quote from Hydraulic Controls Inc. placed the price at $2,457.26 not including the hydraulic rotary actuator and its associated flow control valve [14]. The budget for the entire project is $1,500, deeming this unacceptable. An attempt to redesign the system by combining the valves offered by Hydraulic Controls Inc. with cheaper rams and a cheaper motor/pump combination yielded a price of $1,210 [14], [15], [16]. This price represented 80.7% of the budget which was too much considering only the thrust and radial loading requirements would be fulfilled. Attempts to spin the shaft and electronically record force data would exceed the budget. An attempt was made to receive additional funding from Honeywell, but this failed, therefore a new paradigm was pursued.

7.2.2.1 First Car Jack Concept

The team determined to utilize car jacks to cut costs. These are jacks that utilize a pump and motor, or motor and lever to exert hydraulic pressure. The first iteration of this design was proposed to Honeywell, it is shown in figure 12. The advantages of this design revolve around simplicity and size. These jacks are thin and inexpensive allowing budget issues to be resolved, volume constraints could be easily met, and bending within the shaft could be kept to a minimum. However, these jacks are not readily available off the shelf. They are modified hand operated jacks. To make these a reality, extensive machining and would need to be done to mount a motor, make a crank, and modify the hand pump to accept those components. There was not enough time to do this, so an off-the-shelf jack was found, and a new design was based around it.

7.2.2.2 Final Car Jack Design

Most off-the-shelf electro-hydraulic jacks utilize a motor and pump. Thus, a force application system utilizing these jacks was designed. It is shown in figure 23 below.

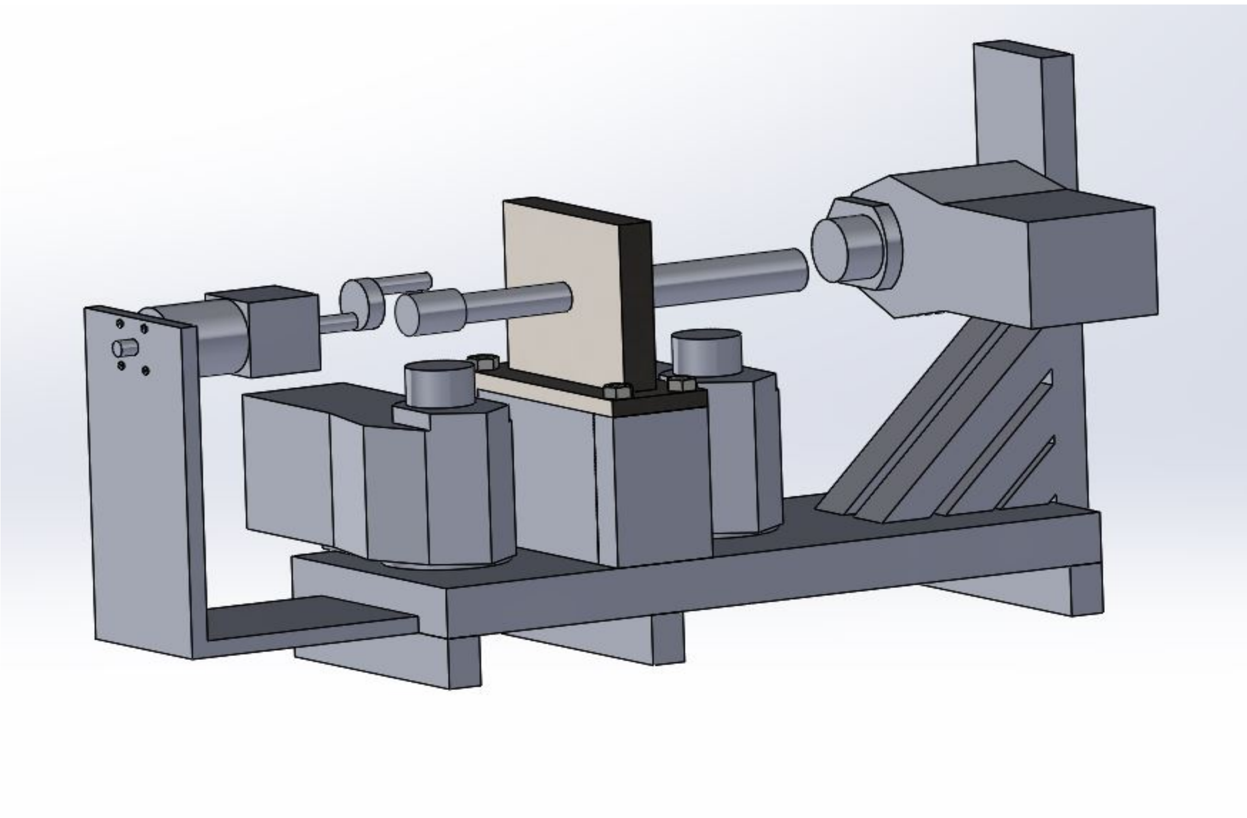

Figure 23: Final Electro-Hydraulic Iteration

Along with being more readily available than motor-and-crank operated jacks, these jacks are shorter. This results in the moments experienced within the bearing housing supports and thrust load frame are reduced. Additionally, underneath the plastic cover, the three plugs shown in figure 24 offer access to the hydraulic fluid.

Figure 24: Jack Plugs

This enables one to more easily maintain the jacks assuming there is a malfunction necessitating the evacuation of air or addition of hydraulic fluid. These jacks exhibit many advantages, but they are wide. Thus, a longer frame is required to fit the jacks and the radial forces are applied further away from each other leading to greater bending in the shaft. Nevertheless, the advantages of these jacks outweigh the disadvantages and are immediately available. Overall this system costs $252 not including the steel, thereby freeing up the budget for other expenses. These jacks are rated for 3-tons and will be able to meet all loading requirements.

7.2.3 Force Measurement System

The team has research multiple methods of force measurements and adjusted the design accordingly. Initially, the team wanted to use strain gages because they are cheap and readily available. However, while the gages themselves are inexpensive, an application kit is required which increases the price. Also, due to the rotation of the shaft, a simple perpendicular orientation of strain gages is not sufficient. The strain gage measuring the radial force would measure a different strain with every degree of rotation resulting in inaccurate data. Another design was considered: putting strain gages on the bolts that hold the bearing support. However, since the axial force imparts a moment on the support, each bolt would measure the radial force of the shaft and this moment which complicates the system. For these reasons, the team is going to tap the electro-hydraulic jacks to obtain force readings. This is done with a pressure transducer measuring the pressure of the hydraulic fluid and converting that to a current between 4mA and 20mA or a voltage between 0V and 5V. This output can be read by an Arduino unit and converted to a force. The pressure transducer is approximately $100 [1]. Three pressure transducers will be ordered so that the force can be determined in each jack independently.

