# Honeywell Bearings

# Midpoint Report


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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 and Dr. Trevas for dedicating time to helping us with this project.

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# 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 input for the machine must be adjustable to 4 different bearing sizes: 38, R8, R12, and R14. The thrust and radial load on the bearing depends on the overall size. The R14 bearing, the largest bearing, must be capable of measuring the torque with a maximum of 8000 lbs radial load and 4000 lbs thrust load. As the bearing diameter decreases, the radial and thrust loads will also decrease. The rotation requirement for the inner race of the bearing is between 1 and 10 degrees/sec. One of Honeywell's original requirements for the project was to have an ability to test the bearings within their temperature extremes ranging from -65 degrees Fahrenheit to 1000 degrees Fahrenheit. After speaking with Honeywell and faculty advisors, this requirement was eliminated as it was deemed too difficult to achieve within reasonable costs. This project must output a real time plot that relates torque to the applied loads. The team must also solve for and display the static and kinetic friction experienced by the bearing under the specified loads. Finally, Honeywell stated that the volume of the testing machine must be less than 9 ft3 and had to include a burst shield enclosure for safety.

# 2 REQUIREMENTS

## 2.1 Customer Requirements (CRs)

Table 1: Customer Requirements and Constraints

| Requirements | Constraints |
|---|---|
| Must meet loading requirements | Adjustable radial Load from 0 to 8,000lbf<br>Adjustable thrust load from 0 to 4,000lbf<br>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 $90^o$ |
| Affordable | Cost less than $1,500 |
| Easy Maintenance | No lube, cannot be fragile |

Table 1 outlines the properties the team must consider and their corresponding parameters. 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 the requirement for the product to be mobile. If put on wheels, the team determined that a weight of less than or equal to 300 lbs 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 on 0.315in, 0.5in, 0.75in and 0.875in with a tolerance of +- 0.005in.

## 2.2 Engineering Requirements (ERs)

Table 2: List of engineering requirements

| Engineering Requirements | | |
|---|---|---|
| Property | Unit | Value |
| Radial Load | lbs. | 8,000 |
| Thrust Load | lbs. | 4,000 |
| Weight | lbs. | |
| Shaft Rotation | $^0$ | 90 |
| Cost | USD | 1,500 |
| Voltage | V | 120 |
| Current | Amps | 15 |

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## 2.3 Testing Procedures (TPs)

1. Radial Loads (Up to 8000 lbs)

Radial loads will be applied using hydraulic jacks. The reservoir used will contain enough fluid to reach the 8000 lbf. This will ensure that the 8000 lbf requirement is met. To test this the team will first test the shaft to make sure it can withstand this load. The team will use two chucks to hold the shaft and apply increasing known loads up to 8000 lbf plus a factor of safety of 100 lbf. The team will then place the shaft within the bearing supports and conduct the same test. If both shaft and the supports can handle these loads the team will assemble the hydraulic system and ensure it is applying a maximum load of 8000 lbs utilizing pressure transducers.

2. Thrust Loads (Up to 4000 lbs)

Thrust Loads will be applied using hydraulic jacks. The team must ensure that the shaft can handle the 4000 lbs of compressive force. The team will place the shaft in a vertical clamp and apply increasing known loads up to 4000 lbf load plus a factor of safety of 100 lbf. After the shaft passes the 4000 lbs load test, the team will place the shaft within the bearing supports and conduct the same test. If both shaft and the supports can handle these loads the team will assemble the hydraulic system and ensure it is applying a maximum load of 4000 lbs utilizing force sensors

After procedures 1 and 2 have been completed, the team will apply all loads simultaneously and test for errors caused by this.

3. Weight (less than 300 lbs)

The system weight at or below 300 lbs. To test this the team will weigh the system after it has been built and tested for safety when applying the radial and thrust load. The system will be weighed with no hydraulic fluid to verify it is under 300lbs.

4. Volume (9 $ft^3$)

The system needs to fit within a 9 cubic feet space. For this reason, the shafts will be short to minimize the volume the system needs. Externals such as the supports will also be streamlined to ensure they do not take up extra space. Hydraulic jacks are being used because they can of apply the required loads in a small form factor. To test this requirement the team will measure the length, width, and height occupied by the system. This means the system fits in a rectangular box of 9 $ft^3$.

5. Rotation (Up to 90°)

The shaft will be rotated using hydraulic rotary actuator. To test if the system will be able to rotate the shaft under the radial and thrust loads, a setup will be constructed using a shaft, 3 hydraulic jacks, and ball bearings. The shaft will then be loaded with the radial and thrust loads. The team will run the system to determine if the hydraulic rotary actuator can rotate the shaft the desired 90°.

6. Cost (Under $1500)

The team has made multiple adjustments to the design to ensure the cost does not exceed $1500. The force application and force measurements systems have been redesigned to cheaper alternatives reducing the overall price of the system. The current iteration is $1322 which is safely under budget with allowance for minor changes.

7. Voltage (120V)

The team has been given a standard outlet so any power that is required needs to be able to run off 120V. All electronics for the project will be rated for 120V. This will be proven by using a voltmeter to ensure the voltage across electronic systems fall below or meet the 120V limit.

8. Current

The team has been given a standard outlet so any power that is required needs to be able to run off 15A. Similar to the voltage restraint, this requirement will be proved by using a multimeter to test all electronics to make sure the amperage draw is at or below the 15A constraint.

9. Shaft Size

Honeywell wants the system to be capable of testing 4 different bearing sizes including r8, r12, r14, and 38. To do this, the team will make a steel shaft for each size to ensure each bearing can be tested. To test this requirement, the team will slide the bearing on to each shaft and make sure the bearing and the shaft have a snug fit.

## 2.4 House of Quality (HoQ)

To create the quality functional deployment shown in appendix B, 10.2.1, 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 materials is unknown currently and will be determined later.

# 3 EXISTING DESIGNS

## 3.1 Design Research

There are no existing designs the team can reference which is what Honeywell prefers. They want the team to have a fresh perspective, therefore, they withheld information about their current design. However, Honeywell did provide advice on how the team's design should operate. They want the bearings to experience loads similar what they experience on butterfly valves when in service. Honeywell provided the schematic, shown in figure 1, to help the team visualize how they wanted the loads applied

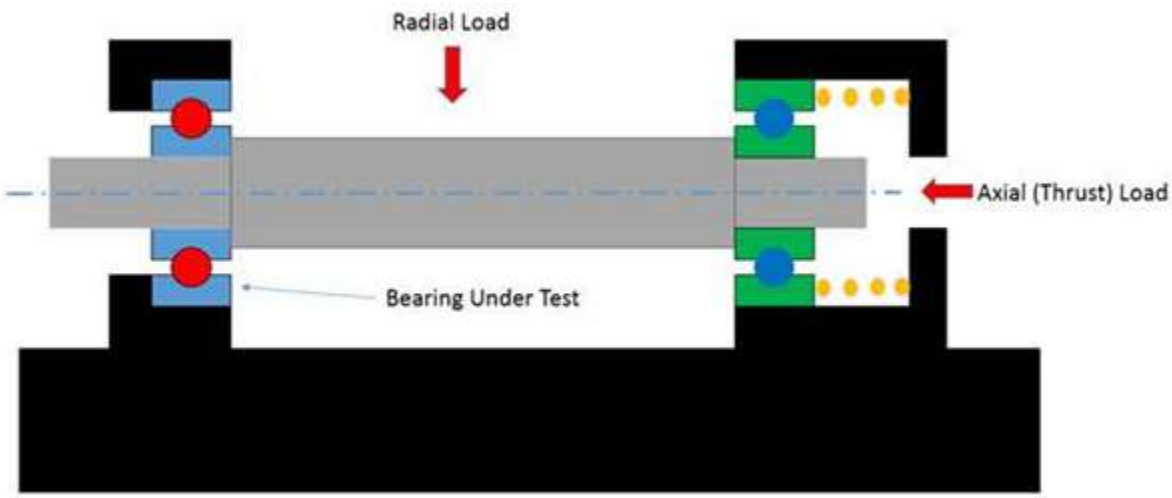


Figure 1: Load application

Figure one reveals that all loads must be applied to the shaft rather than directly to the bearings. For this reason, all concepts in which the loads were imparted directly to the bearing were rejected.

## 3.2 System Level

### 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.

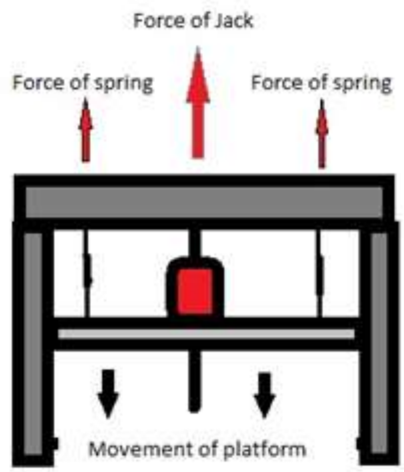


Figure 2: Hydraulic press design

Within figure 2, 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.

### 3.2.2 Shaft Analysis

To ensure the safety of the operator the team performed a load analysis on the shaft. To do this, the team create a MATLAB code that would run various size shafts through their max loading and found the max length and minimum material strength required. The formulas used are seen below.

$$\sigma = F/A$$

Shear= F*L*d/2/(PI/64*(d/2)4)

Using these two formulas in an interactive process enabled the team to determine the maximum allowable distance between supports was in and the minimum material strength required was equivalent to that of stainless steel.

Then the client requests an FEA analysis done in Solidworks to double check the MATLAB code.

