Team 9: ArcelorMittal Boot Seal Project

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Executive Summary

ArcelorMittal’s I/N Tek Manufacturing facility, asked us to design an oil boot seal for their I/N TEK cold rolling mills with a lifetime of at least 18 months, due to previous premature failures of their oil boot seal designs. ArcelorMittal would also like us to create a test fixture capable of testing boot seal designs. We have gathered much information about the project, but we have not determined the root cause of the problem.

The I/N TEK facility runs 24 hours a day, 365 days a year; from this we calculated that our boot seal must survive at least 287 million cycles at 405 rpm at 90% uptime. It must also fit into the current mill system without requiring any modifications. The test fixture’s task is to simulate real world conditions experienced by the oil boot seal when attached to the drive spindles. It accomplishes this by applying displacements on the test samples, creating stresses. The seal must accommodate deformation caused by axial and bending stresses while maintaining an operating temp near ambient and containing gear coupling oil. Bending stresses are prominent in the seal, due to the 405 revolutions the seal must complete every minute in combination with a 1 degree working angle, which causes a displacement of 0.0785in. Axial stresses are less frequent and occur occasionally when work rolls are changed.

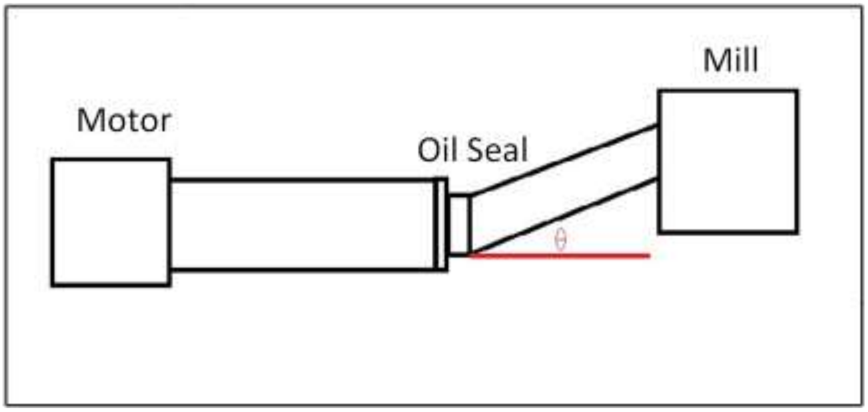
The concept generation process consisted of functional decompositions to define the subfunctions of both the seal and fixture, followed by brainstorming. The functional decomposition generated 4 necessary subfunctions for oil seal and 5 for the test fixture. We generated 18 seal ideas and 8 fixture ideas, which we then combined into concepts to be rated in the concept selection stage. To select a concept, we combined ideas into many design concepts and rated these ideas using a Pugh selection chart. At the end of this process, the top-rated design concept for the seal and the concept for the fixture progressed to the “alpha design” phase. We thoroughly defined each design and prepared them for rigorous engineering analysis. Then we performed the engineering analysis, created a design, defined the prototypes, and prepared a plan to validate them. Finally, we fabricated the prototype according to the design and validated it.

Table of Contents

Introduction

ArcelorMittal's I/N Tek Manufacturing facility has asked us to design an oil boot seal for their cold rolling mills that is capable of lasting through 18 months of use. The facility uses 5500 hp motors to power cold rolling mills. The Motors translate power to the mill through 2 drive spindles, which join at an oil filled gear coupling, shown in Figure 1 below.

Figure 1: Motor powers mill through 2 drive spindles, joined at the oil seal. Oil Seal must accommodate working angle.

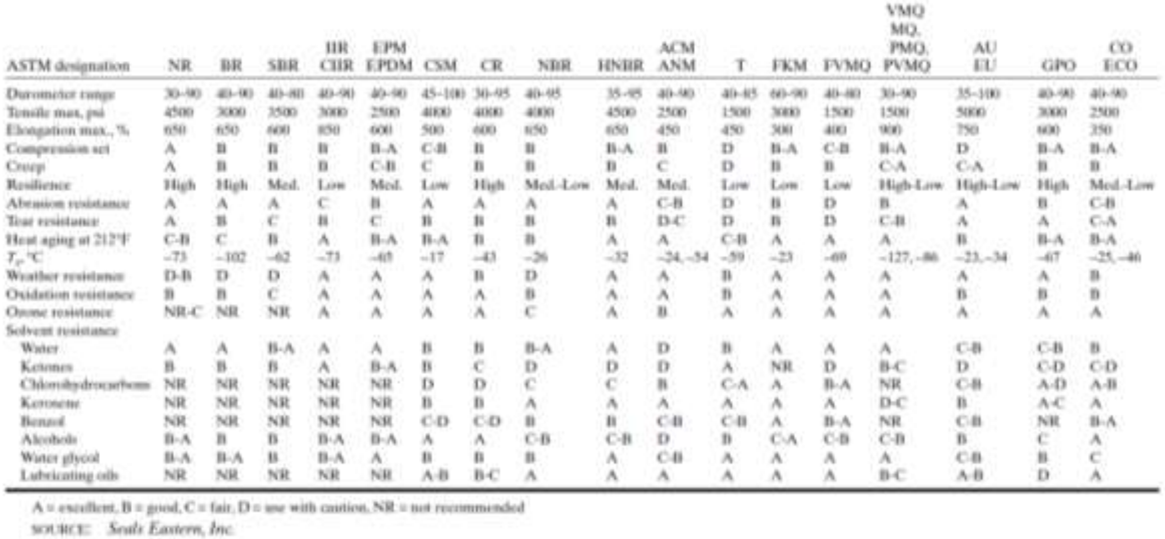


The spindles join at a 1 degree working angle, creating bending stresses in the oil seal during rotation. We have also been asked to create a test fixture that is capable of testing boot seal design and simulating real world conditions experienced by the oil boot seal on the drive spindle. If successful, this seal would prevent unnecessary shutdowns of mills and increase productivity and uptime at the I/N TEK facility, saving the facility approximately $50,000 that is lost every time a seal breaks. We have determined engineering specifications necessary to build a boot seal that meets ArcelorMittal's needs, and have generated designs for a new oil boot seal as well as a text fixture necessary for testing. We have fabricated a prototype of the test fixture.

Information Sources

We have gathered many drawings related to the mill: the original boot design drawing, a gearbox drawing, mill stand drawings, a roll diagram, a diagram for the arrangement of the mill system, the mill drive layout, 2 drawings for the flexible drive spindle, and the drawing for the most recent revision of the boot seal. These drawings all came from our ArcelorMittal contact, Andrew Grasley, and they gave us a general understanding of the system that the boot seal will be a part of. They also gave us details of how the system operates and how the boot seal must fit into the system. The drawing for the most recent revision of the boot seal will aid us in determining the root problem, as it details the dimensions and material used. We searched online the material used, Hydrogenated Nitrile Butadiene Rubber (HNBR), and found that it is very well suited to this application.[1] We have used CES EduPack 2012 to get material properties for HBNR. Figure 2 on page 2 shows a chart from *Harris' Shock and Vibration Handbook* that compares HNBR to other materials.

Figure 2: Comparison of elastomers' properties from *Harris' Shock and Vibration Handbook*



ASTM designation	NR	BR	SBR	IIR CIIR	EPM EPDM	CSM	CR	NBR	HNBR	ACM ANM	T	FKM	FVMQ	VMQ MQ, PMQ, PVMQ	AU EU	GPO	CO ECO
Durometer range	30-90	40-90	40-80	40-90	40-90	45-100	30-95	40-95	35-95	40-90	40-85	60-90	40-80	30-90	35-100	40-90	40-90
Tensile max, psi	4500	3000	3500	3000	2500	4000	4000	4000	4500	2500	1500	3000	1500	1500	5000	3000	2500
Elongation max., %	650	650	600	850	600	500	600	650	650	450	450	300	400	900	750	600	350
Compression set	A	B	B	B	B-A	C-B	B	B	B-A	B	D	B-A	C-B	B-A	D	B-A	B-A
Creep	A	B	B	B	C-B	C	B	B	B	C	D	B	B	C-A	C-A	B	B
Resilience	High	High	Med.	Low	Med.	Low	High	Med.-Low	Med.	Med.	Low	Low	Low	High-Low	High-Low	High	Med.-Low
Abrasion resistance	A	A	A	C	B	A	A	A	A	C-B	D	B	D	B	A	B	C-B
Tear resistance	A	B	C	B	C	B	B	B	B	D-C	D	B	D	C-B	A	A	C-A
Heat aging at 212°F	C-B	C	B	A	B-A	B-A	B	B	A	A	C-B	A	A	A	B	B-A	B-A
T_g, °C	–73	–102	–62	–73	–65	–17	–43	–26	–32	–24, –54	–59	–23	–69	–127, –86	–23, –34	–67	–25, –46
Weather resistance	D-B	D	D	A	A	A	B	D	A	A	B	A	A	A	A	A	B
Oxidation resistance	B	B	C	A	A	A	A	B	A	A	B	A	A	A	B	B	B
Ozone resistance	NR-C	NR	NR	A	A	A	A	C	A	B	A	A	A	A	A	A	A
Solvent resistance																	
Water	A	A	B-A	A	A	B	B	B-A	A	D	B	A	A	A	C-B	C-B	B
Ketones	B	B	B	A	B-A	B	C	D	D	D	A	NR	D	B-C	D	C-D	C-D
Chlorohydrocarbons	NR	NR	NR	NR	NR	D	D	C	C	B	C-A	A	B-A	NR	C-B	A-D	A-B
Kerosene	NR	NR	NR	NR	NR	B	B	A	A	A	A	A	A	D-C	B	A-C	A
Benzol	NR	NR	NR	NR	NR	C-D	C-D	B	B	C-B	C-B	A	B-A	NR	C-B	NR	B-A
Alcohols	B-A	B	B	B-A	B-A	A	A	C-B	C-B	D	B	C-A	C-B	C-B	B	C	A
Water glycol	B-A	B-A	B	B-A	A	B	B	B	A	C-B	A	A	A	A	C-B	B	C
Lubricating oils	NR	NR	NR	NR	NR	A-B	B-C	A	A	A	A	A	A	B-C	A-B	D	A

A = excellent, B = good, C = fair, D = use with caution, NR = not recommended

SOURCE: *Seals Eastern, Inc.*

We also found from searching online that there are no patents for seals in this application. However, we did find some patents on self-sealing fuel tanks [3]. These relate to our self-sealing concept. The process involves putting un-vulcanized rubber in between two other layers. When the tank is penetrated, the rubber absorbs the fuel and expands to fill the hole.

Andrew Grasley also provided us with the names of the oils and lubricants that the boot seals will be exposed to, the spindle operating temperature, the ambient temperatures of the facility, and the displacements the displacements that the seal is subjected to. We have also received thermo images of the spindle that show us the temperature of the rubber when it is in use. These told us about the environment that the boot seal will need to perform in. Mr. Grasley also provided us with a record of failure of past boot seals and steps taken to correct this, which gave us background about the problem we need to solve. Finally, Mr. Grasley gave us the specification that the boot seal must last 18 months running 90% of the time before being replaced in preventative maintenance, which gave us a target to achieve in our design.

The only specification for the HNBR on the drawing that we have is the hardness. We contacted our sponsor to see if he can find out the brand of the rubber used so that we can get more accurate mechanical properties of the rubber, but our sponsor was unable to. We are also unable to find stress vs. lifetime data for HNBR. We have not been able to find this in any of our literary or online searches. To get this, we have emailed HNBR producers (Zeon Chemicals and LANXESS) to see if can provide us with this data, but they have not given us the data.

Engineering Specifications

Boot Seal

Two Quality Deployment (QFD) worksheets were used to determine any engineering specifications not given to us by ArcelorMittal and their influence on ArcelorMittal's requirements. One QFD applied to the seal and one to the test fixture required to seal designs. The QFDs, found in Appendix D and E were useful in relating ArcelorMittal's requirements for the seal and test fixture into targets to the engineering specifications and exploring their relationships. The specifications represent every technical requirement necessary for a successful seal and test fixture to function properly. Each sponsor requirement was given a weight, with a higher weight indicating a greater influence on the completion of a satisfactory design. These weights are the same from the Pugh chart in Appendix G. These weights, which add to 100, were assigned after consulting with ArcelorMittal and uncovering what functions the seal must complete, followed by team deliberation as to the importance of each requirement and the effect each requirement had on building a satisfactory seal and test fixture that met our engineering parameters. The requirements needed by ArcelorMittal that the previous seal could not fulfill were given the highest weights.

With the highest weighted requirement first, ArcelorMittal required that the seal we develop fit their geometric constraints, be flexible, last a long time, be durable, be easy to install, be cheap and run quietly. Our engineering specifications for the seal were cycles to failure and cost.

Each of the specifications was given a score of 0, 3, 6 and 9 based on how it affected the customer requirements with 0 equaling no effect and 9 equaling significant effect. These scores were added and weighted, resulting in a rank for each requirement. These ranks enabled us to focus on key engineering specifications, and assign them reasonable, realistic values that would satisfy sponsor requirements and result in a satisfactory design. Competitive products used for comparison were previous boot seal generations. In Figure 3 below, engineering specifications for the oil boot seal are given.

Figure 3: Engineering specifications for the oil boot seal.

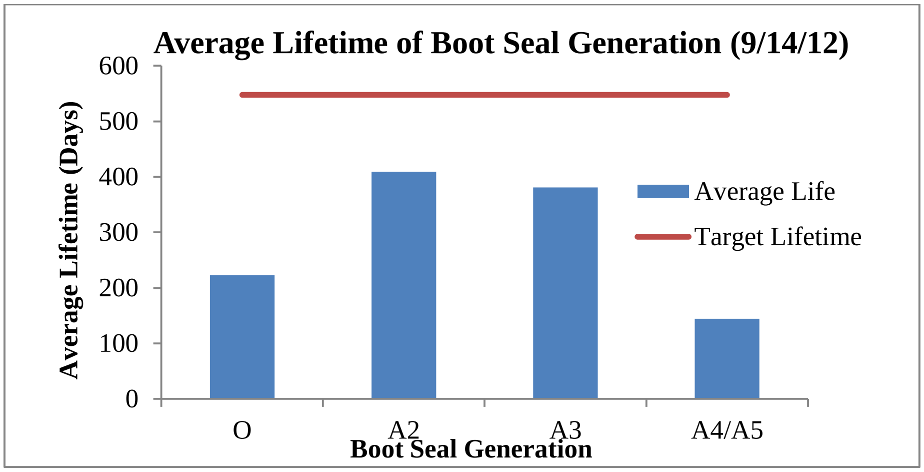
Cycles To Failure	**Cost**	**Noise**	**Geometric Constraints**			**Operating Temperature**	**Ease of Installation**
			Length	Outer Diameter Inner Diameter	Bolt Pattern		
> 287 Million Cycles	< $1000	< 70 dBA	4.50 in. +1.00/-0.59	15.83 in. 11.84 in.	0.53 in. diameter holes equally spaced	70 – 115 deg F	< 2 Hrs.

The new seal must also last at least 18 months or at least 287 million cycles at 405 rpm with an uptime of 90%. The target time of 18 months along with a max spindle speed of 405 rpm, and 90% uptime was enough to calculate the number of cycles that the boot seal should last, as shown below.

$$405\ rpm * \frac{60\ min}{1\ hour} * \frac{24\ hours}{1\ day} * \frac{365\ days}{1\ year} * 1.5\ years * \ 0.9\ = 287\ million\ cycles$$

Products used for comparison sake to the future boot seal were the old boot seal designs, as shown in Figure 4 below.

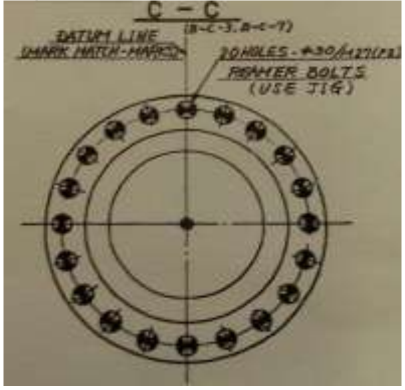
Figure 4: The average lifetime of each boot seal generation falls below the target.



Generations A4 and A5 are the same design but were created with different molds. Many of the A4/A5 generation are in current use, so their lifetimes increase each day. The graph uses data as of 9/14/2012. It is possible to see that the mean average days of operation of each previous boot seal generation has fallen significantly short of the Target of 18 months. It should be noted, that because boot seal generation A4 and A5 are more recent designs and are still in use, the sampling rate of failures for those generations was much smaller.

According to ArcelorMittal, the new seal must fit into the current figuration, with a length of 4.50 in. +1.00/-0.59, an outer diameter of 15.83 in. at its widest point on the motor side and 11.84 inches at its thinnest point on the mill side. The new seal must have twenty 0.53 inch diameter holes equally spaced in the 15.83 inch diameter circle to mount to the system: the same pattern as the old boot seal. This pattern is shown below in Figure 5 on page 5.

Figure 5: Bolt pattern of original seal.



Because the material must be flexible, an elastomer is the required material for the oil seal. Initially, we planned to test several elastomers in the gear coupling oil and complete a tension test to compare the oils effect on each elastomer’s ability to resist fatigue. Due to time constraints, there is not enough time to complete this test. For ease of installation, a value of less than or equal to 2 hours was chosen, which is the same amount of time it takes ArcelorMittal’s current seal design to be installed and is acceptable. The Noise constraint was chosen to be less than 70 dBA, which is equal to the loudness of normal speech level. Because of the noise levels created by the machinery at the IN/Tek facility, noise is not of high concern. A cost of less than $1000 for the oil boot seal was chosen due to the high cost of failure. The operating temperature of the oil seal must be within 70-115 deg f, or around the ambient temperature at the IN/TEK facility. The seal must resist heating up when in use, due to the fact that high temperatures within the drive spindles could indicate several possible failures. When elevated temps are noticed, the line is stopped.

Test Fixture

The text fixture must be able to simulate real world conditions experienced by the oil boot seal on the drive spindle. With the highest weighted requirement first, ArcelorMittal required that the test fixture we develop test quickly, test accurately, hold multiple test samples, maintain the correct system temperature, be durable, hold several types of oil, and be cheap to manufacture. Our engineering specifications for the fixture were operating temperature, cost, testing time, standard deviation of test and percent difference between test and spindle result. As with the boot seal, each of these specifications scores were added and weighted, resulting in a rank for each requirement. In Figure 6 on page 6, engineering specifications for the text fixture are given.

Figure 6: Engineering specifications for the test fixture.

Operating Temp	Multiple Test Samples	Cost	Durability	Testing Time	Standard Deviation of Test	Percent Difference between test and spindle result
70 – 115 deg F	10+	< $3500	Last > 287 Million Cycles	> 114 Days	< 97 Million cycles	<10%

The test fixture must be able to obtain a constant temperature because stretching the rubber test samples will cause them to heat up, which could cause premature failure. We specified that the test fixture should be able to maintain the ambient temperatures experienced in the I/N TEK facility. A test sample size of 10 has been specified for the test fixture. The total volume of the 10 samples will be equal to the volume of the rubber of the full-sized boot seal. The test fixture should be able to complete at least one full test of 287 million cycles, so the test fixture parts made of steel were designed to resist structural failure using buckling equations. We hope to achieve a 10% difference between the test fixture results and the real life spindle performance through maximizing the accuracy of the displacements experienced by the test samples in the test fixture.

Concepts Generated

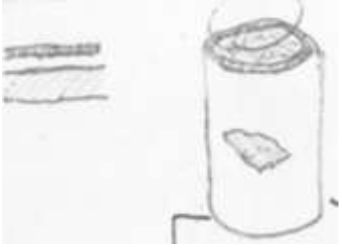
The concept generation process began with functional decomposition, followed by brainstorming. Two functional decompositions were created; one for the seal and another for the required test fixture. These functional decompositions can be found in Appendices F and G on PAGE. These diagrams allowed us to elaborate on the numerous functions that the seal and test fixture must accomplish. They gave insight as to what input initiates a function, what output results from each function and how the functions relate to one another. For instance, two functions our oil boot seal design must accomplish is to keep liquid inside the seal and resist chemical reactions. The relating input to these functions is oil. The relating output is oil. The flow is material. Our functional decomposition depicts the flow of material, energy and information with bold, thin and dotted lines. The resulting diagram allowed us to further generate concepts that better accomplished the sub functions needed for a successful seal and test fixture, as shown in Appendix H on PAGE. Brainstorming led to concepts for several test fixtures and oil seals. The focus of the brainstorming was to generate any possible concept that could solve the design problem, no matter how unfeasible it may have seemed.

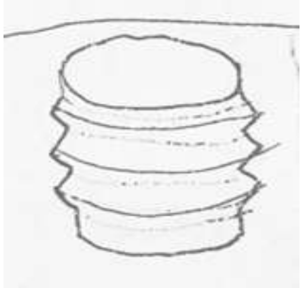
Our concepts for the seal can be divided into the categories of material design and cross section design. Material design alludes to the possible seal material(s) that could accomplish each function. Cross section design alludes to possible geometric shapes for the seal that could accomplish each function. Concepts for the test fixture were all motor driven and can be divided into the categories of singular seal testing and multiple seal testing. Several concepts for the seal are described in figure 7 and test fixture concepts are described in Figure 8. These designs were chosen to highlight the wide variety of concepts generated. Additional designs can be found in the Appendices H and I on PAGE.

Figure 7: Description of varying boot seal concepts.

Seal concept 1, a cross sectional design, uses an S-shaped cross sectional pattern similar to ArcelorMittal's current seal design. In this configuration, the S has been extended in to better distribute forces throughout the length of the cross section.

Seal Concept 2, a material design, uses a several layers of rubber. The inside rubber piece is initially isolated on both sides, yet is designed to react with oil if contact is made with the oil, forming a seal.

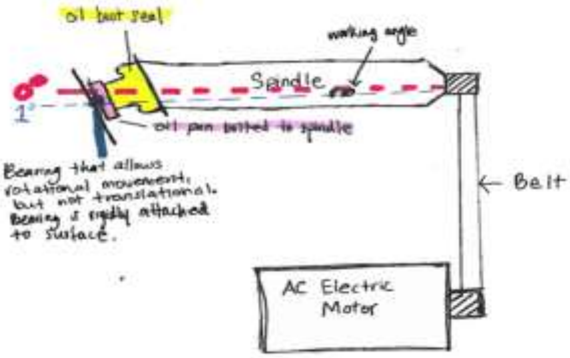




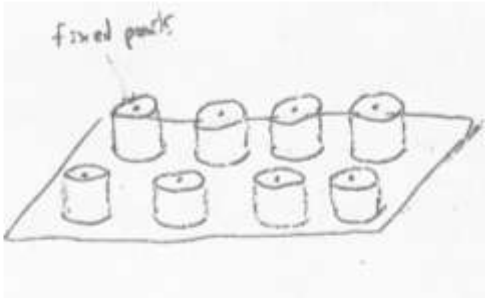
Seal Concept 3, a cross sectional design, uses an "accordion" design. This design is designed to handle large amounts of deformation. Its design is similar to that of an automotive CV boot, which is used to cover constant velocity joints that transmit power through variable angles.

Seal Concept 4, a cross section design, uses a spiral design to distribute forces on the cross section. This design also has the flexibility to stretch long distances.

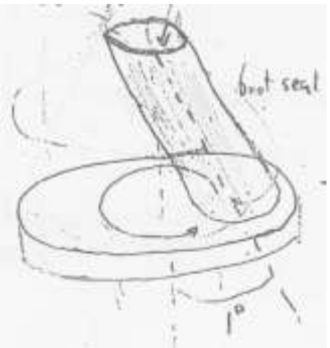
Figure 8: Description of varying test fixture concepts.



Test fixture Concept 1, a singular fixture design, uses a belt driven spindle to rotate a half scale oil boot seal. The seal is attached to a stationary surface using a bearing that allows rotational movement but not translational.

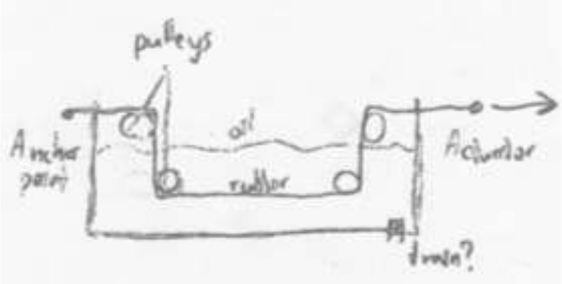


Test fixture Concept 2, a multiple fixture design, features a rotating surface. Attached to this surface are eight miniature seals all capable of rotating simultaneously.