7.2.4 Shaft Design

The shaft is crucial for applying the radial and thrust load to the test bearing. There is a shaft for each bearing size. Originally, all shafts were to match the diameter of the inner race of the bearing and then step down to go through it. The issue with this was that the load bearings would need to match the diameter of the test bearing requiring different load bearings to be purchased for every shaft. Therefore, a cap was designed so the same load bearings could be used for all shafts. This is shown in figure 25.

Figure 25: Initial Threaded Shaft Design

It was later determined that R14 load bearings should be used on every shaft. The shafts the team purchased could already accommodate these, thus, it was determined all shafts could be stepped down to accommodate different bearings and a far smaller cap could be used. Therefore, the design was changed to the one shown in figure 26.

Figure 26: The new design

The cap also features a flap that will enable the crank to rotate the shaft.

7.2.5 Torque Application

One of the requirements outlined by Honeywell is that torque must be measured in real time with respect to the applied loads. To do this the shaft must be spun and the force it takes to do so must be recorded. The system shown in figure 27 was initially envisioned to complete this task.

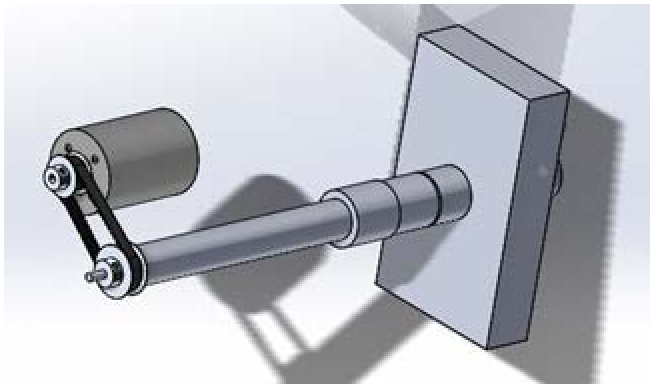

Figure 27: Belt and Pulley System

A motor is connected to a pulley that turn the shaft and force is recorded via a strain gauge on the shaft. However, further analysis of this concept revealed that deflection in the shaft would be so small that strain gauges would be unable to detect it and thereby be unable to record a force. Other methods of torque application were devised.

7.2.5.1 Rotary and Linear Hydraulic Actuators

When a fully hydraulic test fixture was envisioned it was determined that the torque application should also be hydraulically actuated. Initially a hydraulic rotary actuator was considered, but a quote from Hydraulic Controls Inc. revealed it would cost $486.21 [14]. This constitutes a third of the budget pushing the team to pursue a cheaper solution. A hydraulic linear actuator utilizing a $136 ram a clevis rod [15] was proposed. However, this too introduced new challenges. A custom rod and crank would need to be machined to turn the shaft 90º complicating the project. Additionally, a flow control valve and direction control valve would be needed to ensure the rod spins the shaft at 10º/s or more and is pressurized independently from the other rams. These components cost $33.51 and $127.28 respectively placing the total cost of this system at $296.79 not including the steel needed for the mounts and custom parts or the pressure transducer needed to find the force applied to the system [14]. This price was still considered unacceptable, so a new paradigm was explored.

7.2.5.2 Stepper Motor and Crank

A stepper motor was determined to be a better solution than any hydraulic actuator as it was cheaper and could simply be programmed through Arduino to turn 90° at 10°/s. The first iteration of this system is shown in figure 28.

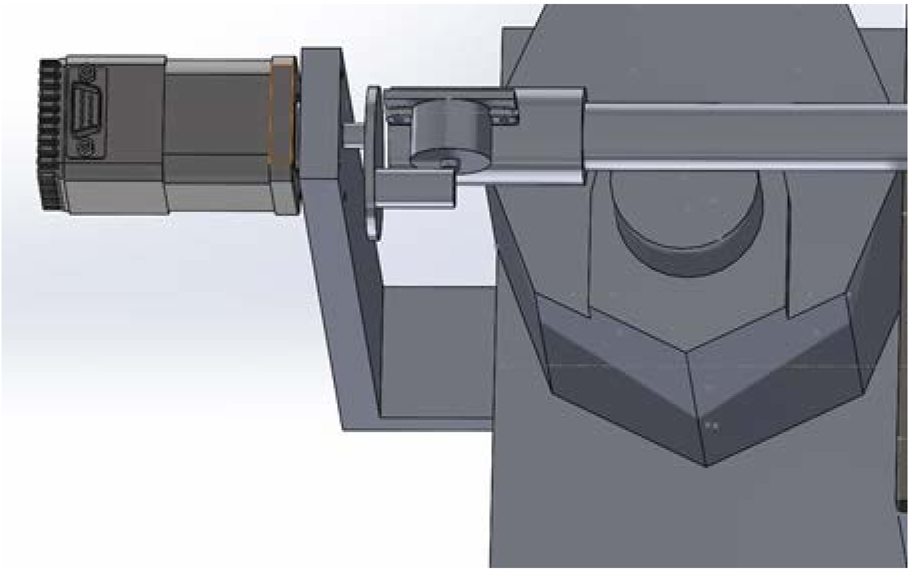

Figure 28: First Crank and Motor Iteration

This torque application system utilized a $430 stepper motor and crank to turn the shaft [18]. This motor was too expensive, but it was used to illustrate the concept. The first crank designed for the fixture is shown in figure 29.

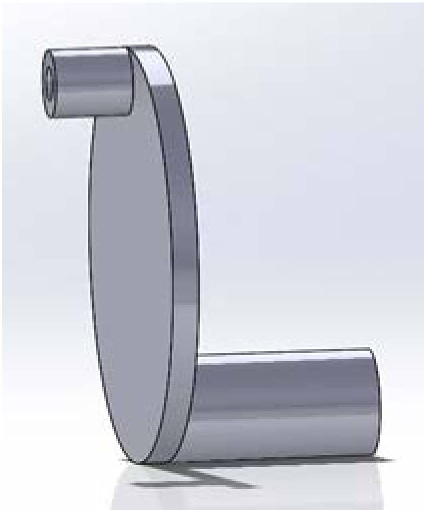

Figure 29: First Crank Iteration

The crank is a one-piece system than consists of a shaft coupling, base and arm that meets the force sensor. It is shaped in this way to minimize the amount of material needed to make the crank while maximizing the arc of the lever arm. Eventually a far cheaper stepper motor was found leading to another iteration of this design shown in figure 30.

