# Biaxial Test Fixture

## Final Report

**Group Members:**
Ben Hainsey
Nicole Walsh
Eric Hebner

**Sponsor:**
Cummins Inc.

**Faculty Advisor:**
Dr. William Oates

**Graduate Consultant:**
Parker Harwood

**Instructors:**
Dr. Kamal Amin
Dr. Chiang Shih

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## 1.0 Abstract

The objective of this project was to design a biaxial test fixture that could be incorporated into an existing MTS machine for the testing of gasket materials for Cummins' engines. This design theoretically accomplishes the goal of equal biaxial tension across all gripping locations of the sample.

## 2.0 Acknowledgement

This project would not have been possible without the enormous amount of advice by Terry Shaw, Parker Harwood, and our faculty advisor, Dr. William Oates. Their continued assistance and encouragement has kept this project running as smoothly as possible. We would like to personally thank the members of the National High Magnetic Field laboratory, including Robert Walsh and Scott Bole, for their extensive help and plethora of advice provided to us.

# 3.0 Project Overview

This project originated because of the inconsistencies that occur while obtaining compression data for the gasket materials used in the engines of Cummins' diesel trucks. While preforming uniaxial compressive tests on gasket materials a mixed state of shear, tensile, and compressive forces is generated in the specimen. This gives inaccurate and inconsistent data. A biaxial tension test is a way to simulate a compression strain. The reason Cummins, Inc. does not purchase a biaxial test machine is very simple; it is incredibly expensive. Some test machines can run upwards of six figures. This is not economical for a company to purchase unless they specialize in that type of testing. They already know their gaskets work, but the need for better understanding and the possibility of broadening the types materials is the inspiration for this project.

## 3.1 Background

### 3.1.1 Polymers

For centuries materials such as wood, rubber, and silk have been used. These naturally occurring materials are polymers. They are inexpensive to produce and are organic in origin. Similar to metals the properties of a polymer is dependent on the structure of the atomic bonding within that material. Because of the organic aspect of the material, the bonds are covalent and molecular chains are formed. The mechanical characteristics of a polymer are very sensitive to temperature, strain rate, and the environment it is exposed to.

A stress-strain curve for a semicrystalline polymer is displayed in Fig. 1. In polymers there are both ductile and brittle modes of fracture possible. They also can experience elongations greater than 1000%. The impact the strain rate has on the material cannot be emphasized enough. In fact decreasing the rate of deformation has a similar effect on the stress-strain curve as increasing the temperature. The stress-strain curve depicts a different type of behavior than seen in metals. Once a small neck forms in the gauged section the molecular chains become oriented; this means that they align parallel with the elongation direction. This inhibits deformation and the neck propagates along the gauged section.

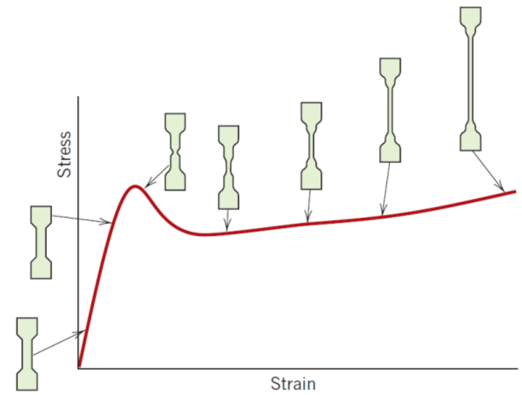



*Figure 1: Tensile stress–strain curve for a semicrystalline polymer[1]*

As seen in Fig. 2 three very different types of stress-strain behaviors are possible.  Curve A is a brittle polymer that fractures as it is deforming elastically. Curve B is a plastic polymer; this material experiences an elastic region before yielding. This is followed by plastic deformation and then fractures. Sometimes fracture occurs at a greater stress than the yield stress. The material for Curve C is an elastomer type material; these experience large recoverable strains at low stress levels.  The rest of the material background was concerned with the deformation behaviors of elastomers and how they are formed[1].

