Productization and Instrumented Testing of a Corrosion Fatigue Test Device

by

Przemyslaw Michal Krol

Submitted to the Department of Mechanical Engineering in partial fulfillment of the requirements for the degree of

Master of Science in Mechanical Engineering

at the

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

June 2017

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Abstract

Corrosion Fatigue has been identified as the limiting factor of submarine propulsion shaft operation intervals. Increasing the inspection interval from 6 to 12 years could save a significant amount of money on procurement and maintenance costs. Corrosion fatigue data is sparse and incomplete and an initial prototype of a fatigue testing device that more accurately reflects the operational loading of sub shafts was designed in a previous thesis. The U.S. Navy has identified the device as improvement on current testing methods. The primary purpose of the fatigue testing machine has been identified within a long-term testing plan for the Navy. In this work, the key aspects of the design have been updated. The manufacturing, setup, operation, and maintenance of the device have been provided. Instrumentation has been as part of an effort to monitor motor health and to explore the possibility of detecting crack initiation within the test shaft. The test device has been used to collect relevant data provide baseline data on artificially pitted samples and unpitted samples test shafts in seawater for the Navy. Artificial seawater was used for testing consistency. A continued testing regime is recommended and outlined. Last, further design updates and ideas are suggested.

Thesis Supervisor: Alexander H. Slocum
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Chapter 1

1. Introduction

The United States Navy has a goal of increasing the scheduled maintenance interval of submarine propulsion shafting from the current 6-year operational interval to a 12-year operational interval. The solution to this complex problem of increasing the operational interval is multi-faceted and involves the design of the submarine as a whole as well as the propulsion shafting system. This thesis came out of and expands upon the work of a previous thesis done by Douglas Jonart. The first chapter of this work gives the context of the problem and previous work done. The second chapter addresses the development, design, and manufacturing of the test device. The third chapter leads through testing of the device itself as well as experimental results obtained by using the test device. The fourth chapter centers on areas of improvement of the device and possible solutions.

1.1 The Problem at Hand and Project Motivation

Current submarine propulsion shafts are made of carbon steel and are coated with multiple layers of protective coatings. Operation and maintenance schedules call for the shaft to be in service up to six years. At the end of every operational interval of the shaft, the submarine is brought into a dry dock. The propulsion shaft is removed from the submarine and all sleeves and coatings are removed to allow for inspection to detect any defects. After all defects are repaired, coatings are re-applied and sleeves are fitted and the shaft is then returned to a rotating stock of shafts ready for use in another submarine. Thus far, all inspections have revealed imperfections, such as corrosion, pits, and in a few cases, cracks in the shaft. All shafts have needed refurbishment after just six years in service, signifying that doubling the operational interval will require improvement in one or more aspects. These changes could include changes of shaft material, improvements in coatings, or changes in the design of the propulsion shafting system itself.
The primary focus of the propulsion shafting system has been on reducing the risk of corrosion fatigue failure, as this is the current limiting failure mode. A six-step failure chain model [1] has been developed as a start to the solution and a beginning to a comprehensive understanding of the life of a submarine shaft. As a further, separate, but related effort, the development of an initial test device, described further in Chapter 2, began before continuing on to develop and test device that can recreate propulsion shaft life conditions for the duration and cycles of a propulsion shaft. Overall, if the propulsion shafting inspection interval was 12 years, the U.S. Navy could save $5-10 billion by buying 1-2 fewer submarines to fulfill the same mission requirements.

1.2 Thesis Motivation and Goal

This thesis stems out of a previous thesis written by Douglas Jonart and focuses on the afore-mentioned initial test device. The device is to be used for testing shafting materials under the same loading conditions as a submarine propulsion shaft. This work seeks to develop the initial prototype into a product that the United States Navy can purchase and use. This machine was to be rapidly deployable and low cost, using mostly off-the-shelf products and very few custom parts. The intended use for this device is to run short-term (up to one year in length) tests of shafting material candidates with various coatings to narrow down choices combinations of materials and coatings that are being considered in the design of propulsion shafting for the Navy’s next class of submarine. After narrowing down the choices, long-term (around seven to eight years) tests would be run on the air bearing test device developed by Jonart.

1.3 Purpose of this Test Device

Previous work identified the need for the test device, which was originally intended only to prove several concepts and was called the Bench Level Prototype (BLP). The BLP was designed to simulate the two loading characteristics of a submarine shaft while part of the shaft is submerged in seawater. The two loading characteristics are: a cyclic bending stress imposed on the shaft by the heavy, cantilevered propulsor and a constant torsion
load. Once demonstrated, the device was identified as a necessity to fulfill two purposes. The first is to generate data for S-n curves for different combinations of shaft materials and coatings that the U.S. Navy is considering. The second is longer-term testing (up to one year) of the shafts and coatings to see the degradation of the coatings over time under constant operating conditions.

1.3.1 Background on Corrosion Fatigue and Pitting
Corrosion fatigue of metals is very complex and is affected by the interaction of many factors, including metallurgical, environmental, and mechanical properties [2]. Mechanical and metallurgical factors include alloy composition, mechanical strength, mechanical working, distributions of allowing elements and impurities, orientation of grains, cyclic loading frequency, maximum stress, and crack size and shape and its relation to component size and geometry. The temperature, electrochemical potential, and pH of the environment and any coatings or inhibitors on the metal all influence corrosion fatigue of that metal [2]. Melchers wrote that even though information on corrosion and corrosion theory is readily available, the tests are often done in ideal and specific conditions [3]. This means that the information is not able to be applied in real-life situations and the interactions are not fully understood in most cases.

One of the most relevant factors to submarines is that corrosion fatigue is frequency dependent [4]. At high frequencies, phenomena at the crack tip may invalidate an extrapolation from testing conditions. Transition from pits to cracks is frequency dependent as well [4]. The change of transition conditions occurs around 5Hz. Submarine shafts typically operates below 5 Hz, so this puts an upper limit on the allowable frequency and limits the ability to accelerate testing. Research has shown that reducing frequency from 20Hz to 1 Hz reduced cycle life by approximates 2 in seawater. Further reducing to 0.2 Hz reduced life by approximately 20%. Decreased cycling frequency causes marked reductions in high-cycle fatigue strength under free-corrosion conditions [2].
1.3.2 Market Availability of test devices.

In the past, the majority of corrosion fatigue experiments have been performed on specimens loaded in bending or specimens loaded uni-axially. Torsion or more complex multi-axial loading states have not been used in corrosion-fatigue work. On the market, most commercial test devices available do either reversed cyclical bending or reversed cyclical torsion, but not both. There are fatigue testing machines available for purchase that test specimens in combined axial and torsional loading, but no machines that can run fatigue tests for a cylindrical shaft with reversed cycling bending and constant torsion. No machines found are able to test with a section of the shaft submerged in an environment. Existing machines simply do not provide the conditions and environment experienced by submarine shafts. Therefore, there is room in the marketplace for such a machine as is described in this work.
Chapter 2

2. Machine Functionality and Prototype

2.1 Evolution of the Design

The fatigue testing machine applies the combined loading characteristics of a submarine shaft: fully reversed bending and constant torsion. In broad terms, the machine consists of two DC gear motors, coupled to each other via a test shaft. Both motors are on pivots, so that a bending moment can be applied via hanging weights. In order to get a constant torsion on the shaft, one of the two motors spins backwards creating a resistance load. Part of the test specimen runs through an enclosure, or tank, through which seawater flows to be the environment mentioned in Chapter 1 of this thesis.