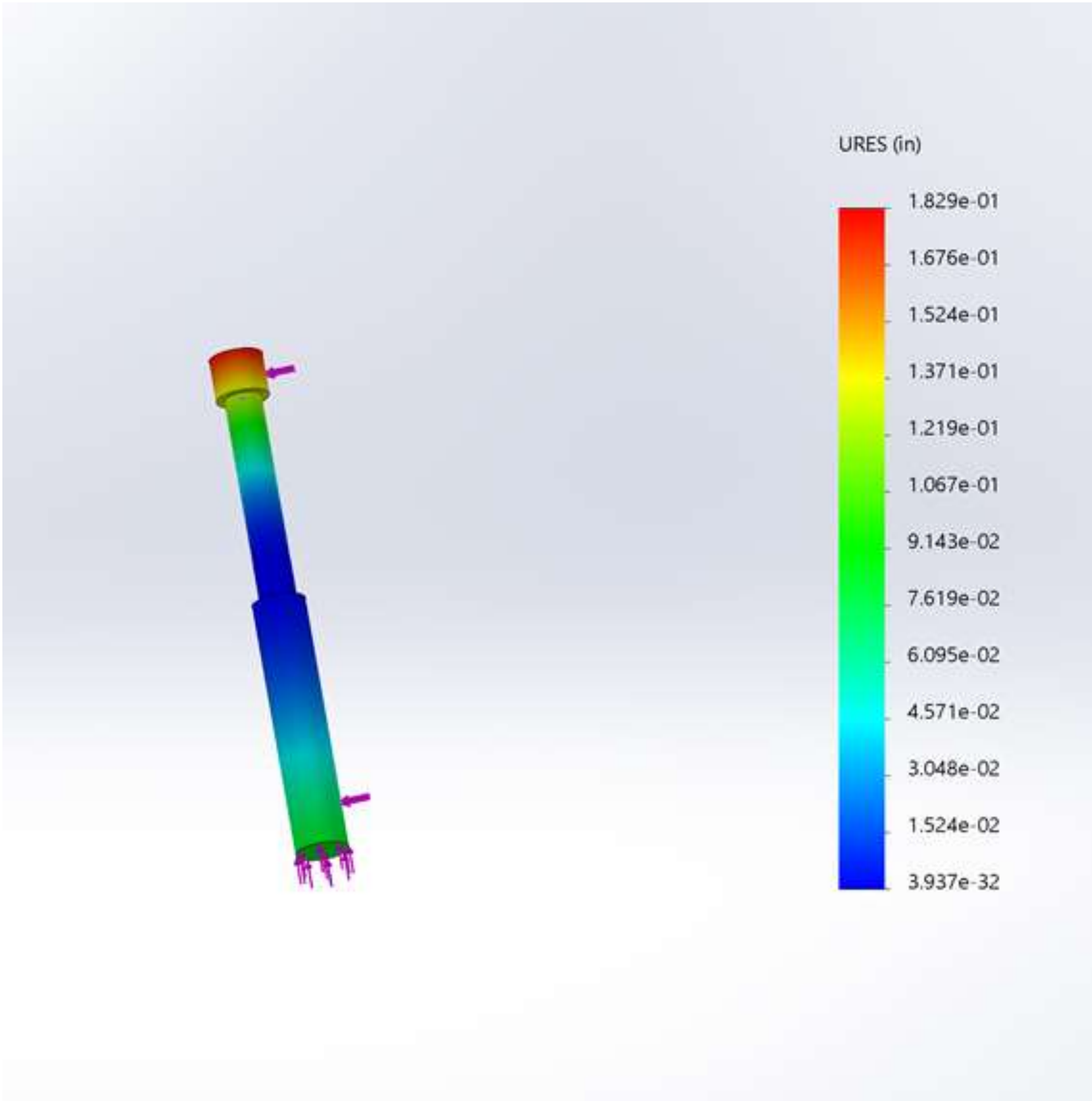


Figure 3: Show the stress analysis.

As seen in figure 3, the shaft will perform in the elastic region for 1018 steel. This proves the MATLAB code is correct and 1018 steel is acceptable metal for this design

### 3.2.3 Friction of Bearings

Bearings are often used in situations where one member needs to rotate while keeping the other stationary. While they often have a low coefficient of friction, there is always a small amount of friction present. If the torque on the shaft is known, the coefficient of friction can be determined using a simple formula.

$$T = F\mu d2$$

In this formula, T is the torque, F is the force on the shaft, μ is the coefficient of friction, and d is the pitch diameter of the bearing [1].

## 3.3 Functional Decomposition

The project began with a problem decomposition in which it was broken down into 6 main components. The first of these are the jacks, which apply the forces to the shaft, another component that 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 and contains provisions for a burst shield. 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

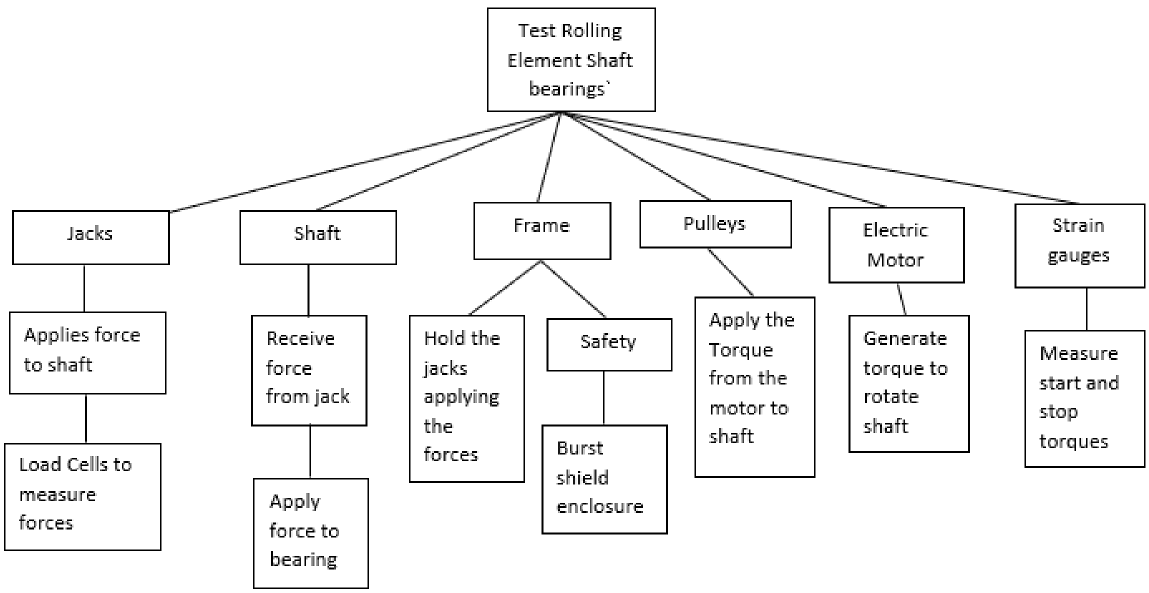


Figure 4. Decomposition of Testing Bearings

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

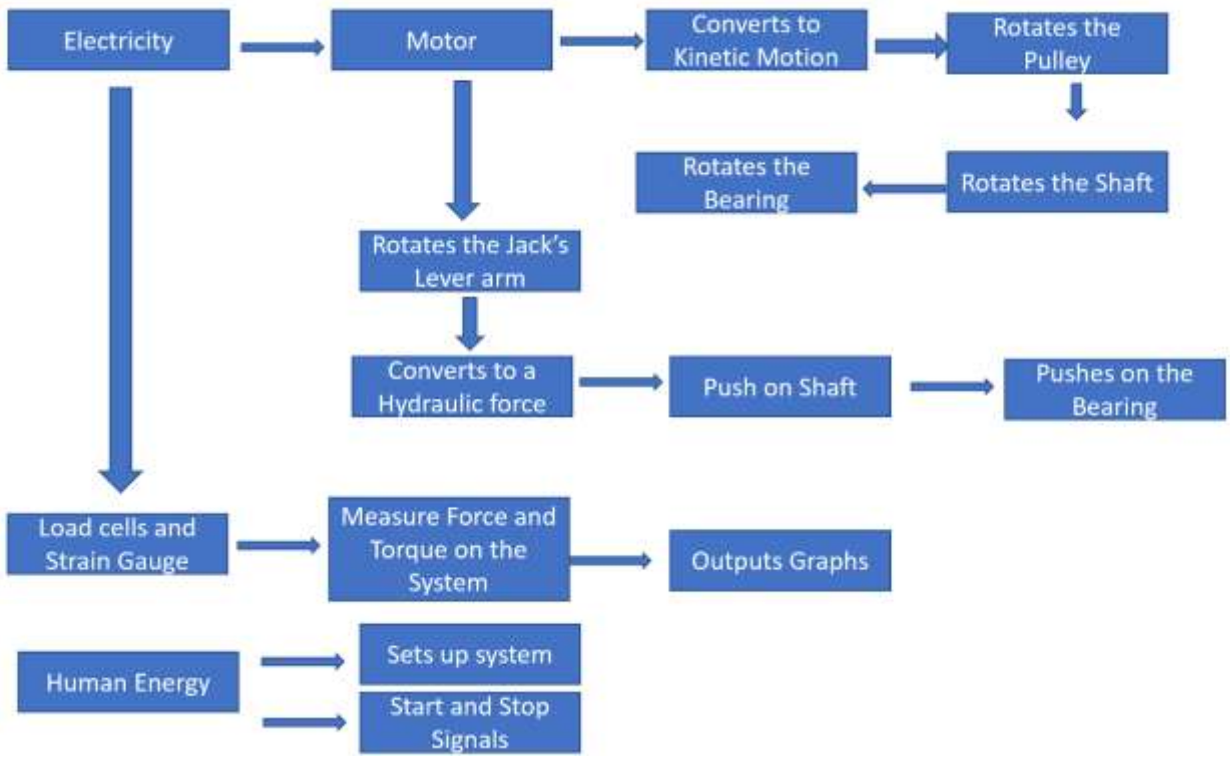


Figure 5: Functional Model

Figure 4 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 will be needed to set up the test fixture, meaning 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.

### 3.3.1 Black Box Model

The team created a black box model to help visualize the material, energy, and signal flows within the system. 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.


Figure 6: Black Box model for Testing Bearing

## 3.4 Subsystem Level

This project deals 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 addresses a specific part of the design requirement

### 3.4.1 Subsystem #1: Shafts

The shaft will provide a place for the bearings to be mounted on and will need 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 thus 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, thus 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 will be used to contain all components; therefore, it must be strong enough to contain all loads. All forces must be absorbed by the frame to keep the system stationary. It is crucial that the frame stays together and does not bend so results will be accurate.

#### 3.4.2.1 Bearing Support

The bearing support system will consist of a bearing support base that will be secured to the frame. There will be 4 interchangeable supports for each bearing size that will slide in and out of the base. The bearing supports must remain stationary and not deform or the bearing load data will be inaccurate. The design has been constructed in SOLIDWORKS and simulated at the max loads of 8000 lbs radially and 4000 lbs axially. The design was able to support the loads and had a max deflection that was small enough to not affect the measurements.

### 3.4.3 Subsystem #3: Torque Application

#### *3.4.3.1 Electric Motor*

The team determined that the shaft will 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 is necessary to spin the shaft; thus, any motor can be used so long as it can be programmed or made to move 90º at 10º/s or less.

#### *3.4.3.2 Hydraulic Rotary Actuator*

The team should be able to utilize an existing hydraulic circuit to implement a hydraulic rotary actuator which will enable the team to spin the shaft 90° without the use of a separate electric motor. There are many different actuators designs the team can consider including rack and pinion systems, scotch and yoke systems and enclosed piston crank actuators, all of which should be capable of meeting the team's goal of turning the shaft. These systems should be able to accommodate a shaft, pulley, or thrust bearing via a coupling or keyway.

### 3.4.4 Subsystem #4: Power Transmission

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

#### *3.4.4.1 Belt and Pulley*

A flat belt and pulley combination exhibit a high efficiency of about 98% [1], which is desirable as transmission losses will have a minimal effect on test bearing friction measurements. Due to elastic creep, however, the belt moves at a speed slower than the pulley [1], therefore adjustments will have to be made to ensure the shaft turns the full 90º. Another advantage to consider is 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 be had for around $22.00 [2], and an accompanying drive pulley will cost around $13.00-$22.00 [3].