Test fixture Concept 3, a singular fixture design, uses the same concept as concept 2, but does so on a larger full size scale. The surface that rotates a seal at an angle is attached to the base of the seal.

Test fixture Concept 4, would load a sample of the rubber that is submerged in fluid. This would not test the shape but would test how the material reacts with the fluid.



Concept Selection

After creating our list of design concepts, we created Pugh selection charts for the seal concepts and the fixture concepts using a 1 to 5 resolution scale. The datum, ArcelorMittal's current design, is rated at 3. 1 is much worse than the datum, 2 is worse than the datum, 3 is the same as the datum, 4 is better than the datum, and 5 is much better than the datum. The Pugh charts are shown below in figure 9, with the top five design concepts shown and rated. The full Pugh chart is shown in Appendices E and F on PAGE.

Figure 9: The "Big radius" concept and the"10-slot" fixture concept rated best in their respective charts. The Datum for the boot seal is the current design and for test fixture is actually putting the seal on the spindle.

Boot Seal Concept		Datum	The bulge	Spiral	Big radius	Rubber Accordion
Criteria	Weight					
Long lasting	70	3	4	4	5	4
Easy to install	5	3	3	3	2	3
Robust	15	3	4	2	4	4
Cheap	10	3	4	1	4	3
Total	100	300	395	335	460	385

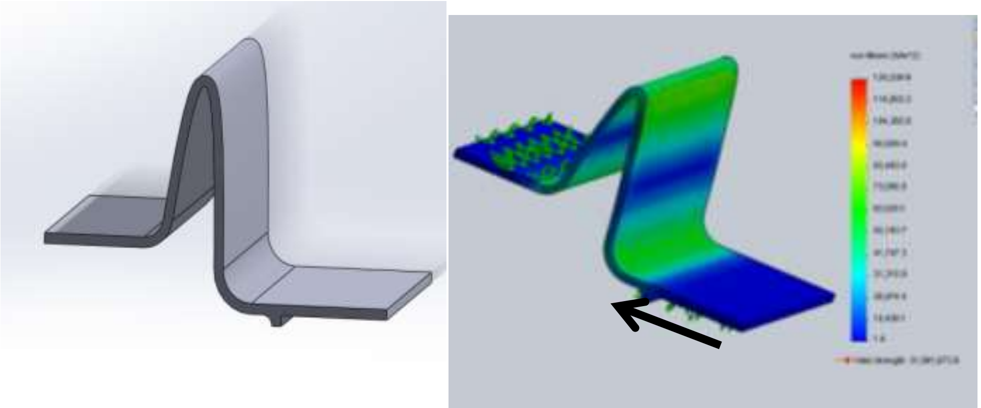
Test Fixture Concept		Datum	Slots	Chem box	Turntable	Many table	Vertial Turntable
Criteria	Weight						
Tests quickly	20	3	5	5	4	5	4
Tests accurately	20	3	1	1	2	1	2
Repeatable testing (Complexity)	10	3	5	5	4	4	4
Cheap	5	3	5	5	4	4	4
Durable	10	3	2	2	2	1	2
Multiple oil types	5	3	4	5	3	2	3
Keeps system at correct temperature	10	3	4	4	3	2	3
Easy to swap seals	5	3	5	4	4	3	4
Tests multiple samples	15	3	5	3	3	5	3
Total	100	300	375	345	310	310	310

The spiral received a 1 on the "cheap" criteria because this shape is likely very difficult to manufacture. The Many table design received a 1 in durability due to the intricate system that would need to be created for all the moving parts.

Boot Seal

For the boot seal selection, we used the four criteria long lasting, robust, easy to install, and cheap. Long lasting describes the seal's survival, and is measured in cycles to failure. This criterion dominates the design process and is weighted at 70%. We scored our seals on their relative ability to resist fatigue. To do this, we used finite element analysis. First, we created CAD models of the cross sectional profile of each seal design in SolidWorks. Next, we used the software to simulate the real-world displacements the seals would encounter in service and measured the maximum stresses occurring in each simulation for each seal. An example of a CAD model and its simulation is shown on the next page in Figure 10.

Figure 10: The seal shows stress concentrations on the curves in this test. The arrow in the picture shows which surface moved and which direction



We simulated displacing one end of the model up and down relative to the other end to simulate the regular fatigue flexing the seal would encounter due to its one degree bending in service. We also simulated tension and compression by stretching the seal 1in and compressing it 0.25in, which are irregular stresses that the seal will encounter occasionally and must survive in service. We used the maximum stresses from the regular flexing simulations to determine which seal would last longest in service. We used the simulation results for the irregular stresses to determine the robust score for each seal. We defined Robust as the seal's ability to resist irregular stresses: both the stresses we simulated as well as other accidental stresses encountered in the real world. Robustness is weighted at 15% because if the seal fails due to the occasional unusual stress, it will compromise the primary objective of being Long Lasting. Easy to Install is the time needed to mount the boot seal, and is considered of low importance because any setup will be relatively short compared to the 18 months of operation in between seal swaps. Additionally our contact verified the low importance of this criterion. Finally, cheap is the cost of producing each boot seal and is weighted at only 10% because of the relatively high cost the boot seal is addressing. Each boot seal failure presently costs the factory about $50,000. A costly seal that works well is preferable to a cheap, unreliable seal.

In comparison to the boot seal datum, rated at 3 for all criteria, the "Big Radius" concept scored highest. It rated 5 in long lasting due to very low simulated stress concentrations, 2 in easy to install because the orientation of material used in the design may increase difficulty in mounting it, 4 in robustness because it compresses and stretches at lower stresses than the datum, and 4 in cost, because it requires less material and a simpler mold than the "s" shape used by the datum. The seal's ability to weather the stresses well, particularly the regular flexing, makes it the best concept design. The "Bulge" concept scored second highest. It scored 4 for long lasting because of lower simulated stresses. It scored 4 in robustness because the extra material allows the design to stretch and compress easily, and 4 in cost because it uses less material and a simpler mold than the datum. The "Rubber Accordion" concept scored third highest. It scored 4 for long lasting because of lower simulated stresses and 4 for robustness because its shape allows it to handle compression and tension well. The "Spiral" concept scored fourth highest because of low predicted stresses during normal use, but its unusual shape cause it to be a less robust design meaning that it experience high stresses under high displacements and more difficult to manufacture due to the fact that this shape would require additional actions in the molding process. Some sort of additional rolling step before the rubber was fully hardened would like have to be added. All other designs scored below the datum.

Test Fixture

The criteria used in the fixture Pugh chart are tests quickly, tests accurately, repeatable testing, cheap, durable, multiple oil types, keeps system at correct temperature, easy to swap seals, and tests multiple samples. Tests quickly is scored by how long a fixture needs to test samples, which is judged the feasibility of each fixture to accelerate testing. Tests accurately is defined as how closely the test fixture replicates the conditions on the factory floor where the boot seal will be in use. These first two are each rated at 20% because we consider them critical to producing useful results. Repeatable testing is used to judge the consistency of the results that the test fixture will produce, as defined by the standard deviation of cycles to failure. This is considered somewhat important to the usefulness of the results, but not as much as the first two; the standard deviation of the results is due in part to the seals tested, so the goal is to minimize the contribution of the test fixture to this number. The data to judge this is not available because these fixtures are currently only concepts, so we scored this category based on the relative complexity of each concept; our reasoning is that a more complicated fixture will have more variables affecting its results and therefore less precision. The Cheap rating is simply the cost needed to produce the fixture, and is considered low-priority due to the relatively high cost of the problem the seal is to address, which costs approximately $50,000 per failure. Durability reflects the robustness of the fixture, and is considered somewhat important because the fixture must remain functional until testing of its samples are finished in order to produce valid data. Multiple oil types indicate how easy it is to place different lubricant solutions into the test fixture. This is of low importance because presently we are only interested in the effect of one particular oil on the samples. Keeps system at correct temperature defines the ability of the test fixture to keep the tested seal at real-world temperature when the testing is accelerated. This is reminiscent of the tests quickly and tests accurately criterion, but we felt that it deserved its own category because it rates the fixtures ability to integrate these two functions, and we therefore consider it somewhat important. Easy to swap seals is judged by the time needed to change one set of test samples, which is of relatively low importance because of the relatively long time the seals must be tested before producing results: Even with accelerated testing, testing for 18 month survivability will take significantly longer than the setup. Finally, tests multiple samples is scored by how many test samples the fixture can test simultaneously and is considered important because this enables us to output more useful data each time the fixture is run. One fixture testing multiple samples is better than multiple fixtures testing one sample each because the majority of cost is in the generation of the motion.

The fixture datum used is the factory floor where the seals are currently used, as this is currently the only way to test the boot seals. The “Slots” concept scored highest. It scored 5 for tests quickly because it could easily accelerate testing of the samples. It needs only a small motor and features a robust drive system. The system is also relatively simple and so scores 5 in repeatable testing. The simplicity and small actuator combine to make the fixture much cheaper than the datum, so here it also scores 5. The design tests rubber strips formed to the cross sectional profile of the actual boot seal. This means that it is easy to swap the test samples and also allows for the testing of multiple samples at once, so the fixture scores 5 in both easy to swap seals and tests multiple samples. However, testing strips also means that the seal suffers in tests accurately because it does not test a full boot seal like the datum does. The oil bath the strips are tested in can be used to cool the samples during accelerated testing, so the “Slots” concept scores well in the keeps system at correct temperature category. The advantages of the “Slots” concept greatly outweigh the disadvantages and make this the best concept for the test fixture. The “Chem Box” scored second highest for many of the same reasons because it is very similar to the “Slots” concept. It is better for changing the type of fluid used in testing, but can only test one sample at a time and so scores lower in Tests Multiple Samples. The “Turntable,” “Vertical Turntable,” and “Many Table” concepts all scored equally at third. “Turntable” and “Vertical Turntable” are nearly identical, varying only in their orientation to the earth, and score well because they test full-scale boot seals in a manner similar to the factory, but of course they cannot match the factory for the accuracy of test conditions. A downside of these two concepts is that they cannot test more than one seal at once. The “Many Table” design excels at testing multiple seals at once and could easily accelerate testing, but suffers in accuracy because it uses scaled-down boot seals. Its many small parts also make it less durable.

After choosing which test fixture design to go with we broke down the test fixture into sub-functions and created concepts to accomplish each of these sub-functions. After that we created a Pugh Chart scoring each concept. The sub functions were providing mechanical energy to samples, keeping liquid in, converting energy to mechanical energy, removing oil and applying oil. The following figures show a comparison of the different concepts.

Figure 11: Pugh chart of concepts for the "providing mechanical energy to samples" sub function

Providing mechanical energy to samples	Allow for individual samples to be removed easily	Securely hold samples in place	Keeps ends of samples oriented correctly	Even force distribution	Doesn't damage samples	Sum
weight	10	35	30	15	10	100
Hooks	3	3	3	3	3	300
Clamps	**2**	**4**	**5**	**5**	**5**	**435**
screw into samples	2	5	5	3	3	420
glue to samples	1	5	5	5	1	420

Figure 12: Pugh chart of concepts for the "Keeping liquid in" sub function

Keeping liquid in	Ability to resist leaks	Transparency	Longevity	Sum
weight	60	10	30	
sealant	3	3	3	300
painted surface	2	1	2	190
plastic lining	**4**	**1**	**3**	**340**
fish tank	3	5	2	290

Figure 13: Pugh chart of concepts for the "Converting energy to mechanical energy" sub function

Converting energy to mechanical energy	Efficient	Quiet	Reliable	Top speed	Achievable accuracy	Sum
weight	10	5	40	35	10	
"scotch Yoke"	**3**	**3**	**3**	**3**	**3**	**300**
tradition eccentric	**2**	**3**	**3**	**3**	**4**	**300**
linear servo motors	2	4	4	1	4	275
pneumatic	1	2	2	2	4	210
stepper motor	3	3	3	2	2	255

Figure 14: Pugh chart of concepts for the "Removing Oil" sub function

Removing Oil	Quick	Reliable	Complexity	Usability	Sum
weight	30	30	20	20	
drain at bottom	3	3	3	3	300
lower/ raise test samples	4	2	1	4	280
drain spout on	**3**	**3**	**3**	**4**	**320**

side					
suction device	3	2	2	2	230

Figure 15: Pugh chart of concepts for the “Applying Oil” sub function

Applying Oil	Even coating	Realistic coating	Ability to coat one side	Complexity	Sum
weight	30	30	20	20	
Immersion	**3**	**3**	**3**	**3**	**300**
Apply coating at setup	3	1	4	3	260
pump spray	3	1	5	1	240
pump trickle on	2	2	4	1	220

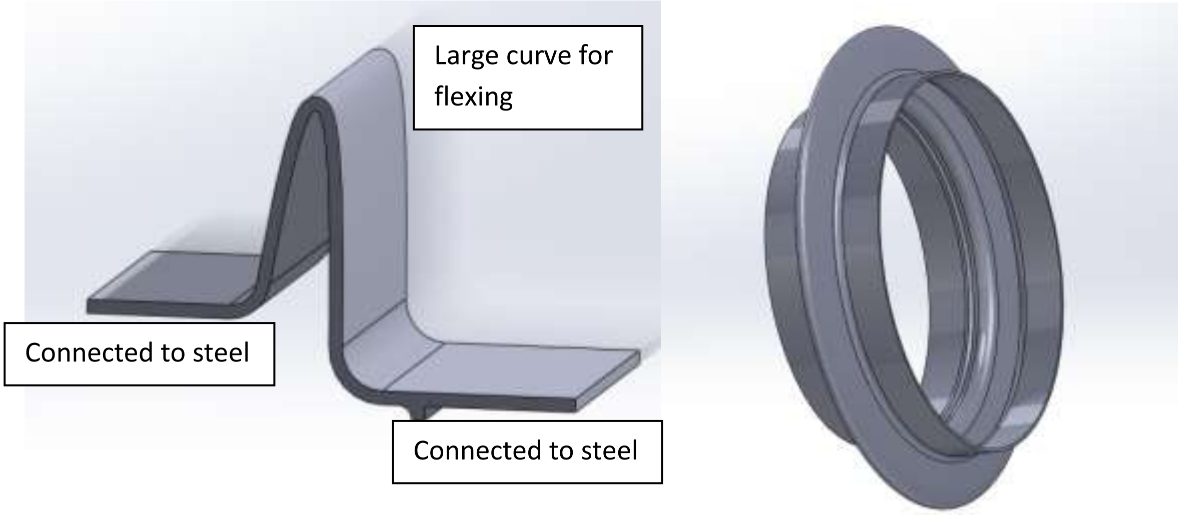
When trying to determine how to generate the require motion, we looked in to linear actuators. Although these showed some promise, we could not find any fit our needs.

Selected “Alpha Design”

Boot Seal

The selected design for the boot seal features a curve with a large radius to handle the necessary displacements while maintaining low stresses. A rendering of the seal’s cross sectional and revolved profiles are shown in Figure 16 below.

Figure 16: The profile of our design (left) is revolved to create the boot seal (right).

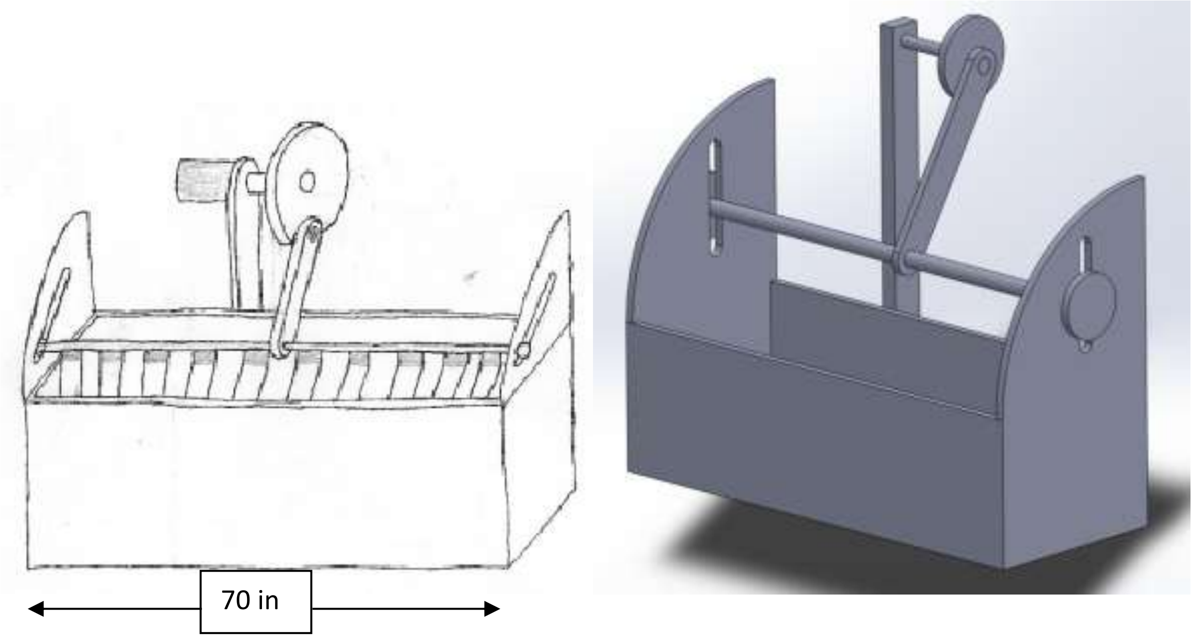


The curved midsection of the design is made from rubber, specifically HNBR, so that it may flex to accommodate displacement while still containing the oil in the gear coupling it houses. The ends shown in the figure are also HNBR: they are molded to the steel connectors on each side by which the boot seal will attach to the rest of the system in which it is to be used.

Test Fixture

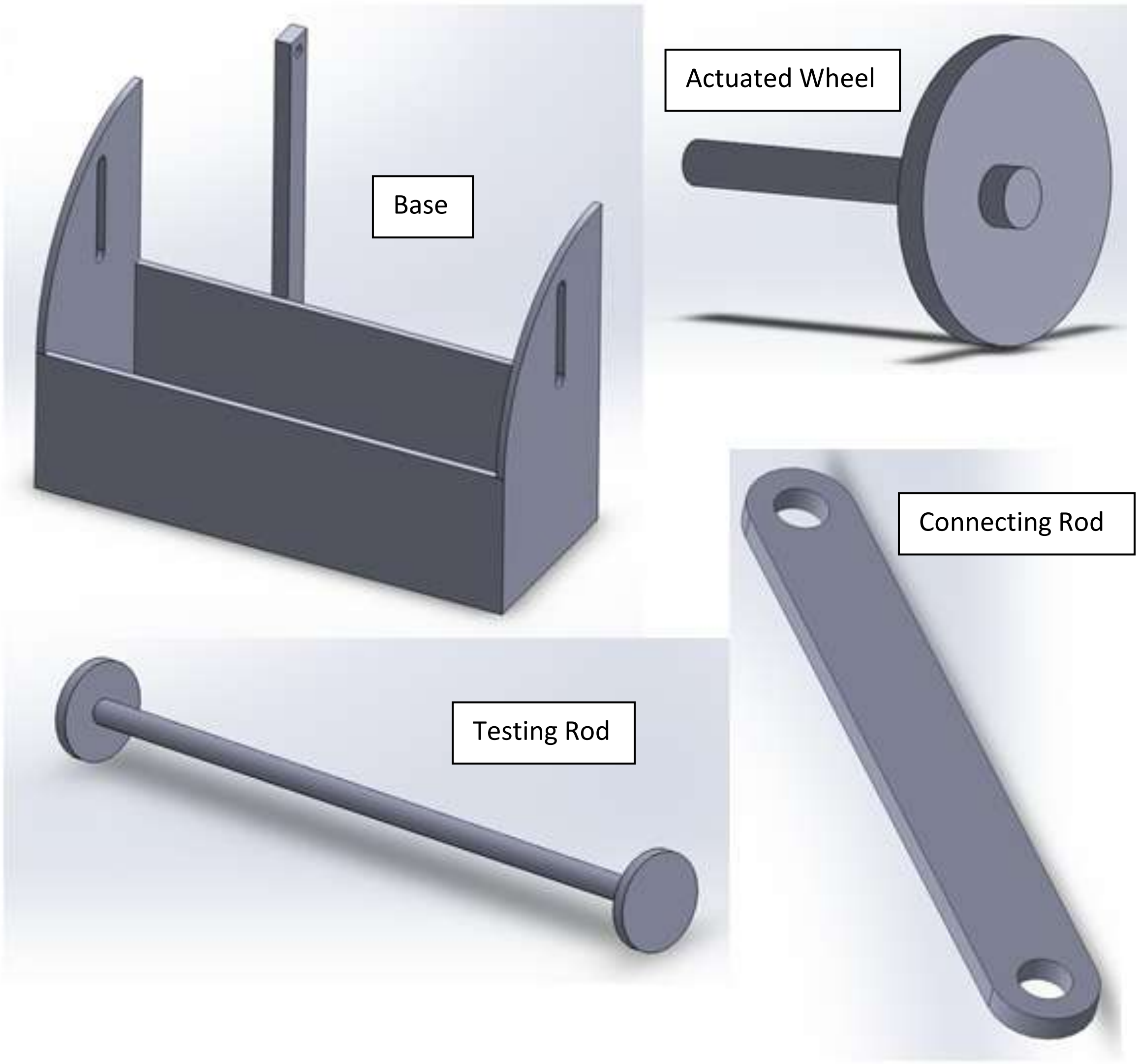
The fixture design tests a set of 10 rubber strips at once. The strips are 4.4 inches wide and have the cross sectional profile of the boot seal that is to be tested. The total volume of the 10 samples will be equal to the volume of the rubber of the full-sized boot seal. Volumetric defects propagating through the material cause the failure; thus, we expect that the statistical time to failure of the full boot seal is equal to the time of 1 of the 10 samples failing. That is, we expect that the full boot seal is 10 times as likely to fail as one of the samples. We can accelerate this failure by increasing the stress on the samples, which in turn is caused by increasing the displacement forced on them in testing. The boot seal profile will match the rendering in Figure 16. A CAD rendering of the fixture concept design is shown on the next page in Figure 17.

Figure 17: These images show the concept design for the test fixture.



In order to be tested, these test samples must be mounted at both ends. One end will be anchored at the bottom of the test fixture. This area of the test fixture is built as a simple box and serves to immerse the test samples in the lubricant that the samples will encounter in the factory. The second end of the test samples will be mounted to a single rod that applies the same displacement to all of them. This rod is located above the fill line of the lubricant, so only the test sample is immersed. A shortcoming of this design is that, in the real steel mill, only one side of the boot seal is immersed in the lubricant, while the other side is exposed to air. The rod is mounted in a slot that allows it to move in only one dimension, namely the direction of testing displacement desired. This rod is connected to another rod is connected to an actuated wheel. The drive system moves like the piston-crankshaft system of a car, but in this case it is the wheel that is driving and the rod that is driven. As the wheel is driven, it pulls the rod connected to it by moving the connection point in a circle. This pulls on the rod constrained to one dimension, moving it forward and back in the guiding slot. This motion applies the desired displacement to all the test samples while immersing them in oil. Each of the parts described is shown in Figure 18 shown on the page 16.

Figure 18: The parts for the test fixture are shown, top left, top right, bottom left, bottom right: The base of the test fixture, the actuated wheel, the testing rod, the connecting rod.



Engineering Design Parameter Analysis

When analyzing the boot seal we performed finite element analysis. Doing FEA with elastomer isn't an exact science. To validate the analyses that we did, we ran these simulations on past models of the seal and compared where the max stress was with where the failures were occurring in real life. The Figures 19-21 show both where the seals were failing and where the maximum stresses are.

Figure 19: Max Stresses and failure area for the 2th design (first of the "S" shape). The surface moved is moved left in this picture for the "In" displacement and down for the "down" direction.

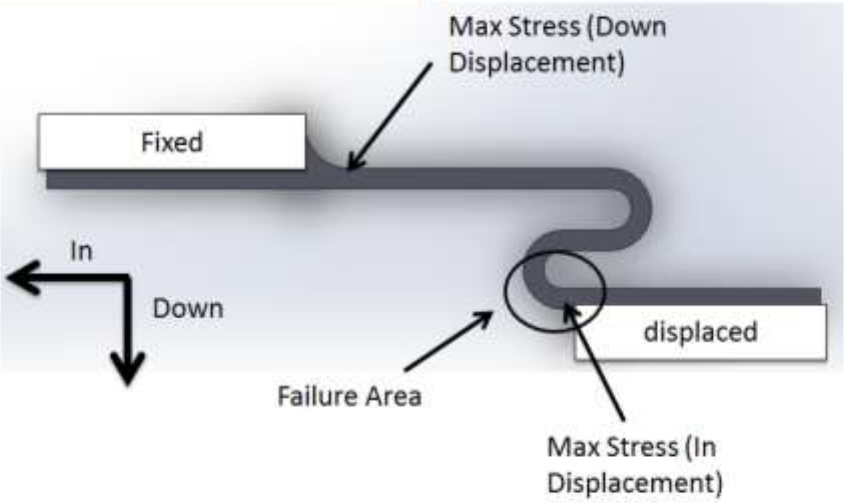


Figure 20: Max Stresses and failure area for the 3th design The surfaces moved are moved left in this picture for the "In" displacement and down for the "down" dispalcement.