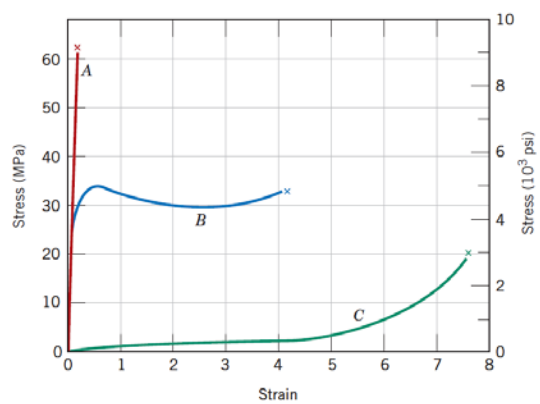



*Figure 2: Stress-strain behavior for a polymer (A) brittle (B) plastic (C) elastomer[1]*

### 3.1.2 Elastomers

The gasket material that will be tested in the biaxial test fixture is a nearly incompressible elastomer; that is capable of handling high temperatures. Elastomers have the ability to achieve large deformations and then elastically spring back into shape. The modulus of elasticity is quite small and varies linearly since the curve is no longer linear.

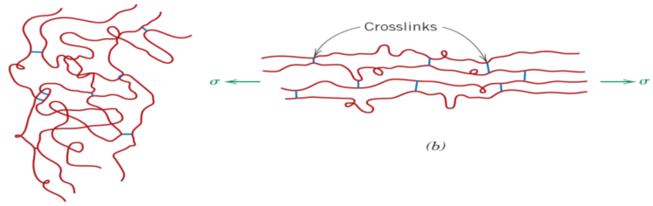


*Figure 3: Representation of the crosslinked chains molecules found in polymers, and how they react with applied stress[1].*

Fig. 3 shows the crosslinked polymer chains that makeup the structure of elastomers.  While unstressed the crosslinked chains are coiled and kinked; once a stress is applied the elastic deformation occurs by the straightening and unfurling of the chains. When the elastomer is released the coils snap back into the original shape.  There are several criterions that a material must possess if it is to be considered an elastomer. First, it cannot easily crystalize. Second, the chain bonds must be relatively free to move and respond to the applied force. Third, for the elastomer to experience the huge elastic deformations that they do the plastic deformation must be delayed.

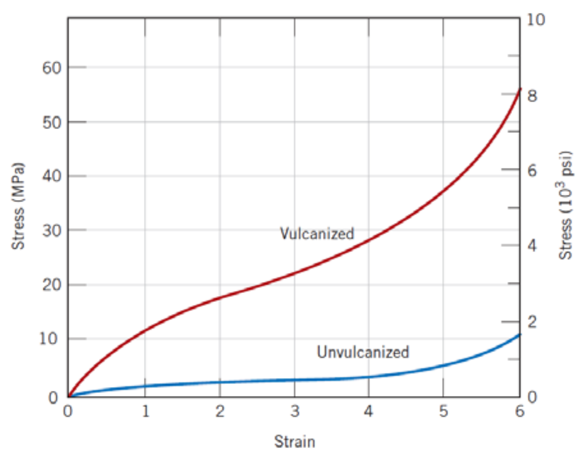


*Figure 4: Stress-strain curve to 600% elongation for natural rubber both vulcanized and unvulcanized[1].*

Crosslinks are formed by a process called vulcanization. This is normally an irreversible chemical reaction that is carried out at elevated temperatures. Vulcanization enhances the tensile strength, modulus of elasticity, and resistance to degradation. Fig. 4 shows a stress-strain curve for vulcanized and unvulcanized rubber[1].

### 3.1.3 Material Testing

To accurately predict the behavior of elastomers, three basic component properties are needed. The first is uniaxial tension data; this can be gathered quite easily by a standard tensile test. The second need is to understand how the material will behave in pure shear. It has been found that a planar tension test will produce shear values with excellent accuracy. The third property component is uniaxial compression. These results have proven to be highly inaccurate in the standard compression test due to the frictional effects between the specimen and the loading plates. In fact, when the compression specimen was analyzed a mixed state of compressive, shear, and tensile strain were found present. Friction is the main obstruction in gathering accurate data for the elastomers; because friction is a function of the normal force it increases as the compressive load increases[1].