#### *3.4.4.2 Chain and Sprocket*

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

### 3.4.5 Subsystem #5: Force Measuring System

It is necessary to collect data on how much force is being applied to the system to ensure correct parameters are attained, and to utilize those measurements to calculate friction.

#### *3.4.5.1 Strain Gauge*

A strain gauge 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 can tell how much a material has deformed, and back out a force measurement based on that. The team can take advantage of this phenomenon to obtain the torque required to spin the shaft and determine how much thrust and radial force has been imparted upon the system. The strain gauge kit will cost about $332 [7].

### 3.4.5.2 Load Cell

The team also investigated compression load cells, which include four strain gauges packed into a small steel casing [8]. This device therefore works on the same principle as strain gauges do. When the steel casing compresses, the strain gauges inside it either stretch or compress, changing resistance, and thereby giving a strain value that can be used to back out force. These devices are only 1in in diameter and 1in tall and can withstand forces up to 10,000 lbs [9], therefore the team can directly mount them to the force application system. This would allow the team to get a direct measurement of the force applied in real time. However, these devices are expensive as they cost around $350 each [9], and the team will likely need 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 are applying the forces can be tapped with a pressure gauge. As the hydraulic applied a force, this pressure will change. 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 must impart a maximum of 8000 lbs of force radially, and 4000 lbs of force axially onto the bearings. Honeywell is providing 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. There are bottle jacks rated at 4-tons available for about $20.00 [10]. Additionally, these jacks have a minimum height of 7.625in and a maximum height of 15.125in [10] which means these fit well within the team's 9ft3 constraint, thus allowing for design flexibility. A potential drawback with these systems is that these jacks must be hand cranked potentially reducing the precision of the measurements, and they do not have a pressure readout therefore a sensor will need to be used to determine how much force they are imparting. Tapping these could give more precise readouts, however, this also has drawbacks. In the worst case, tapping the jack could break it rendering it unable to impart its maximum force. Assuming tapping is done successfully, delivering a pressure near four tons could require a manometer that is too tall and unwieldy to be practical.

### 3.4.6.2 Electric Jacks

Electric jacks utilize an electric motor to twist a power screw which in turn provides a lifting force. These jacks are considerably smaller than the hydraulic 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 are also subject to issues with directly measuring force, just as bottle jacks are.

## 3.4.7 Subsystem #7: Shaft Retainers

This is necessary for mounting the shaft to the system and allowing for a way for uniform shafts to be interchanged.

### 3.4.7.1 Lathe Chuck

A lathe chuck could be used to very quickly change out shafts. It is made to be strong and will have no problem withstanding the applied loads. Furthermore, a belt can be wrapped around it creating an easy way for the torque to be transferred from the motor to the shaft. However, when heavy loads are applied the could dig into the shaft, compromising the integrity of the shaft. Additionally, some chucks may not be 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 is ideal as it will likely hold up to the axial forces subjected to it through the shaft. It is also a cheap option costing around $51.00 [13]. However, given the inner diameter of this bearing cannot be changed, an adaptor would have to be made to accommodate different shafts.

# 4 DESIGNS CONSIDERED

## 4.1 Design #1: Funnel Design

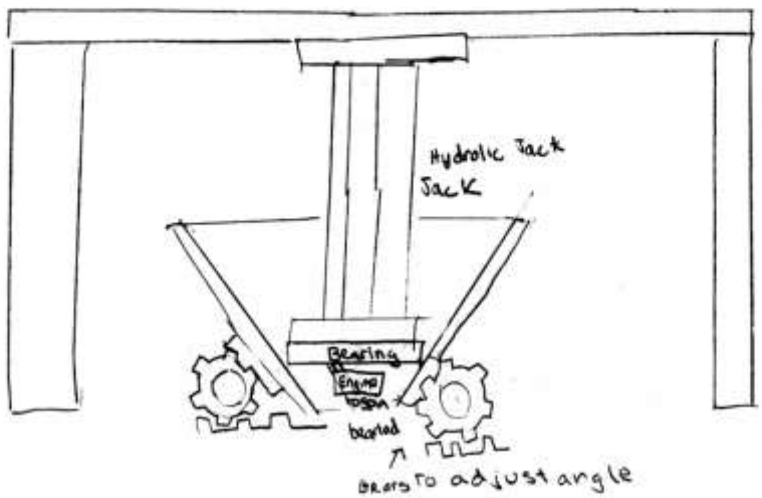


Figure 7: Funnel deign

This design would place the bearing in the funnel that is adjusted with a gear system. The angle is changed to adjust how the load it applied. Once the bearing is in place the jack applies a load to the face of the bearing. Based on the angle the funnel a resultant force will exist whose components will make up the thrust and radial loads. This was the cheapest of all the designs, as it does not require the operator to change shafts and would not risk breaking shafts due to deflection. The biggest problem with this design it that the test fixture applied both loads to the outer race bearing and the client wants the load directly on the inner race of the bearing.

## 4.2 Design #2: Shaft Step

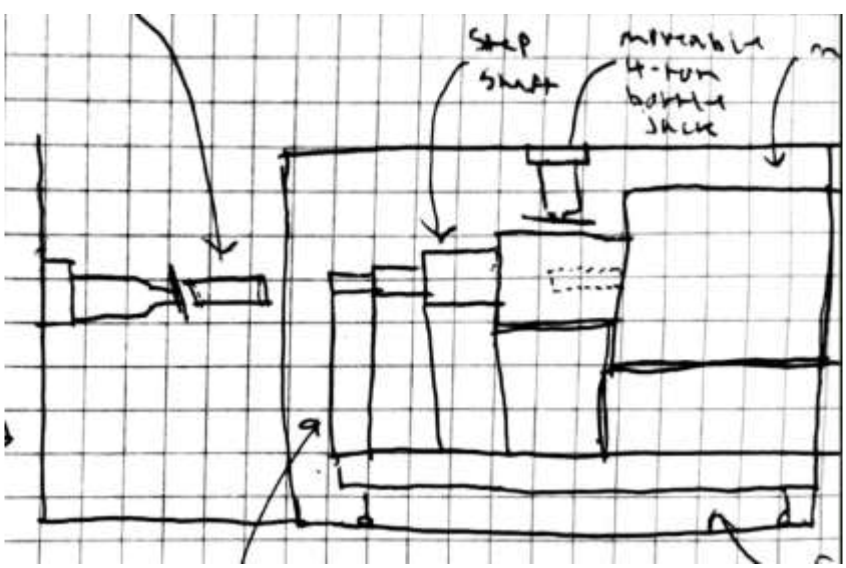


Figure 8: Stepped Bearings

This design utilizes a step shaft directly mounted to an electric motor to mount the bearings without the need to switch out shafts. A bracket will support the bearings so when a force is imparted upon them a moment will not be experienced by the shaft. The forces will be imparted by two bottle jacks. The first jack will impart a force directly to the outer race of a bearing, as it is expected that the inner race will experience a reaction force of the same magnitude. This jack will be moveable so one can select which bearing to test. The second jack will be mounted horizontally and will push 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 needs to impart. Therefore, load cells would be mounted to each jack instead. This design allows for each bearing to be quickly tested as there is no need to switch out shaft assemblies. Additionally, calculations are made easier as the forces are applied directly to the bearings, and no transmission losses would have to be considered given the shaft is directly attached to the motor. Because of this, however, the motor will experience the full thrust force which could break it. Honeywell ultimately determined that this method of testing the bearings is 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

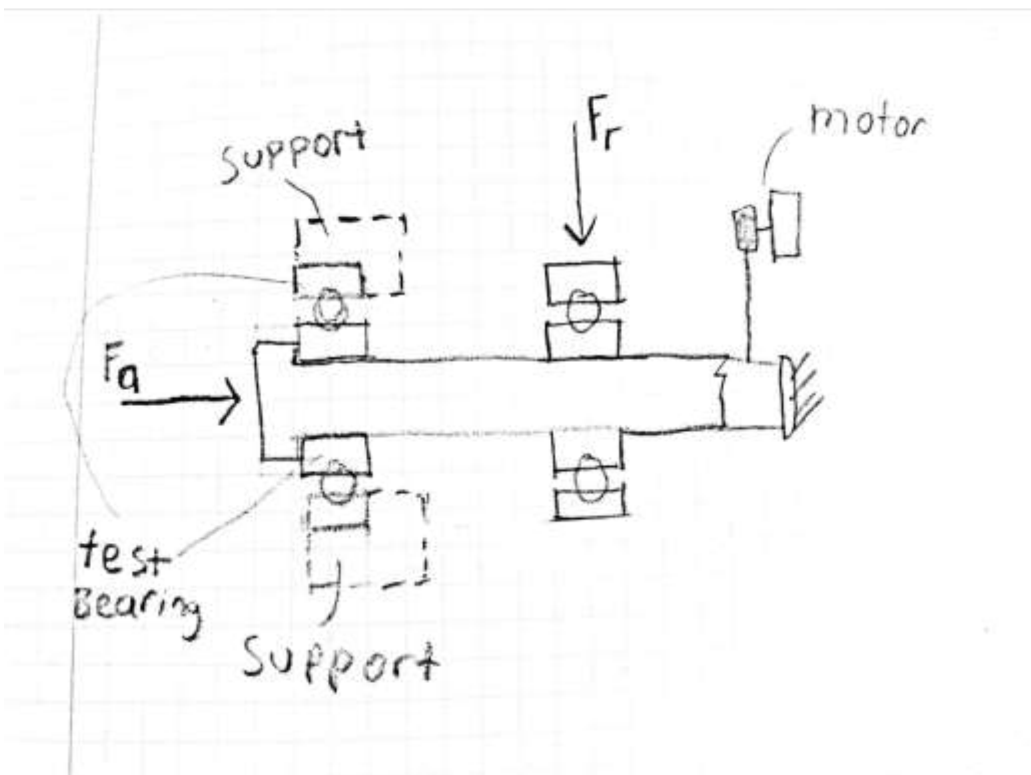


Figure 9: Inner Race Design

The rationale behind this design is that loads are applied directly to the shaft to ensure that the forces act on the inner race of the test bearing. The thrust load is applied to the shaft and the radial load is applied to another bearing which deflects the shaft thus ensuring the radial load is applied to the inner race of the test bearing. The design is supported by the frame which connects the bearings. An oversight on this iteration of the design is that the system is supported where the radial load is applied. To ensure that the radial load is applied to the shaft, the shaft must be supported elsewhere. However, since this design does 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