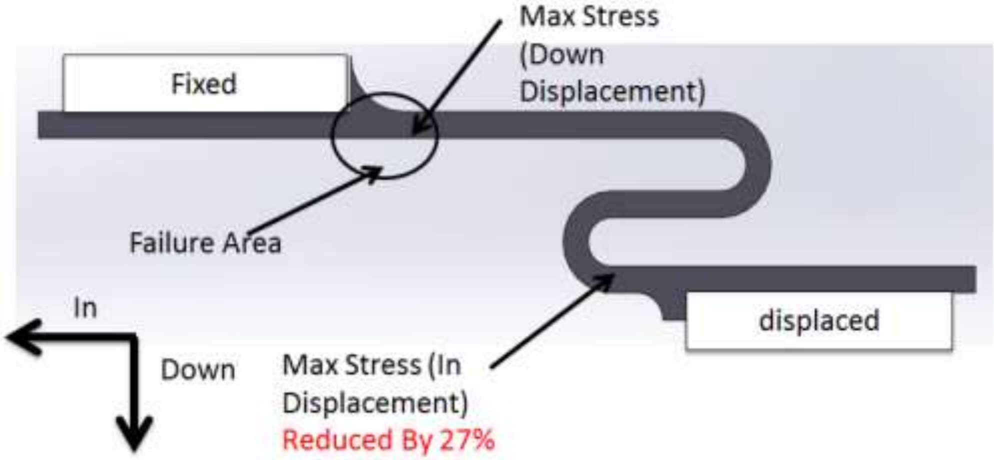
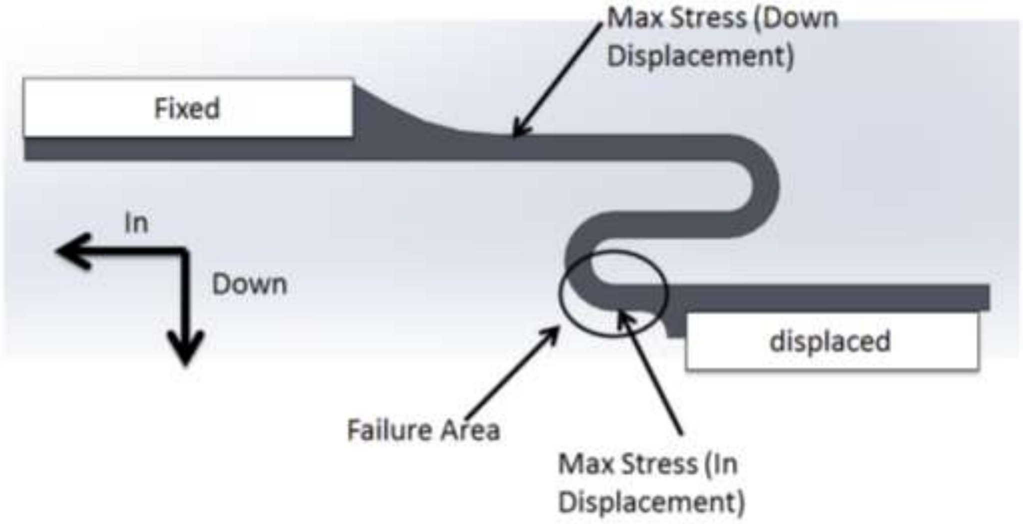


Figure 21: Max Stresses and failure area for the 4th design. The surfaces moved are moved left in this picture for the "In" displacement and down for the "down" dispalcement



As shown in the Figures 19-21 as the maximum stress are in the same areas that the seal are fail when in use. When comparing the 3rd design to the 2nd, it's noteworthy that max stresses from both the inward displacement and the downward displacement in the same area. This validates our concern that both axial and radial displacements are causing the failures, and that the location of the max stress moves with each redesign.

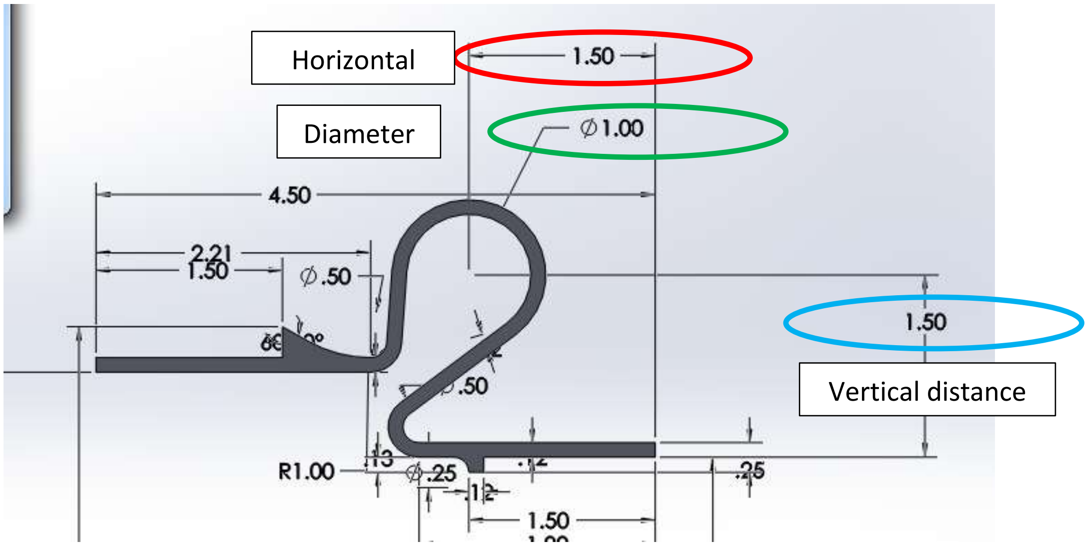
To allow us to compare our boot seal designs, we wanted to find the maximum stress and stains on the boot seal when it undergoes various displacements. We did this by performing finite element analysis. We used SolidWorks instead of Hypermesh for the FEA because SolidWorks is more user friendly and is used in the industry. We loaded models of the seal designs in multiple ways that simulated the displacements that it has to withstand. A chart showing how our design concept compare the current design in use is provided below in Figure 22.

Figure 22: Maximum Von Misses Stress in FEA model

Design	In 0.25"(Pa)	Out 0.25"(Pa)	Up 0.0785"(Pa)	Down 0.0785"(Pa)	Out 20.75"(Pa)
Current	473,568	443,736	73,498	73,153	1,285,595
Big curve	118,488	125,260	49,572	49,572	394,745

To determine the ideal shape of the boot seal, we ran a design optimization study in SolidWorks varying three parameters to find the lowest von misses stresses when the was input 0.25" in and 0.0785 down; the direction for these to inputs are shown below. The parameters varied are as follows and are shown in the figure below, horizontal distance: .75"-1.5" with step size 0.2", vertical distance: 0.75-1.5" with step size 0.2" and the diameter: 0.4"-1" with step size 0.15". The step sizes where chosen to keep the number of simulations and the time required to run them at a reasonable level.

Figure 23: This figure shows the three parameters of the design that were varied



This optimization record the maximum von misses stress from the possible displacements. Each possible scenario was then scored using equation 1 where σ_{in} is the max stress in when pushed in 0.25”, $\sigma_{Avg.in}$ is the average of the σ_{in} for all the scenarios, σ_{down} is the max stress in when pushed in 0.25”, Wgt_{in} is weight of σ_{in} in the score, $\sigma_{Avg.down}$ is the average of the σ_{down} for all the scenarios, and Wgt_{down} is weight of σ_{down} in the score. In this case 50% was used for each of the weights. The scenario with the lowest score was then selected.

$$\frac{\sigma_{in}*\sigma_{Avg.in}}{\sigma_{Avg.in}}*(Wgt_{in})+\frac{\sigma_{down}*\sigma_{Avg.down}}{\sigma_{Avg.down}}*(Wgt_{down})=Score \qquad \text{(Eq. 1)}$$

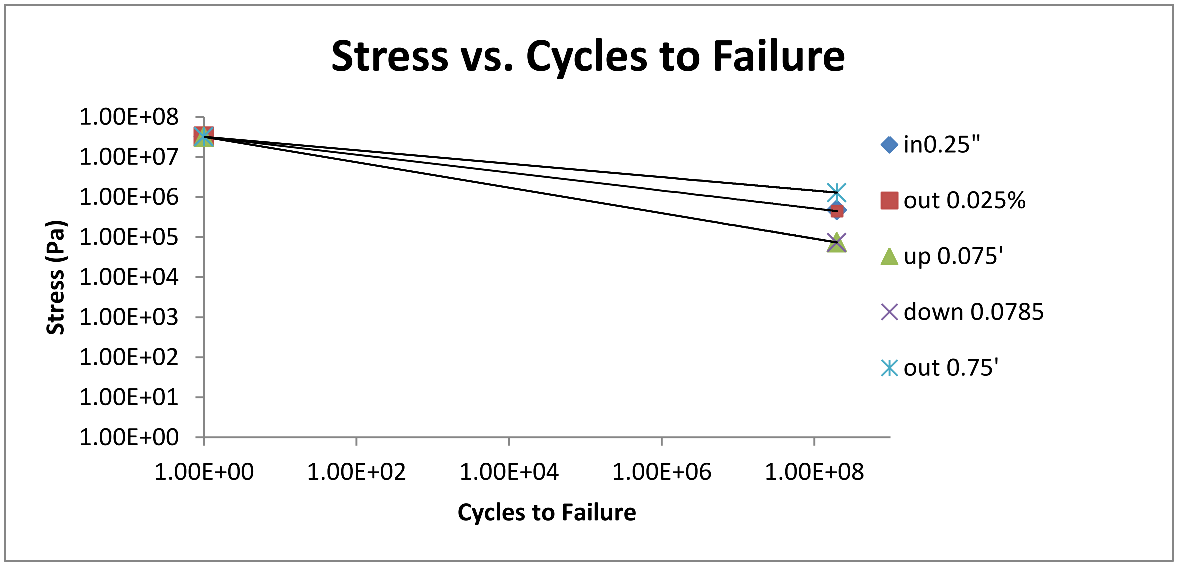
Another parameter that was looked at was the thickness of the rubber portion of the seal. This done varying the thickness of the design and comparing the simulation results.

Since we were unable to get any fatigue data on the material that is used for the seal, a very crude method would be to use the limited data that we have and do a power fit. The data that we have is the ultimate tensile strength of the material which would give us the stress to break in one cycle and the average lifetime of the current seal and the stress from the FEA. Since there are different maximum stresses for each of the different displacements it could undergo, a safe method would be to compare the lifetimes if each possible displacement was sole root cause of the failure.

Figrure24: Shows that data behind the fatigue chart

Cycles to failure	In 0.25”	Out 0.25”	Up 0.0785”	Down 0.0785”	Out 0.75”
1.0	3.2*10^7	3.2*10^7	3.2*10^7	3.2*10^7	3.2*10^7
199 million	473,568 Pa	443,736 Pa	73,498Pa	73,153 Pa	1,285,595Pa
Exponential fit	$\sigma = 3.2*10^7 N^{-.224}$	$\sigma = 3.2*10^7 N^{-.224}$	$\sigma = 3.2*10^7 N^{-.32}$	$\sigma = 3.2*10^7\ N^{-.32}$	$\sigma = 3.2*10^7\ *N^{-.168}$

Figure 25: Stress vs Cycles to Failure graph depending on which displacement causes the failures.



To analyze the material we would like to perform tensile tests on a series of strips of rubber that have been soaked in each of the fluids and compare them to tests done on samples left untouched. Doing this has a lot of barriers. The first is time. In order to do this we would have to order samples of the rubber and the two fluids, wait for them to arrive, soak the samples, and finally test the samples on a tensile testing machine. The less time we spend soaking and the samples the less conclusive our results will be. This test would also require an apparatus to load the samples and the use of a tensile testing machine. Additionally, the test involves some possibly dangerous chemicals. Because of this we have discussed this test being done by ArcelorMittal at their I/N Tek facility.

One difficulty that we face is creating the test fixture so that it will be able to model the displacements to the accuracy that we want. Another issue we face making sure that the text fixture is safe while running at higher speeds. Finally, ArcelorMittal has told us that they will not be able to fabricate our final oil boot seal design in time for the Design Expo.

Designsafe was used to access the risks and hazards of the completed prototype. The majority of the danger from our prototype comes from either the moving parts or the liquid the samples are to be tested in. These things cannot be eliminated easily. Therefore, steps such as creating a cage around the moving parts and having a lid in the tube that contains the liquid have be implemented to protect the user from danger. By adding these safety features we were able to reduce the risk down to “Negligible”. The design safe results can be found in appendix K.

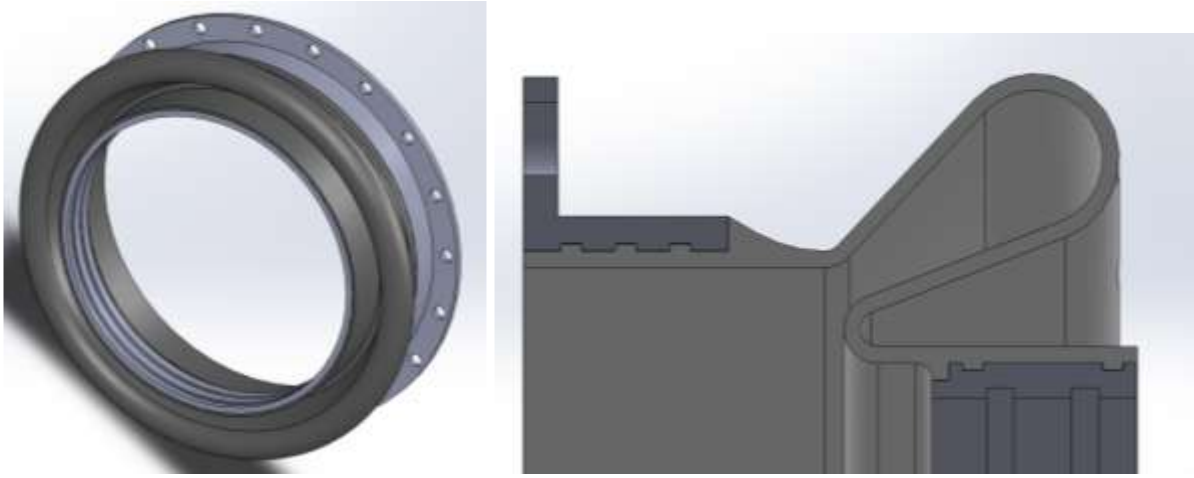
The SimaPro analysis indicates that the manufacturing of 1kg of steel and aluminum have minimal impact on the environment. The largest effect of production of these materials is the mineral utilization necessary for the production of steel and aluminum. More information can be found in appendix C.

Final Design Description

Boot Seal

After executing the analysis described in the parameter analysis section, the optimal design for the boot seal is shown in the figure below. Although lowering the thickness of the rubber portion lowered the maximum strain, we do not know enough about this material to know how its thickness affects its lifetime. This is something that could be further investigated with our test fixture.

Figure 26: Final Boot Seal design Isometric view cross section view



The charts below shows compares the seal currently in use, our alpha design and our final design.

Chart 2: Shows how the final design compares to the alpha design and the current design.

Design	In 0.25”(Pa)	Out 0.25”(Pa)	Up 0.0785”(Pa)	Down 0.0785”(Pa)	Out 0.75”(Pa)
Current	473,568	443,736	73,498	73,153	1,285,595
Alpha	118,488	125,260	49,572	49,572	394,745
Final	116,690	100,198	31,011	32,080	287,202

Using the data from the FEA and real the seal currently in use, the lifetime assuming each case is the only source of failure is shown in the chart below.

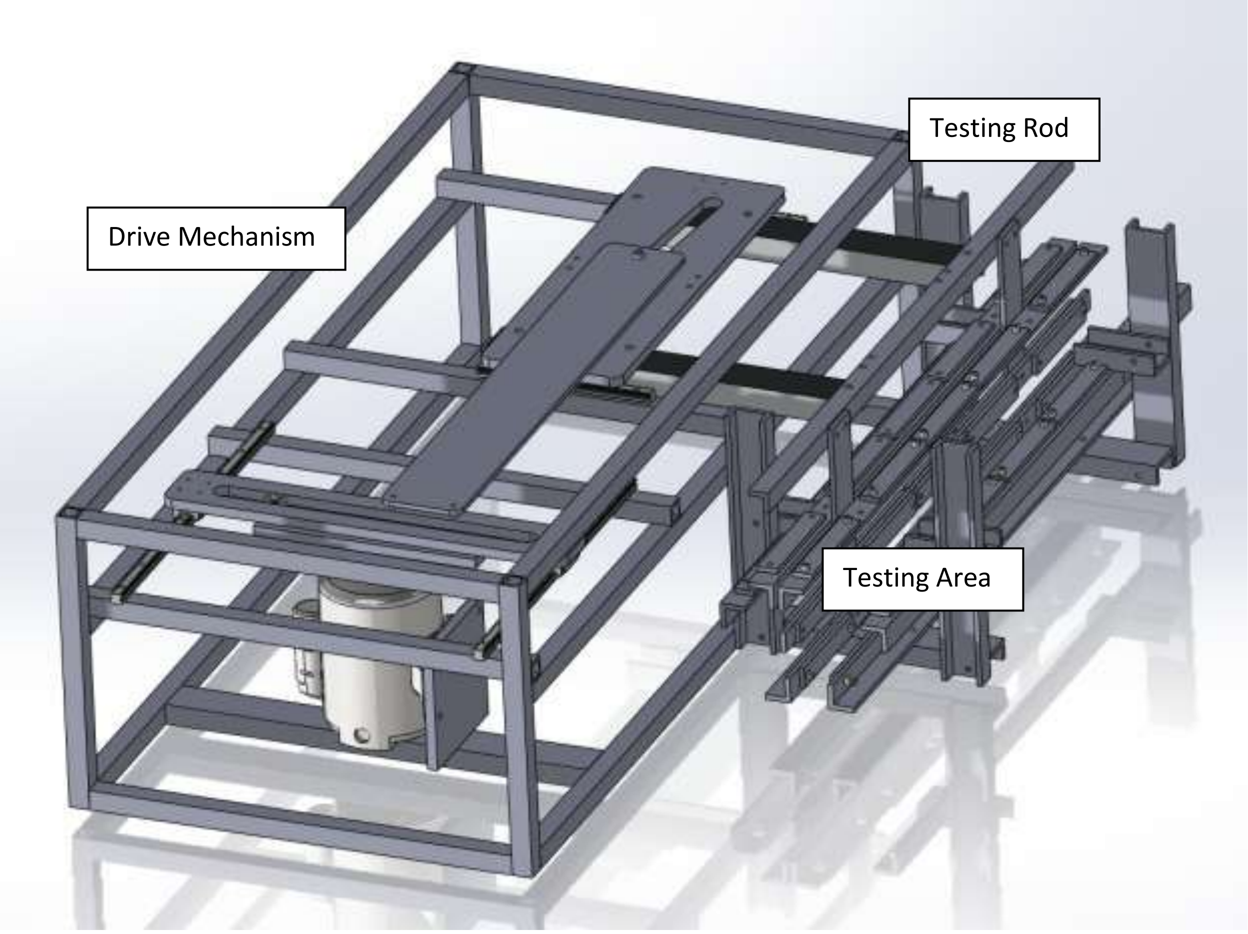
Chart 3: Show the potential lifetime for each type of cyclical loading.

Displacement	In 0.25”	Out 0.25”	Up 0.0785”	Down 0.0785”	Out 0.75”
Lifetime (millions of cycles)	76,639	151,311	2,998	2,695	1,528,000

Here in the worst case this seal stills has an 8.4 safety factor. It is important to keep in mind that these are very rudimentary numbers. However, this good of perforce is a good indicator that this seal will meet the requirements and these could be realistic considering the dramatic drop in the stresses the boot seal undergoes.

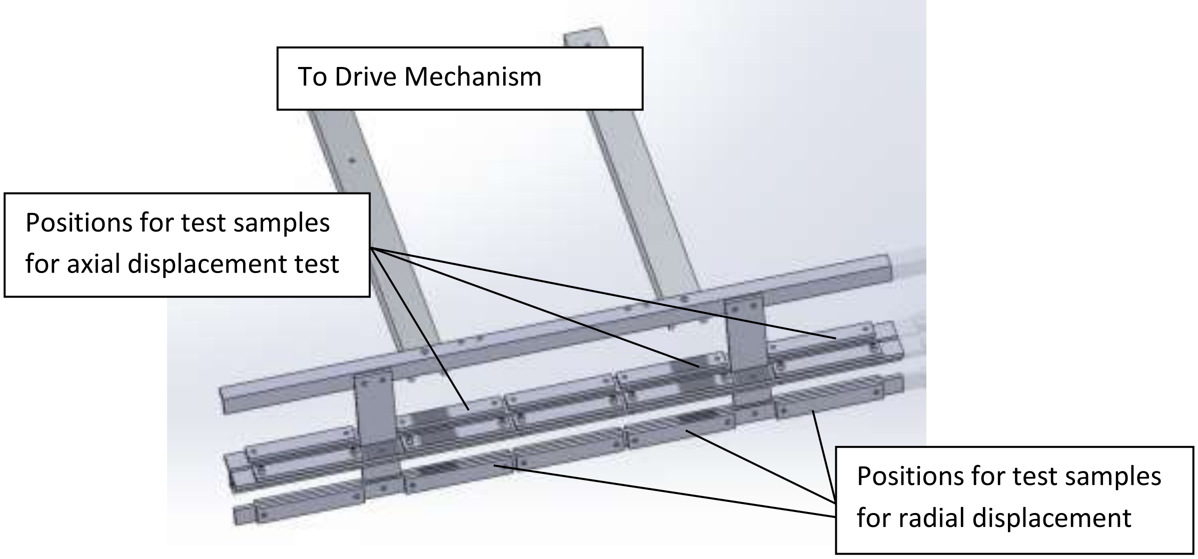
Test Fixture

Figure 27: The overall fixture design with the Drive Mechanism, Testing Rod, and Testing Area labeled



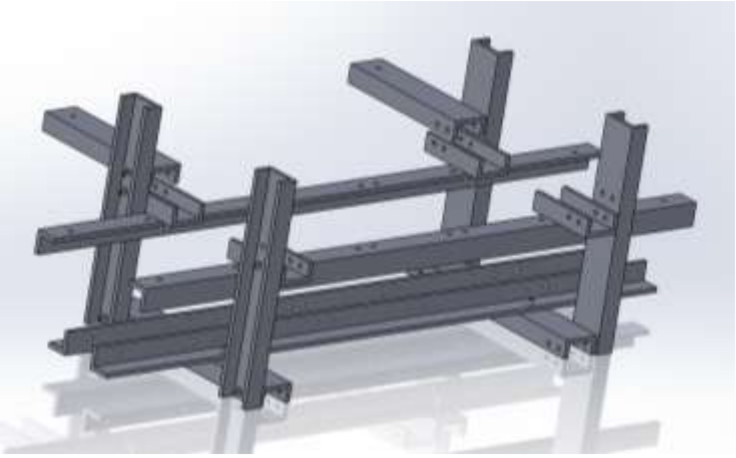
The fixture design tests a set of 10 rubber strips at once. The strips are 4.4 inches wide and have the cross sectional profile of the boot seal that is to be tested. The total volume of the 10 samples will be equal to the volume of the rubber of the full-sized boot seal. In order to be tested, these test samples must be mounted at both ends. One end of the test samples will be mounted to a single rod that applies the same displacement to all of them. A rendering of this testing rod is shown in Figure 27.