The need for compressive data, for proper modeling, is the driving force for the development of the biaxial tensile test fixture. An equibiaxial test fixture can be used to achieve pure compressive strain. As a specimen is pulled in equal tension along the entire perimeter a special case of Mohr's circle is formed and the stress state becomes a point circle located on the stress axis. This eliminates the resulting shear forces seen in the traditional compression test. The gasket material is nearly incompressible, and Poisson's ratio is close to 0.5. During this test the specimen remains a constant volume. As seen in Equation 1; Poisson's ratio is the ratio of lateral and axial strains[2].

$$\upsilon = -\frac{\epsilon_z}{\epsilon_x} \tag{1}$$

This allows for compressive strain to be measured while pulling in equibiaxial tension. As seen in Fig. 5 the specimen is pulled in the horizontal direction causing the lateral height to change. This is change in height is

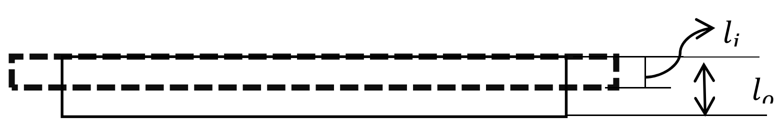



*Figure 5 A schematic representation of the loading effects for an incompressible elastomer.*

equivalent to the compressive strain the elastomer experiences. It also eliminates the frictional effects in the standard uniaxial compressive test.

#### 3.1.4 Gasket Material

The objective for this project is to accurately produce radial stress and strain values for a range of gasket material. The basic properties for some of the given material are found in Table 1. In addition to these materials rubber and paper gaskets will also need to be tested. Even though the tensile strength for the materials is relatively low; the elongation is much greater than that of metals. The binder material has been chemically enhanced to achieve desired properties, such as temperature tolerance and better resistance to degradation. They are typically in the family of rubbers. The material ranges in its sealing properties and the designated uses[6].

#### 3.1.5 Finite Element Models

Finite element analysis is a computer modeling protocol that predicts the behavior of materials of a given composition and geometry based upon their material properties. The gaskets which are the focus of our project are rubber and rubber coated fiber materials which are generally analyzed as incompressible or nearly incompressible. There are great challenges in analyzing these types of materials. Their properties and performance are much less well understood than the properties of other materials such as metals, plastics and fibers. Many different mathematical models have been developed to attempt to predict their behavior. The most common are the Neo-Hookean, Mooney-Rivlin, and Ogden models. However, there are even many subsets within these models. Fortunately, they all require the same data in order to achieve the best results.

*Table 1: Gasket Material Properties*

| | MP-15 | N-8092 | TS-9003 | Standard of Testing |
|---|---|---|---|---|
| Density, g/cc(lb/cu.ft) (min.) | 1.54 (96) | 1.20 (75) | 1.44 (90) | ASTM F 1315 |
| Compressibility, % (at 34.5MPa) | 13 - 25 | 15 - 30 | 15 - 30 | ASTM F 36 |
| Recovery, % (min.) | 50% | 35% | 20% | ASTM F 36 |
| Tensile Strength, AMD, MPa(psi) (min.) | 10.34 (1500) | 11.03 (1600) | 6.90 (1000) | ASTM F 152 |
| Binder Type | Polychloroprene | Nitrile Butadiene | Styrene Butadiene | |

Models predicting the behavior of incompressible or nearly incompressible materials all produce the most accurate results when given three fundamental data sets. These are the stress-strain relationship of the material in pure shear, uniaxial tension and uniaxial compression. Given an MTS machine it is very easy to develop the test sample geometry and proper gripping techniques to acquire both the pure shear and uniaxial tension data. Unfortunately due to frictional forces developing shear and tension forces it is impossible to get reliable data from a uniaxial compression test. Given this problem, a pure equal biaxial tension test has been developed. Due to the nature of incompressible materials if a sheet of incompressible material is radially stretched in all directions equally, the material will compress. Given the stress and strain data from the equal biaxial test one can rather easily compute the stress strain relationship for compression using Mohr's Circle. In order to get the best data it is important to induce a pure stress state in as large of an area as possible.

# 4.0 Design and Analysis

## 4.1 Biaxial Test Fixture Design

In producing a viable design, several different approaches were examined. There were two main theories used in designing. The main choice was to have the design self-driven or to have it integrated into an existing MTS machine. The first design incorporated linkages. This design was ultimately ruled out due to the inability to alter the linkage without difficulty. However, the extreme advantage to this design is that linkages are very easy to design and machine. Also, the linkages would theoretically be equal all the way around the baseplate allowing for equal strain at all desired locations.

This is what then led to the development of our second design, a system integrated with pulleys to facilitate the straining process. This design was much more desirable in the sense that cables are much easier to work around than linkages. The main drawback, however, is that the cables needed to be the exact same length or else there could be unequal tension around the baseplate which leads to unequal strain around the specimen. If not addressed, this problem could lead to inaccurate data and meaningless results.