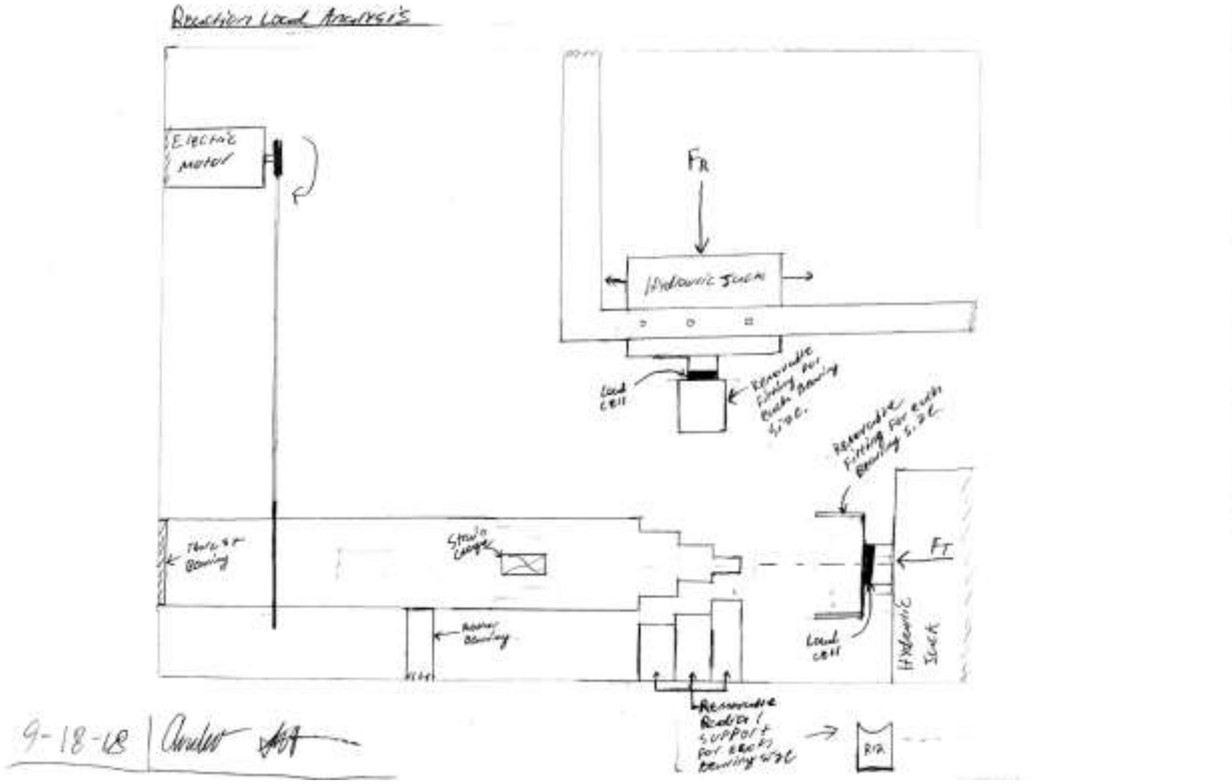


Figure 10: Reaction Load Analysis

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

## 4.5 Design #5: Lever Arm

This design, shown in Appendix A, 10.1.1, utilizes a test fixture shaft that would be forced apart three ways to apply the radial load. The thrust load would be 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 was cost of this design. 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 is meant to directly connect several uniform shafts to the bearing. These shafts, with the bearings attached, will be mounted to a carousel that will allow the user to quickly select a bearing to test, and connect it to the motor via a spline, or keyway. Everything will be connected to a slide, so the user can center each bearing and prepare it for a test. A bottle jack mounted to a slide on top of the frame will impart the radial load. The bearing will be placed on a bracket to avoid imparting a moment on the shaft. Another jack mounted horizontally will push a hollow pipe into the inner race imparting the thrust load. The problems with this design are 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 once has been centered, must be devised. Another problem is the input shaft on the electric motor may be too large to fit some of the smaller shafts. Finally, the electric motor will have to absorb the entire thrust force, which could lead to failure. The only advantage this design exhibits is that fact that uniform shafts are 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, applies the thrust to the test bearing using a prong system attached to a jack. It also applies the radial load directly to a test bearing. However, Honeywell wants both loads applied to the inner race of the bearings so both methods of application do not satisfy the requirements. The design would be able to apply loads to the bearing, but since it does not meet Honeywell's requirements, the idea was not considered.

## 4.8 Design #8: Gear Turner

This design shown in Appendix A, 10.1.3, is very similar to other designs in which two jacks are used to apply the forces to the shaft which are then reacted onto the inner race of the bearing. The gears are then attached to the motor input shaft and the testing shaft to apply the correct amount of torque needed. The main problem with this design is that the torque for the shaft may be large and could potentially damage the gears. It also isn't simple enough for the team to feel comfortable designing because determining the gear ratios may be too cumbersome and extra gears may need to be used.

## 4.9 Design #9: Air Compress

This design, shown in Appendix A, 10.1.4, focuses 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 is that the forces needed cannot be imparted by air compressors. The remaining portions of the design are 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, utilizes a bolt and washer that would pull 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. Attached to washer and bolt would be weights. This design was one of the cheaper designs and used a unique thrust load delivery system. However, due to volume constraints and the safety the design made this design impractical.

# 5 Design Selection

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 | + | | + | | | | - | | - |
| Need Safety Equipment | | | + | - | - | | - | | + | + |
| Needs to be portable | A | | - | | | + | | + | - | |
| Needs to be compatable with the 4 Bearong sizes | | - | + | + | + | | + | | | |
| Needs to output torque vs applied load graphs | T | + | | | | | | | | |
| Output friction vs applied load graph | | + | | | - | | | | | |
| Needs to fit in an office Space | | | | | | | - | - | - | - |
| Electric power | U | | | - | | | | | | |
| Need to rotate bearing | | | | - | + | | | | | |
| Easy to maintain | | + | | - | - | | - | - | - | - |
| Affordable | M | | + | | | | - | | | |
| sum + | | 4 | 3 | 2 | 2 | 1 | 1 | 1 | 1 | 1 |
| sum − | | 1 | 1 | 3 | 3 | 0 | 4 | 3 | 3 | 3 |
| sum | | 3 | 2 | -1 | -1 | 1 | -3 | -2 | -2 | -2 |

To start the selection process the team put all ten designs into a Pugh chart using its favorite design, the reaction load analysis design, as the datum. The team then compared how well this design meets the customer requirements as compared to all other designs. Then the team then eliminated any design that received 3 or more negative signs in the Pugh chart. The team set this as the cut off for consideration for the final design because it did not want show Honeywell anything that was worse than our Datum. The Datum, and the three design that passed this criterion were passed on to Honeywell and were considered within the team's decision matrix.

Table 4: Decision Matrix

| | Ease of Assembly | Cost | Size | Ease of Maintaince | Ease of Use | Precision | |
|---|---|---|---|---|---|---|---|
| | number of moving parts | number of parts that are expensive | how big | number of wear out parts | number of steps to operate | source of error | |
| Weight | 4 | 6 | 6 | 5 | 4 | 7 | Total(Lowest Score wins) |
| Reaction Load Anaylis | 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 depicted by table 4 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 out parts, number of steps to operate, and sources of error. Do note, a higher score meant a design had more negative attributes, therefore, design with the lowest score wins. 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 winning design is shown in figure 10. It is a modified version of design #3. Since Honeywell liked that design the team decided to modify it based on the suggestions they gave. The axial load is applied to the shaft imparts a force upon the inner race of the test bearing. The bearing is connected to a support on its outer race to ensure that the bearing is stationary. The radial load is applied to another bearing so that the shaft can continue to rotate as the shaft is deflected imparting a radial load to the inner race of the test bearing. Torque will be measured with strain gauges attached to the shaft. The shaft is rotated by way of a using a hydraulic rotary actuator. The shaft is supported at the far end by either a chuck or a thrust bearing. Since each shaft will be tailored to each bearing, they need to be easy to remove. If the thrust bearing is chosen, the shaft would have to be threaded so it can screw into an adapter connected to the thrust bearing supported on its other end. This design covers all requirements set by Honeywell and is relatively simple. Complexity could arise when trying to quantify the friction within the test bearing since there is another bearing that also succumbs to friction. The team will need to determine how much friction comes from the extra bearing and take that into account when doing calculations.

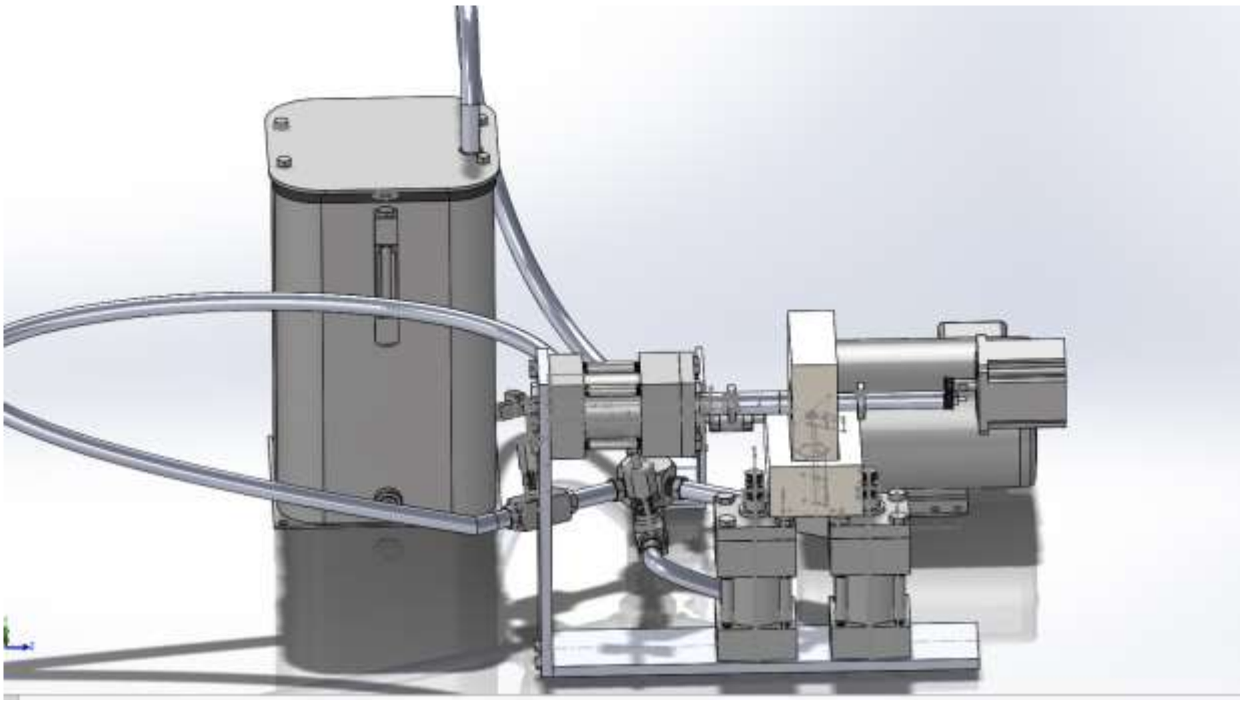

Figure 11: Solidworks Model of Design

This 3D model shows how the major components of the team's design will come together. Those components include the shaft, supports, hydraulic system, and power source. The loads will be applied to the bearings through forces imparted upon the shaft via the hydraulic rams. The rams will act on both sides of the shaft to keep bending moments from being imparted upon the bearing. The horizontally oriented ram will apply the thrust load directly to the shaft. Do note, this is not the final iteration of the team design. In the final iteration, the shaft will be free to move along the x-axis but will be stopped by the bearing which will be stopped by the support. Therefore, the inner race of the bearing will experience the full thrust load. Additionally, a hydraulic servo will be used to impart the rotary motion to the shaft instead of the stepper motor and the reservoir will be replaced by one Honeywell stated they will supply. Finally, a return will be added to the hydraulics system. For the prototype, the team plans to machine the shafts and supports. The team will outsource the hydraulic components as the team lacks the expertise to engineer these items on its own. The software will be coded via Arduino and coded by the team. An exploded view of the current iteration is depicted within figure 10.