Figure 28: The testing rod with 10 positions for samples indicated



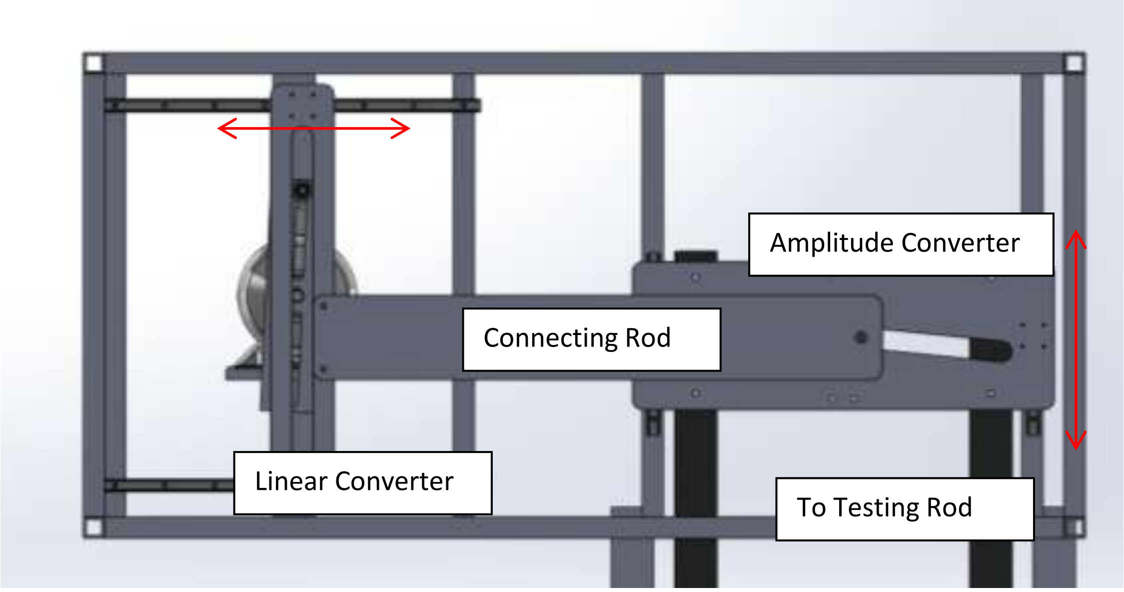
One end of the test samples will be placed in the available positions on the testing rod. Then, on top of each sample, a clamping piece will be placed and bolted to the rod, securing the test sample between them. All 10 clamping pieces are individually attached so that each test sample can be attached or removed independently of the others. Note that this same clamping method is used at the other end of the test sample to be discussed in the testing area. The testing rod is connected to the driving mechanism and is structured to apply the necessary loading to all the test samples without bending. The rod is built from steel. It moves in the single degree of freedom given it by the driving mechanism. It has 20 surfaces where test samples can be mounted: 10 for radial-displacement testing, and 10 for axial displacement testing. These two surfaces are perpendicular to each other, so that the single degree of freedom motion can be used for either testing condition. The second end of the test samples will be anchored on a surface of the testing area. Samples in the lubricant that the samples will encounter in the steel mill. The test area will be in tub. A rendering of the testing area (excluding tub) is shown in Figure 29.

Figure 29: The testing area



There will be two available types of mounting positions in the testing area to match those on the testing rod: one will be in line with the motion of the testing rod, and the other will be mounted perpendicularly to this motion and below the testing rod, allowing for test samples to be tested in both testing conditions. It should be noted that the fixture is designed to test only one set of 10 samples at a time, although there are 20 positions available. It may be possible to use a stronger actuator so that 20 samples can be tested simultaneously, with 10 in each loading condition. However, bi-modal testing is not possible. The testing area is built from bent sheet steel on the sides and bottom, and features a plexiglass splash guard on the top. Mounted to the back side of the testing area is the drive mechanism, shown in Figure 30.

Figure 30: The drive mechanism: The red arrows describe the directions of motion.



The drive mechanism has several components, each made of steel. The motor rotates a slotted wheel. The slot in the wheel allows for positioning of a rod that serves to make the wheel an eccentric. Before beginning the experiment, the user selects the radius from the wheel's center to position the shaft and then tightens it into place; the radial position is proportional to the displacement applied to the test samples during the experiment. At the other end of the shaft is a ball bearing, which allows the rod to slide back and forth in the slotted piece above the wheel, called the linear converter, while reducing friction and wear. As the wheel revolves, the bearing shaft is free to slide up and down in the slot of the linear converter, but it is not free to slide left and right. This causes the linear converter to slide back and forth along its single degree of freedom defined by the carriage rails it is mounted to. The interaction between the eccentric wheel and the linear converter serves to convert the rotational motion of the wheel to cyclic linear motion. A connecting rod is attached rigidly to the linear converter. On the rod's other end is another bearing rod identical to the one used between the wheel and the linear converter. This second bearing rod is allowed to slide in the angled slot of the piece it is connected to, called the amplitude converter. The amplitude converter is attached to a carriage rail, like the linear converter, which allows only one degree of freedom: perpendicular to the movement of the linear converter. The interaction between the connecting rod and the amplitude converter changes the direction of motion by 90 degrees, but that is not the primary function of the amplitude converter. The angled slot in the amplitude converter is angled such that it allows 10 times as much motion along the linear converter's line of motion than along the amplitude converter's line of motion: the result is that the amplitude of the motion output by the amplitude converter is one tenth that of the motion input. The purpose of this is to convert the coarse adjustment of the slotted wheel into finely adjusted small displacements to be applied to the test samples. This allows for greater precision. It also solves for another problem: without the amplitude converter, the slotted wheel would need to be too small. Its bearing rod would need to be mounted 0.1 in from the axis of rotation, leaving no room for the drive shaft of the motor to connect. Smaller parts would also be less able to conduct the necessary power through the drive mechanism. Finally, the testing rod is rigidly connected to the amplitude converter. The drive mechanism is housed in a sheet steel box, like that used in the testing area, to protect users from moving parts.

Scaling

We are considering scaling down the test fixture to test scale-size samples. The design described in the previous subsection would be 70 inches long if it were to be manufactured full-scale. Scaling down the system could allow for easier assembly and use while still delivering the necessary experimental data. The test fixture can be made smaller; the issue in question is whether or not we can scale down the rubber samples and if this scaling will affect the accuracy of the test results. To learn about the effects of scaling on elastomers, we consulted several published articles [4 - 6]. Each of these articles confirmed that scaled testing of elastomers is valid, provided that the strain is 50% or lower, which matches the conditions in our test fixture. The other issue to be addressed is the feasibility of scaling down the rubber samples. A full-scale sample is only 0.12 inches thick. A 1:10 scale, for example, would drop this thickness down to 0.012 inches, which casts doubt on the feasibility of manufacturing these test samples. On Friday, November 2nd, we called our sponsor to investigate this feasibility, but the meeting must be rescheduled for November 5th or November 6th. If scaling is feasible, even to a degree lesser than 1:10, we plan to pursue scaling down our test fixture design.

Prototype Description

Our prototype for our fixture is our focus; in our case, it is identical to the final design because it is a one-off test fixture for laboratory experiment. Its performance will be its own validation, and we plan to provide it to the sponsor upon the project's completion. Due to the constraints of the manufacturing processes involved, the boot seal will not be fabricated by design expo. We investigated the possibility of an elastomeric rapid-prototype of the boot seal, but did not have time to pursue this.

Fabrication Plan

Materials needed to fabricate the test fixture include 1/8" thick steel sheets, plexiglass, linear converter shafts, a motor, a thermometer, ball bearings, bolts, washers, nuts and sealant. More details can be found in the bill of materials in Appendix G. The motor support that will sit under the motor will be shaped on a mill. The Eccentric wheel attached to motor will be shaped on a lathe and bored on a mill. The wheel slot that connects the eccentric wheel to connecting rod; the connecting rod; the test sample clamps; the testing rod that pulls on the test samples; and the amplitude converter will all be cut to shape using water jets and drilled and/or bored on a mill. The test fixture enclosure as well as the drive mechanism enclosure will be built from bent sheet steel that is cut using a band saw. A hole will be drilled in the test fixture enclosure using a drill press, on a vertical side near the bottom, to allow for easy draining of the oil. The method of plugging of this drain hole has yet to be determined. Once all pieces of the test fixture have been built, they will be assembled using bolts, washers and nuts to hold them together. A sealant will be applied where the plexiglass meets the steel enclosure to keep oil from leaking out of the test fixture during operation. The best sealant for this application has yet to be determined. Assembly should not be difficult, because we plan on using bolts, washers and nuts to hold together the test fixture.

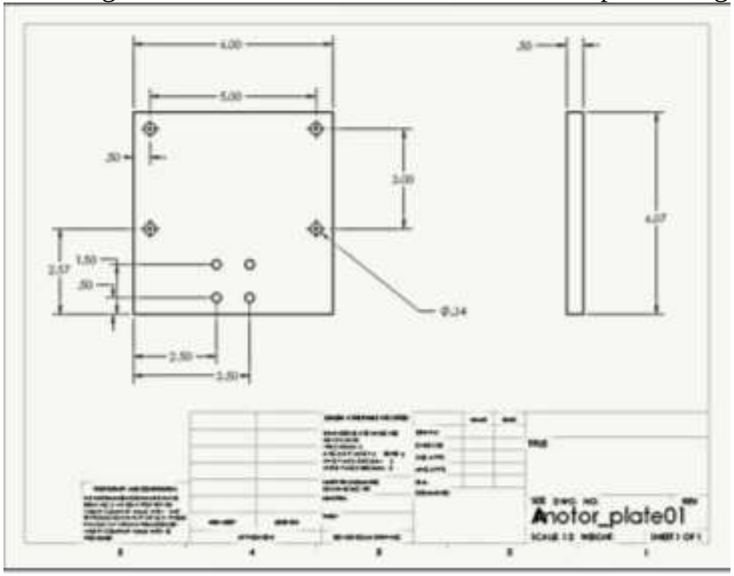
Several parts of the test fixture require small tolerances and are therefore key surfaces in building a successful test fixture. These parts are the parts that translate the motors angular rotation to vertical displacements on the test samples and include the eccentric wheel connected to the motor, the wheel slot that connects the eccentric wheel to the connecting rod, the connecting rod, the amplitude converter and the testing rod that pulls on that test samples. Because the displacements applied to the test samples are small, we plan to use a mill to create accurate parts and limit the variation in the needed dimensions. Tolerances for the test fixture and drive mechanism enclosure as well as the test sample clamps are less important because these parts do not translate motion.

Cutting speeds were found in the Turning/Boring/Drilling Guide found in the Wilson Center machine shop. Any drill bit speed that was not listed on the chart could be determined using the following equation

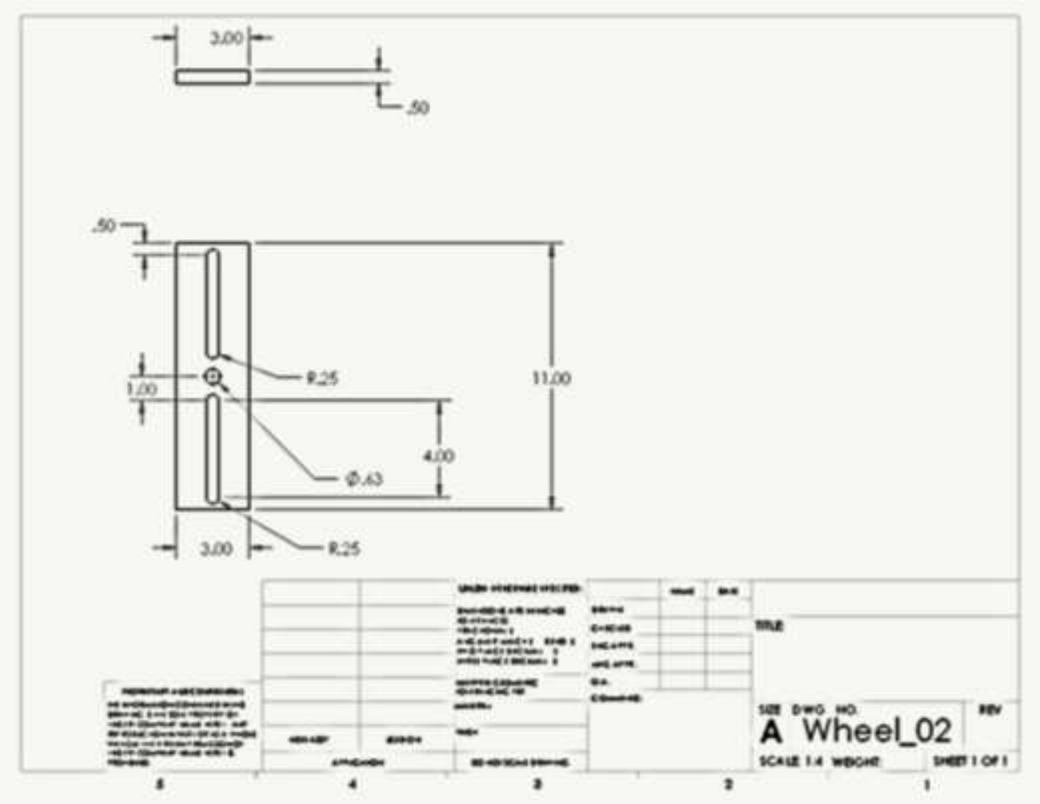
$$\text{RPM} = \frac{12 * V}{\pi * d}$$

Where v is the cutting speed if ft/min, d is the diameter of the drill bit and S is the speed in RPM. Lubrication is required for every turning, boring and drilling operation and will be used in generous amounts. All aluminum stock is sourced from McMaster.

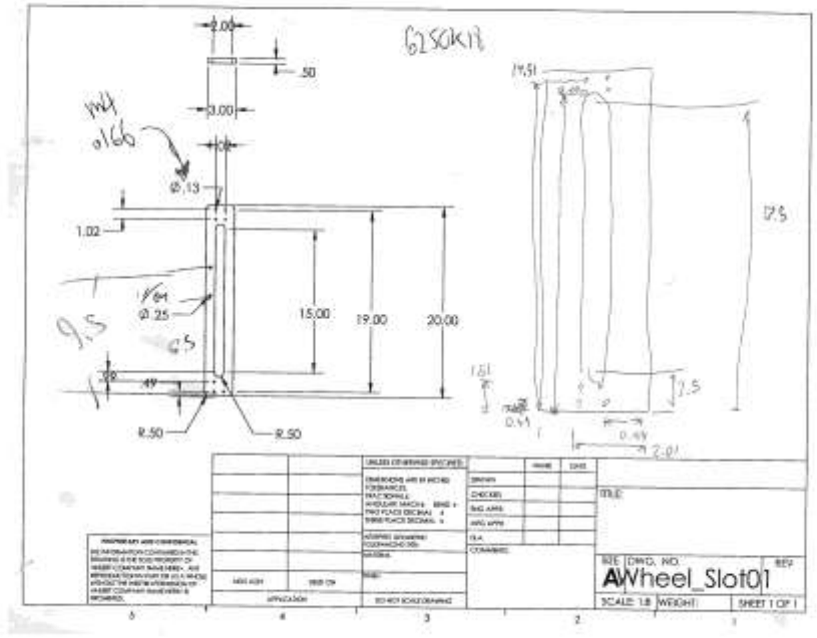
1) Motor Mounting Plate will be cut from the 1/2”x12”x36 aluminum stock on a band saw at a speed of 300 FPM. It will then be milled to its dimensions on a mill using a 1 inch drill bit at 150 RPM. The mounting holes for the motor will be drilled in the plate using an R, size drill bit at 1500 RPM.



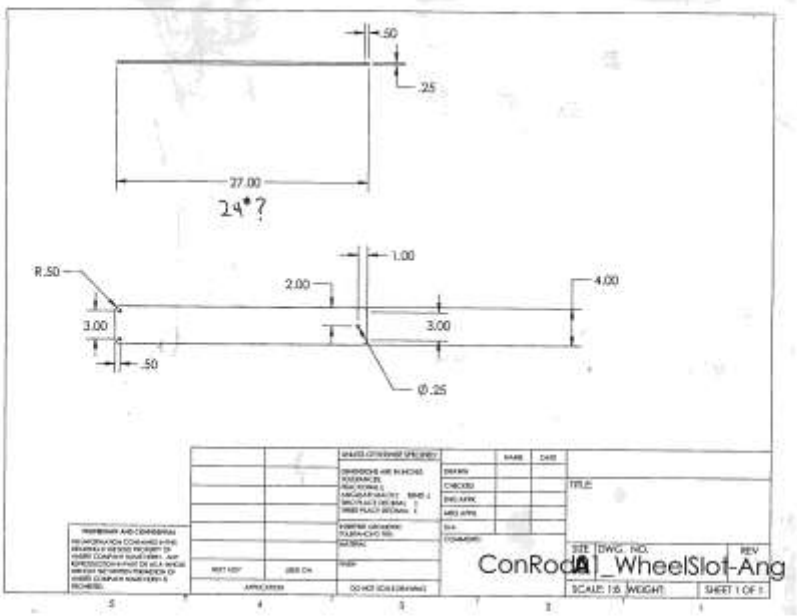
2) The drive wheel will be cut from the 1/2”x12”x36 aluminum stock on a band saw at a speed of 300 FPM. The slots will be created using an endmill size of ¼” at 1200 RPM. The center hole will be drilled on the lathe with a 5/8 drill bit at 600 RPM.



3) The scotch yoke will be cut from the 1/2”x12”x36 aluminum stock on a band saw at a speed of 300FPM. It will then be and bored on a mill using a 1/2 end mill at 1200 RPM to create its inner slot. The end wholes for the scotch yoke for the guide rail tracks to attach as well as the connecting rod will be placed on the drilled on the mill using a #80 drill bit at 1500 RPM.

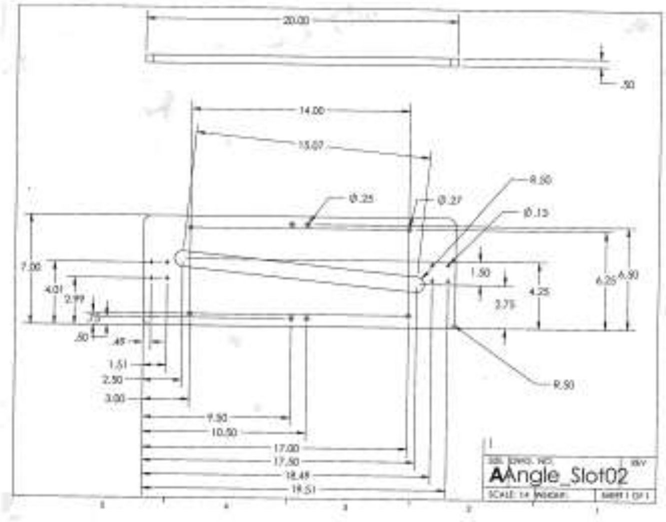


4) The connecting rod will be cut from the 1/4”x4”x36” aluminum stock on the band saw at a speed of 300 FPM. It will then be shaped on a mill using a 5/8 drill bit at 1200 RPM. It will have its holes where it meets the bearing and the scotch yoke drilled on the mill. The holes connecting to the scotch yoke will be drilled using a 17/64 drill bit at 1500 RPM. The bearing hole will be drilled using a E drill bit at 1500 RPM.



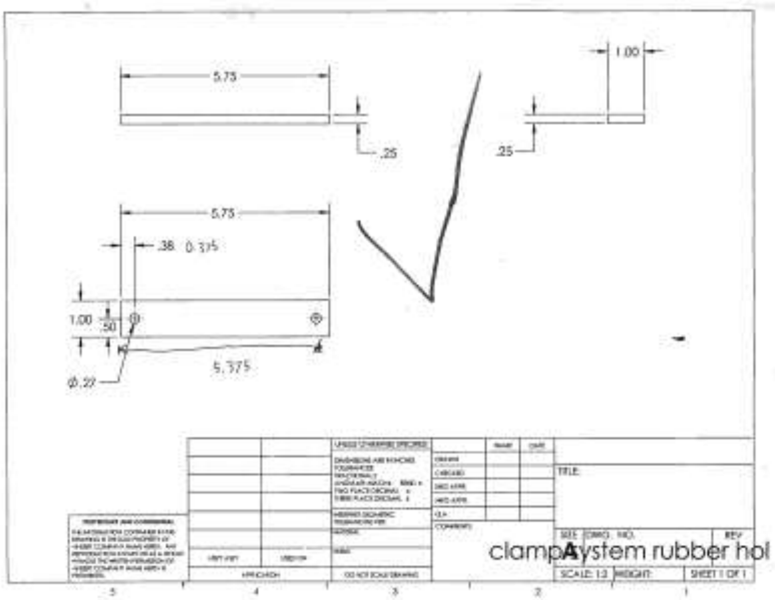
5) Amplitude Converter

The amplitude converter will be cut to shape from the 1/2”x12”x36” aluminum sheet on the band saw at a speed of 300 FPM. The piece will then be bored using a CNC mill as well as the wholes where the testing rod assembly connects. . A CNC mill is capable of holding very tight tolerances, which are needed for the amplitude converter because of its task of creating accurate displacements. The specifics of how this piece will be milled are not yet finalized, and as a result this piece will be manufactured last.



6) Testing Rod Clamps

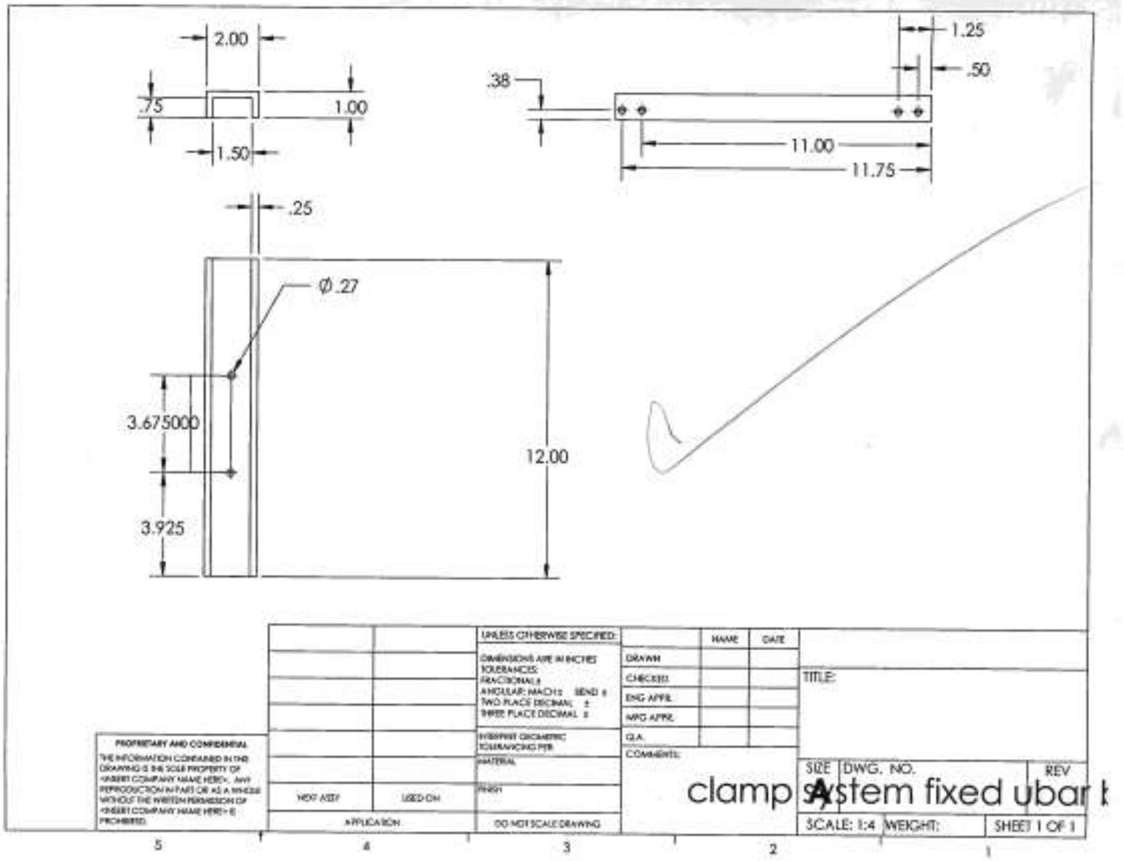
The testing rod top and bottom clamps will be cut to shape from the 1/2”x1”x36” aluminum sheet using a band saw at a speed of 300 FPM and drilled on a mill using a 17/64 drill bit at 1500 RPM. The edges on the parts will be further cut to shape using a mill to ensure that tolerances are kept to a minimum and parts are close to their specified size. 20 of these are needed.