Many of the design decisions were influenced by the availability of parts and ease of setup while fulfilling the requirements. The biggest decision was to use gear motors. These were selected due to the availability, the specifications that allow them to run at the test conditions desired, and the ease of being able to set up the whole test device. With the motors, the two sides of the machine would be mirror images, and require the same mounting interface and shaft collars. Any maintenance required and replacement parts would be the same for both.

To meet the functional requirements of simulating the loading of a submarine shaft, the same shear stress from the constant torsional load needs to be applied. This was determined to be 67 in-lbs for a ¼" test shaft and was the main driving factor in selection of the motors. The motors are Leeson DC Gear Motors that rated for ¼ hp and 100in-lbs of torque at 110 rpm. The calculations for torque required and how it relates to shaft size and motor size can be seen in Appendix A. As can be seen in the calculations, if the test shaft diameter was to be increased, larger motors would be required, which increases the cost, the size, and the weight of the testing device.
Each DC gear motor is attached to a custom motor mount, which is turn is attached to a frame. The attachment point is used as a pivot point. The first prototype of the test consisted of two DC gear motors that were attached to bushings via shoulder bolts. Lightbulbs were used to create a higher torsional load than just the motor spinning backwards. A picture of the prototype can be seen below in Figure 1. Initial tests showed a rocking motion of the motors as the test shaft spun around, with one motor rocking further on its pivot than the other. This rocking motion could be attributed to the misalignment of the motors or lack of shaft straightness. In order to prevent this “hop” and remove any parasitic loads on the shaft, flexures were designed by Jonart [1] to allow for motion in both horizontal directions, but still meeting the requirements for stiffness in the other directions. A picture of the flexure is attached for reference and can be seen in Figure 2. For vertical misalignment or runout in the shaft, the flexures can bend forward and backward together to accommodate the motion that the motors go through. For any horizontal misalignment, one flexure can move forward and one can move backward to accommodate any twisting. In such a way, any runout is accommodated for the full rotation of the test specimen. The prototype flexures are made out of aluminum and are manufactured using a waterjet. Aluminum was chosen as the material due to its resistance to corrosion while being strong enough to support the stresses from the weight of the motor assemblies and additional weight. Due to the complexity of the shape and the low cost and quick manufacturing speed of a waterjet, the flexures are made using the waterjet.

![Figure 1. Photo of the First Prototype](image-url)
Further testing showed that the initial water tank design was not ideal. The tank was a 3x3 aluminum channel with a hole for a barbed tube fitting on the bottom for water to be pumped in. There were two more holes near the middle to allow for the test shaft to pass through. The tank can be seen in Figure 1 and was not rigidly attached to the frame, which resulted in movement due to the vibrations of the machine. This caused lots of water to be leaking as the test went on. To combat this, a rigidly attached water tank was designed and can be seen below in Figure 3. This tank was to be a hanging design in which it was to be attached to a 2" x 2" square aluminum extrusion from the top. The tank consists of a 2" x 2" aluminum channel with one hole at the bottom, two holes side-by-side near the top, and one hole on each side near the middle to allow the test shaft to pass through. Each hole has a barbed tube fitting to attach a hose. Water is pumped into the tank through the bottom hole and drains back into the reservoir through the two top holes. In order to prevent water flowing down through the holes that the test specimen passes through, there is a rubber sealing washer pressed up against the tank wall on both sides of the tank.
The last design change made was to the weighting system. Each motor mount has a 5/8" threaded rod screwed into it directly in the center below the output shaft. Originally, each motor had a hanging weight bar attached to. Each motor was weighted equally to get the same moment loading on the shaft. As the bending load gets heavier, the flexures flex more and more during the rotation of a shaft. While running tests with high bending loads, the weights would start swinging around. In order to mitigate the swinging, a combined weighting system was devised. The combined weighting system consists of an aluminum bar that has 5 holes drilled into it. Looking at the bar from a front view as can be seen in Figure 4, there are two holes on near the top and close to the outer edges, while three holes are on bottom, two near the outer edges and one directly centered. The weight bar is connected to the threaded rod on the motor mounts via wire rope lanyards. The weight bar is hanging off of 90° shackles. There are three holes so that more weight can be hung for a higher bending load. The weight bar on the prototype machine can be seen in Figure 5.
2.2 Device Assembly

This part of the thesis leads through the process of assembling the current test device. All supporting drawings of custom parts and a bill of materials are included in appendix B.

2.2.1 Physical Assembly

2.2.1.1 Motor and Tank Assembly

The motor and housing assembly is a sub-assembly of the overall machine. Figure 6 can be used as reference.

1. Take off the screws and faceplate that come with the motor from the manufacturer.
2. Align the motor shaft and surrounding extrusion with the 1.55” hole on the housing. Use the screws from the manufacturer to attach the housing bracket.
3. Insert the 5/8” threaded rod into the corresponding hole of the housing bracket.
4. Attach a flexure on each side of the housing bracket using ¼”-20, 1” long socket head screws. Include the flexure spacer in between the housing and the flexure.
The steps for assembling the water tank are as follows:

1. Insert a barbed tube fitting in each holes on the 3"x2" aluminum tubing as shown in Figure 3.
2. Attach the hose to each of the barbed tube fitting and make sure the barbed tube fitting are still tight.
3. Press-fit both the marine-grade polyethylene pieces into the aluminum channel.
4. Attach the top mount onto the 2"x2" 80/20 Aluminum Extrusion as shown below. Now this sub-assembly can be attached onto the frame assembly as described in section 2.2.1.2.
2.2.1.2 Frame Assembly

1. These are the steps to assemble the frame for the fatigue testing device. A reference can be seen in

2. Cut all 80/20 extruded aluminum bars to size.

3. Once all the 80/20 extruded aluminum pieces are cut to size, use the 80/20 hardware to attach two 27.5" long 2"x1" pieces and two 13" long 2"x1" pieces in a rectangular fashion as shown in Figure 7.

4. Take the four remaining 13" long 2"x1" pieces and attach them vertically in the locations shown. Do not fully tighten the fasteners yet, as some adjustment may be necessary later.

5. Take the two 25.5" long pieces and attach each one to two of the vertical 13" bars as shown.

6. Attach two 7" long 1"x1" pieces to the 25.5" long pieces as shown.

7. After the Tank Assembly in Section 2.2.1.2 is complete, attach the 2"x2" piece in between the 1"x1" pieces on each side as shown below.