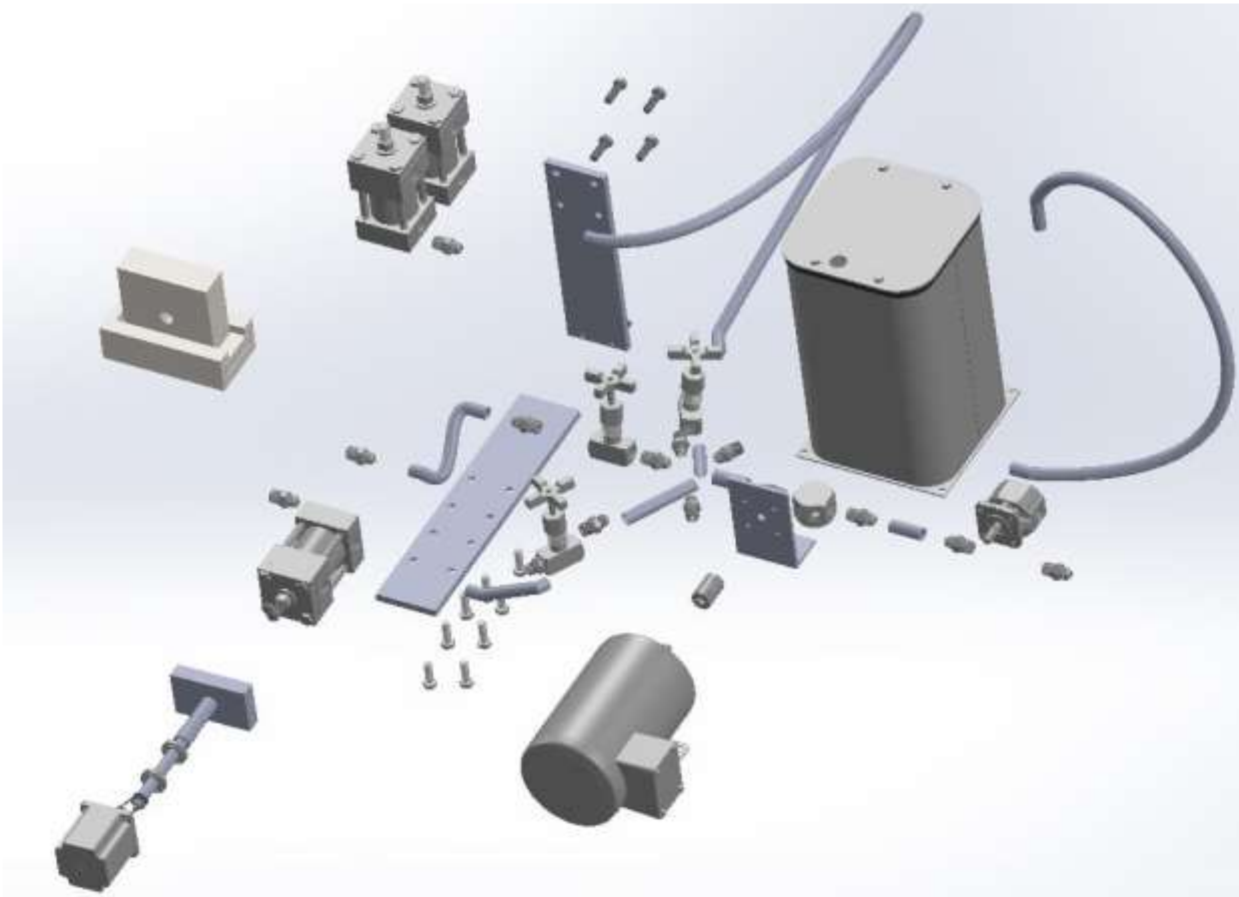

Figure. 12 Exploded view of CAD model

To build a prototype, the team plans to machine the shaft, supports, and lever arm. The team decided to outsource the hydraulic system since building them is complicated and dangerous when not done properly. The software will be done by Arduino and coded by the team.

The budget was then based off the team building part of the project and outsourcing the hydraulic system. The team's current design budget is over the team's max budget by about 1200 dollars. The team will need to look to cut money in the hydraulic system.

Table 5: Preliminary 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 |

The schedule is a moving schedule base off the client needs but the current schedule can be seen below.

Table 6: 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 |

Both the team and Honeywell thought this design would best meet the design requirements. Cost was not considered. Cheaper rams, motors, pumps, and actuators were considered to cut costs. All remaining parts would have been machined or sourced from Hydraulic Controls Incorporated.

# 6 CONCLUSIONS

The design selected for this project was determined by the group and Honeywell to be the most practical to meet the design requirements. At this moment the team's purpose was to propose a quality device and not be focused on the cost. The team is aware that the current total is more than expected, and a further analysis will be needed to determine areas of which costs will be decreased. The largest costs of the projects are in hydraulics systems. The cost of the system can be significantly reduced by using a different method of force application. Another approach to decreasing the total budget will be for the team to machine the parts. By machining the parts, the team will only need to buy the base metal which is much cheaper than each part. Some of the hydraulics will need to be outsourced to other companies to assist with the pumps, gauges, and other more complex portions.

# 7 Implementation

## 7.1 Manufacturing

All necessary design work for machinable parts has been completed except for the truss system required for the axial load bearing frame. This has yet to be analyzed. In the meantime, detailed drawings of each part have been rendered along with g-code if needed. Additionally, work orders for the bearing housing and its supporting structures have been turned into the NAU machine shop. While designing these parts the team considered how they may be machined. It is expected the shafts will be turned on the lathe while everything else is milled. Holes larger than 1.25 inches will be machined on Tormach CNC mills. All jacks, the motor, and parts of the bearing housing structure will be assembled via fasteners, whereas large plates, trusses and the remaining parts of the bearing housing will be welded. Table 7 outlines which components are needed to build the test fixture, how many of each are needed, and how much they cost.

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 |

Missing is a shaft coupling which costs an additional $36.82 from Mcmaster Carr, as well as the necessary fasteners which the remaining $141.18 can be used for. Table 8 outlines the team's current schedule.

Table 8: Current 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 | 3/29/19 |

As the schedule suggest, everything will hopefully be ready for assembly within the next three weeks.

## *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.

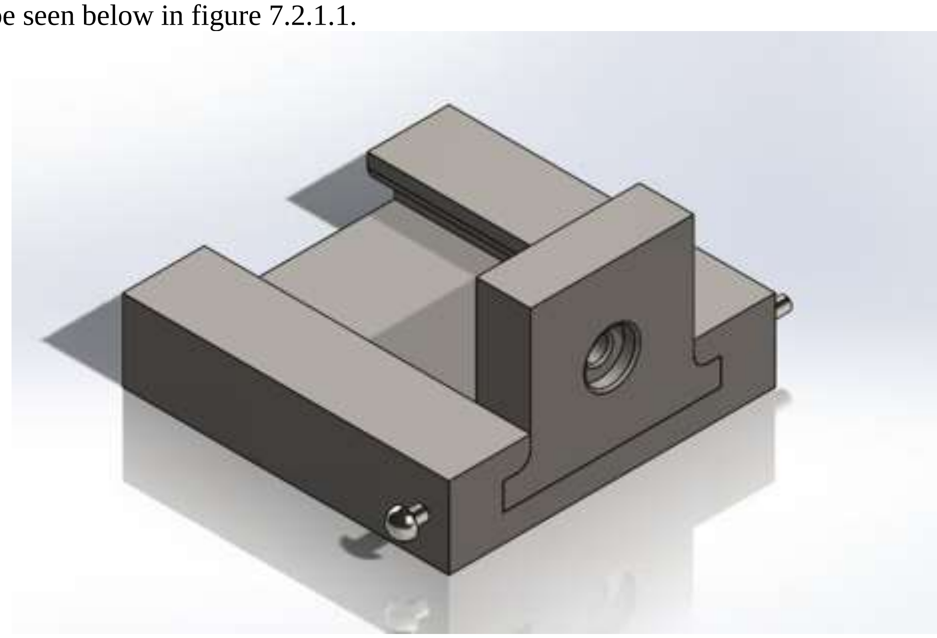

Figure 13: Sliding Frame and Bearing Housing

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 14 below shows the current design.

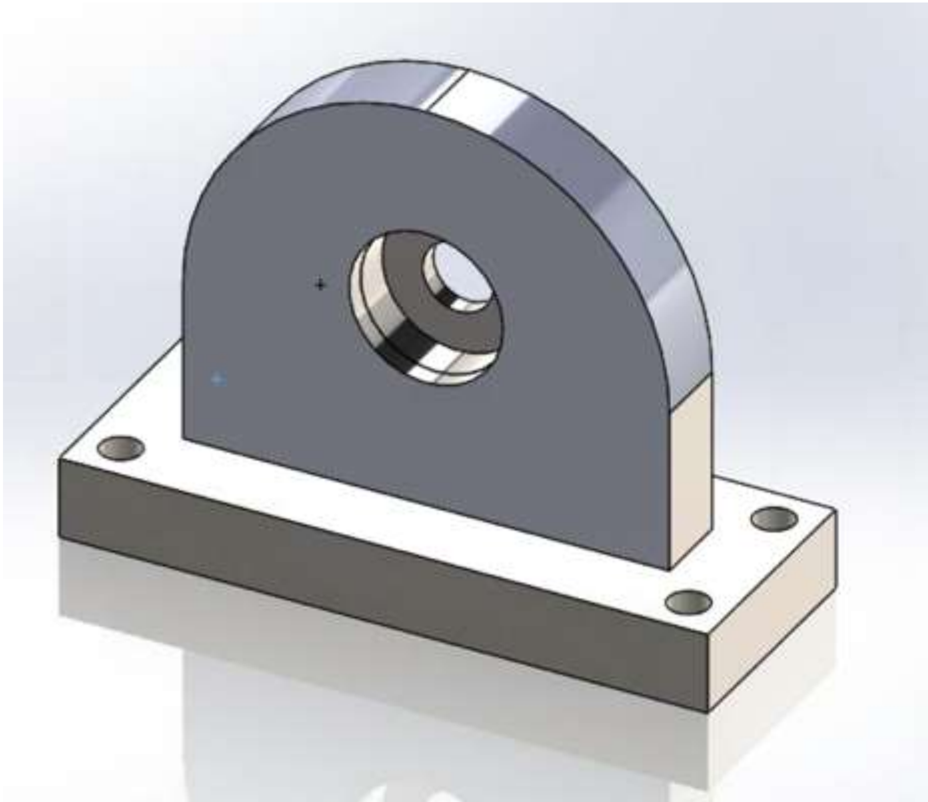
Figure 14: Bolt-on Bearing Housing

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.