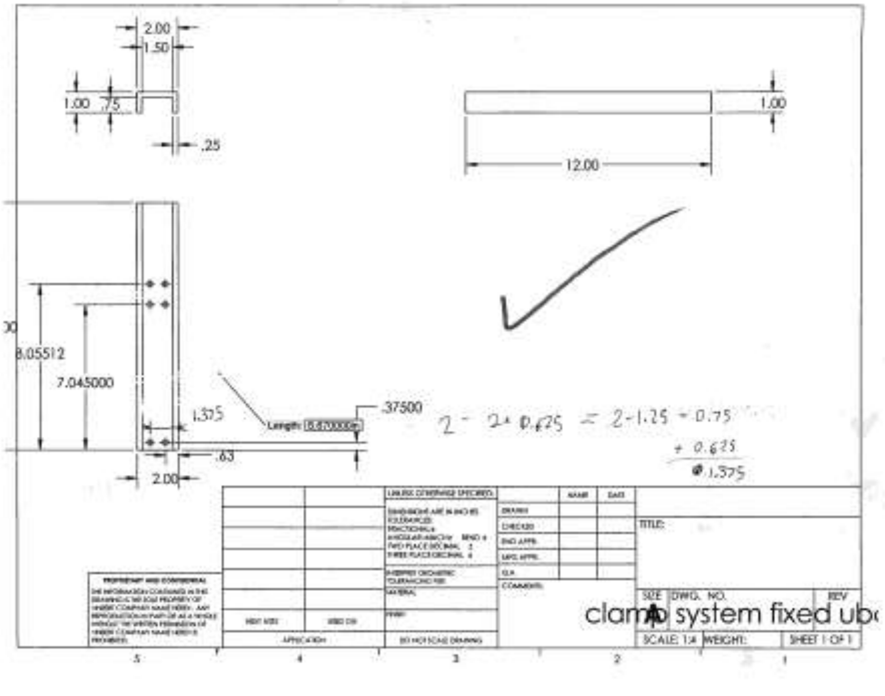
7) The Aluminum Extrusions will be cut to length using a band saw at 300FPM. The lengths of the 1”x1” extrusions that we need are as follows:4x45.5” 4x23”, 4x21”, and 4x15”.

8) The Aluminum U-bars will be cut to length using a band saw at 300 FPM and have the holes drilled using a 17/64” drill bit on a mill at 1500 RPM. Two of the base U-bars, four of the vertical U-bars, four 4” U-bars, two base-cage Ubars and two long U-bars are needed.

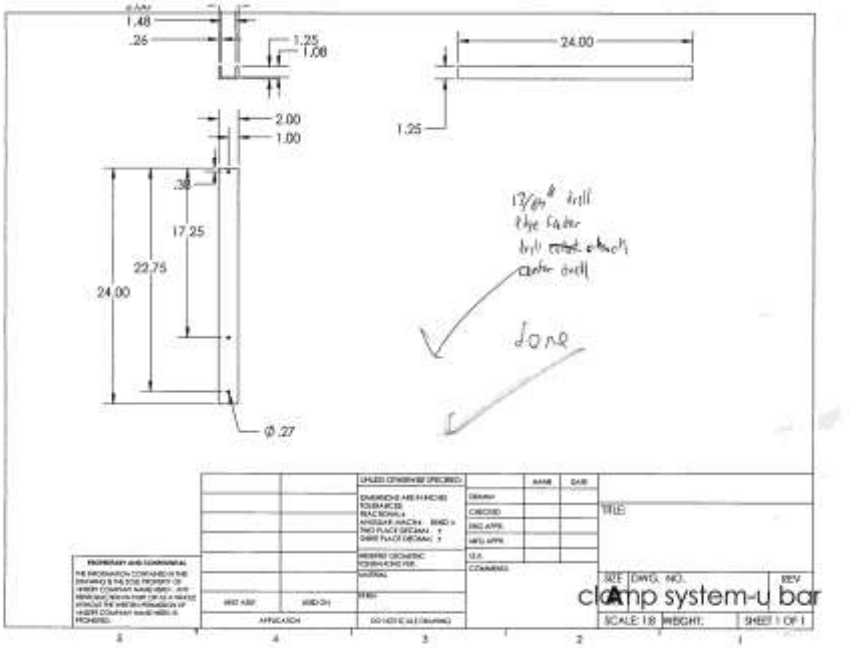
Base U-bar



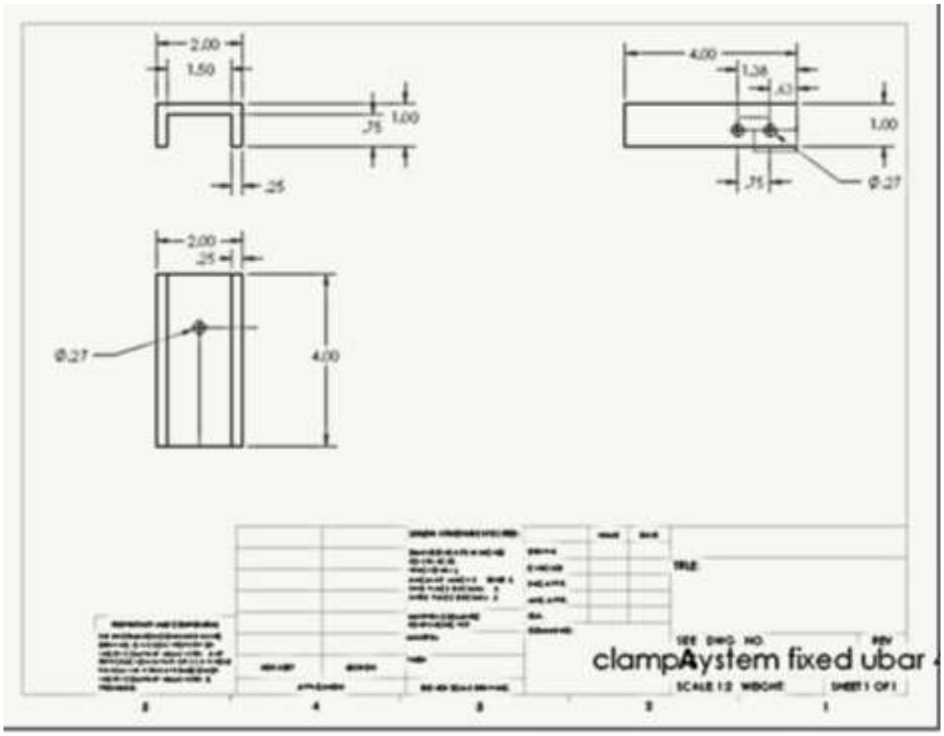
Vertical U-bar



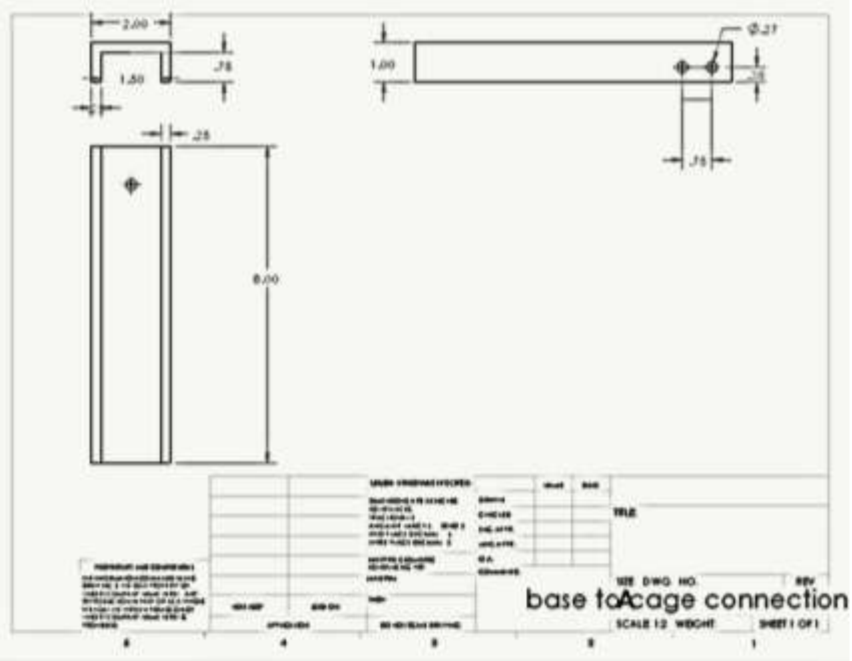
Long U-bar



4” U-bar

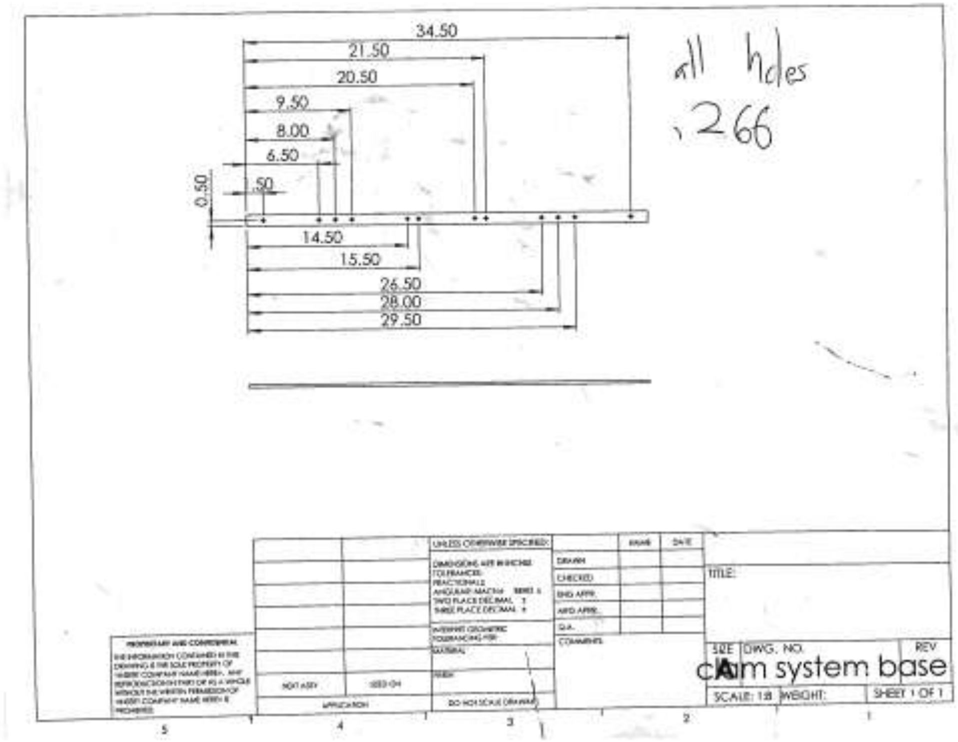


Base-Cage U-bar



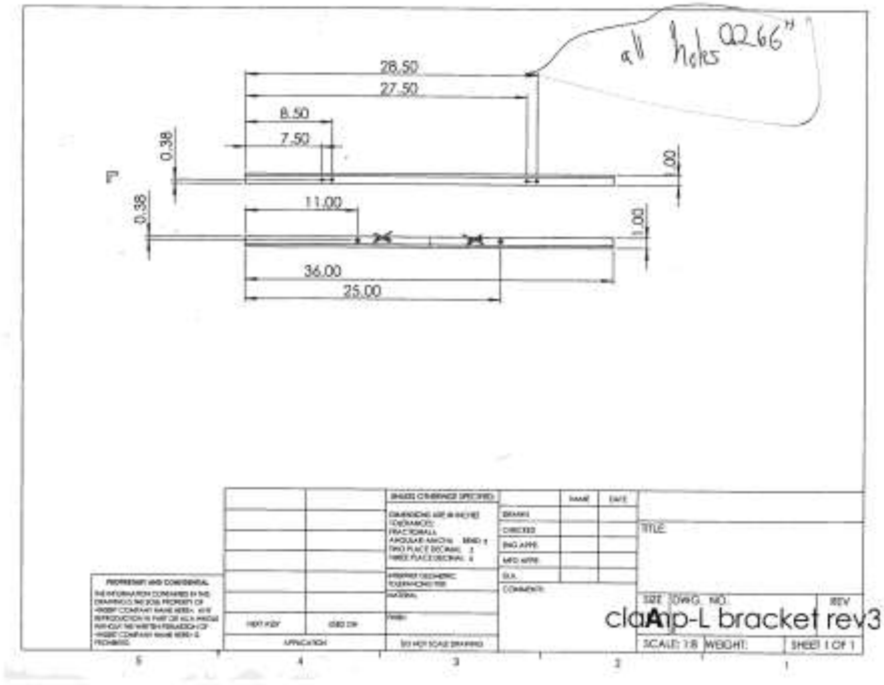
9) The Plexiglas panels will be cut using a band saw at 100 FPM. The holes will be drilled using a dulled 9/32" dill bit in a hand drill. The drill bit must be dulled so that the Plexiglas doesn't crack.

10) The clamp base came in the correct length. Therefore only holes need to be drilled in it. It will be drilled with a 17/64" drill at 1200 RPM.

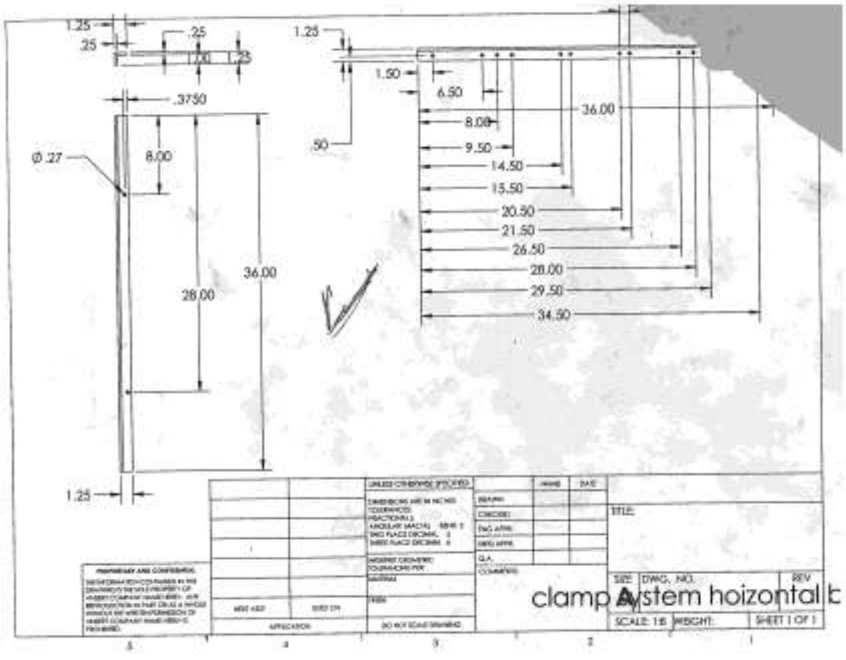


11) the L-bracket and L bases will need to be cut to length using a band saw at 300 FPM and have holes drilled in them with a 17/64 drill bit at 1200 RPM.

L-mount



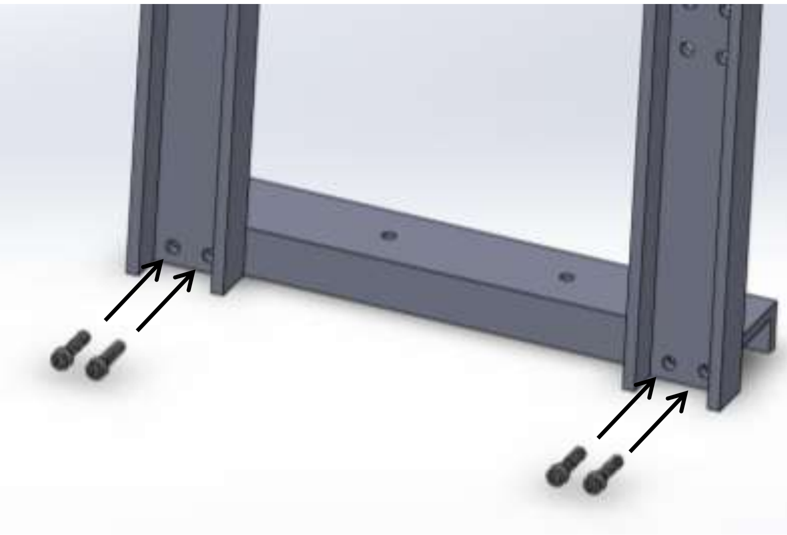
L-base



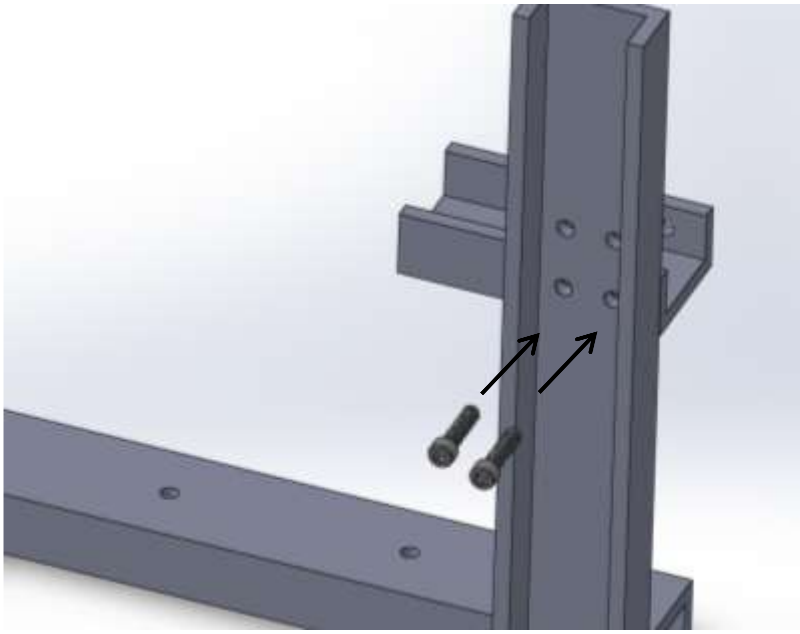
The Assembly process for our test fixture involves creating three sub-assemblies, the arm, base and cage, then assembling them together.

Base Assembly

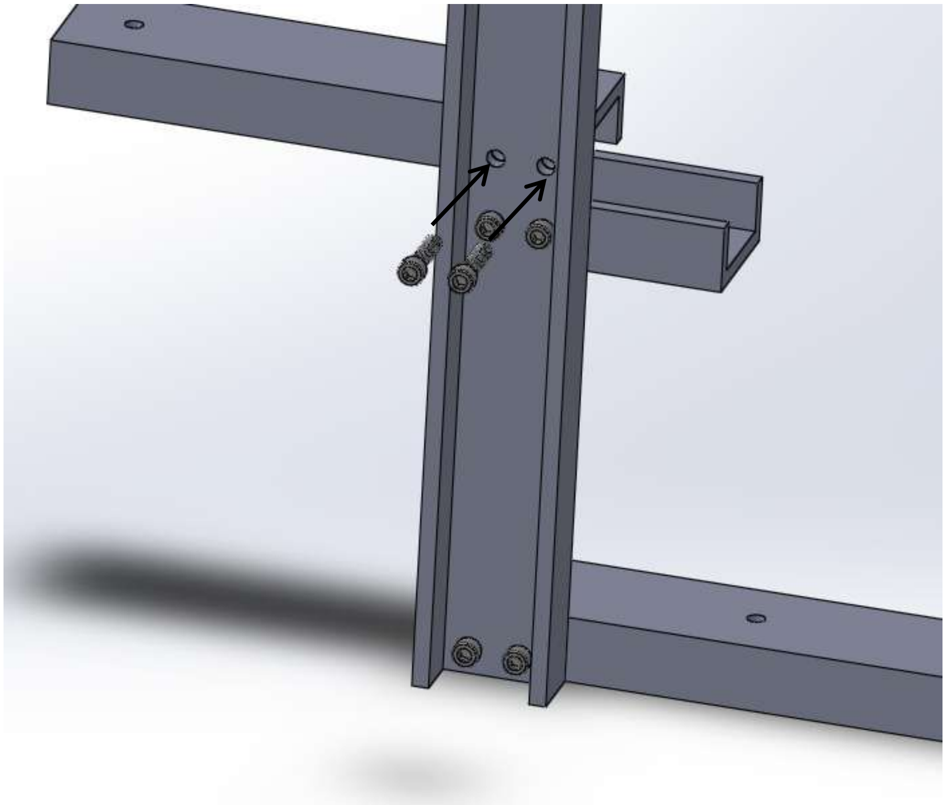
1. Line up the holes in vertical u-bar and the horizontal u-bar and fasten with 1/4”-20 bolts and nuts as shown. This is done twice. The bolts must be inserted from the direction shown.



2. Line up the holes from the short u-bars and the vertical u-bars. The end of the short u-bar without holes should face inward. They are fastened with 1/4”-20 bolts and nuts as shown. This is done twice. The bolts must be inserted from the direction shown.

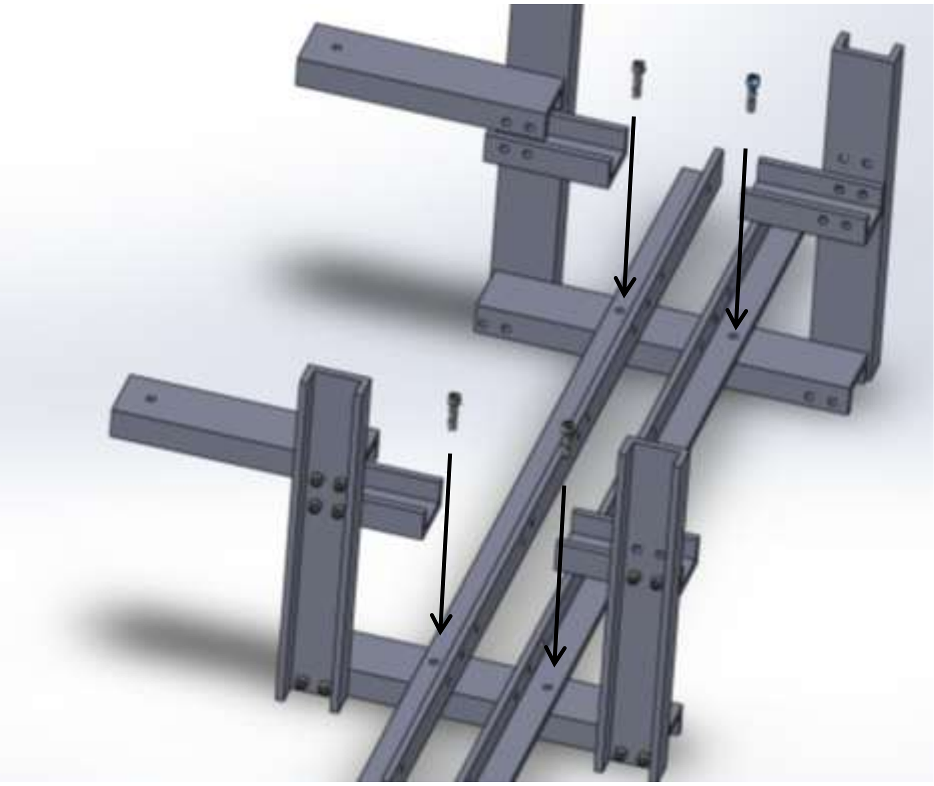


1. Line up the holes from the short cage-base connector and the vertical u-bars. The u-bar without holes should face point outward. They are fastened with 1/4”-20 bolts and nuts as shown. The bolts must be inserted from the direction shown.

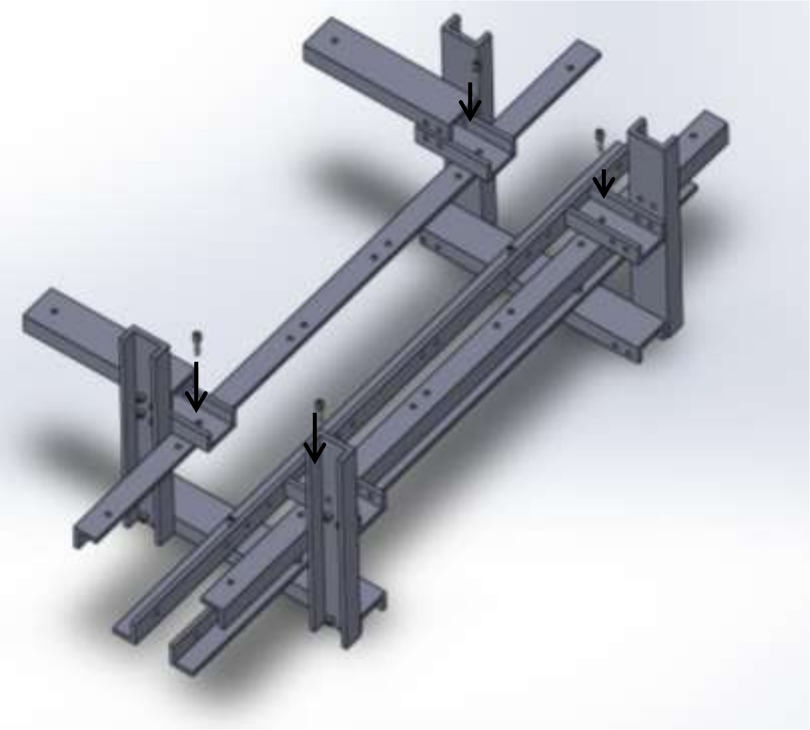


2. Repeat steps 1-3 but do a mirror image

3. Line up the holes from horizontal u-bars and the lower bases. The vertical part of the base should be on the inside. They are fastened with 1/4”-20 bolts and nuts as shown. This is done twice.