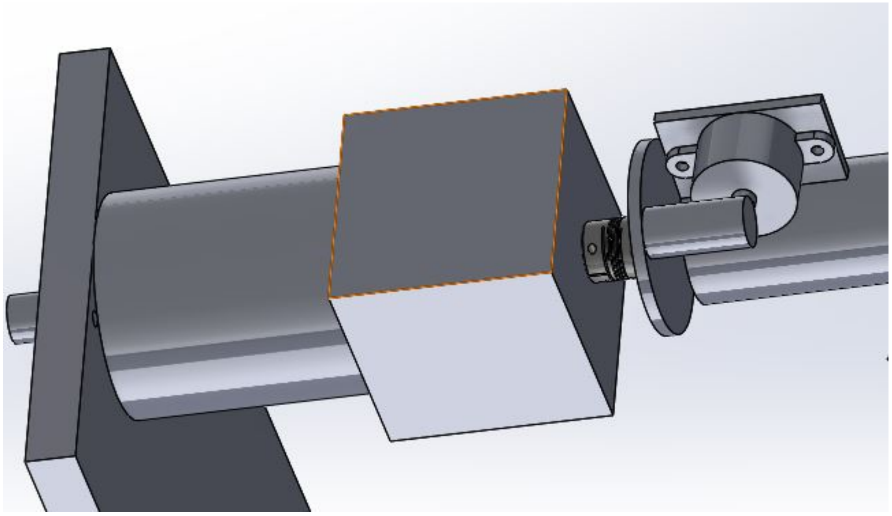

Figure 30: Second Crank and Motor Iteration

This stepper motor is larger but has an output shaft too small to be placed on the back of the ½ in motor mount, therefore it was placed on the front of it. The back of this motor was exposed to allow a rotary optical encoder to be mounted. This would have enabled the system to record the speed and position of the shaft. The crank was slightly redesigned as shown in figure 31.

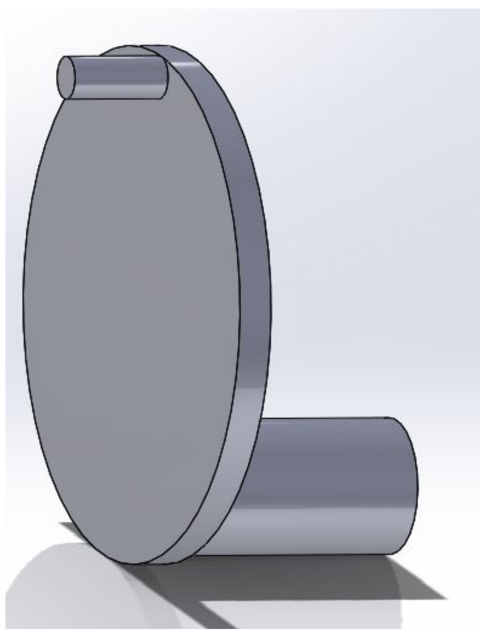

Figure 31: Final Crank Iteration

The shaft coupling was replaced with a uniform shaft that fits into a separate shaft coupling as this is expected to provide a better connection between the crank and motor. This shaft was made short enough to accommodate the worst case +0.7mm length tolerance listed for the stepper motor's output shaft [20]. The motor, optical rotary encoder, shaft coupling, and force sensor cost $201.82. This system was within budget and could satisfy all requirements. However, the team could not find the time to order a shaft coupling. Additionally, the distance between the crank and shaft was found to be too great necessitating a redesign. This is shown in figure 32.

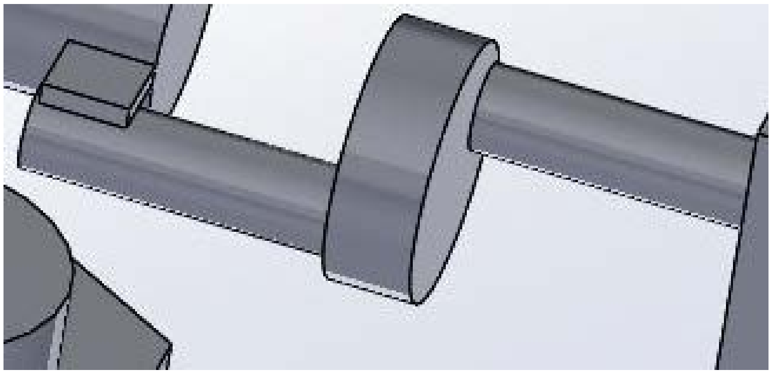

Figure 32: Final Crank Design

The crank was ultimately 3D printed and a shaft coupling was added back onto it to save time. The final crank iteration appears as shown in figure 33.

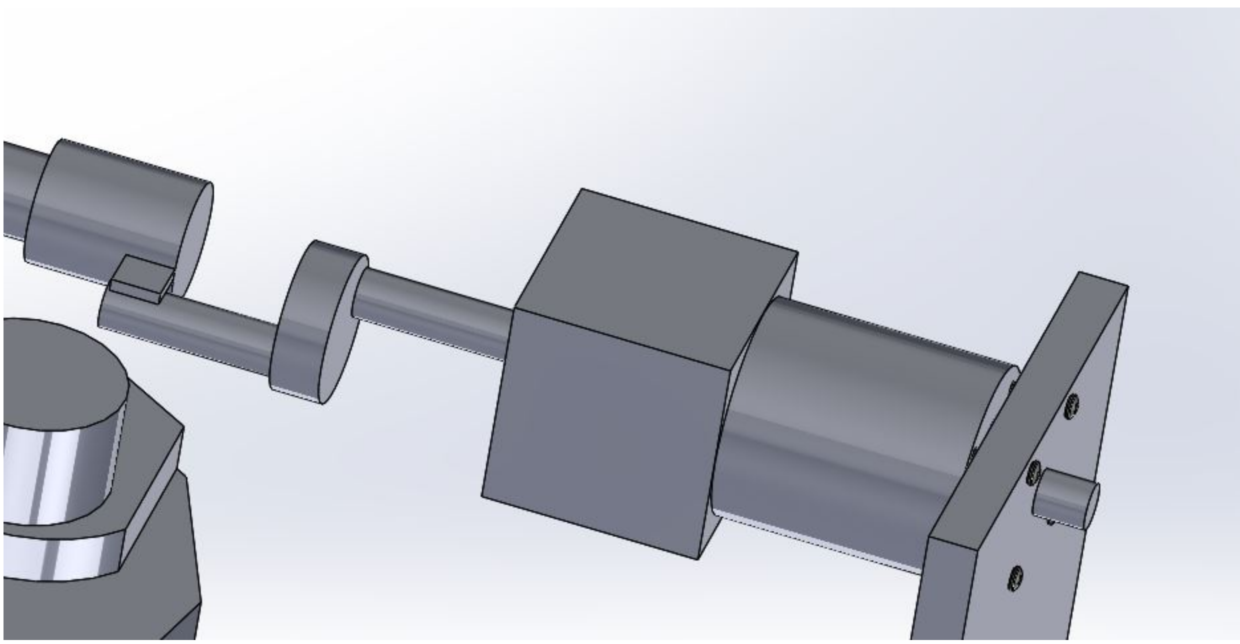

Figure 33: Final Crank and Motor Iteration

This design proved capable of spinning the shaft under a light load or no load but struggled under heavy loads as the shaft would bend.