![Figure 7. Photos of the Steps Taken to Assemble the Frame](image_url)
2.2.2 Control Panel and Electrical Wiring

This fatigue testing device is wired in such a way that it shuts down automatically once a test shaft is broken. Using a solid-state switch requires a voltage signal from both the power outlet and a second source (in this case the motor that is spinning backwards is generating electricity) for an electric current to go through. As long as the machine is running and the test shaft is intact, the motor is back-spinning and the control panel is powered. This eliminates the need for a turn-off switch that gets pressed once the test is complete since the shaft is broken and the motors are no longer mechanically coupled, so the back-spinning motor is no longer generating electricity. In order to start the machine, the override switch must be pressed and then the motor controller is powered on. The wiring diagram and control panel can be seen below in Figure 8 and Figure 9.

![Wiring Diagram of the Fatigue Testing Device](image)

Figure 8. Wiring Diagram of the Fatigue Testing Device

![Photo of the Control Panel](image)

Figure 9. Photo of the Control Panel
2.3 Device Operation and Maintenance

2.3.1 Operation Procedure
To start the fatigue testing device, first make sure to turn the speed knob to 0. Then turn the power on to the motor controller unit. Next, hold the override switch while slowly turning the knob to 100. Now the device is at operating conditions.

To turn the LabVIEW program on, open the BLP DAQ VI. Open the ‘Bock Diagram’ window. For a new test, double click the ‘Open Data Storage’ block and change the name of the file that data will be recorded to. In the ‘Front Panel’ window, the motor current, motor temperatures, and vibrations are displayed in graphs. To start saving data, click the green toggle switch corresponding to the data that you want to save. All the files get saved as a TDMS file, which can be opened using Excel or LabVIEW.

2.3.2 Maintenance
The test device needs maintenance in only one regard – replacing the carbon brushes in the motor. The carbon brushes wear out on these DC motors as the motor is used. The brushes on the powering motor tend to wear out quicker than those on the generating motor and need to be replaced about every two and a half months of constant use. The carbon brushes on the generating motor need to be replaced about every four months of constant use. The required brushes for these motors are Leeson M1900010.47 carbon brushes.

2.3.3 Test Shaft Installation and Removal
The procedure for installing and removing test specimens requires the removal of water tank and can be a bit complicated at first. It can be seen in Figure 10 and is as follows:

1. Loosen end-feed fasteners that connect the 2"x2" 80/20 aluminum to the 1"x1" 80/20 aluminum pieces
2. Insert test specimen through both sides of the water tank
3. Slide a Rubber Sealing Washer on each side of the shaft
4. Slide the neoprene rubber tube onto each side of the shaft
5. Slide a worm-drive hose clamp onto each neoprene rubber tube
6. Slide a shaft coupling onto both ends of the test shaft. Make sure the full length of the coupling has gone over the shaft.

7. Position the 2"x2" 80/20 aluminum so that the couplings and test shaft nearly align with the output shafts of the motors.

8. Slide the couplings into position on the output shafts of the motors. Tighten the bolts.

9. Position the 2"x2" 80/20 aluminum so that the test shaft is nearly centered in the through-holes of the tank. Tighten the end-feed fasteners holding the 2"x2" 80/20 aluminum to the 1"x1" 80/20 aluminum pieces.

10. On each side, slide the rubber washer, the rubber tube, and the worm-drive hose clamp to the aluminum water tank. While pressing these items firmly against the side of the tank, tighten the screw on the hose clamp to keep these firmly in place.

11. Test can now be started.

Figure 10. Photo Instructions of the Installation of a Test Specimen

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2.4 Recommended Changes for an Improved Device

The fatigue test device has used to collect data on 12 tests accumulating 24 million cycles and has been proven to work as intended. In order to make the machine applicable to more situations and increase user friendliness, a couple changes are suggested. The first suggestion is to change the design of the flexures and use a different material. Even though aluminum is resistant to corrosion, it does not have an endurance limit and will break after a finite number of cycles. A new design using spring steel is suggested in Chapter 4. The other suggestion is to modify the way the constant torsion is applied to the shaft. With the current setup, the torque regulation is very coarse and depends on the use of lightbulbs or other constant resistance items. Ideas for a different method of imposing a torque on the shaft are further discussed in Chapter 4. The third suggestion is the alternate the design of the water tank to allow for an easier method of installing and removing test specimens.
Chapter 3

3. Instrumentation and Experimental Data

3.1 Testing Durability of the Gear Motors

3.1.1 Background on Testing
This chapter is focused on the use of instrumentation to monitor the health of the two DC gear motors used on the test device, pictured below in Figure 11. The instrumentation, described further in section 3.1.2, is being used for three intended purposes. The first is to determine the useable life of the motors as well as a maintenance schedule for the BLP. The second is to determine whether measuring electric current or temperature can be used as a simpler and cheaper alternative to accelerometers as a method of monitoring motor health and predicting failure. The third is to determine whether electric current or vibrations of the motor can be used to determine when a crack has started to propagate within the shaft.

3.2.2 Instrumentation

3.2.2.1 Equipment
The instrumentation being used is purchased from National Instruments. The data acquisition system pairs well with the NI LabVIEW software and allows for compatibility and reliable implementation. Temperature, current, and vibration were the chosen
parameters to be monitored to get a complete view of what is happening with the motors. The instrumentation used for testing is as follows:

- 1 National Instruments CompactDAQ 9178
- 1 NI 9401 5V/TTL Digital I/O Module
- 2 NI 9230 3-Channel IEPE, AC/DC Analog Input Accelerometer Modules
- 1 NI 9227 4-Channel Current Input Module
- 1 NI 9227 4-Channel Current Input Module
- 1 NI 9227 4-Channel Current Input Module
- 2 K-type Thermocouples
- 4 Industrial 2-pin IMI Accelerometers, 100mV/g
- Monarch ROS-P Optical Sensor with SPSR-IM Self-powered sensor interface module

A LabVIEW program is made to monitor temperature, current, and vibration and to save the data for further analysis. The program can be seen in Appendix C for reference. Accelerometers were chosen to monitor vibrations of the bearings in the gear motors to determine if there is any damage. Current is directly related to the amount of torque being generated by the motor. An increase in the current for the same testing parameters can be a sign of bearing damage. Temperature of the motors can be an indication of damage as well if the motor is working harder to perform at the desired speed. In order to get a good idea of the motor health, all three are being monitored and analyzed for long-term trends.