### 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, thus this was deemed 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 Electro-Hydraulic Jack Concept*

In an attempt to cut costs an attempt was made to utilize electro-hydraulic car jacks for force application.

These are jacks that utilize a pump and motor, or motor and lever to exert hydraulic pressure. The first iteration of this design, shown in figure 14, utilizes the later system.

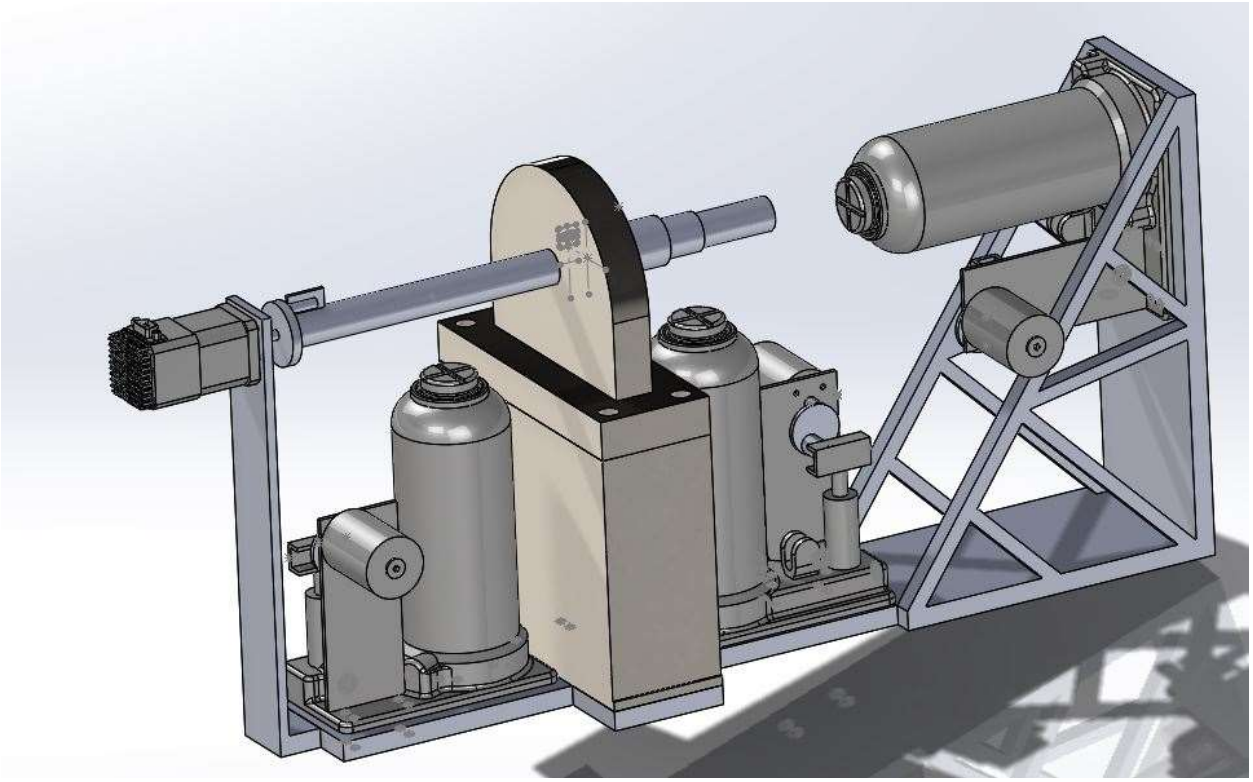

Figure 15: First Electro-Hydraulic Iteration

The advantages of this design revolve around simplicity and size. These jacks are thin and inexpensive thus budget issues could 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. The ones shown in figure 15 are modified hand operated jacks. Thus, 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, thus an off-the-shelf jack was found, and a new design was based around it.

#### 7.2.2.2 Final Electro-Hydraulic 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 15 below.

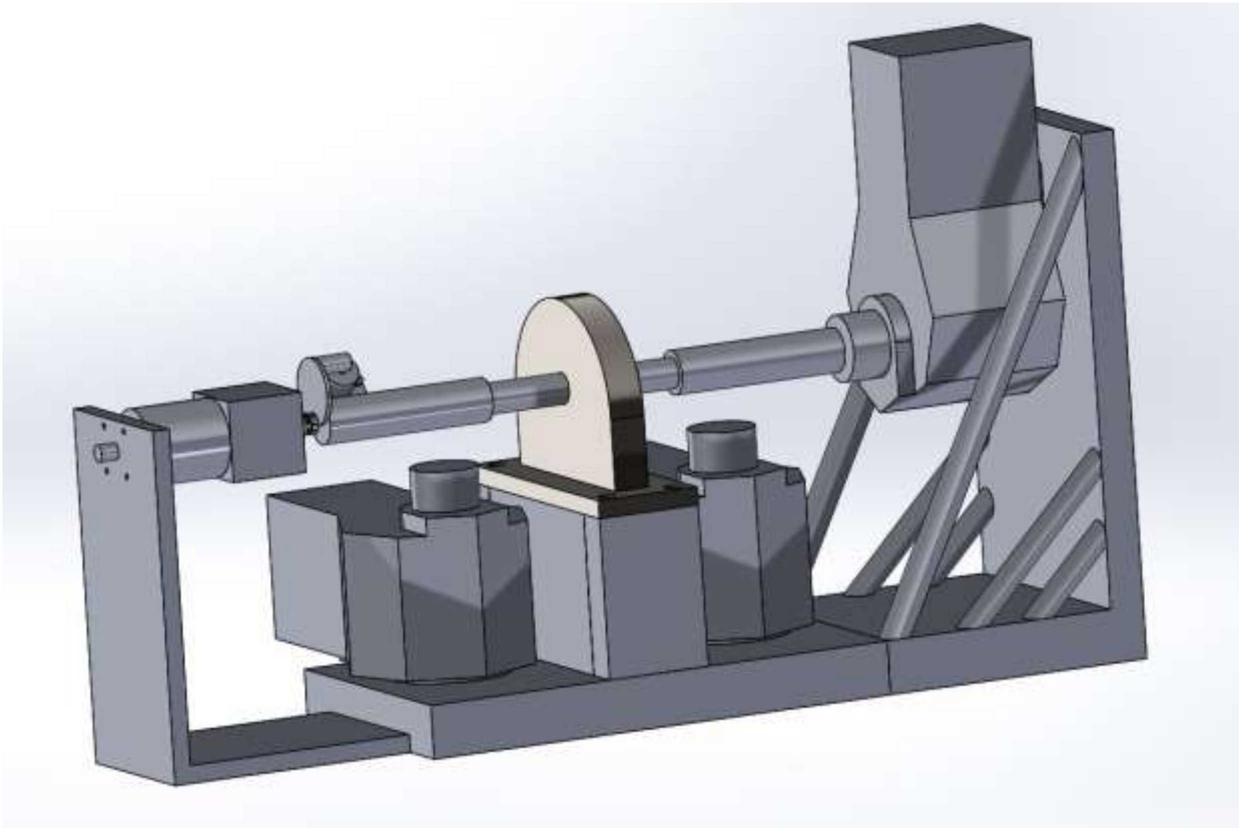

Figure 15: Final Electro-Hydraulic Iteration

Along with being more readily available than motor-and-crank operated jacks, these jacks are shorter. Thus, 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 16 offer access to the hydraulic fluid.

Figure 16: Jack Plugs

The largest plug likely leads to the reservoir, but the small plug or screw likely lead to the pump where the fluid is pressurized. Therefore, a pressure transducer can be easily installed by threading it into one of these ports either directly or via an adapter. 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 thus 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 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. The original design used a two-part shaft design for each design where one end of the shaft would be the diameter of the inner diameter of the test bearing and the other inside diameter. The screw in this design was for a universal holder that would stay the same for each design. The issue with this design was that the bearing at the end of the shaft that takes the radial load from the jack would have to have the same diameter of the test bearing. This makes the design more expensive having to purchase four different sized bearings for each shaft. The new design is a two-piece shaft that has uniform diameters at each end. This makes the design uniform and purchasing the bearings less expensive. We got rid of the universal holder idea since it was not necessary.

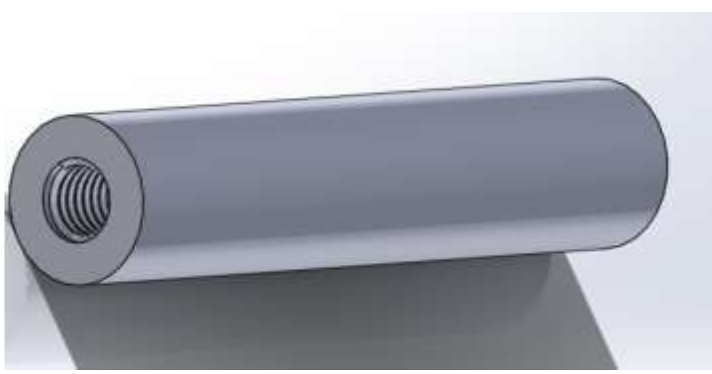

Figure 17: Shaft Case

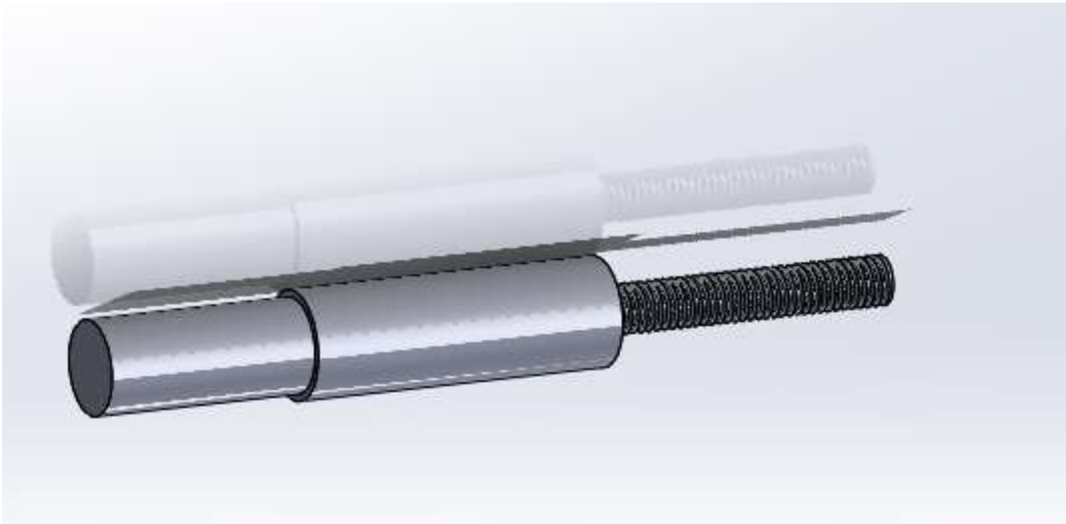

Figure 18: Old Design for Comparison

Figure 19: The new design

In figure 19 the left end will unscrew to allow the shaft to slide into the bearing support. Each end will use the same bearing that has an inner diameter of 1.24 inches.