8 Testing

Section 2.3 discussed the testing procedures the team would take to ensure requirements could be met. This section will discuss the results of those tests and their implications. The first tests were the FEA analysis done in Solidworks. This analysis was to apply to the shafts though it was conducted for the bearing housing and thrust frame as well to ensure no bending would occur. The results of the shaft analysis are shown in figure 34.

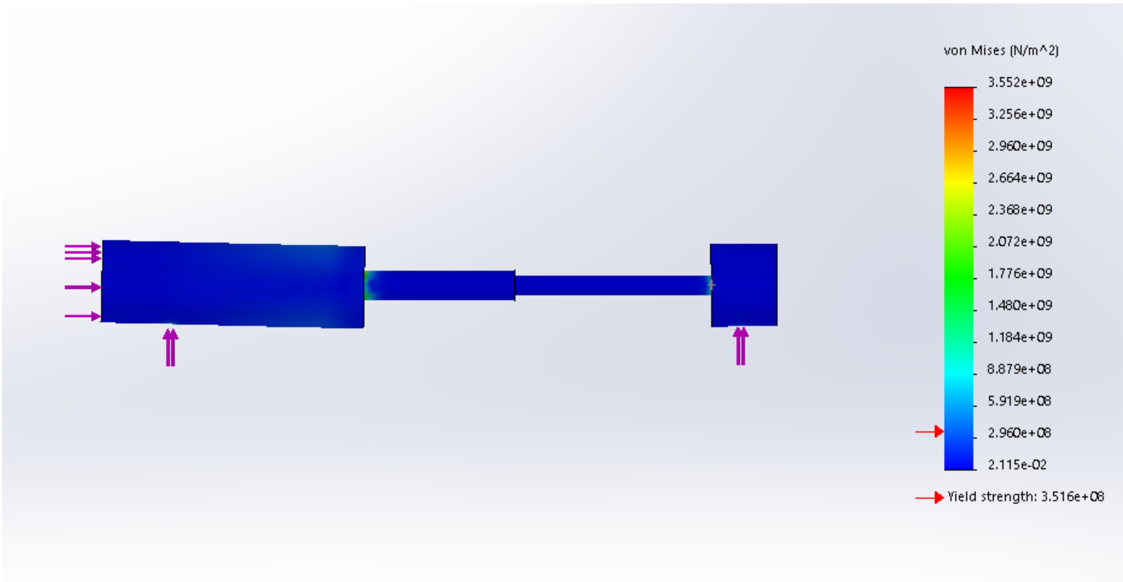

Figure 34: Shaft Analysis Results

The stresses indicated by this analysis suggest that the shaft never yields as the maximum stress it experiences is $2.9\text{x}10^6$ N/m^2 while the yield strength of 1018 low carbon steel is $3.516\text{x}10^8$ N/m^2. These results were further confirmed by a MATLAB program utilizing the following formula.

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

$$\tau = \left(FL\left(\frac{d}{2}\right)\right) / \left(\left(\frac{\pi}{64}\right)\left(\frac{d}{2}\right)^4\right)$$

Where σ is stress, F is force, A is the cross sectional are of the shaft, τ is shear, L is the length of the shaft, and D is the diameter of the shaft. The results of this analysis also determined the shaft would not yield. Certain assumption was made. The shafts were to be loaded 3.5in away from the bearing housing and had about 1 in of contact for support. Loads were also simulated with 1020 steel instead of 1018 steel. These assumptions proved too optimistic. The 1in of contact support was diminished after it was determined the shaft could not contact it if the shaft was to spin. As a result, the R12 shaft bent when loaded slightly above its maximum radial load of 3,800 lbf. FEA analysis was conducted on the bearing housing as well. The results of this are shown in figure 35.

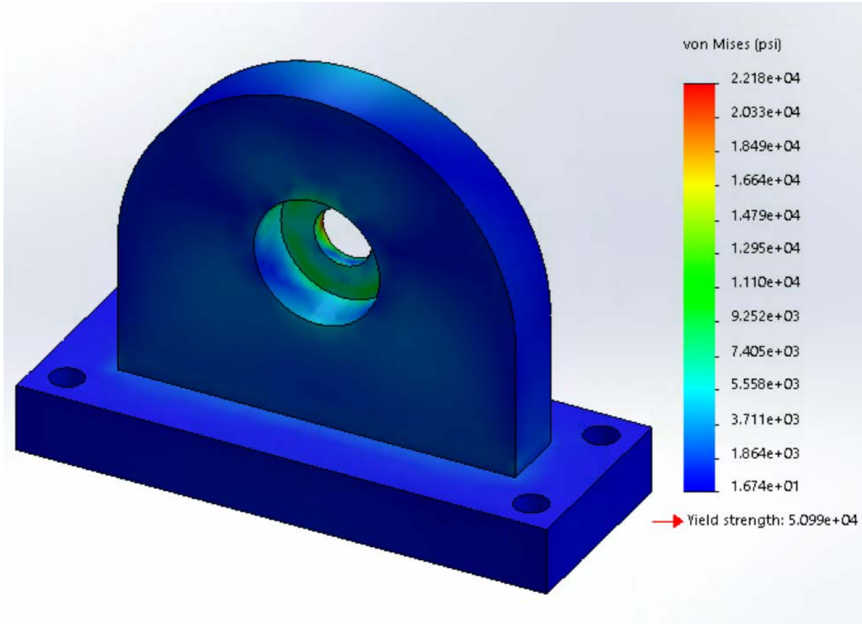

Figure 35: Bearing Support Simulation

This analysis predicted the housing would not fail as the maximum stress it experienced is around 9.25×10^3 psi and the yield strength is 5.099×10^4 psi. When loaded in reality the housing had no trouble taking the load. The thrust frame was also tested in Solidworks FEA, the results are shown in figure 36.