### 3.2.2.2 Layout

#### 3.2.2.2.1 Mounting Accelerometers

The accelerometers are mounted above the gears of the gear motor as can be seen in Figure 12, as close to the bearing on the output shaft as possible. Figure 13 is an image of the Leeson DC gear motor and shows where the bearing is located. Due to the size of the accelerometer and the curved surface around the bearing, placing the accelerometer on top of the bearing is not possible.
Figure 12. The accelerometer is mounted on top of the gearbox of the motor

Figure 13. Image of the Leeson DC Gearmotor

3.2.2.2 Thermocouple locations
The thermocouples are placed at the end of the gear motor as can be seen in Figure 14, as close to the electrical motor as possible. This is the hottest part of the motor housing and is a good representation of the temperature of the DC motor.
3.2.2.2.3 Wiring of Electric Current Measurement wires
The current draw of the motor is being monitored by the NI 9401 Current Module. It is wired in between the motor and the Leeson Motor Controller as can be seen in Figure 15.

3.1.3 Software and Data Collection
A custom LabVIEW program has been made to use in combination with the NI CompactDAQ. This program allows for continuous monitoring of the vibrations, current, and temperatures and allows the data to be saved. The live data shown includes an FFT of the acceleration data to see the frequencies that are the most common. The accelerometers data is collected at 6.4 kHz, the current data is collected at 2 kHz, and the
temperature data is collected at 2 Hz.

3.1.4 Data and Results
All tests completed in this paper were run at a shaft bending load of 41,500 psi and 75 in-lbs of torque. These conditions were chosen because the BLP is going to be run at similar conditions when used to gather data for the S-n curves. This bending load is causing a greater stress on the motor and bearings than the long-term testing at 6,000 psi and 75 in-lbs of torque that these motor will be running at during long-term testing. Thus, it will give a good idea of a lower limit on how long these gear motors will last when used by the Navy. The motor on the right side in Figure 11, called the Powering Motor, is getting power from the outlet while the left motor, called the Generating Motor, is back-spinning and generating power.

3.1.4.1 Cold-Start Warm up
3.1.4.1.1 Temperature

When the test is started up, it takes about 15,000 cycles before the gear motors get up to temperature. In this example, the powering motor temperature increases up to around 65 °C before staying about constant. The generating motor gets to about 45 °C and then levels off. This can be seen above in Figure 16.
3.1.4.1.2 Current

Figure 17. Graphs of motor current upon startup

Figure 17 shows that there is a spike in current in the powering motor at the very beginning of a test. This is due the initial torque that the powering motor has to exert in order to get the back-spinning motor turning. As can be seen in Figure 17, the current draw then gradually decreases until it levels out to about 1.6 Amps on average after about 3000 cycles and stays around that average for the rest of the test.

Figure 18. Steady-State current of the powering motor

The current of the powering motor at normal operating conditions can be seen in Figure 18. Throughout one revolution of the motor, the current goes up to around 1.64 amps and then decreases to about 1.57 Amps. The graph of current looks the same with and without a bending load.
3.1.4.2 Vibrations
A Fast Fourier Transform (FFT) is used to analyze the vibration data collected. Past research has been done that determine that the FFT is a reliable way of determining damage within a bearing through changes in the vibrational frequencies [5] [6]. Figure 19 shows the FFT for vibrations of the powering motor at 7.5, 10, 12.5, 15, 17.5, and 20 million cycles, respectively. All three graphs show the largest spikes at 129.35Hz and 258.7Hz. The graphs also show smaller amounts of vibration at 468Hz, 814 Hz, and 863Hz. There are small differences between the three graphs, but damage cannot be determined.

![Figure 19. Graphs of the Fast Fourier Transforms of the Vibrations of the Powering Motor at A) 7.5 B) 10 C) 12.5 D) 15 E) 17.5 F) 20 Million Cycles](image)

Figure 20 shows the FFT for vibrations of the generating motor at 7.5, 10, 12.5, 15, 17.5, and 20 million cycles, respectively. There are no discernible differences between the vibrations. Each graph shows spikes at 120, 260, and 517 Hz. From this vibration data, there is no sign of bearing failure detected in the testing thus far.
3.1.4.3 Trends over Time

3.1.4.3.1 Temperature

Figure 21 shows the average temperature of the gear motors over time as well as the ambient temperature at certain points in the testing. Throughout this testing, the
bending load and torsion on the test shaft have been constant at 45,000 psi and 75 in-lbs, respectively. The ambient temperature is the only external variable that has changed. Ambient temperature cannot be controlled for this test setup because it is set for the entire building and cannot be changed for each room individually. As can be seen in Figure 21, the running temperatures of both motors increased initially until about 10 million cycles. The motor temperatures have remained relatively stable since. The ambient temperature has fluctuated between 21°C and 28°C. The powering motor started at 54 °C and now runs at about 65 °C. The generating motor started at 38°C and now runs at about 45 °C on average. The increasing temperature of the motor can be related to two things: the increased ambient temperature and possible damage to the bearings and gears.

3.1.4.3.2 Current

![Figure 22. Average current throughout the testing](image)

Due to the normal fluctuations of current, the average current of a revolution is used as a means of comparison. The average current throughout all the tests can be seen in Figure 22. The current has stayed about constant at 1.6 Amps on average up until about 17.5 million cycles. After 17.5 million cycles the current has been trending upwards until a sharp increase followed by a decreasing trend, which then lead back into an increasing trend. Overall, there is a noticeable increase in the average current at which the motor is being powered.
3.1.4.4 When a Test Shaft Breaks

Figure 23. The electric current of the powering motor when a test shaft is breaking example 1

Figure 24. The electric current of the powering motor when a test shaft is breaking example 2

Figure 25. The electric current shown in Figure 24, focused on the last 5 cycles
Two graphs of the motor current when a shaft is breaking are shown in Figure 23 and Figure 24. As can be seen in the first example, the electric current amplitude increases steadily until it drops to zero within 3 revolutions. After a crack develops in the test shaft, it grows with each cycle. As it grows, the motor has to exert more torque on one side of the rotation and less torque on the other side of the rotation. This can be seen in the increasing amplitude of current during the breaking process. In other samples, such as in example 2, the crack propagates very quickly and the shaft breaks within one revolution. This is shown in Figure 25.

3.1.5 Continuing Work and Changes to Testing
As testing continues, the data collected will continue to be analyzed and trends will be determined. A third thermocouple is recommended to be added to monitor the ambient temperature of the room. The data from the third thermocouple will be compared with the temperatures of the motors to better understand why the motor temperature is fluctuating between and during tests. A voltage sensor is desired to be added to monitor the voltage draw of the powering motor. The voltage of the motor is directly related to the speed of the shaft. Thus, a voltage sensor will allow for comparison between the voltage and current going through the motor. We can see if the voltage has the same fluctuations as the current and can determine whether the fluctuations in torque are due to bending in the shaft, run-out of the shaft, or another reason.

3.2 Materials Testing Data
3.2.1 Test Specimen and Environment Setup
The aforementioned environment used in testing is artificial seawater. It is made from sea-salt, ASTM D1141-52 Formula a. It was always made per instructions on the packaging. While running the tests, pH was checked to be at 8.2 and adjusted accordingly if otherwise. Temperature of the seawater solution was kept at 21 °C.