4. Line up the holes from short u-bars and the and the upper bases. The vertical part of the base should be on the outside. They are fastened with 1/4”-20 bolts and nuts as shown. This is done twice.

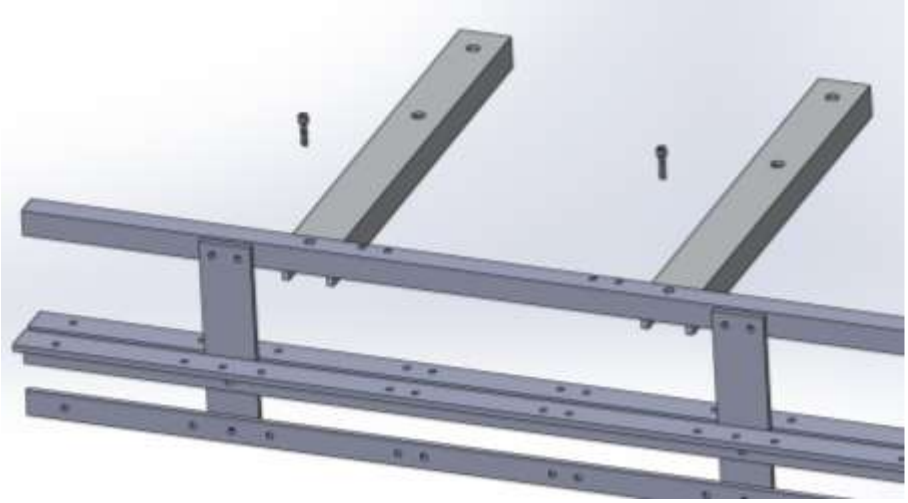


1. Line up the holes of the L-bracket and the clamp system arm. The arm should be oriented as shown. Fasten with 1/4”-20 bolts and nuts. This is done twice(other arm not shown).

2. Line up the holes of the base bar and the clamp system arms. The arm should be oriented as shown. Fasten with 1/4”-20 bolts and nuts. Connect base to system arms.

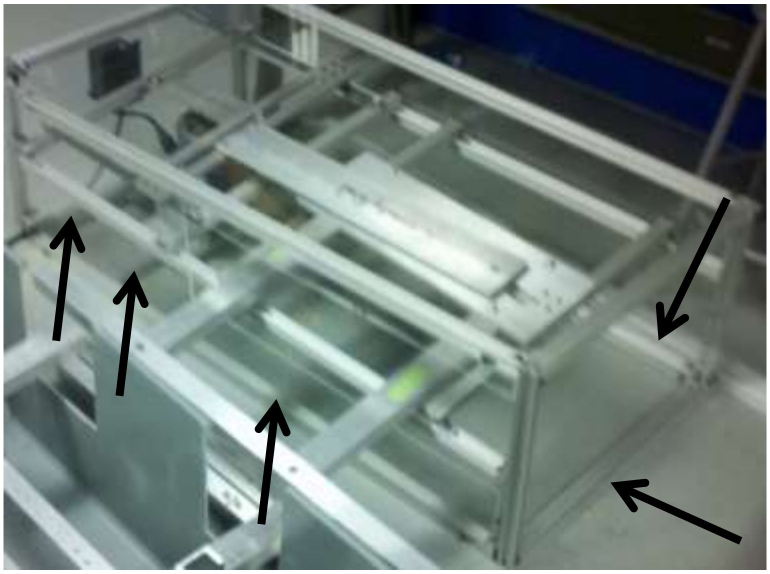
3. Line up the holes of the horizontal bases and the clamp system arms. The bases should be should be oriented as shown with the horizontal closest to the L bracket. Fasten with 1/4”-20 bolts and nuts. Connect base to system arms.

4. Line up the holes of the connecting arms and the L bracket. The connecting arms should be should be oriented as shown. Fasten with 1/4”-20 bolts and nuts.

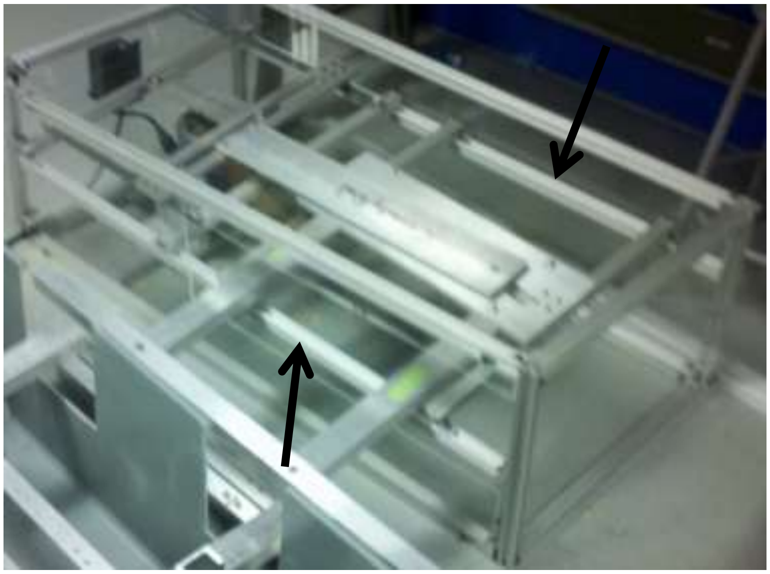


Cage Assembly

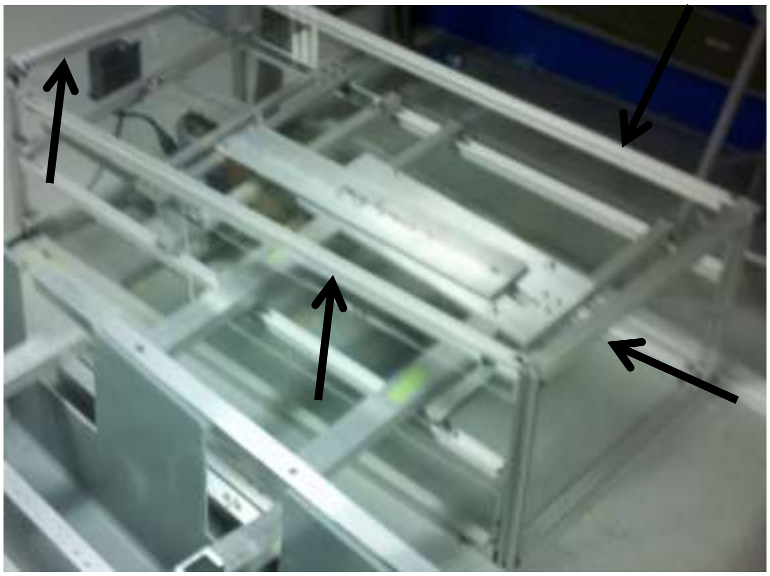
1. Connect lower T–slot extrusions (pointed to in picture) to the vertical T–slot extrusions. As shown in this picture. Use three L-brackets per corner and two L-brackets for each end of the double T-slot extrusion. Add two extra T-nuts to the outside of the two longer extrusions that to hold the Plexiglas later.



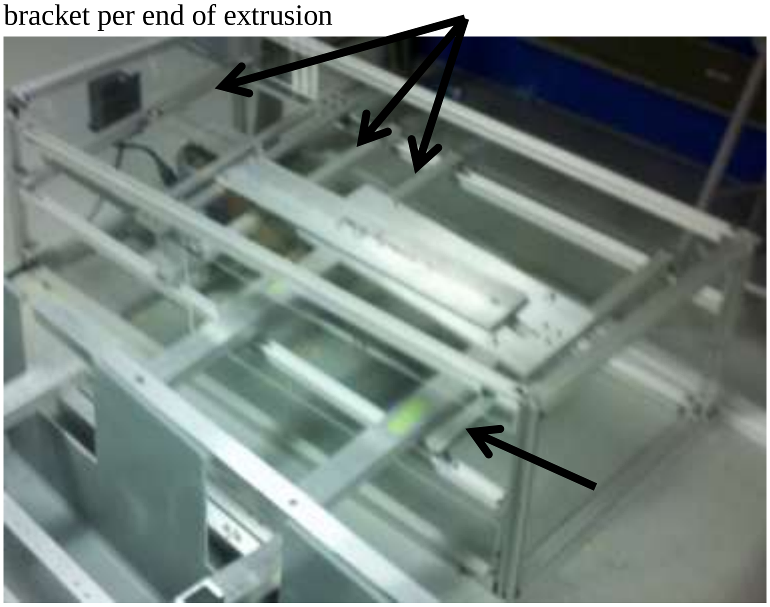
2. Connect middle long horizontal t-slot extrusions(pointed to in picture) to the vertical t-slot extrusions. Use one bracket to attach the bottom side of these to the vertical extrusions. Add 6 t-nuts to the top side of the extrusions before install.



3. Connect upper T–slot extrusions (pointed to in picture) to the vertical T–slot extrusions. As shown in this picture. Use three L-brackets per corner Add two extra T-nuts to the outside of the two longer extrusions that to hold the Plexiglas later.

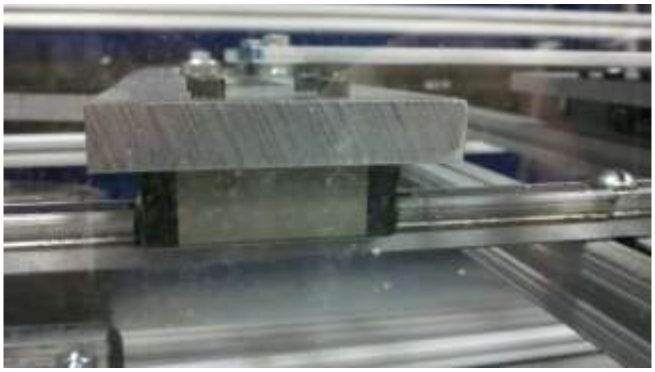


4. Connect middle horizontal short t-slot extrusions to the long middle t-slot extrusions. Use one L-bracket per end of extrusion



5. Slide the carriages onto the guide rails
6. Connect guide rails to t-slot extrusions. The longer guide rails are perpendicular to the middle short horizontal extrusions and the short guide rails are parallel to the middle short horizontal extrusions.

7. Connect the Scotch yoke and the amplitude converter using M4 screws.



8. Connect motor mount and supports using the double L-brackets and the motor base bracket.

9. Connect motor to motor mount using 7/16th bolts and nuts

10. Connect wheel to motor by screwing in the ¼”-20 set screw.

11. Connect ball bearing assembly to transfer arm and wheel

12. Connect transfer arm to scotch yoke/amplitude converter using ¼”-20 screws and nuts a 1/4” worth of spacers.

Final Assembly and calibration

1. Put Base in tube.
2. Adjust the height so that the base can attach to the middle level to connecting ubars

3. Connect the arm assembly to the amplitude converter using ¼”-20 bolts and nuts.

4. Adjust the position of the all four of the guide rails so that the arm and base are in the proper relative positions.
5. Add the Plexiglas panels to the cage
6. Connect the power cord from the controller to the motor. And attach controller to the side panel using 5/16” nuts and bolts.

Validation Plan

Boot Seal

ArcelorMittal informed us that they would not be able to fabricate our final oil boot seal design in time for the design expo. As a result, we will not be able to prove that the engineering specifications for the boot seal have been met. But, demonstrating that the engineering specifications have been met for the seal can be easily achieved. The cost will be known after the seal mold has been manufactured and bonded to the inner steal rings. The oil seal can be measured to prove that it fits the geometric constraints for length, inner and outer diameter and bolt pattern. If the seal is installed on the gear coupling in between the two drive shafts, then ease of installation can be determined if the installation time is measured with a stop watch. Noise can be tested for with a digital sound level meter. Because the operating temperature of the gear coupling is normally recorded during operation at the IN/TEK facility, the operating temperature of the oil seal will be known. Cycles to failure can be tested for experimentally, by running the seal for 18 months or longer before it is replaced due to preventative maintenance.

Test Fixture

Not all of the engineering specifications for the test fixture design can be tested for due to the time constraints of building the fixture, obtaining test samples, and testing the samples to failure before the design expo. We plan to test for the fixtures operating temp by installing a thermometer within the test fixture that will show the temperature of the oil in side. The specification for the multiple test samples can be validated through visual inspection, and the cost is known due to the bill of materials in Appendix G. All other engineering specifications can be tested for experimentally. If we were to run the test fixture for a total of 114 days, or 287 million cycles, then our testing time will be validated as well as the durability of the test fixture if it is able to last the whole test without failure. The standard deviation of the failure of the test samples can be found experimentally, using the equation below.

$$s_N = \sqrt{\frac{1}{N}\sum_{i=1}^{N}(x_i - \overline{x})^2}.$$

Where $\{x_1, x_2, \ldots, x_N\}$ are the observed cycles to failure for the test sample, $\bar{x}$ is the mean value of the cycles to failure for the test samples, and N is the sample size of 10. The 10% difference between the test fixture results and the real life spindle performance specification can be tested if the test fixture and our oil seal designs are allowed to complete one full cycle of 287 million cycles. Then their failure times can be compared. We plan to test for as many cycles as we have time for, as it will be less than the full 287 million due to our time constraints.

Validation Results

Seal: FEA results from Final design section

We were not able to produce a prototype of our boot seal, so we could not conduct experiments to validate it. However, the FEA and lifetime analysis conducted to create the design, thoroughly discussed in SECTION, are very encouraging. They strongly suggest that the boot seal design will be successful, as evidenced by the long lifetime predicted.

The test fixture developed to the oil boot seal was successful and works as designed, applying displacements to multiple oil seal test samples. Running the test fixture caused the motor to vibrate at accelerated speeds. Additional bracing would eliminate this problem. To begin the experiment, we powered on the motor controller and adjusted to motor speed, initiating the drive mechanism's motion. The rotational speed of the motor was measured using a tachometer and reflective tape on the motor shaft. Rubber test samples were prepared and cut to a size resembling the cross sectional area of the production oil seal, due to the unavailability of test seal samples being produced in time. The samples were loaded into the test fixture and clamped in place. With the samples in place and with a tachometer, we were able to estimate the cycles each sample endured.

Only 3 samples tested due to time constraints, and only for a short time: control groups and chem.
Result

To validate the test fixture, we powered the system and proceeded to run it without test samples loaded. We began by running the motor at its lowest speed, and then we gradually increased the speed to determine the maximum speed at which the motor could be safely run. We found that at relatively low speeds of approximately 60 rpm, which corresponds to the “5” position on the motor controller, the motor was not mounted securely enough and vibrated as a result, which indicated that it should not be run at higher speeds. However, at the safely low speeds the drive mechanism moves as designed and applied displacement any samples mounted in the testing area.

After establishing that the system moves as intended at low speeds, we prepared use the fixture to test samples. To do this, we produced ten test samples. Eight test samples were made of 1/8” thick NBR and were cut to approximately 4.4”x4.5”. To seven of these samples we caused varying damage, cutting or scoring them to weaken them. One NBR sample was left undamaged as the control sample. The remaining two test samples were made of latex rubber and were cut to approximately 4.4”x4.5” from latex cleaning gloves. One of these two samples was coated with petroleum jelly to induce a chemical degeneration of the test sample, while the other latex sample was an uncoated control sample. Photographs of the test samples are shown in Figure 31 below.

FIGURE 31: The undamaged control samples for NBR (left) and latex (right)

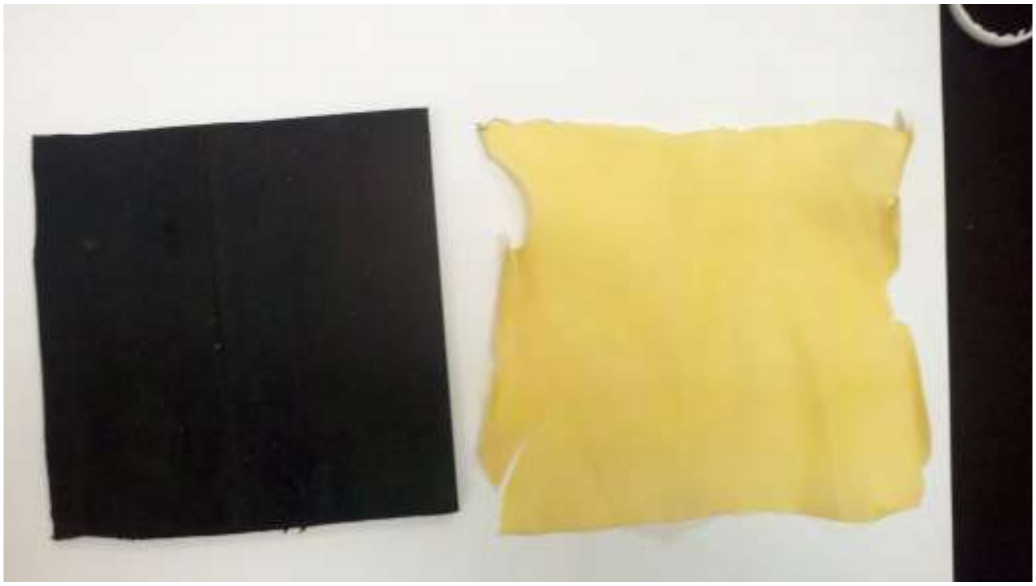


FIGURE 32: The NBR samples with 1” cuts

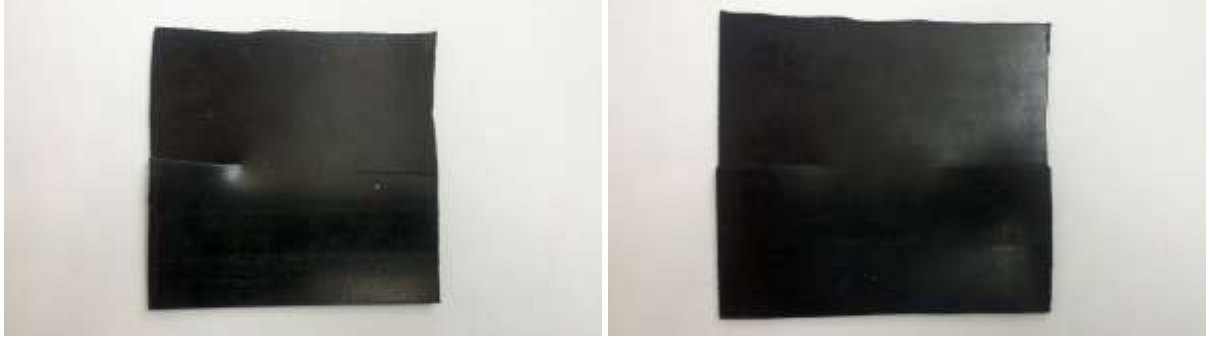


FIGURE 33: The NBR samples with DIMENSION

Due to time constraints, we were only able to test the NBR control sample and the two latex samples. The three samples were tested simultaneously and for the same displacement and number of cycles. These results indicate that our text fixture induces fatigue on tested samples, as intended.

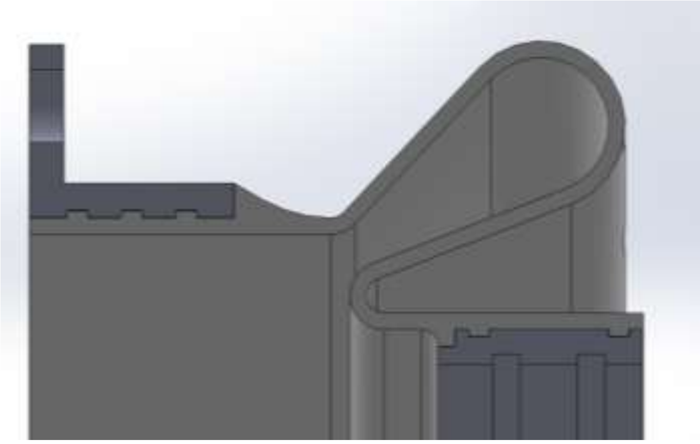
Discussion

Seal:
Improve: manufacturability?
Strength: fits within current setup, uses same materials, reduced stress, highly improved lifetime
Weakness: more difficult to manufacture than current seals

We produced a strong design for the boot seal. The seal's strengths are that it fits within the current setup, uses the same materials, shows significantly lower stresses than the current design under the same conditions, and therefore has a highly improved lifetime in service. Because the seal fits within the current system and uses the same materials, it should be compatible with I/N Tek's current system for obtaining boot seals and therefore that much easier to put into service.

A weakness of the seal design is that it is more difficult to manufacture than the current boot seal. The process used to create the current generation of boot seals involves molding the rubber onto the steel. The rubber shape used in the current seal allows for easier access of the mold, as shown in Figure 34 below. Our redesign of the boot seal may require an additional action to be used in the molding process.

FIGURE 34: Our redesign does not allow mold access as easily as the current seal design.



The manufacturability of our design could likely be improved. If it is not possible to incorporate mold actions into the manufacturing process, then the design would need to be modified to be very similar to the current design in shape: the difference would be that the size of the two halves of the “S” shape would be optimized to reduce stress just as we did earlier in the design process to produce our design. The difference would be the new constraint imposed to require no mold actions.

Fixture:
Improve: motor mounting, air flow to cool motor and system, balance of system, keep system from “walking,” improve upper deck cuts, attach test mounts to tub to prevent rubbing
Strength: robust design, simple controls, 10 test samples
Weakness: large, unbalanced, heats up, can slide on floor, oil sealing, parts rubbing

Strengths of our test fixture design include a robust design, simple controls, and the ability to test 10 test samples at once. The sturdy aluminum design can withstand incidental punishments that it might incur during operation in the steel mill. The motor controller features a switch and a simple dial, labeled 0 to 10, enabling quick and easy operation and allowing for emergency shut-off. Testing 10 test samples simultaneously greatly reduces experimental error among the 10 samples; if each of the 10 samples were tested separately, the results would be less reliable. Testing for 10 samples also allow for easier statistical analysis: the first sample to fail gives the statistical L10 life for the experiment, and more samples provide better data than few samples.

The fixture design also has weaknesses. The motor is not secured rigidly enough. The fixture is large, 55”x45”x16”, and so should not be operated on a tabletop. The large displacements of the scotch yoke and connecting rod make the drive mechanism unbalanced, and so the test fixture has the potential to “walk” during operation, especially at higher operating speeds. The large motor used and the friction in the system cause heating in the drive mechanism even after a short time. The current tub is slightly too small and is not rigidly connected to the test mounts placed in it; this can result in rubbing between the two as the test fixture vibrates. The cut in the tub that allows the test mounts to be placed in it is too low; the uncut depth of the oil tub is too low, and would allow oil to splash over the side. Finally, the test fixture does not have a readout for rpm or for cycles run.

This list of weaknesses show that the fixture design has much room for improvement. The motor must be better secured: the top of the motor must be rigidly fastened, and the output shaft must be constrained by bearings to prevent the shaft from incurring moments that the motor is not designed for. The system heating could be improved by greater ventilation. A possible solution is to add a fan to cool the motor, and to cut ventilation slots in the plexiglass casing that houses the drive mechanism. To satisfactorily improve the balance of the drive mechanism would require a redesign, though the balance could be helped more simply by reducing the mass of the moving components. To prevent the system from “walking” during operation, rubber feet could be added to the fixture’s bottom in addition to improving the balance. Finally, the oil splash hazard requires a redesign. A possible solution is to raise the drive mechanism and lengthen the test mounts such that the test fixtures and therefore the oil level in the tub are significantly below any openings through which oil might escape. The motor controller currently used could be replaced by a variable frequency drive that would show the rpm of the system and allow for more precise control. A digital cycle counter could also be added.

Recommendations

We believe that our boot seal design is strong and offers great value to I/N Tek. We recommend that I/N Tek investigates our seal design and compares it to the current seal design. If the redesign compares favorably, consider testing it to reduce risk and then put it into use in the mill to reduce breakdowns that halt production.