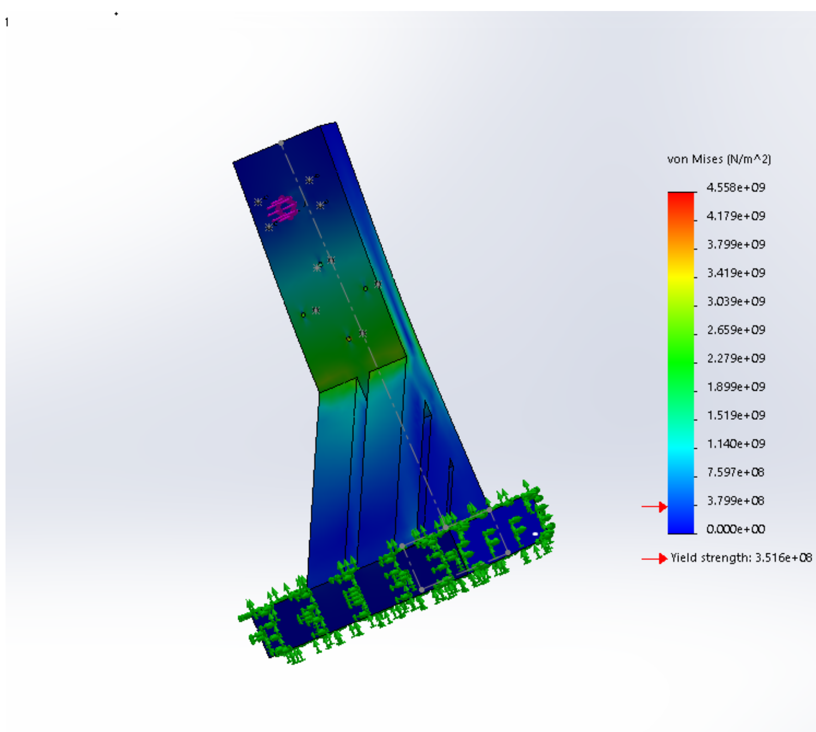

Figure 36: Jack Support Simulation

The assumptions made for this scenario were pessimistic. The load was placed several inches higher than where it would be in reality. However, the simulation still showed that the design never yields under the maximum thrust load of 4,000 lbs. When loaded in reality, the thrust frame performed as expected.

The team then tested jacks to ensure they could apply the maximum loads. The pressure transducers had to be calibrated first, however. To do this Nestor, who weighs approximately 150 lbs, stood on one jack while another member monitored the pressure displayed on the laptop. This pressure was corrected. To verify this worked, the team checked to make sure the pressure returned to zero after Nestor stepped off. After this was done the team attempted to impart a load of 3,800 lbs. to the R12 shaft. The team did not have an R14 bearing so it could not attempt to impart an 8,000 lb. load. Ultimately the team ended up imparting a 4,000 lb. load which was 200 lb. over max load. This deformed the R12 shaft which was unexpected, however the jacks performed as expected.

To test the weight constraint two team members picked up and maneuvered the fixture. This was done successfully, though the team cannot recommend having two people move the fixture for extended periods of time.

To test the volume the constraint the fixture was measured. It is about 28 ¾ inches in length, 12 ½ inches in width, and 15 ¼ inches in height. Therefore, the volume amounts to 5,480.5 in^3 or 3.17 ft^3, which is well within the volume constraint.

To ensure the fixture could be powered by a wall outlet the jacks were hooked up to a converter which was then plugged into the wall. Upon being turned on, they started up. This confirmed that everything could run off a 120V 15A electrical outlet.

The final test was to ensure the motor could rotate the shaft. An attempt to rotate the shaft using the stepper motor with no load was attempted first. The machine passed this test. The team then attempted to rotate the shaft under load. The motor was able to rotate the shaft until it began deforming. The team expected to pass this test since under simulation the shaft did not deform, and the motor provided the torque needed to rotate shaft under load as claimed by the client. After testing, the team realized the shafts needed to be made with a stronger material or the loads needed to be located closer to the bearing housing. The team was unable to make these changes due to time constraints.

Due to the destruction of one of the shafts, and the inability to spin the shaft the team was unable to quantify friction and produce a plot of friction vs. the applied loads.

9 Conclusions

The team failed in its goal of quantifying friction within Honeywell's bearings, however, it is aware of what went wrong and how it could be fixed given the opportunity. Many valuable lessons were learned, especially in regard to manufacturing. This section will discuss what organizational aspects worked within the team, what could have used improvement. It will also go over lessons learned and provide an overview of results.

9.1 Contributors to Project Success

As stated in the charter the team's goal was to design and provide a machine that quantifies the frictional forces present in a variety of bearings. The team ultimately failed to reach that goal. The final product was unable to rotate the shaft; thus, it could not determine the frictional forces. Due to budget constraints the team had to alter the original design which affected the outcome. Many design aspects had to be modified to stay under budget, forcing the team to think creatively. The force application system was redesigned to use car jacks over a custom hydraulic system. The team reverse engineered the jacks to determine the best placement for a pressure transducer and determined a modification would be needed to do this. The simplest solution for determining the force output by the jacks was to place a load cell on the ram, but this was not feasible due to the limited budget.

When it came to team dynamics the group excelled at communication and following the team ground rules. Everyone understood that people were busy but also understood completing this project was apriority. Everyone kept this in mind and put as much time into the project as possible. When it came time for manufacturing, each member had a role. It was common for most, if not all, of the team to be working together in the machine shop. Issues seldom arose within the team dynamic. The coping strategies stated within the team charter were often more than enough to solve any problems.

The team performed best during the design and problem-solving processes of the project. Constant communication between all members enabled a fluid design process which helped to solve problems quickly and effectively. However, the group exhibited poor performance in manufacturing. The lack of previous experience in manufacturing proved to be the team's greatest downfall. Aware the team's inexperience, the group sought the help of the machine shop managers. The knowledge and expertise provided by the shop managers proved to be invaluable to the team.

Overall, each team member learned valuable skills which helped with this project and will help in future projects. A major skill learned was the use of FEA. This tool allowed the team to conduct accurate stress analysis on parts and was useful throughout the whole project. The team learned about hydraulic systems, force measuring devices, and data acquisition systems. These are all prevalent in engineering and the experience gained during this project will prove useful in the future.

9.2 Opportunities/Areas For Improvement

The team was unable to make a fixture that quantified rotational friction as a function of radial and thrust loads. This was the result of a few design flaws. To improve the project the loads would need to applied closer to the bearings, the shafts need to be made of a stronger material, and an automatic cut-off would need to be coded into Arduino to prevent overloading the shafts. Thinner jacks or hydraulic rams would be needed to reduce the distance between the bearing and load application to around 1in if the same shaft material is used. This would reduce the amount of stress and deflection the shaft experiences. After testing, the team realized the distance between the applied loads and the bearing holder were too great and that a shaft made from 1018 hot-rolled steel could not possibly handle the increase in stress associated with this. Making the shaft from a stronger steel, such as 4130 steel, would mitigate this issue. The last major change the team would make is implement an automatic cut-off for the jacks. When the team tested the jacks, it was discovered that they impart forces sporadically in a manner that is tough to control. This resulted in the jack being susceptible to damaging the bearing and shaft. A safety cut-off would prevent this.