Each test shaft is artificially pitted using the pitting procedure described in Appendix C.
3.2.2 S-n Curves

Figure 26 shows the graph of all fatigue life tests done while this thesis and the thesis this work expanded on was in progress. The Navy wetter and dry curves were created from data point obtained by the Navy. The “Flat” designation signifies that those tests were flat-plate specimens run on the LFE-150 fatigue tester. The Navy Class 1 samples were run on the BLP in the combined loading situation with part of the shaft in the environment. Initial tests show that flat plate specimens that are pitted and wetted with seawater do not last as long as those tested by the Navy. The pitted, dry specimens align well with the Navy’s dry curve for a bending stress of 60,000 psi, but otherwise follow the trend of the wetted curve rather than the dry curve. Initial testing on the BLP show that the pitted and wetted test shafts lie directly on or near the Navy’s wetted curve, which can be used to validate the design and testing capability of the fatigue testing device. The test shafts tested on the BLP are separated into two categories. The “Navy Class 1” data is from test shafts that were manufactured out of a section of a submarine shaft sent to the test location by the Navy. The “New Navy Class 1 Samples” are test shafts that were manufactured by the U.S. Navy in January 2017 specifically to be used for tested on this device.
Figure 27 shows a graph of only the test shafts that were tested on the BLP. These include the Navy Class 1 samples, the New Navy Class 1 Samples, and unpitted samples made from 12L14 Steel that were used in the interim period when no Class 1 samples were available. Thus far, only 1 Navy Class 1 sample and 1 New Navy Class 1 sample has been tested at a bending stress of 41,500 psi and a torque of 36 in-lbs. The Navy Class 1 sample lasted 2.7 million cycles, which is more than double that of any of the test shafts tested at 75 in-lbs of torque. No conclusions can be drawn from this test, though, as one data point is not enough to determine any trends. The 12L14 samples did not have an artificial pit when tested, so these samples rotated through more cycles than the class 1 steel. If there had been a bit, it would be expected that the 12L14 would go through less cycles than the class 1 test shafts. Further testing is currently in progress and recommended in section 3.2.4.
3.2.2.2 Comparison of Properties of Steels Tested

The chemical properties of Navy steel and 12L14 steel are shown in Table 1 and Table 2, respectively. The mechanical properties of Navy steel and 12L14 steel are listed in Table 3 and Table 4, respectively. Both the yield strength and the ultimate strength of the 12L14 steel are lower than that of the class 1 steel.

Table 1. Chemical Compositions of Navy Steel

<table>
<thead>
<tr>
<th>Element</th>
<th>Class 1</th>
<th>Class 2</th>
<th>Class 3</th>
<th>Class 4</th>
<th>Class 5</th>
<th>Class 6</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carbon</td>
<td>0.24</td>
<td>0.26</td>
<td>0.35</td>
<td>0.10</td>
<td>0.27</td>
<td>0.30</td>
</tr>
<tr>
<td>Manganese</td>
<td>0.15 -- 0.45</td>
<td>0.15 -- 0.45</td>
<td>0.30 -- 0.90</td>
<td>0.20 -- 0.45</td>
<td>0.20 -- 0.45</td>
<td>0.20 -- 0.45</td>
</tr>
<tr>
<td>Phosphorus</td>
<td>0.010</td>
<td>0.020</td>
<td>0.030</td>
<td>0.020</td>
<td>0.015</td>
<td>0.015</td>
</tr>
<tr>
<td>Sulphur</td>
<td>0.015</td>
<td>0.015</td>
<td>0.015</td>
<td>0.015</td>
<td>0.015</td>
<td>0.015</td>
</tr>
<tr>
<td>Silicon</td>
<td>0.35</td>
<td>0.35</td>
<td>0.35</td>
<td>0.35</td>
<td>0.35</td>
<td>0.35</td>
</tr>
<tr>
<td>Nickel</td>
<td>2.75 -- 3.50</td>
<td>2.75 -- 3.50</td>
<td>1.50 -- 2.00</td>
<td>1.50 -- 2.00</td>
<td>1.50 -- 2.00</td>
<td>1.50 -- 2.00</td>
</tr>
<tr>
<td>Chromium</td>
<td>0.34</td>
<td>0.50</td>
<td>0.50</td>
<td>0.50</td>
<td>0.50</td>
<td>0.50</td>
</tr>
<tr>
<td>Molybdenum</td>
<td>0.25 -- 0.60</td>
<td>0.25 -- 0.60</td>
<td>0.40 -- 0.60</td>
<td>0.40 -- 0.60</td>
<td>0.40 -- 0.60</td>
<td>0.40 -- 0.60</td>
</tr>
<tr>
<td>Vanadium</td>
<td>0.03</td>
<td>0.03</td>
<td>0.03</td>
<td>0.03</td>
<td>0.03</td>
<td>0.03</td>
</tr>
<tr>
<td>Copper</td>
<td>0.35</td>
<td>0.35</td>
<td>0.35</td>
<td>0.35</td>
<td>0.35</td>
<td>0.35</td>
</tr>
<tr>
<td>Tin</td>
<td>0.030</td>
<td>0.030</td>
<td>0.030</td>
<td>0.030</td>
<td>0.030</td>
<td>0.030</td>
</tr>
<tr>
<td>Arsenic</td>
<td>0.025</td>
<td>0.025</td>
<td>0.025</td>
<td>0.025</td>
<td>0.025</td>
<td>0.025</td>
</tr>
<tr>
<td>Titanium</td>
<td>0.02</td>
<td>0.02</td>
<td>0.02</td>
<td>0.02</td>
<td>0.02</td>
<td>0.02</td>
</tr>
<tr>
<td>Antimony</td>
<td>0.025</td>
<td>0.025</td>
<td>0.025</td>
<td>0.025</td>
<td>0.025</td>
<td>0.025</td>
</tr>
<tr>
<td>Boron</td>
<td>0.01</td>
<td>0.01</td>
<td>0.01</td>
<td>0.01</td>
<td>0.01</td>
<td>0.01</td>
</tr>
<tr>
<td>Bismuth</td>
<td>0.003</td>
<td>0.003</td>
<td>0.003</td>
<td>0.003</td>
<td>0.003</td>
<td>0.003</td>
</tr>
<tr>
<td>Cadmium</td>
<td>0.003</td>
<td>0.003</td>
<td>0.003</td>
<td>0.003</td>
<td>0.003</td>
<td>0.003</td>
</tr>
<tr>
<td>Lead</td>
<td>0.003</td>
<td>0.003</td>
<td>0.003</td>
<td>0.003</td>
<td>0.003</td>
<td>0.003</td>
</tr>
<tr>
<td>Zinc</td>
<td>0.003</td>
<td>0.003</td>
<td>0.003</td>
<td>0.003</td>
<td>0.003</td>
<td>0.003</td>
</tr>
</tbody>
</table>

1/ The percentage of phosphorus and sulphur together shall not exceed 0.03 percent.
2/ The chemical composition shall be adjusted for section size and heat treatment within the maximum limits in order to meet mechanical properties. The requirements for all classes are percent maximum unless a range is shown.
3/ When vacuum carbon deoxidation is used, the silicon maximum shall be 0.12 percent.
4/ Percentages of all elements in the table where limits are shown shall be recorded.
5/ Not required for heat analysis (see 4.4.2.1).