### 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 17 was initially envisioned to complete this task.

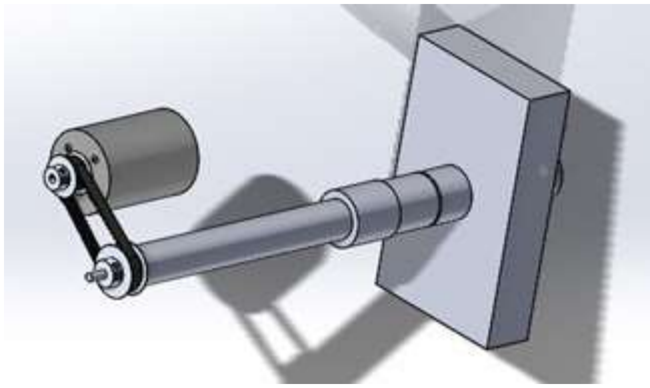

Figure 20: Belt and Pulley System

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 thus 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 thus a cheaper solution was pursued. A hydraulic linear actuator utilizing a $136 ram a clevis rod [15] was thus envisioned. 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

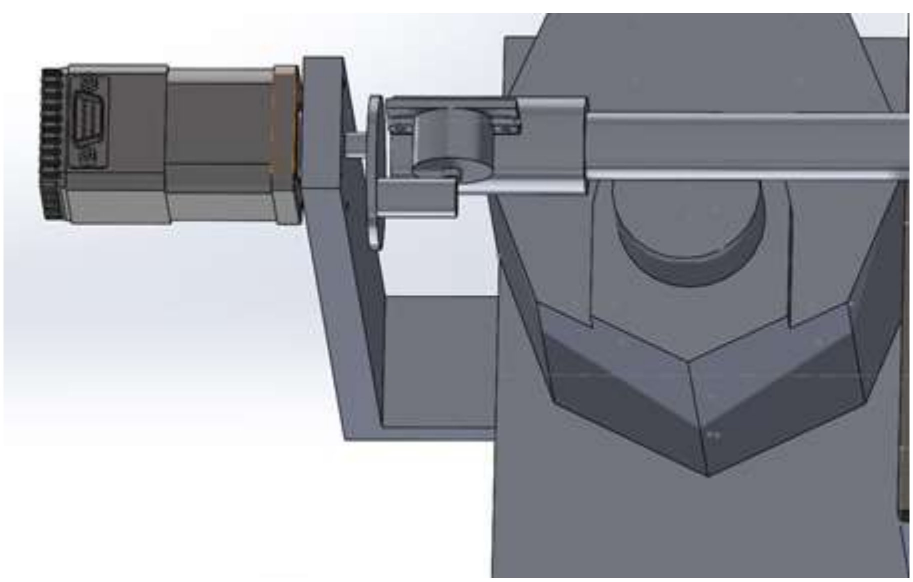

Figure 21: First Crank and Motor Iteration

This 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 crank is shown in figure 22.

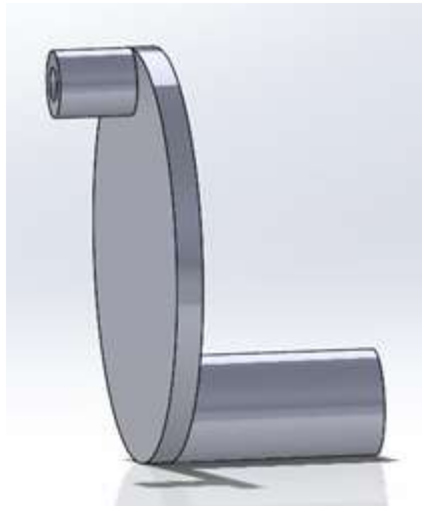

Figure 22: First Crank Iteration

The crank is a one-piece system than consists of a shaft coupling, base and arm that contacts 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 the final iteration of this design shown in figure 23.

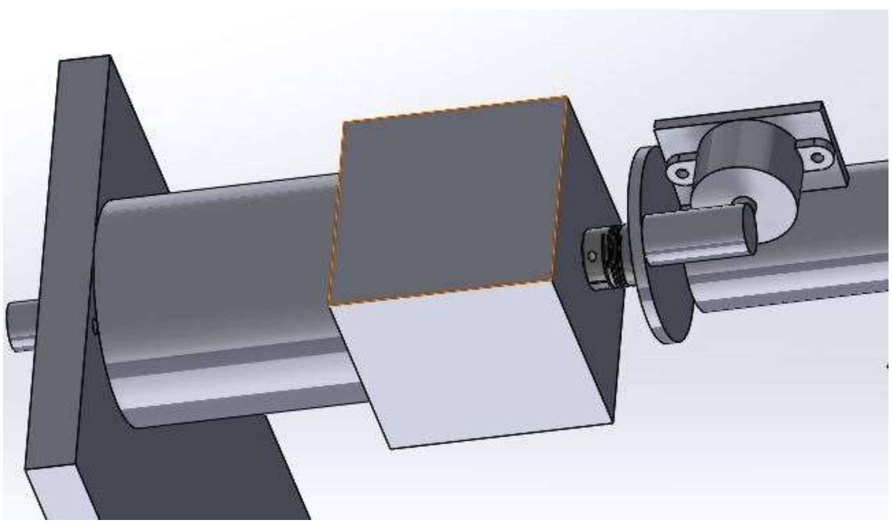

Figure 23: Final Crank and Motor Iteration

This stepper motor is larger but has an output shaft too small to be placed on the back of the 1/2in motor mount, therefore it was placed on the front of it. The back of this motor is exposed to allow a rotary optical encoder to be mounted. This will enable the system to record the speed and position of the shaft. The crank was slightly redesigned as shown in figure 24.

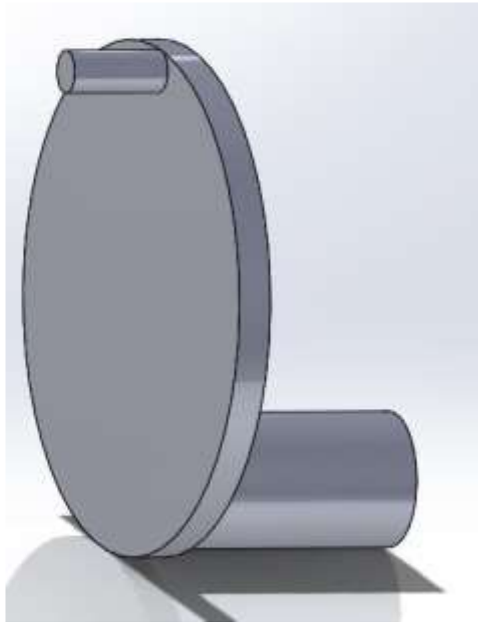

Figure 24: Final Crank Iteration

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 is within budget and can satisfy all requirements. It is also possible the crank could be 3D printed eliminating machining time and additional costs for steel.

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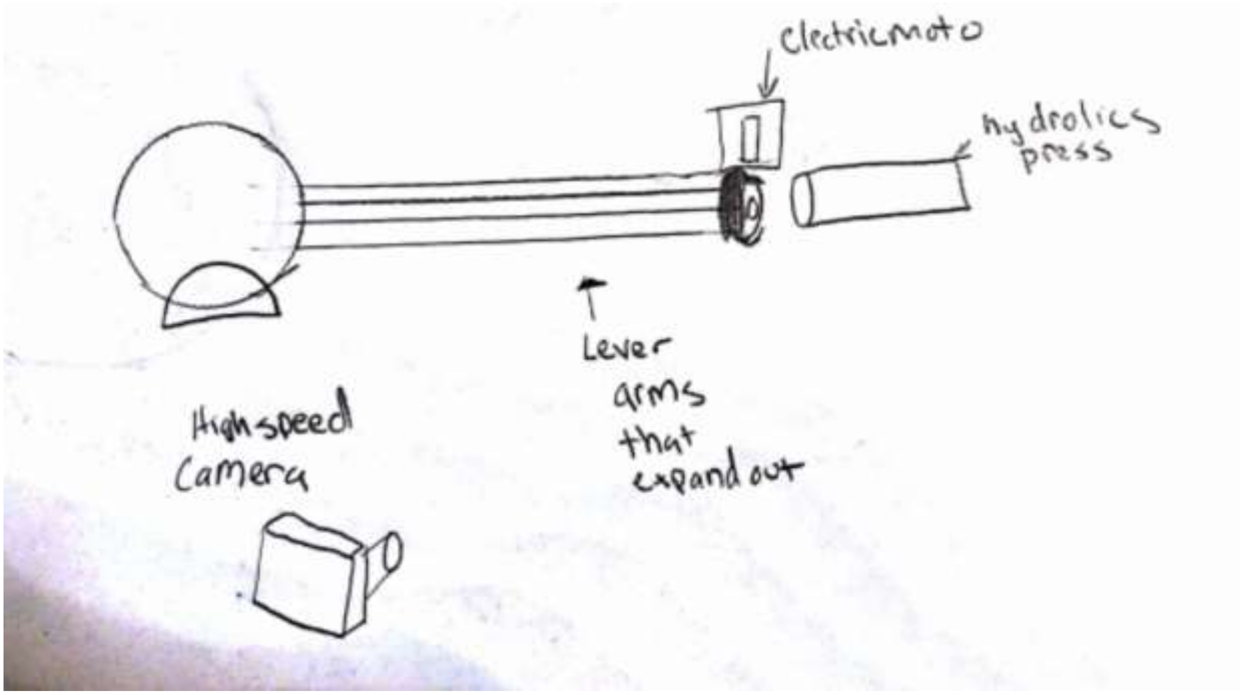
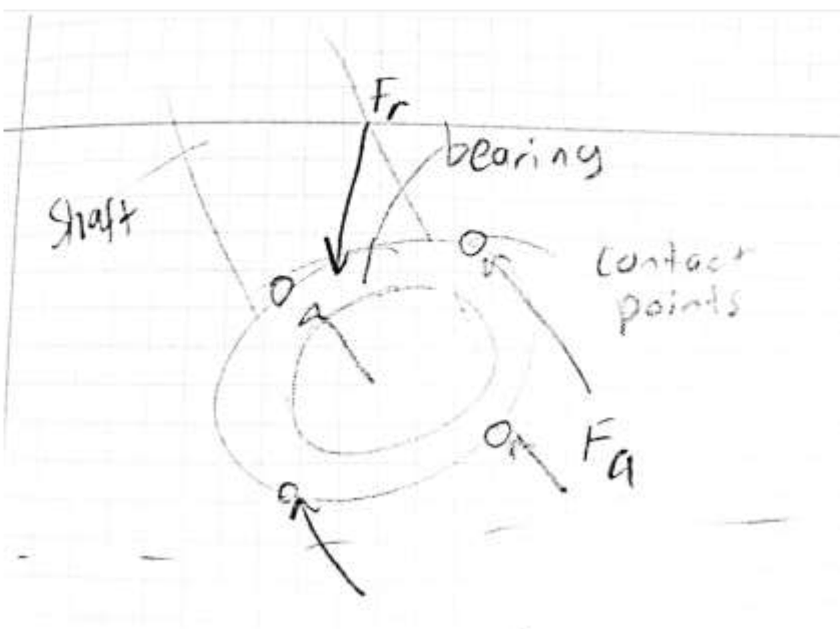
# 9 APPENDICES