There were also some obstacles encountered during the manufacturing process. The jacks needed to be reverse engineered which took a lot of time as it required some trial-and-error. The team initially tapped a low-pressure system on the jack, which required repairs. This same jack was used as the axial jack and succumbed to many issues. The jack had to be re-oriented and a zip tie had to be used to force hydraulic fluid into the pump. This partially resolved the issue, but the situation could have been avoided entirely if a better jack was selected, or if the selected jack was reverse engineered earlier.

While the team communicated effectively overall, peer collaboration could have made the project run smoother. Many issues in design could have been avoided if the team collaborated more frequently. Analysis were conducted individually and blindly accepted by all team members. This was not a problem for the most part though there were times errors in methodology were discovered far later than they should have been. As described earlier, the shaft deformed during testing due to overly optimistic assumptions being accepted. This may have been prevented if the team looked at the results of the analysis critically before building the shafts.

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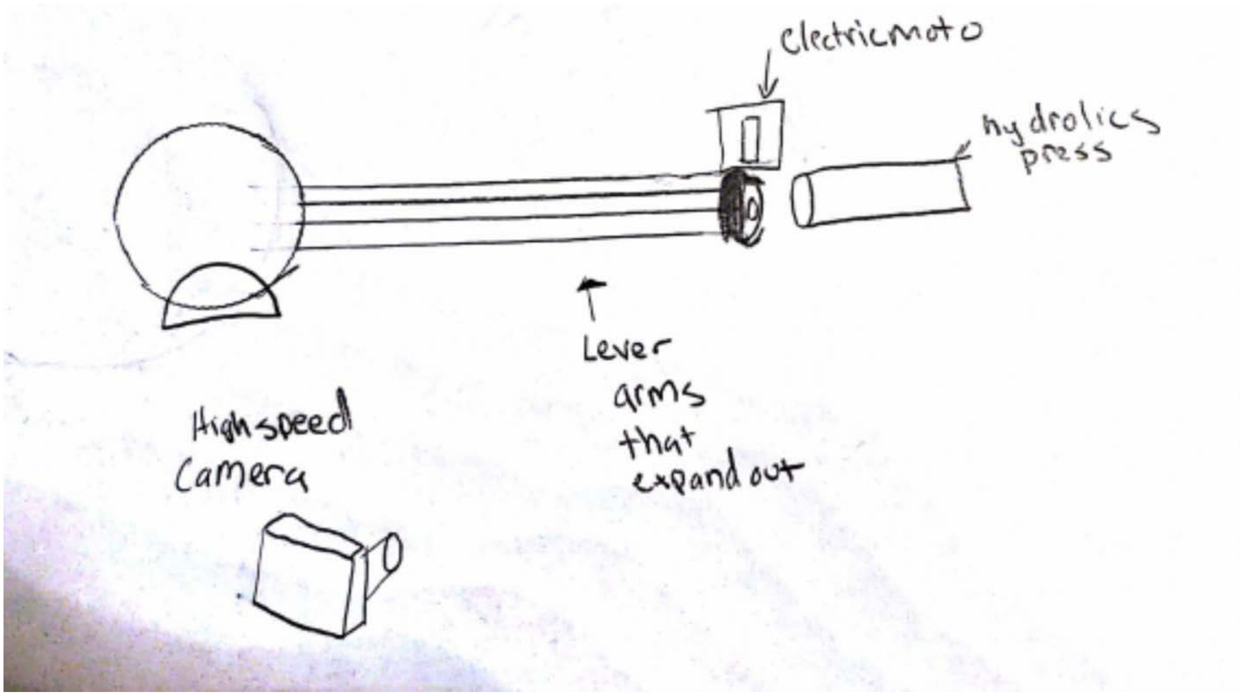
11 APPENDICES

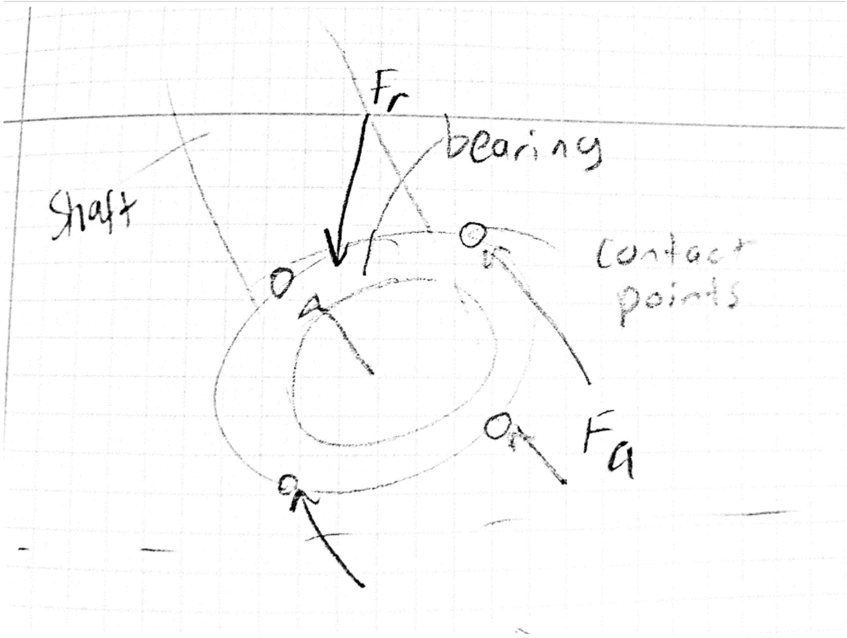
[Use Appendices to include lengthy technical details or other content that would otherwise break up the text of the main body of the report. These can contain engineering calculations, engineering drawings, bills of materials, current system analyses, and surveys or questionnaires. Letter the Appendices and provide descriptive titles. For example: Appendix A-House of Quality, Appendix B- Budget Analysis, etc.]