For the test fixture, the motor and its drive shaft must be better secured. We recommend rigidly mounting the top of the motor. Also consider using an even faster motor to reduce test time. The aluminum used in the test mounts is able to be immersed in oil. However, if the aluminum test mounts were to be immersed in the rolling fluid used in the mill, they would disintegrate. Consider replacing aluminum parts in contact with test fluid with stainless steel to increase the number of testing fluids that the test samples can be tested in. Cut ventilation holes in plexiglass casing to cool the system, and consider installing fan to go with motor. Install rubber feet on bottom to make the system more stable. We also recommend replacing the motor controller with a variable frequency drive; however, if this is not done, I/N Tek could calibrate the current motor controller for more precise control of the system.

Conclusion

ArcelorMittal would like us to design an oil boot seal for their I/N TEK mills with a lifetime of at least 18 months. We have gathered much information about the project, but we have not determined the root of the problem. We calculated from the operating conditions that our boot seal must survive at least 287 million cycles at 405 rpm. It must also fit into the current mill system without requiring any modifications. We have created a plan to complete this project as shown in Figure 29. We have generated many design concepts, selected the best concepts for both the boot seal and the fixture, and defined these concepts into “alpha designs.” We then performed an engineering analysis, created a design, defined the prototypes, and prepared a plan to validate them. Finally, we manufactured a prototype of the test fixture and validated it.

Acknowledgements

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Mark Stock

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Appendix A

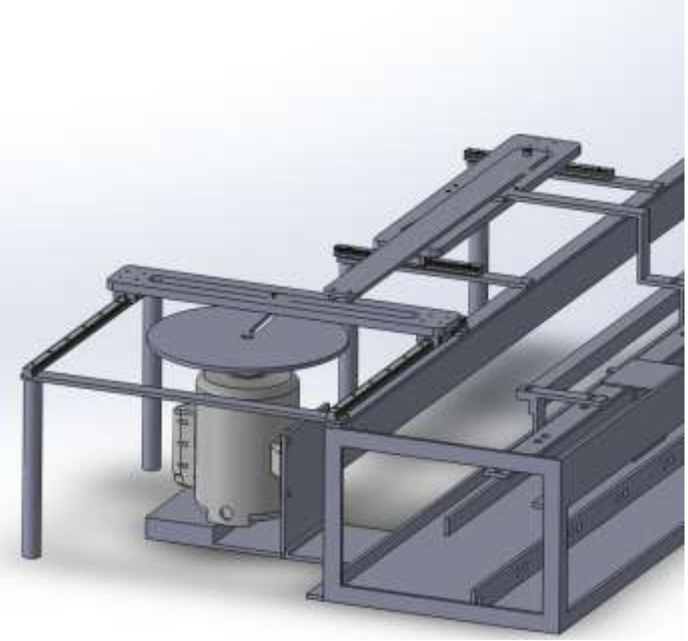
Part #		Part name	Qty	Material	Price(per unit)	Price(total	Manuf. Process
5990K26		Motor	1		$287.46	$287.46	McMaster
9246K64	1/2"x12"x36"	1	Al	96.51	96.51	McMaster	9246K64
8975K424	1/4"x4"x36	1	Al	30.49	30.49	McMaster	8975K424
91274a172	1/4"-20 1" screws X 50	0	Steel	0	0	McMaster	91274a172
948044a029	1/4"-20 nut X 100	0	Steel	0	0	McMaster	948044a029
6383K22	Bearing	2	Steel	4.19	8.38	McMaster	6383K22
6250K1	460mm Slide Rail	2	Steel	138	276	McMaster	6250K1
6250K1	220mm Slide Rail	2	Steel	66	132	McMaster	6250K1
6250K18	Slide Carriage	4	Steel	128	512	McMaster	6250K18
91290A164	M4 x 18mm X 100	1		7.44	7.44	McMaster	91290A164
6750K161	1/2" Diameter 12" long shaft	1	Aluminum	6.29	6.29	McMaster	6750K161
47065T101	1"x1"x120" T slot	7		$31.59	221.13	McMaster	47065T101
47065T223	90 degree Brackets	40		3.98	159.2	McMaster	47065T223
47065T107	1"x2"x24"	1		12.85	12.85	McMaster	47065T107
47065T176	90 degree 8 hole bracket	1		5.96	5.96	McMaster	47065T176
47065T169	90 degree 4 hole bracket	1		5.58	5.58	McMaster	47065T169
8560K263	48"x48"x1/8" plexiglass	1		96.85	96.85	McMaster	8560K263
8560K262	24"x48"x1/8" plexiglass	1		53.27	53.27	McMaster	8560K262
47065T142	End-fed fasteners (4-pack)	15		2.3	34.5	McMaster	47065T142
1630T472	2"x1"x36" U-beam	5	Al	22.18	110.9	McMaster	1630T472
8982K314	1.25"x1.25"x48" beam	6	Al	23.79	142.74	McMaster	8982K314
8975K24	.25"x1"x72" Strip	2	Al	16.02	32.04	McMaster	8975K24
1630T473	2"x1"x12" U-beam	2	Al	9.51	19.02	McMaster	1630T473

Appendix B

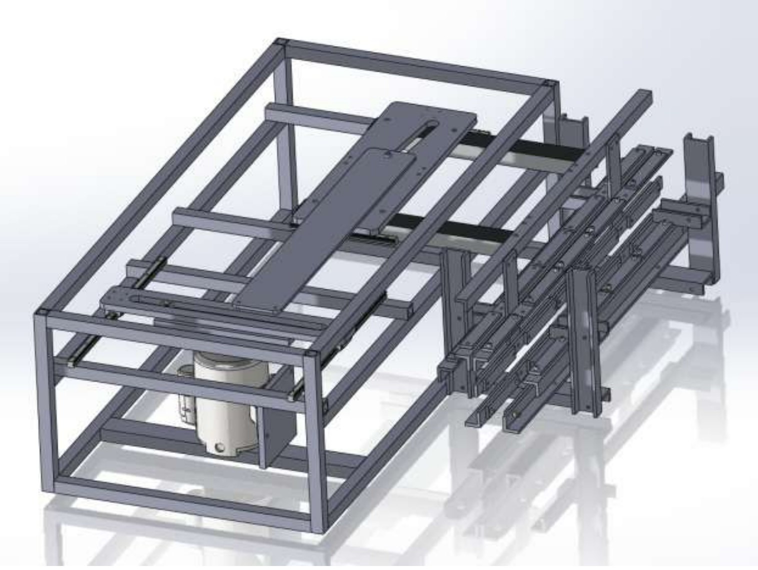
The design of the boot seal has reminded unchanged. This is due from not having 3 months to do testing of the design.

The design of the set fixture has changed greatly since Design Review #3. The first change is how the motion generation portion is supported. The original design used individual supports for each feature. The redesign utilizes T-slot extrusions. This change was made because it more securely held the components in place, it allows us to adjust the position of components such as the guide rails without any additional machining and it provides a way to mount panels around moving parts.

Old:



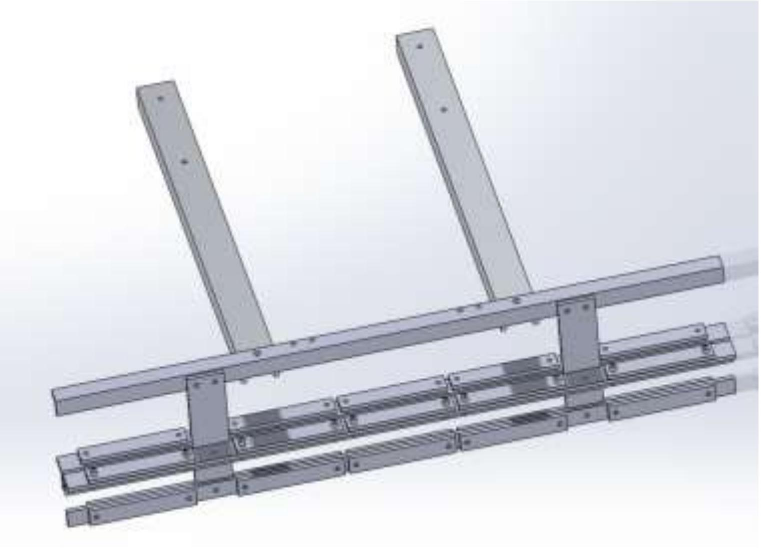
New:



The second change is with the test arm. The test arm got cut in half and now has the ability to mount samples of both sides it. This was done to make the size more practical. We made so that there are two bars instead of one connecting the arm to the amplitude converter. This was done so that so that the arm could withstand the torque from uneven loading.

Old:

New



The last change that was we switched from fabricating a metal tub to buying a plastic tub. This switch was made because it reduced manufacturing, was cheaper and was less likely to leak.

Old:

New:

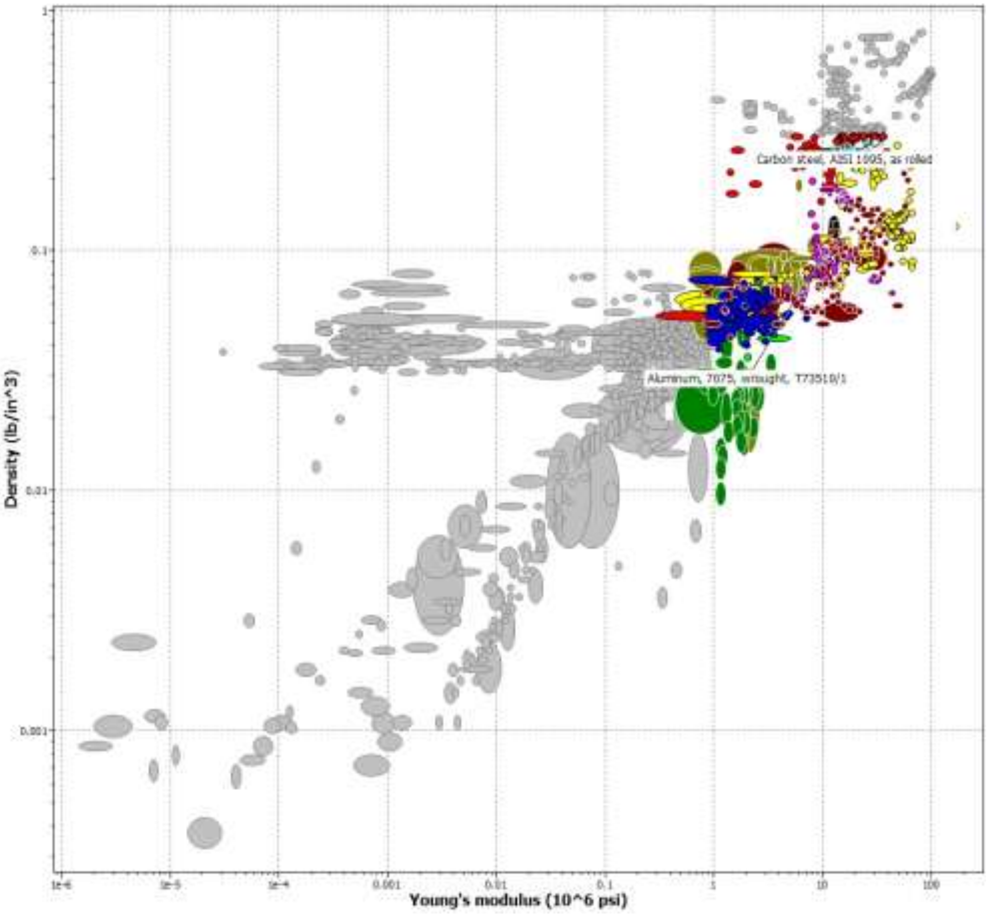
Appendix C

Material Selection Assignment (Functional Performance)

The major engineering criterion of our prototype was weight, and stiffness. Working with these goals in mind we established the necessary Young's Modulus necessary to support the 16 lbs of force necessary to operate the drive mechanism, with a safety factor of 2. given our acceptable deflection, yield strength and weight. We analyzed these limits using the CES software.

The minimum Young's modulus needed for the connect rod to move the amplitude converter and attached testing rod without failure is ________. The structure has a weight of ______. Using a safety factor of ____, the calculated modulus was _________ . We than calculated the minimum yield strength needed to accommodate a displacement of _______, which was ________. ________. Using the graph below in figure XX allowed us to choose a suitable material for the application.

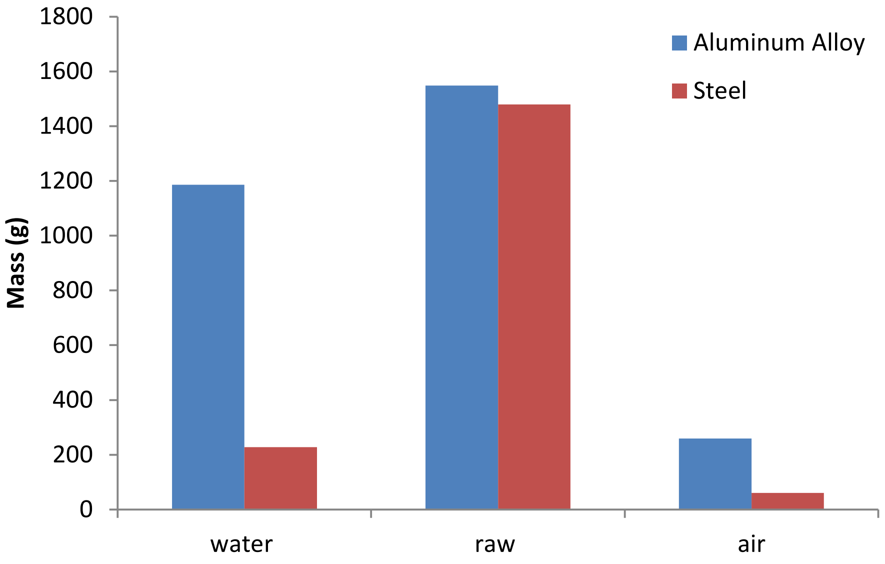
The two material choices that met our criteria selected were ___aluminum and ___ steel. . Aluminum is a lighter material than the steel. The extra strength provided by steel was unnecessary and the added benefit of the weight of aluminum.



Material Selection Assignment (Environmental Performance)

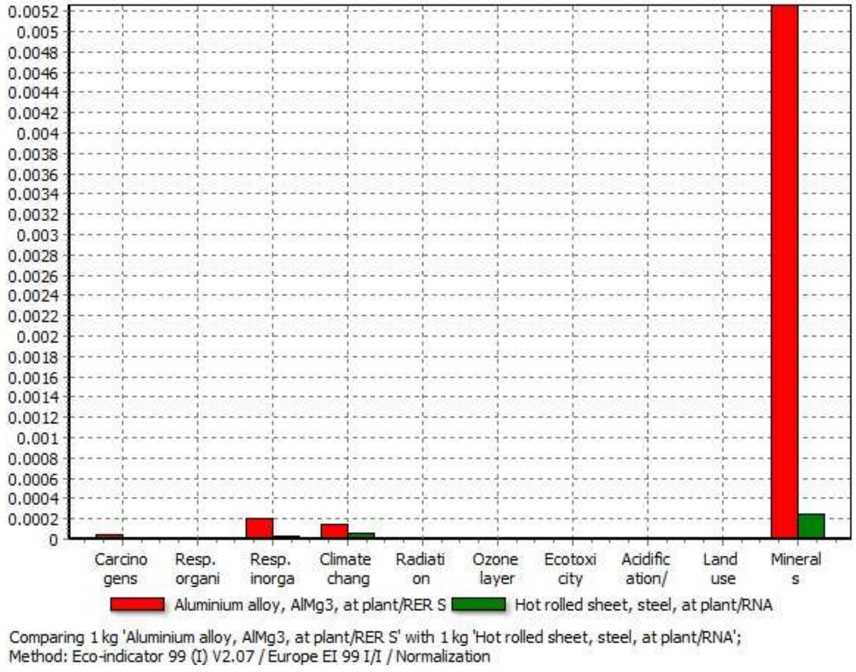
The two materials chosen in the CES analysis above were further analyzed for emissions and environmental impact using SimaPro. The material choices are approximations due to SimaPro's limited database of materials. The following simulation was completed on 1 kg of steel and 1 kg of aluminum. Based on the EcoIndicator 99 damage classifications, the aluminum has a larger impact on the environment, as seen below in Figure 1: While both materials require a lot of raw material to produce; the aluminum pollutes the air and water more during the manufacturing process. The manufacturing process for aluminum alloy is energy intensive, requiring large amounts of combustion and releasing more air emissions than the steel.

Figure 1: Aluminum pollutes the water and air more than steel in addition to requiring more raw materials



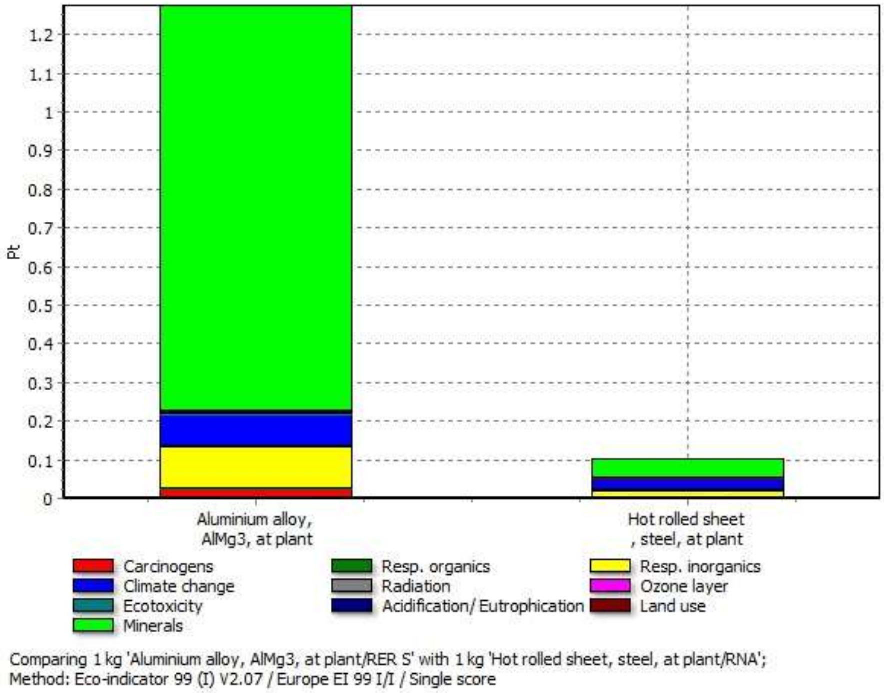
The SimaPro analysis indicates that the manufacturing of 1kg of steel and aluminum have minimal impact on the environment, as seen below in Figure 2. The largest effect of production of these materials is the mineral utilization necessary for the production of steel and aluminum shown in figure 3.

Figure 2: Normalized Environmental damage caused by aluminum alloy and steel production



Comparing 1 kg 'Aluminium alloy, AlMg3, at plant/RER S' with 1 kg 'Hot rolled sheet, steel, at plant/RNA';
Method: Eco-indicator 99 (I) V2.07 / Europe EI 99 I/I / Normalization

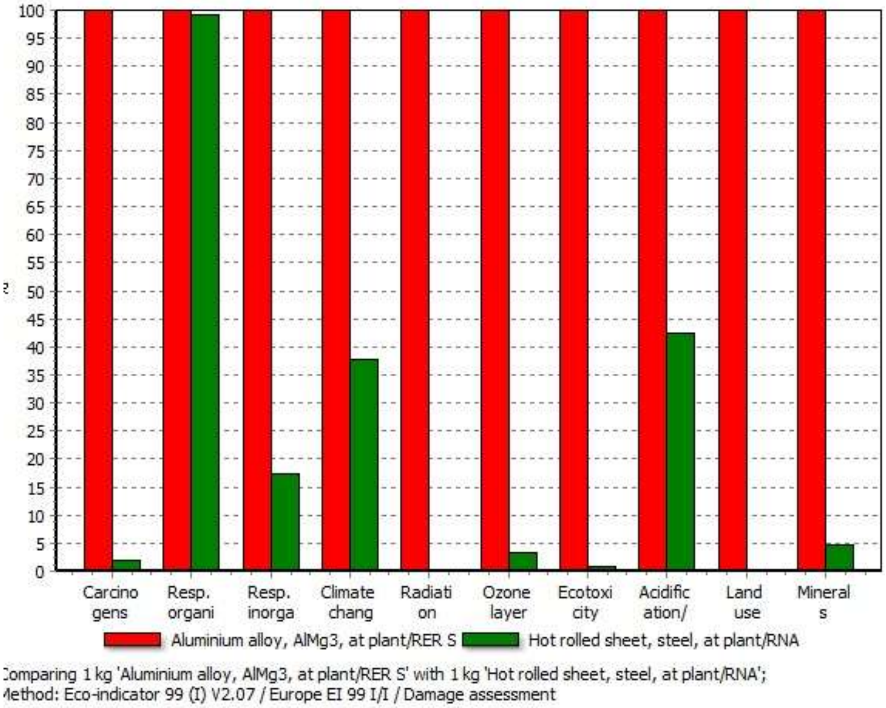
Figure 3: Comparison of aluminum alloy and steel environmental effects.



Comparing 1 kg 'Aluminium alloy, AlMg3, at plant/RER S' with 1 kg 'Hot rolled sheet, steel, at plant/RNA';
Method: Eco-indicator 99 (I) V2.07 / Europe EI 99 I/I / Single score

The figure below, which shows the impact of the materials in relation to one another The SimaPro analysis also shows that the aluminum has the higher negative impact overall when compared to the steel, as seen below in figure 3.