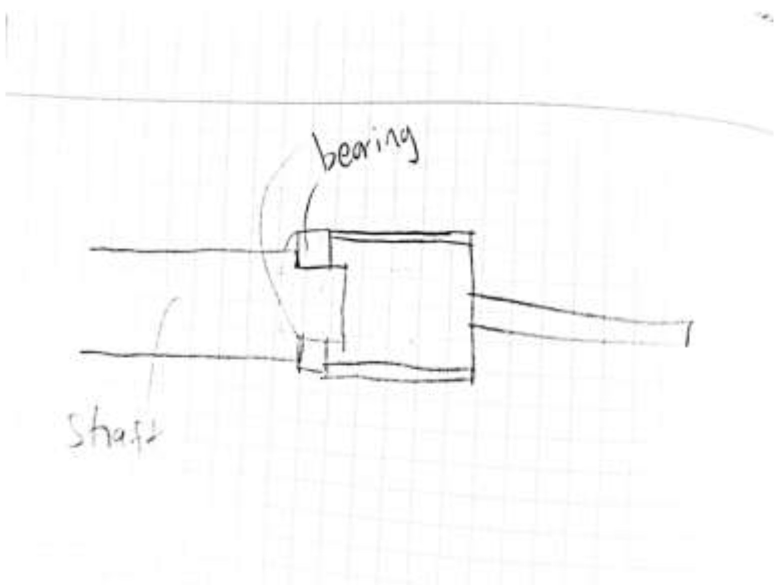
## 9.1 Appendix A: Rejected Designs

### 9.1.1 Lever Arms

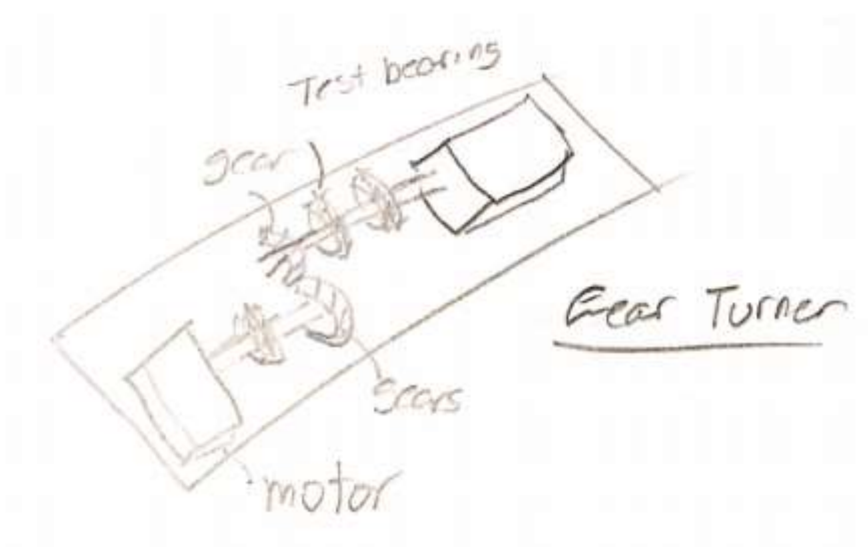



### 9.1.2 Four Points

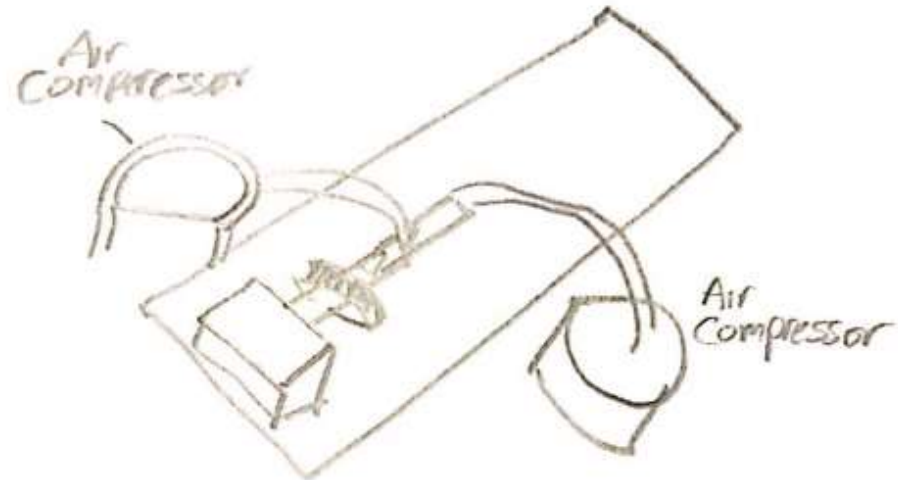




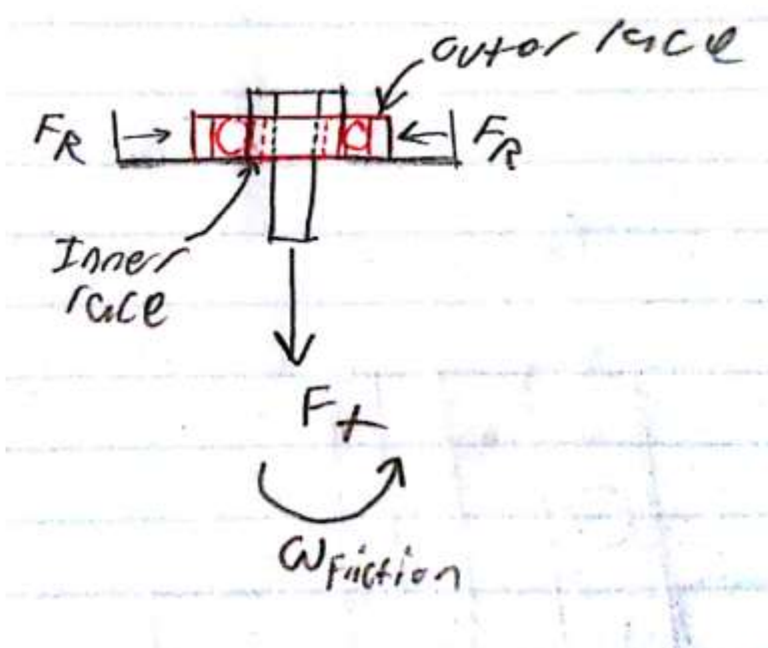

### 9.1.3 Gear Turner



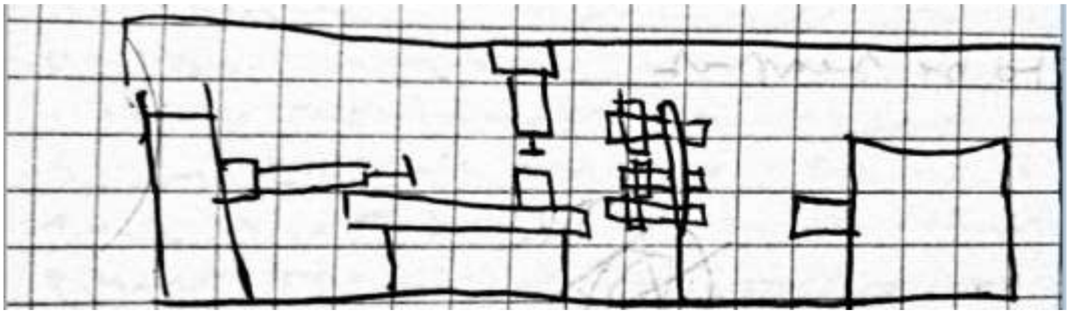
### 9.1.4 Air Compress




### 9.1.5 Gravity Assist



### 9.1.6 Merry-Go-Round



## 9.2 *Appendix B: Design Components*

### 9.2.1 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 | 8 | 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 |

### 9.2.2 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 | 8"x8"x8" | 175 | [illegible] |
| 2 | Hose | 6 | Contains the hydraulic fluid and bring the hydraulic fluid to the appropriate locations | Holds fluids | Rubber | 6 ft | 30 | https://www.mcmaster.com/general-purpose-hose |
| 3 | Pump Motor | 1 | Provide power to the pump | Provides power | Steel | 3"x4"x2" | 380.5 | [illegible] |
| 4 | Piston | 4 | Converts the hydraulic pressure from the pump to a radial and thrust force on the shaft | Converts hydraulic pressure to a force | Steel | d=2" | 30 | https://www.mcmaster.com/#pistons |
| 5 | Molecular coating | 200 | Will be used to make the shaft and stand | Holds and transfers loads | Stainless steel | 300in^3 | 0.05 | [illegible] |
| 6 | Bearing | 3 | Reduces friction from the pistons | Optimize the system | Steel | Varies | 7 | [illegible] |
| 7 | Shaft | 1 | Holds the shaft | Holds | Steel | | 43.73 | https://www.mcmaster.com/shaft-couplings |
| 8 | Pressure Relief Valve | 1 | Stops the hydraulics system from being overpressurized | Safety | Steel | d=1/2" | 33.99 | https://www.mcmaster.com/pressure-relief-valves |
| 9 | Flow control valves x4 | 4 | Allows the team to vary the pressure of the hydraulic system to different pistons | Control | Steel | d=1/2" | 84.61 | https://www.mcmaster.com/flow-control-valves |
| 10 | Hose fittings x16 | 16 | Stops hydraulic leakages | Safety | Rubber | 1/2" | 1.43 | https://www.mcmaster.com/hose-fittings |
| 11 | Steel tubes (6ft) | 1 | Hold hydraulic fluids | moves fluid | Steel | 6ft | 17.90 | 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" | 31.31 | https://www.mcmaster.com/check-valves |
| 13 | 1/2"-20 screws (pack of 50) | 1 | Hold the hydraulic system together | Supports | Steel | 1/2"-20 | 5.27 | [illegible] |
| 14 | 5/8"-11 screws (pack of 50) | 1 | Hold the hydraulic system together | Supports | Steel | 5/8"-11 | 5.10 | [illegible] |