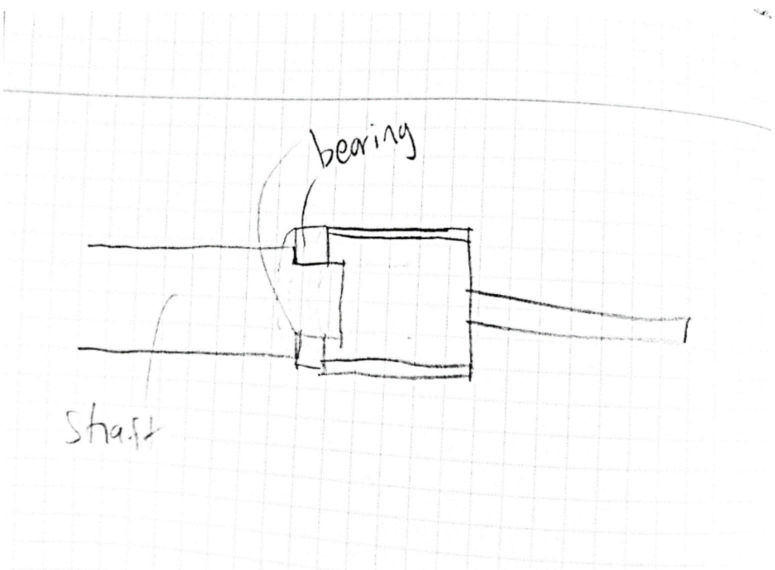
11.1 Appendix A: Rejected Designs

11.1.1 Lever Arms

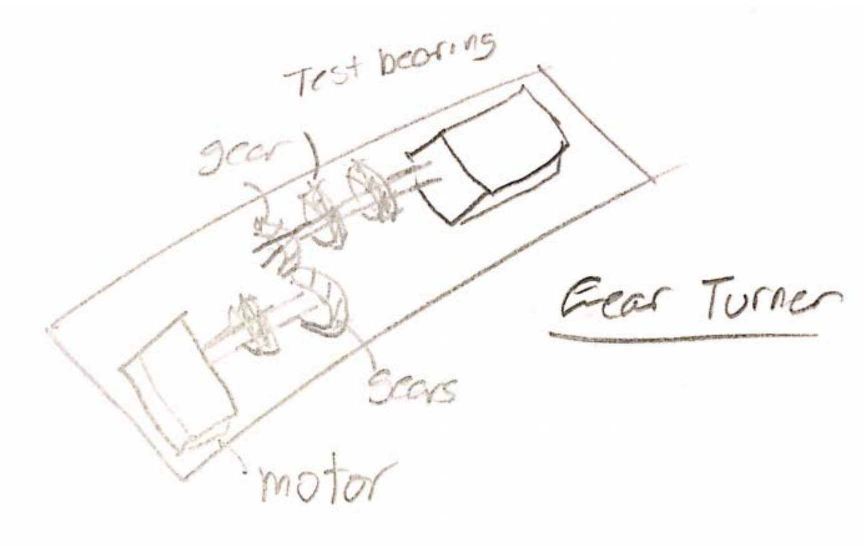


11.1.2 Four Points

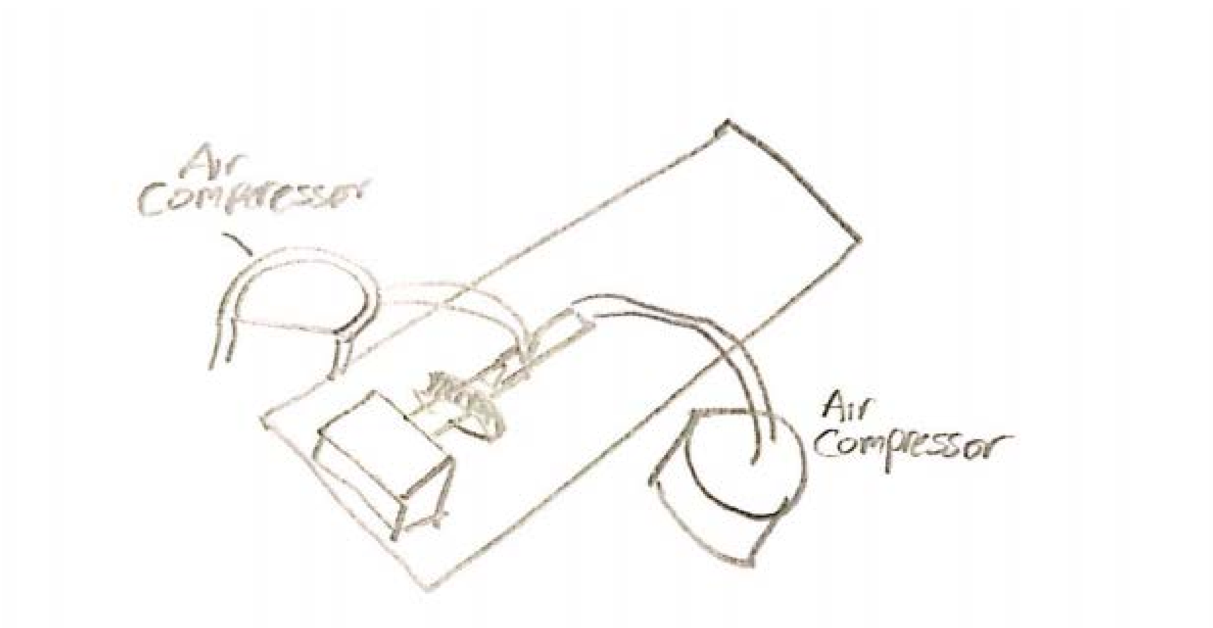




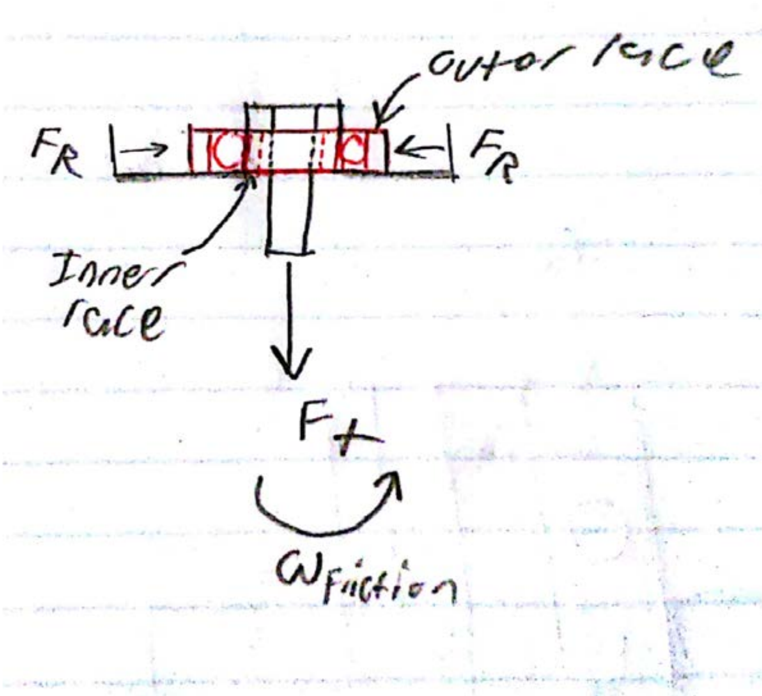
11.1.3 Gear Turner



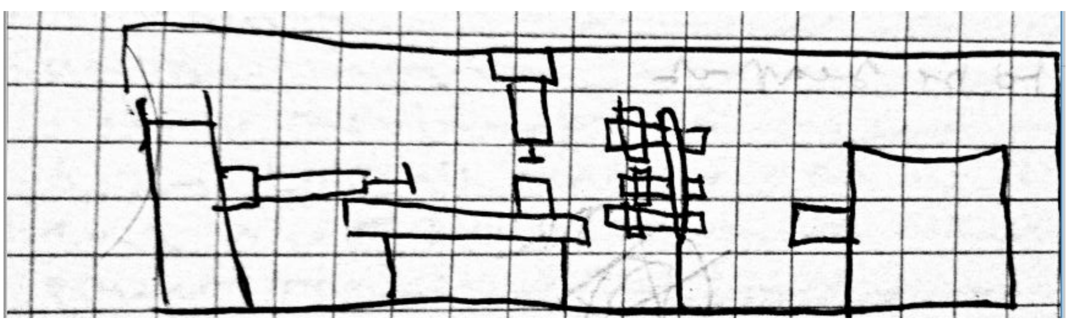
11.1.4 Air Compress



11.1.5 Gravity Assist



11.1.6 Merry-Go-Round



11.2 *Appendix B: Design Components*

11.2.1 QFD

System QFD

Project: **Honey Bear**
Date: 9/20/2018

#		Radial Loads	Thrust Loads	Weight	Volume	Speed	Cost	Voltage	Current	Material Strenght	Shaft size
1	Radial Loads										
2	Thrust Loads	++									
3	Weight	+	+								
4	Volume	+	+	++							
5	Speed										
6	Cost	-	-	--	-	-					
7	Voltage					+	-				
8	Current					+	-	+			
9	Material Strenght	+	+	++			++				
10	Shaft size			+	+					-	

Engineering Requirements

#	Customer Needs	Customer Weights	Radial Loads	Thrust Loads	Weight	Volume	Speed	Cost	Voltage	Current	Material Strenght	Shaft size
1	Meets Loading Requirement	4	9	9	3			1			9	
2	Needs Safety Equipment	4			3	3	1	1			9	1
3	Needs to be portable	2	3	3	9	3		3			3	3
4	Needs to be compatable with different bearings	4			1	3						9
5	Needs to output torque vs loads applied	3	3	3				3				
6	Output friction vs applied loads	4	3	3				3				3
7	Needs to fit in office space	2				9						3
8	Needs to be electrical power	2							9	9		
9	Needs to rotate the bearing	3					9					
10	Affordable	1	1	1	1	1		9			9	
11	Easy Maintaince	3			3	3					3	9
	Engineering Requirement Units		Lbf	Lbf	Lbf	ft^3	Degrees/sec	$	V	Amps	lbs/in^2	in
	Engineering Requirement Targets		8000	4000	300	9	10	1500	120	15	TBC	Varies
	Absolute Technical Importance		64	64	56	58	31	44	18	18	96	91
	Relative Technical Importance		4	4	6	5	8	7	9	9	1	2

11.2.2 BOM of Design

Part #	Part Name	Qty	Description	Functions	Material	Dimensions	Cost per unit	Link to Cost estimate
1	Pump	1	Powers the hydraulic system that will make radial and thrust loads	Move fluid	Steel	6"x6"x6"	120	https://www.azmut.com/c/motors/hydraulic-gear-pumps?p[illegible]hydraulic-gear-pump-3-4-16-inlet-port-size-0.129-cu-in-rev-displacement-MzQyNDM5&gclid=CjwKCAiAuMTfBRAcEiwAV4SDkahyIR2[illegible]C4_E[illegible]