Table 2. Composition of 12L14 Steel [7]

Chemical Composition

The following table shows the chemical composition of AISI 12L14 carbon steel.

<table>
<thead>
<tr>
<th>Element</th>
<th>Content (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Iron, Fe</td>
<td>97.91 - 98.7</td>
</tr>
<tr>
<td>Manganese, Mn</td>
<td>0.85 - 1.15</td>
</tr>
<tr>
<td>Sulfur, S</td>
<td>0.260 - 0.35</td>
</tr>
<tr>
<td>Lead, Pb</td>
<td>0.15 - 0.35</td>
</tr>
<tr>
<td>Carbon, C</td>
<td>0.15</td>
</tr>
<tr>
<td>Phosphorous, P</td>
<td>0.040 - 0.090</td>
</tr>
</tbody>
</table>
Table 3. Mechanical Properties of Navy Steel [8]

<table>
<thead>
<tr>
<th>Material Specifications</th>
<th>Density (g/cm³)</th>
<th>Elastic Modulus (GPa)</th>
<th>Shear Modulus (GPa)</th>
<th>Ultimate Tensile Strength (MPa)</th>
<th>Yield Strength (MPa)</th>
<th>Fatigue Limit (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steel, forged</td>
<td>7.8</td>
<td>200</td>
<td>80</td>
<td>540</td>
<td>415</td>
<td>150</td>
</tr>
<tr>
<td>Class 1 MIL-6-212A</td>
<td>7.8</td>
<td>200</td>
<td>80</td>
<td>540</td>
<td>415</td>
<td>150</td>
</tr>
<tr>
<td>Class 2 MIL-6-212A</td>
<td>7.8</td>
<td>200</td>
<td>80</td>
<td>540</td>
<td>415</td>
<td>150</td>
</tr>
<tr>
<td>Class 3 MIL-6-212A</td>
<td>7.8</td>
<td>200</td>
<td>80</td>
<td>540</td>
<td>415</td>
<td>150</td>
</tr>
<tr>
<td>Class 4 MIL-6-212A</td>
<td>7.8</td>
<td>200</td>
<td>80</td>
<td>540</td>
<td>415</td>
<td>150</td>
</tr>
<tr>
<td>Class 5 MIL-6-212A</td>
<td>7.8</td>
<td>200</td>
<td>80</td>
<td>540</td>
<td>415</td>
<td>150</td>
</tr>
</tbody>
</table>

Table 4. Mechanical Properties of 12L14 Steel [7]

**Mechanical Properties**

The mechanical properties of the cold drawn AISI 12L14 carbon steel are outlined in the following table.

<table>
<thead>
<tr>
<th>Properties</th>
<th>Metric</th>
<th>Imperial</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tensile strength</td>
<td>540 MPa</td>
<td>78300 psi</td>
</tr>
<tr>
<td>Yield strength</td>
<td>415 MPa</td>
<td>60200 psi</td>
</tr>
<tr>
<td>Bulk modulus (typical for steel)</td>
<td>140 GPa</td>
<td>20300 ksi</td>
</tr>
<tr>
<td>Shear modulus (typical for steel)</td>
<td>80.0 GPa</td>
<td>11600 ksi</td>
</tr>
<tr>
<td>Elastic modulus</td>
<td>190-210 GPa</td>
<td>27557-30458 ksi</td>
</tr>
<tr>
<td>Poisson’s ratio</td>
<td>0.27-0.30</td>
<td>0.27-0.30</td>
</tr>
<tr>
<td>Elongation at break</td>
<td>10%</td>
<td>10%</td>
</tr>
<tr>
<td>Reduction of area</td>
<td>35%</td>
<td>35%</td>
</tr>
<tr>
<td>Hardness, Brinell</td>
<td>163</td>
<td>163</td>
</tr>
<tr>
<td>Hardness, Knoop (converted from Brinell hardness)</td>
<td>184</td>
<td>184</td>
</tr>
<tr>
<td>Hardness, Rockwell B (converted from Brinell hardness)</td>
<td>84</td>
<td>84</td>
</tr>
<tr>
<td>Hardness, Vickers (converted from Brinell hardness)</td>
<td>170</td>
<td>170</td>
</tr>
<tr>
<td>Machinability (based on 100 machinability for AISI 1212 steel)</td>
<td>160</td>
<td>160</td>
</tr>
</tbody>
</table>
3.2.3 Potential Issues with Tests
Although the results of the testing align well with the previously made S-n curves for wet and dry specimens acquired by the Navy, the tests for the shafts may not fully represent what is happening. The artificial pit is hemi-spherical with a depth of about 300 microns and can be seen in Figure XX below. For a ¼” test shaft, this pit takes up a significant portion of the cross-section of the shaft. This means that the ratio of pit size to cross-sectional area is much greater than that of a full-size submarine shaft of 21” OD. The stress concentration that this causes to the shaft is significantly greater than that would affect the full-size propulsion shaft.

The other potential issue is that while the pitting procedure was followed strictly, there is still a range of pit sizes. This could cause the shafts with larger pits to break earlier and faster, not necessarily due to corrosion fatigue, but just regular material fatigue. This also ties back to the problem of the ratio of pit size to overall cross-sectional area.

3.2.4 Recommended Further Testing
Much testing has been done to start the verification of the S-n curve for a wetted specimen. The magnitude of the influence of torsion on the shaft is still unclear and further testing is recommended. The following Navy Class 1 Steel test shafts are recommended to be run:

- Two pitted shafts at 41,500 psi bending load and 36 in-lbs of torsion
Completing these tests would give the Navy a more complete reference and a baseline when testing new materials and coatings. It would also allow for a better understanding of how the constant torsion influences the shaft life of a pitted and unpitted shaft. Running unpitted test shafts would give a good idea of how much the life of a shaft is reduced by the presence of a pit. It is recommended that tests on a cyclical torsion device are run as well and the data compared to pure bending tests and the tests run on this device in order to get a complete look at the effect of the influence of torsion on test shafts. To do all these tests however, ideally each test should be done at least three times to have confidence in the data, will take many years, or tests need to be run in parallel on multiple machines.