Figure 4: Impacts in damage categories

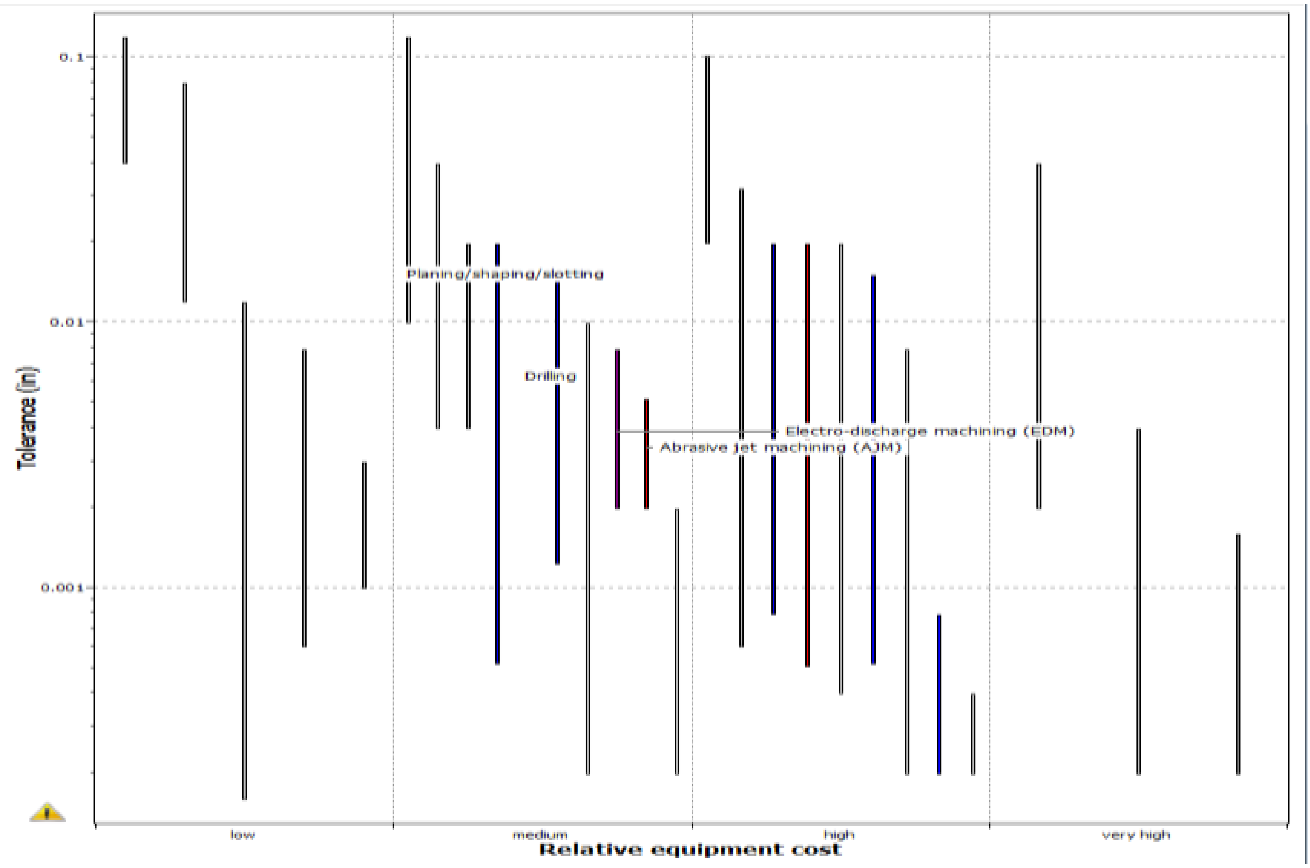


Comparing 1 kg 'Aluminium alloy, AlMg3, at plant/RER S' with 1 kg 'Hot rolled sheet, steel, at plant/RNA';
Method: Eco-indicator 99 (I) V2.07 / Europe EI 99 I/I / Damage assessment

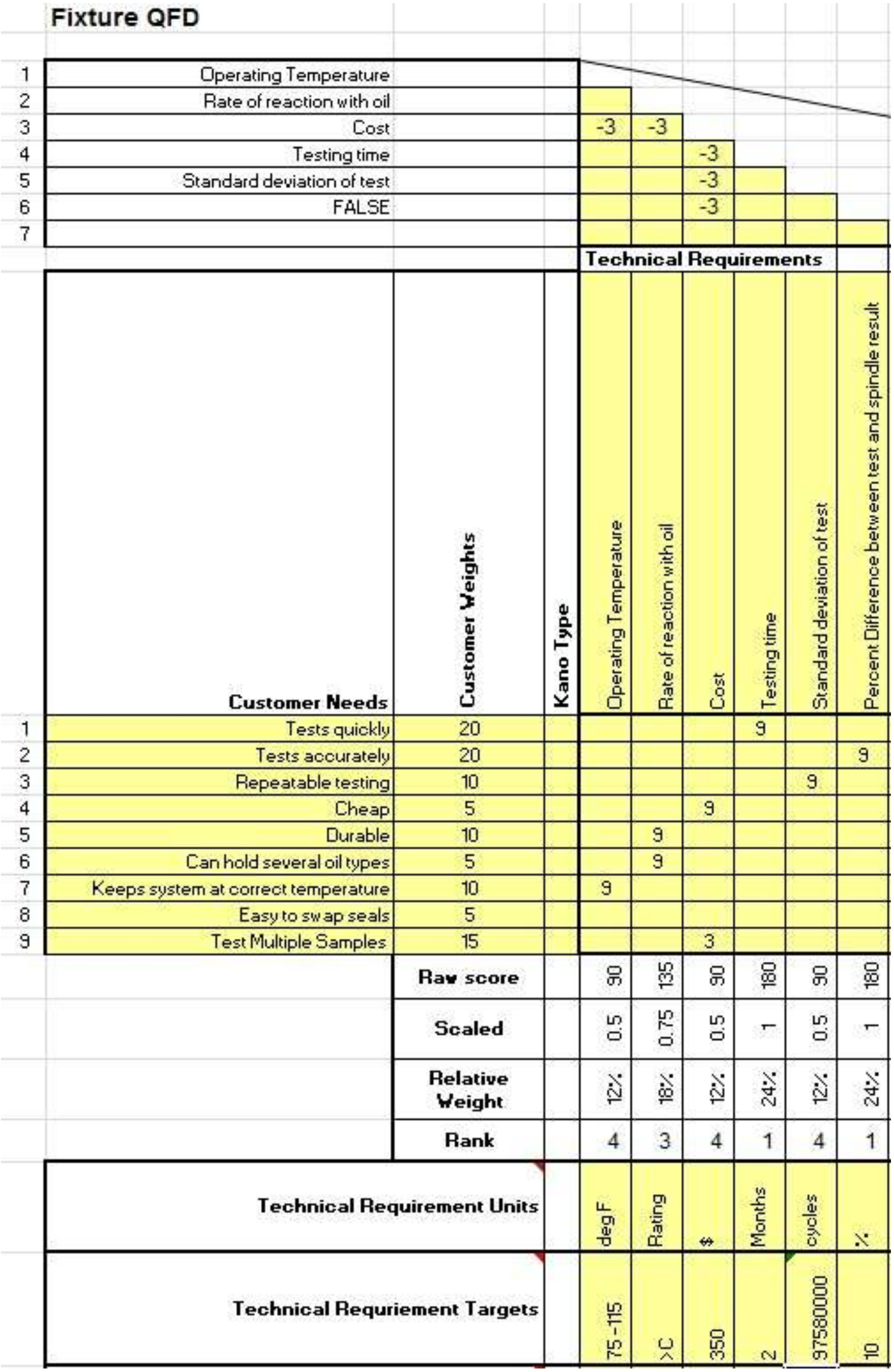
The SimaPro analysis indicates that the aluminum has the higher negative environmental impact overall when compared to the steel. However, weight is a crucial aspect in the drive mechanism of our test fixture due to the load this places on the motor. Steel would not be the better material to use, and because aluminum is relatively cheap and the low volumes of material needed for the test fixture, aluminum is the best material choice for this project.

Manufacturing Process Selection Assignment

The real-world product volume is likely around 100. This could be used in labs throughout the world. Both These components will use either drilling as their manufacturing process. When looking at batch size of 100 and tolerances of less than .005in. with that we than looked at the equipment cost. The cheapest is planning/slotting but this can't to circular shapes. The next cheapest is drilling. Below is the CES chart.



Appendix D



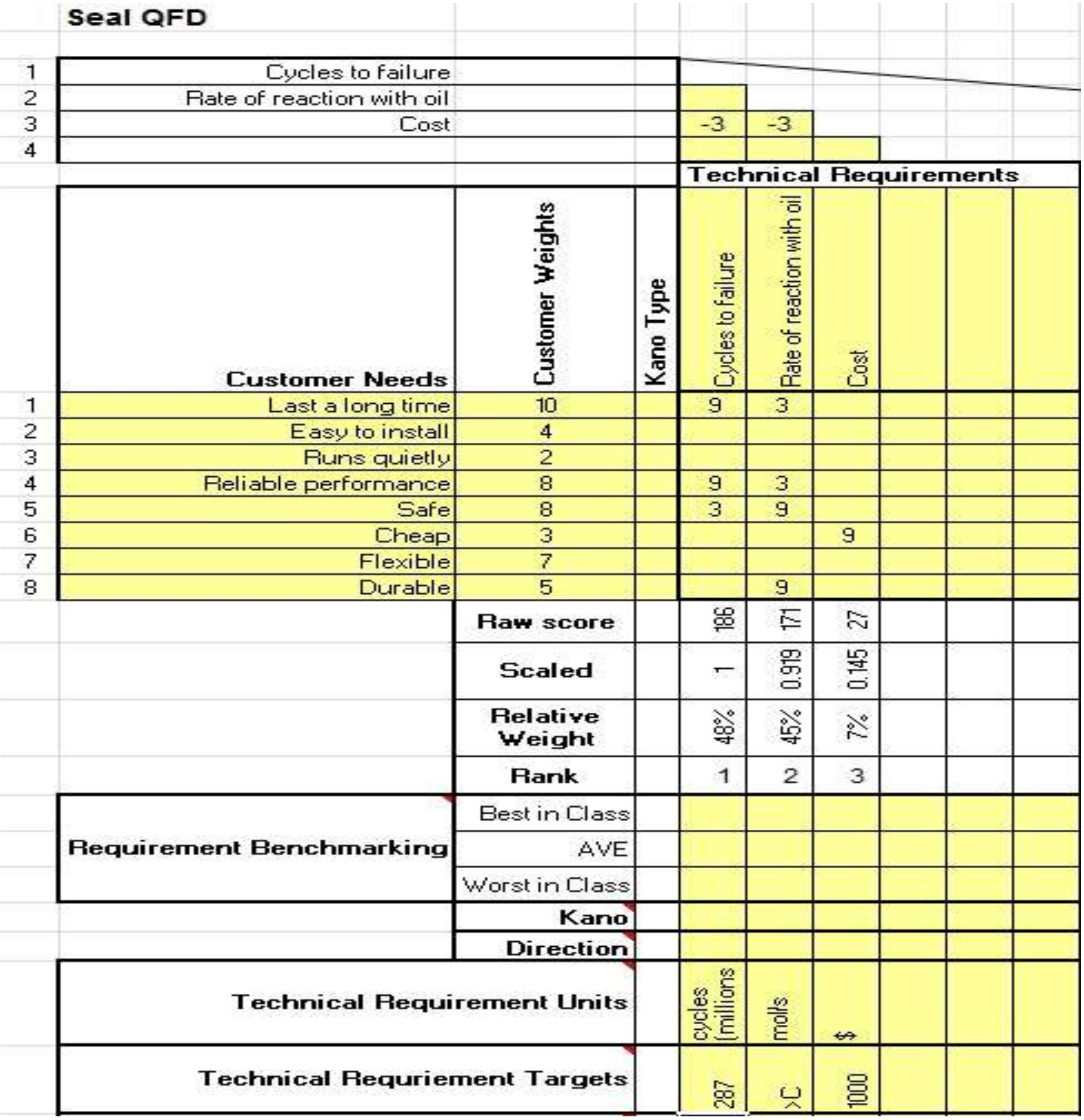
Fixture QFD

1	Operating Temperature						
2	Rate of reaction with oil						
3	Cost	-3	-3				
4	Testing time			-3			
5	Standard deviation of test			-3			
6	FALSE			-3			
7							

Technical Requirements

	Customer Needs	Customer Weights	Kano Type	Operating Temperature	Rate of reaction with oil	Cost	Testing time	Standard deviation of test	Percent Difference between test and spindle result
1	Tests quickly	20					9		
2	Tests accurately	20							9
3	Repeatable testing	10						9	
4	Cheap	5				9			
5	Durable	10			9				
6	Can hold several oil types	5			9				
7	Keeps system at correct temperature	10		9					
8	Easy to swap seals	5							
9	Test Multiple Samples	15				3			
		Raw score		90	135	90	180	90	180
		Scaled		0.5	0.75	0.5	1	0.5	1
		Relative Weight		12%	18%	12%	24%	12%	24%
		Rank		4	3	4	1	4	1
	Technical Requirement Units			deg F	Rating	$	Months	cycles	%
	Technical Requriement Targets			75 -115	>C	350	2	97580000	10

Appendix E

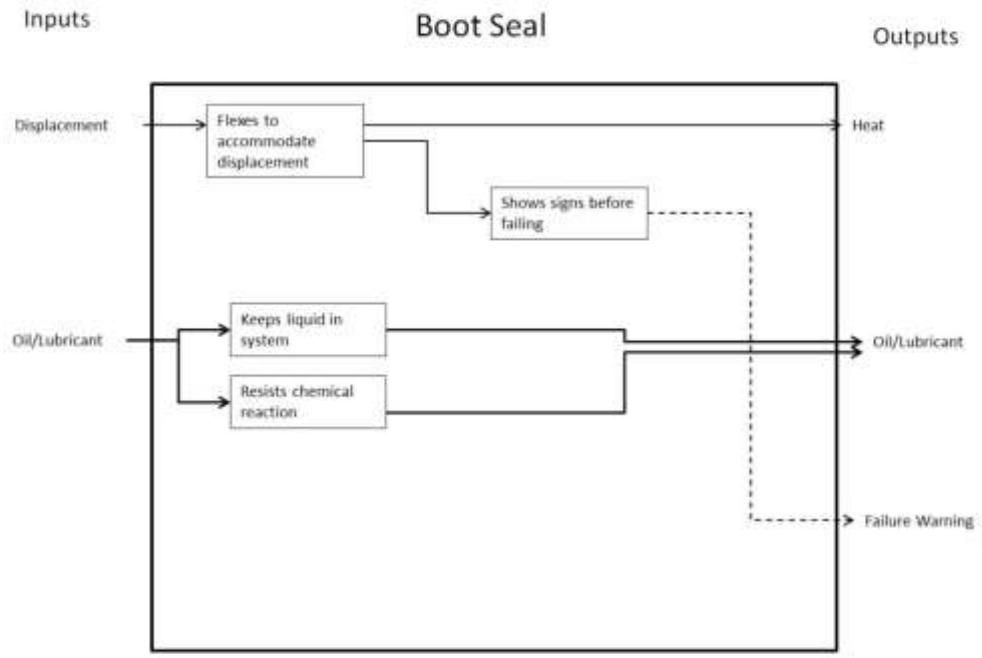


Seal QFD

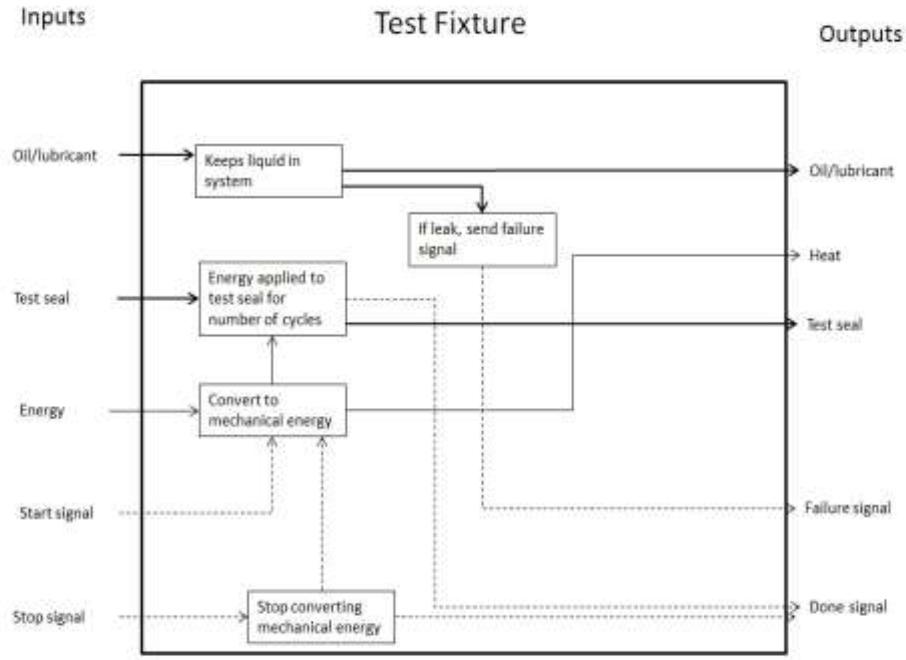
1	Cycles to failure			
2	Rate of reaction with oil			
3	Cost	-3	-3	
4				

	Customer Needs	**Customer Weights**	**Kano Type**	Cycles to failure	Rate of reaction with oil	Cost			
1	Last a long time	10		9	3				
2	Easy to install	4							
3	Runs quietly	2							
4	Reliable performance	8		9	3				
5	Safe	8		3	9				
6	Cheap	3				9			
7	Flexible	7							
8	Durable	5			9				
		Raw score		186	171	27			
		Scaled		1	0.919	0.145			
		Relative Weight		48%	45%	7%			
		Rank		1	2	3			
	Requirement Benchmarking	Best in Class							
		AVE							
		Worst in Class							
		Kano							
		Direction							
	Technical Requirement Units			cycles (millions	mol/s	$			
	Technical Requriement Targets			287	>C	1000			

Appendix F

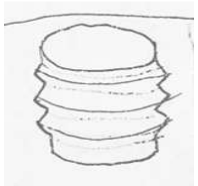

Inputs
Boot Seal
Outputs
Displacement
Flexes to accommodate displacement
Heat
Shows signs before failing
Oil/Lubricant
Keeps liquid in system
Resists chemical reaction
Oil/Lubricant
Failure Warning

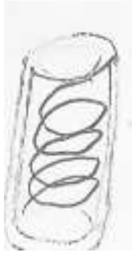
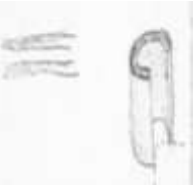
Appendix G


Inputs
Test Fixture
Outputs
Oil/lubricant
Keeps liquid in system
Oil/lubricant
If leak, send failure signal
Heat
Test seal
Energy applied to test seal for number of cycles
Test seal
Energy
Convert to mechanical energy
Start signal
Failure signal
Stop signal
Stop converting mechanical energy
Done signal

Appendix H

Below and on the next page is the Pugh chart for the oil boot seal containing all of the concepts generated. Some clarifications of the criteria are as follows; long lasting means how well the seal can handle the everyday cyclical loading and robust means how well it can withstand the less frequent large displacements that it could undergo.

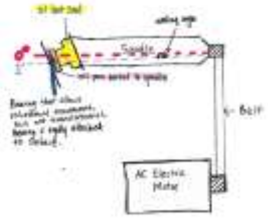
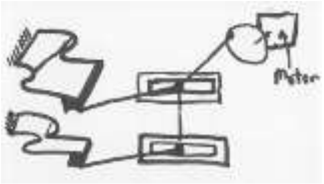
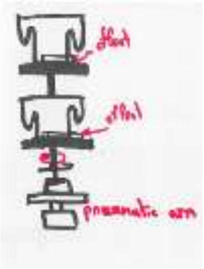
Descripton		Datum	Helix	Straight	The bulge	Spiral	Long S	Big radius	Rubber Accordion
Sketch									
Criteria	Weight								
Long lasting	70	3	1	1	4	4	2	5	4
Easy to install	5	3	1	3	3	3	2	2	3
Robust	15	3	4	2	4	2	3	4	4
Cheap	10	3	2	5	4	1	3	4	3
Total	100	300	155	165	395	335	225	460	385

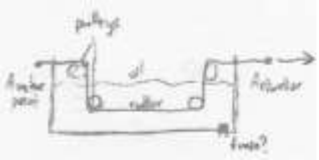
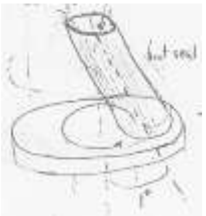
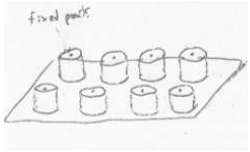
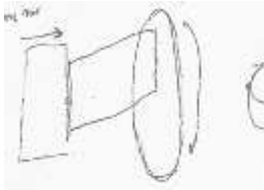
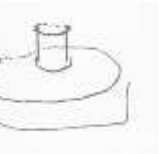
Descripton		Particle Patching	Beyond Steel	Spring Support	Double Layer	Warning layer
Sketch						
Criteria	Weight					
Long lasting	70					
Easy to install	5					
Robust	15					
Cheap	10					
Total	100					

Additional designs; Cloth, fiber supported, oil recycler, external sealant, self sealing tank, metal curvature

Appendix I

Below and on the next page is the Pugh chart for the test fixture containing all the concepts that we generated. Some clarifications of the criteria are as follows; test quickly means how many cycles can the concept go through in a certain amount of time, tests accurately means how well does the test fixture represent what the seal is undergoing in real world conditions, and repeatable testing means how likely is it that this test fixture is going to operate the same way from test to test.

Descripton		Datum	Belter	Slots	Thrust-rotate motor	The stack
Sketch						
Criteria	Weight					
Tests quickly	20	3	4	5	4	3
Tests accurately	20	3	2	1	2	2
Repeatable testing (Complexity)	10	3	4	5	4	4
Cheap	5	3	4	5	4	4
Durable	10	3	2	2	1	1
Multiple oil types	5	3	3	4	3	2
Correct system temperature	10	3	3	4	3	3
Easy to swap seals	5	3	4	5	4	3
Tests multiple samples	15	3	3	5	3	4
Total	100	300	310	375	300	285

Descripton		Chem box	Turntable	Many table	Vertical Turntable	Little portable
Sketch						
Criteria	Weight					
Tests quickly	20	5	4	5	4	5
Tests accurately	20	1	2	1	2	1
Repeatable testing (Complexity)	10	5	4	4	4	4
Cheap	5	5	4	4	4	5
Durable	10	2	2	1	2	1
Multiple oil types	5	5	3	2	3	2
Correct system temperature	10	4	3	2	3	3
Easy to swap seals	5	4	4	3	4	4
Tests multiple samples	15	3	3	5	3	3
Total	100	345	310	310	310	300

Appendix J

	allow for individual samples to be removed easily	securely hold samples in place	keeps ends of san
weight	10	35	30
Provides Mechanical energy to samples			
Hooks	3	3	3
Clamps	2	4	5
screw into samples	2	5	5
glue to samples	1	5	5

	Ability to resist leaks	Transparency	Longevity
weight	60	10	30
Keeps Liquid In			
sealant	3	3	3
painted surface	2	1	2
plastic lining	4	1	3
fish tank	3	5	2
Plexiglass	3	5	4

	Efficient	Quiet	Reliable
weight	10	5	40
Coverts energy to mechanical energy			
scotch Yoke	3	3	3
tradition ecentric	2	3	3
linear servo motors	2	4	4
pneumatic	1	2	2
stepper motor	3	3	3

	Quick	Reliable	complexity
weight	30	30	20
Remove Oil			
drain at bottom	3	3	3
lower/ raise test samples	4	2	1
drainspout on side	3	3	3
suction device	3	2	2

	Even coating	realistic coating	ablity ot coat one
weight	30	30	20
Apply Oil			
Immerse	3	3	3
Apply coating at setup	3	1	4
pump spray	3	1	5
pump trickle on	2	2	4

Appendix K

Final Assessment Severity Probability	Risk Level	Item Id	User / Task	Hazard / Failure Mode	Risk Reduction Methods / /Comments	Initial Assessment Severity Probability	Risk Level
Moderate Remote	Negligible	1-1-1	All Users Common Tasks	mechanical : drawing-in / trapping / entanglement The fixture is designed to have 1/2 hp rotating motor. If something gets caught in there it could do a lot of damage	We have designed that the motor and the mechanism with be inclosed in a cage.	Moderate Unlikely	Low
Minor Remote	Negligible	1-1-2	All Users Common Tasks	mechanical : pinch point When running the fixture could pinch someone if he/she touched where either of the bearings are	We have designed that the motor and the mechanism with be inclosed in a cage.	Minor Unlikely	Negligible
Moderate Remote	Negligible	1-1-3	All Users Common Tasks	mechanical : stabbing / puncture We have a rod that moves lineararly between 6-12" at a high RPM.	We have designed that the motor and the mechanism with be inclosed in a cage.	Moderate Unlikely	Low
Moderate Remote	Negligible	1-1-4	All Users Common Tasks	mechanical : fatigue This fixture is designed to put samples through hundreds of millions of cycles.	Have chosen beams with higher second moments of inertia to reduce to maximum stress placed on the material	Moderate Unlikely	Low
Moderate Remote	Negligible	1-1-5	All Users Common Tasks	mechanical : break up during operation This fixture is designed to put samples through hundreds of millions of cycles.	Have chosen beams with higher second moments of inertia to reduce to maximum stress placed on the material	Moderate Unlikely	Low

Final Assessment Severity Probability	Risk Level	Item Id	User / Task	Hazard / Failure Mode	Risk Reduction Methods / /Comments	Initial Assessment Severity Probability	Risk Level
Moderate Remote	Negligible	1-1-6	All Users Common Tasks	electrical / electronic : shorts / arcing / sparking The motor is powered by AC. There is inherent risk when working with elctronics in general	The presense of exposed wire will be avoided	Moderate Unlikely	Low
Moderate Remote	Negligible	1-1-7	All Users Common Tasks	electrical / electronic : water / wet locations the samples are tested in a liquid it is possible of some of the liquid to get on the floor and create a slipping hazard.	The fixture has a lid to protect against slashes. A faucet has been included to allow for easy and controlable drainage. The presense of exposed wire will be avoided	Moderate Unlikely	Low
Minor Remote	Negligible	1-1-8	All Users Common Tasks	slips / trips / falls : slip the samples are tested in a liquid it is possible of some of the liquid to get on the floor and create a slipping hazard.	The fixture has a lid to protect against slashes. A faucet has been included to allow for easy and controlable drainage	Moderate Unlikely	Low
Moderate Remote	Negligible	1-1-10	All Users Common Tasks	chemical : skin exposed to toxic chemcial The chemicals that the samples are tested in could irritate skin.	The fixture has a lid to protect against slashes. A faucet has been included to allow for easy and controlable drainage	Moderate Likely	Medium
Minor Remote	Negligible	1-1-11	All Users Common Tasks	fluid / pressure : fluid leakage / ejection the samples are tested in a liquid. It is possible for some of the liquid to get out	The fixture has a lid to protect against slashes. A faucet has been included to allow for easy and controlable drainage	Minor Unlikely	Negligible
		1-1-9	All Users Common Tasks	environmental / industrial hygiene : corrosion The fixture will be exposed to various fluids.(Sponsor has verified material will suffencent)		Moderate Very Likely	High