2	Hose	5	Contains the hydraulic fluid and bring the hydraulic fluid to the approoriate locations	holds fluids	Rubber	6 ft	20	https://www.mcmaster.com/general-purpose-hose
3	Pump Motor	1	Povide power to the pump	Provides power	Steel	3"x4"x2"	586.2	http://www.electricmotors.com/2hp-3600rpm-56c-frame-3phase-motor.html
4	Piston	3	Coverts the hydraulic pressure from the pump to a radial and thrust force on the shaft	Convert hydraulic pressure to a force	Steel	d=1"	30	https://www.mcmaster.com/~1fpexvm
5	Metal per cubic in	200	Will be used to make the shaft and stand	Holds and transfers loads	Stainless steel	200in^3	0.65	https://www.amazon.com/SF-IT3444358-Steel-Bench-Block/dp/B008C08DIM/ref=sr_1_1?ie=UTF8&qid=1543619138&sr=8-3&keywords=block+of+steel
6	Bearing	5	Reduce friction from the pistons	Optimize the system	Steel	Varies	7	https://www.mcmaster.com/r14-ball-bearings
7	Shaft	1	Holds the shaft	holds	Steel		41.73	https://www.mcmaster.com/shaft-couplings
8	Pressure Relief Valve	1	Stops the hydraulics system from being overpressurized	safety	Steel	d=1/2"	52.07	https://www.mcmaster.com/pressure-relief-valves
9	Flow contrl. valves x3	3	Allows the team to vary the pressure in the hydraulic system to different pistons	Control	Steel	d=1/2"	94.91	https://www.mcmaster.com/flow-control-valves
10	Hose fittings x15	15	Stops Hydraulic leakages	Safety	Rubber	1/2 "	7.42	https://www.mcmaster.com/hose-fittings
11	Steel tubes (6ft)	1	Hold hydraulic fluids	moves fluid	Steel	6ft	22.38	https://www.mcmaster.com/steel-tubing
12	Check Valve	1	Check the pressure in the hydraulic system and activate the pressure relief valve	Safety	Steel	d=1/2"	15.24	https://www.mcmaster.com/check-valves
13	1/2"-20 screws (pack of 10)	1	Hold the hydraulic system together	Supports	Steel	1/2"-20	5.67	https://www.amazon.com/Hillman-Group-48107-LumberTite-12-Pack/dp/B000JGT1MW/ref=asc_df_B000JGT1MW/?tag=bingshopping-20&linkCode=df0&hvadid={creative}&hvpos={adposition}&hvnetw=o&hvrand={random}&hvpone=&ll
14	3/8"-16 screws (pack of 50)	1	Hold the hydraulic system together	Supports	Steel	3/8"-16	9.18	https://www.racksolutions.com/rack-hardware.html?utm_source=connexity&utm_medium=cpc&utm_campaign=Connexity