3.3 Possibility of Integration of LabVIEW and Operation of Machine
Currently the operation of the machine and the running of the LabVIEW program described above are two separate procedures. Integrating the two could provide benefits to testing and gathering data to start filling in the data of the failure chain described by Jonart [1]. If the LabVIEW program could control when the machine stops rotating, experimental data on the number of cycles a shaft rotates through from pitting to crack
initiation could be obtained in some cases. The amplitude of the current of the powering motor starts to increase as the shaft is breaking as seen in Figure 23 in section 3.2.4.4. If the LabVIEW program could shut the machine off when this starts to happen, the test can be stopped when a crack initiates. Then the test shaft could be continued to see how many cycles it takes from crack initiation to shaft failure.
4. Conclusions and Recommendations

4.1 Conclusions

This thesis seeks to expand on the design of a corrosion fatigue testing device, originally called the basement level prototype (BLP), and make it ready to be deployed to begin immediate testing. The BLP was originally a prototype to prove a concept and was selected to be used by the Navy. Many of the original design decisions were in the interest of cost, part availability, and design simplicity. Key areas of the design have been updated to counter some of the original flaws noticed in the functionality and the ability to run consistent tests. It has been determined that this machine's primary purpose will be for testing uncoated shafts to failure, primarily to supplement the creation of S-n data on shafts of various materials and treatments. Several copies of the BLP should be used in parallel to gather data and fill in the S-n curves for not only Class 1 Steel, but for all other materials of interest to the Navy. It would also be of value to the Navy and to the community of researchers to continue to perform testing that quantifies the effects of torsion in order to determine whether the torsional loading is necessary for testing. This testing has been started, but there has not been enough data collected yet to make any conclusions. Quantifying the effects of torsion would allow the Navy to first decide whether torque needs to be considered in future design analyses and testing and then to consider the effects into the design of the shafts themselves. The BLP has been outfitted with instrumentation in an effort to monitor the health of the motors as tests have running continuously. This has been to determine how long the motors will be usable before needing replacement due to failing bearings or a failed gearbox. While data thus far has shown that no damage has been detected yet, the instrumentation could prove to provide useful data for the Navy and there is a possibility of integrating the controls for the machine with the software for the data collection to gather data for filling out the failure chain.
4.2 Suggested Product

4.2.1 Idea 1
The current prototype has a small frame and in order to get clearance for the weights as well as the water reservoir below, it is attached to a large yellow tub. Pictured in Figure 29 the frame has been changed to get rid of the tub and still allow for clearance for the weights to hang above the tube. This design allows for easier access to the water tub. On the current design, refilling the water reservoir, testing the water for its pH level, and cleaning the reservoir can be a challenge. Since it is done relatively often, having easier access is of importance and is accounted for in this design.

4.2.2 Idea 2
A different idea of making this device a table-top tester can be seen in Figure 30 below. This design necessitates a hole to be cut in the in the table for the weights to fall through. The water reservoir, pump, and filter would be placed below the table for easy accessibility. This design also allows the control panel to be next to the machine, rather than on a separate table. Cord management would be more organized and easier in this case.
4.2.3 Control Panel
A new control panel is modeled in order to consolidate all the different pieces that the current prototype has. For ease of use and space savings, all of the wiring can be contained in one panel. The rough render of the panel is shown in Figure 31 below. The wires and cabling would be more organized and not exposed to the environment in this control panel as well. The control panel features a power switch with an LED, a speed control knob, an override switch, and a small display. The display would show the cycle count and rotational speed.
4.3 Additional Recommended Design Changes

4.3.1 Flexure Change
Due to fatigue problems with the aluminum flexures as test cycles accumulate, it is recommended to transition to steel blade flexures. The first set of flexures reached their limit and one broke. The design was re-evaluated and redesigned slightly to account for higher loading than initially intended. A new set was manufactured on the waterjet and installed in the interest of time and cost. In the meantime, a new set of steel blade flexures, pictured in Figure 32 below, has been designed. Using these flexures would add about $1500 of machining costs, but would guarantee that the flexures last the life of the machine without a chance if interrupting and possibly invalidating a test being run. Drawings and design calculations for these flexures can be found in Appendix B.

![SolidWorks Model of the New Flexures](image)

4.3.4 New Braking System
Before any changes to the braking system are considered, it is necessary for the Navy to determine if having a variable torqueing system is a requirement for testing. A variable torqueing system could prove to be useful if tests are to be run at a range of torque for one shaft or if the Navy wanted to simulate the torque profile of a submarine. Changing the braking system from the current one to an adjustable system could add complexity to the mounting system and coupling to the test shaft as well as maintenance
and operation. A few ideas are considered and include magnetic resistance or fluid resistance. Magnetic resistance could be linear or it could be done in steps. A fluid resistance unit, similar to those on indoor bicycle trainers could also be used. Other ideas include a variable electric resistance, such as a potentiometer or rheostat.

4.3.4 Additional Items for Redesign
Two more significant changes are possible to make daily use easier, but would require significant changes to the design. The water tank could stand to be redesigned to allow for easier changing of test shafts. Changing test specimens between tests requires many steps. While not critical, a different water tank design could make a swap of test shafts easier for the operator. The attachment interface between the motor mount and the flexures could be redesigned for easier and quicker attachment. In case a motor needs to be taken off or inspected, the current design does not have a quick or easy way of taking the motor mount assembly off of the flexures or alternatively, taking the motor off of the motor mount. While infrequent, removing and then reinstalling the motor proves to be a difficult process for a single operator. A clever interface would pose an interesting design challenge. Design changes of other components, such as the water reservoir, but have not been pursued due to the added complexity, cost, and time it would take to create.
Appendix A: Calculation of Torques

Power is defined as:

\[ P = \frac{F \cdot d}{t} = \frac{T / R \cdot \frac{2\pi \cdot R}{\text{Revolutions}} \cdot \text{Minutes}}{\text{Revolutions}} = T \cdot 2\pi \cdot \text{rotational Speed} \]

Maximum shear stress in a circular shaft is given by:

\[ \tau_{\text{max}} = \frac{T_{\text{max}} \cdot R}{J} \]

Where the \( \tau_{\text{max}} \) is the maximum shear, \( T_{\text{max}} \) is the maximum torque, \( R \) is radius of the shaft, and \( J \) is the polar moment of inertia of the shaft. The polar moment of inertia of a hollow shaft with inner diameter \( \text{ID} \) and outer diameter \( \text{OD} \) is given by:

\[ J = \frac{\pi \cdot (\text{OD}^4 - \text{ID}^4)}{32} \]

The current class of submarine uses engines rated at 60,000 shaft horse power. Using this power and an assumed speed of 120 RPM, torque on the shaft can be calculated to be \( 2.626 \times 10^6 \) ft-lbs or \( 3.15 \times 10^7 \) in-lbs. Navy guidance in Mil-H-2189 says to increase the torque by 20% to account for resistance from water when turning. This gives a torque of \( 3.15 \times 10^6 \) ft-lbs or \( 3.78 \times 10^7 \) in-lbs. The outer diameter of a submarine shaft is about 21 inches while the inner diameter is about 11 inches. Following the rest of the calculation gives the maximum shear stress on a current submarine shaft to be \( 2.19 \times 10^4 \) psi. This value should be matched for the scaled test shaft. For a ¼" test shaft, the torque required is 67.3 in-lbs, or a 1/8 horsepower motor at 120 rpm. For a 3/8" test shaft, that number jumps up to 227 in-lbs or ½ horsepower at 120 rpm.
Appendix B: Relevant Drawings and Materials

SOLIDWORKS Student Edition. For Academic Use Only.
Tank Bottom

SOLIDWORKS Student Edition.
For Academic Use Only.
Motor Mounting Bracket

B

A

2

1

Drill and tap for 1/4-20 UNC

Drill and tap for 3/4-10 UNC

Ø 1.550

Ø 2.50 typ

1.285

5.00

3.625

4.375

10.000 Ref

4.250 ref

Motor Mounting Bracket

SOLIDWORKS Student Edition. For Academic Use Only.
Sketch for Test Shaft Samples

Material: Shaft Steel

P. Krol

Title: Sketch for Test Shaft Samples

Drawn By: P. Krol

Material: Shaft Steel

Date: 11 Nov 16

Tolerances Unless Otherwise Specified:

- Angles: ±0.175
- Decimals X: ±0.015
- Decimals XX: ±0.010
- Decimals XXX: ±0.005

Break All Sharp Edges
Remove All Burrs

SOLIDWORKS Student Edition.
For Academic Use Only.
TOLERANCES UNLESS OTHERWISE SPECIFIED:
Angles: ± 1/8
Deviations: X ±0.0125
Deviations: XX ±0.010
Deviations: XXX ±0.005

BREAK ALL SHARP EDGES
REMOVE ALL BURRS

Sketch for Test Shaft Samples

P. Krol
Material: Shaft Steel
22 Mar 2016
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Spring Steel for Flexure

Date: 2 May 2017

Material: ASTM A36 Steel

Scale: 1:2

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### Material Specs:

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<tr>
<td>Young's Modulus (psi):</td>
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### Setup:

| Unclamped Length (in): | 1.5         |
| Width (in):            | 0.75        |
| Thickness (in):        | 0.01        |
| MotorAssembly Weight (lbs): | 21.4      |

### Calculations:

| $l_{xx}$ (in$^4$) | 6.25E-08 |
| $l_{yy}$ (in$^4$) | 0.0028125 |
| $l_{zz}$ (in$^4$) | 0.000000125 |
| Stiffness x (lb/in): | 1.288888889 |
| Stiffness y (lb/in): | 464000    |
| Stiffness z (lb/in): | 8700000   |
| Tensile Stress from Motor assembly (psi): | 1426.7    |
Appendix C: LabVIEW Program for Collecting Instrumentation Data
Appendix D: Pitting Procedure

1. Turn on Agilent 6611C Power Supply shown in Figure C1.
2. Press the "Voltage" button
3. Press the "Input button"
4. Press the button twice. Using the scroll wheel, scroll clockwise until the voltage reaches its maximum value.
5. Press the "Current " button
6. Press the "Enter Number" button. Type in ".028."
7. Make sure the apparatus is set up as shown in Figure C2.
8. Connect the positive voltage plug onto the end of the test shaft and shown in figure C3.
9. Align the test shaft directly under the nozzle of the solution
10. Raise the stand until the nozzle is firmly presses against the test shaft. The test shaft should be centered under the nozzle as shown below:
11. Turn the pump on.
12. Open the valve just slightly to allow for solution to flow.
13. Wait approximately 25 minutes.
14. Close the valve for the solution to stop flowing, Turn the pump off, and disconnect the positive voltage alligator clip from the test shaft.

15. Lower the stand height and remove the test shaft.

16. Clean test shaft with DI water

17. Bring to a stereomicroscope and measure the depth of the pit.

18. Record depth of pit in test shaft

Figure C2. Pitting Apparatus

Figure C3. Alligator Clip Connects to Test Shaft and Platform Raised to Firmly Press Test Shaft Up Against Apparatus.
Figure C4. Pump for Solution

Figure C5. Valve to Allow for Solution to Flow
Appendix E: Instructions for Setup and Use of the Air Bearing Machine

Initial Programming:

1. Install AKD Kollmorgen Software on Windows 7 Computer
2. Open Program and Connect to Motor

You will see this window:

On the right in the middle of the page, you will see the motors that are powered on and connected on the same network as the computer. Click the one you want and then click connect.
3. Now Connect to the second motor in the same way that you connected to the first

4. Navigate to Digital I/O Tab of Brake/Torque motor

5. In the row of DIN 6, open up the dropdown menu and select “9-Command Buffer”
Now the rest of the row will be able to be clicked on and customized


7. Type "IL.CMDU 3" under High Command Buffer, and "IL.CMDU 0" under Low Command Buffer. Click OK.

8. Repeat steps 5 and 6 in the DIN 7 row.
9. Type "DRV.DIS" under the High Command Buffer

10. Open the drop-down menu in the DOUT 1 row and select "Absolute velocity greater than." Type "150" in the white box next to the drop-down menu.

11. In the navigation menu on the left side, click "Limits"
12. Under the Current Limits sub-heading, type in 4.00 next to positive peak current, -
4.00 next to negative peak current, and 1.00 next to dynamic break peak current.

13. In the navigation menu on the left side, expand the menu under Spinner.

14. Click on Digital I/O menu. In the DIN 5 row, click on 9-Command buffer in the
drop-down menu. Do the same in the DIN 6 row drop-down menu.
15. Click the “Edit” button in the DIN 5 Row. Type in “vl.cmdu 120” under high command buffer. Under low command buffer, type “vl.cmdu o”, hit enter and type “DRV.DIS”.

16. Click the “Edit” button in the DIN 6 Row. Type in “DRV.DIS” under the high command buffer and leave the space blank under low command buffer.
17. For DOUT 1 and DOUT 2, select “Absolute Velocity Greater than x” in the drop-down menu under mode. In the DOUT 1 Row, type in “10.000” under Param. In the DOUT 2 Row, type in “150.000” under Param.

18. In the left-side navigation menu, select “Limits” under the Spinner expanded menu.

19. Under the “Current” subheading, type in “9.000” next to Positive Peak Current, “-9.000” under Negative Peak Current, and “1.000” under Dynamic Break Peak Current.

20. In the left-side navigation menu, select “velocity loop” under the Spinner expanded menu.

21. Make sure “Ramp Limiter” in the selected sub-heading. Type in “20” next to acceleration and “25” next to deceleration. These numbers may be automatically changed by the program to the nearest value it allows.
How to start a test if the machine is turned off:

1. Turn the power on DC Power master switch

2. Open the Kollmorgen Workbench software

3. Connect to the motors on the software

4. Make sure both of the emergency buttons or not pressed in. There is one on the machine itself and one on the control switches.
5. Press both of the top buttons of the control panel
6. On the software, Navigate to the 'Brake'. Then press 'Enable' on the top toolbar.

7. Navigate to the 'Spinner' and press 'Enable' in the top tool bar.

8. Hook up the air supply to the filters
9. Wait until the pressure sensor shows green.

10. All of the boxes at the bottom of the screen should be green.
11. Press the bottom right button on the control panel. This starts the test.
Bibliography