The last possibility that had to be explored was a system with actuators and load cells at every location. This design offers many opportunities for collecting different types of data. The difficulty with a standalone system is the cost. The cost for the linear actuators and load cells alone exceeds our budget by $300 (excluding shipping). It would also be much more complex due to the need for programming to control the actuators and finding a way to reduce noise in the sensors, unless we broke the bank and found sensors that came that way. Whoever is conducting a test with the device would also need to be trained in how to operate it. Also, if something on the system went wrong, it would be much more expensive to replace an electrical part as opposed to a mechanical part such as in the other two designs.

With all of this in mind, a simple decision matrix was constructed with the critical criteria for our system. It can be seen in Table 2. This shows that a standalone system is out of the question. The other two, however, are very comparable with the pulley design barely edging out the linkage design. So the final design concept choice came with speaking to our sponsor and deciding what he would rather see in the lab at Cummins, Inc. He liked the pulley design the most but made some very valid points and alterations to the design to make it much more reliable and viable for the intended application.

*Table 2: Decision Matrix of Three Design Concepts*

| Decision Factors | | Linkage Design | Pulley Design | Stand Alone Design |
|---|---|---|---|---|
| Criteria | Wt. | 1 | 2 | 3 |

| | | | | |
|---|---|---|---|---|
| Ease of Use | 3.0 | 2 | 4 | 1 |
| Machinability | 4.0 | 5 | 4 | 2 |
| Complexity | 2.0 | 5 | 5 | 1 |
| Cost | 5.0 | 4 | 4 | 1 |
| **Weighted Scores** | | 56.0 | **58.0** | 18.0 |

After the selection of a pulley design, there was some back-and-forth with our sponsor to come to our final design. Items that affected this were the machinability of the device, the required specifications to be able to test the gasket materials, and also the cost of the device as a whole. These are discussed further in Chapter 6 of this report. The final machine drawings for our device can be found in Appendix A.

# 5.0 Prototype Details

## 5.1 Machining

The machining for this design was done at the College of Engineering machine shop and also at the NHMFL machine shop. In coordination with the workers at the machine shop, changes were made to our design to ease the process of machining. In total, there were thirty-seven parts to be made and not enough stock material to remake any parts, so there was a conscious effort to solidify the drawings before turning them over to the machine shop.

## 5.2 Assembly

Assembling the bi-axial test fixture requires only a few basic tools. Before beginning assembly make sure to have:

-A set of SAE hex keys

-A set of SAE box wrenches

-A straight edge

Step 1: Carrier Assembly.

If the cables are not yet assembled, cut the cables to 12in., slide the cut end of the cable through the hole in the top of the carrier so that the crimped end ends up on the side facing the clamping area. Then, attach the cable end according to the instructions supplied with the cable end. Next, slide bushings into the carrier and secure with set screw. Repeat this procedure for each of the eight carriers.

Step 2: Center Plate Prep

Locate the center plate (the ½" thick plate, with many more holes than the other two plates. Support the plate with blocks so that you can use a wrench underneath the plate with the side of the plate with the blue alignment lines facing upward.

Step 3: Center Support

Attach the center support (octagon shape piece with two holes on each face) to the center of the plate using the two hex cap bolts and matching nuts. Make sure that the cap bolts slide into the counter-bored holes in the top of the support. Finger tighten only.

Step 4: Linear Rail Assembly

Locate 2 12” linear rails, one carrier-cable assembly, one exterior support, and two 3/8” x 4 ½” bolts with one nut and two washers for each bolt. Slide the linear rails into the center support, and then slide the carrier onto the linear rods. Next, push the exterior support over the two linear rods, making sure that the correct side of the support is facing up by checking that the rods are parallel to the plate. Secure the exterior support using the 3/8 x 4 ½ bolts with the head of the bolt and a washer on top of the exterior support and a washer and nut on the bottom. Finger tighten only. Your assembly should now look like this:

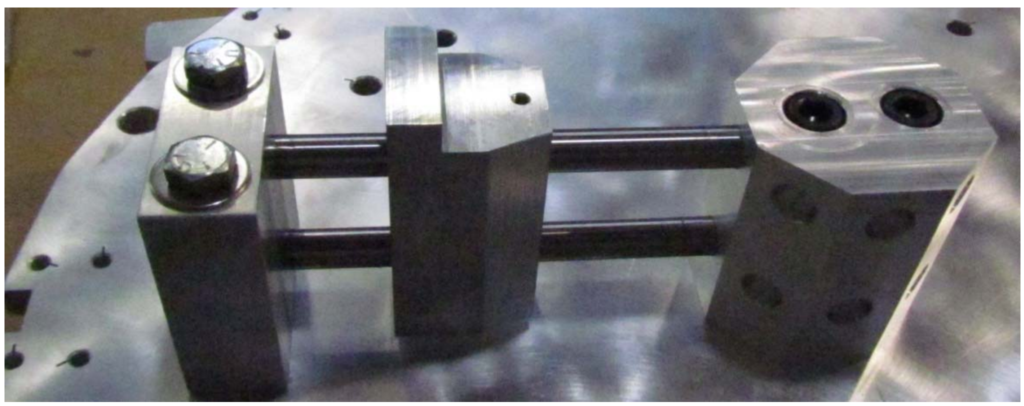


*Figure 6: Carrier on Linear Rods*

Repeat eight times.

Step 5: Alignment

Place a straight-edge against the linear rods and down to the plate, move the exterior support so that the straight edge is 6mm from the blue alignment line. Tighten the exterior support bolts. Repeat for all 8 linear rail assemblies. Once the linear rails have all been secured in their proper places tighten the hex cap bolts of the center support.

Step 6: Pulleys

Gather one pulley, two pulley blocks, and four ¼” x 3” hex cap screws with one nut and two washers each. Place the blocks behind the exterior supports over the through holes in the plate. Place the pulley on top of the blocks. With a washer at the top, slide all four of the hex cap screws through the pulleys, blocks and plate. Once all four bolts are inserted, tighten the nuts down with a washer between the nuts and the plate. Repeat 8 times.

Step 7: Threaded Rods and Top Plate

Gather the eight threaded rods, along with 24 nuts and 20 washers for the rods. Also gather the four square aluminum alignment tubes and the top plate (the ¾" plate with the larger holes). Thread a nut approximately 1.5" onto each of the threaded rods place a washer over every other outside hole in the center plate. One at a time, slide the end of the threaded rods with the bolts through the plate and secure with a bolt and washer on the bottom of the plate, making sure that only a couple of threads are exposed below the bolt. Slide the aluminum alignment tubes over the four threaded rods that do not have washers on top of the plate. Place the top plate on top of the rods and slide the rods through the outside holes. Secure each rod with a nut and washer on top of the top plate. Take care not to excessively torque the nuts on the threaded rods without the alignment tubes to the point where they may warp or crack the top plate (more than 50 ft-lbs).

*Note that this is the furthest assembly should progress if you are not integrating to a Uniaxial Test Machine. Attaching the cables to the bottom plate outside of the test machine may result in severe damage.

Step 8: Bottom Plate

Gather the bottom plate and the hardware you choose to attach the plate to the pneumatic piston of the uniaxial test machine (the plate has a 1" hole in the center so a 1" threaded rod and nut may be used, assuming that is compatible with the piston's attachment method, attachment methods may vary from machine to machine but a 1" threaded hole is normal).

Step 9: Integrating the Top Assembly

Place (3) 2”-3” tall support blocks on the bottom plate and raise the piston to its maximum height. Raise or lower the crosshead so that the bottom of the load cell will have ¼” of clearance when the top assembly is placed on top of the support blocks. Lower the piston at least 6 Inches. Place the top assembly on the support blocks taking care not to pinch or twist the cables. Slowly raise the piston and attach the top assembly to the load cell using your choice of hardware. Lower the piston and remove the support blocks. Raise the piston until it is close to the top assembly but not touching. Attach the cable ends to the bottom plate with one nut and washer above and one nut and washer below the bottom plate. Lower the bottom plate until the carriers are touching the outside supports, but the cables are not pulled tight. Measure the distance between the bottom plate and the machine and install blocks that will prevent the bottom plate from over extending the device when the test machine is turned off, set a displacement maximum so that the machine does not overextend while the test is running.

The Equal Biaxial Test Fixture is now fully integrated. Proceed to the operations manual for instructions on running a test.

## 5.3 Testing

### 5.3.1 Uniaxial Tabletop Data

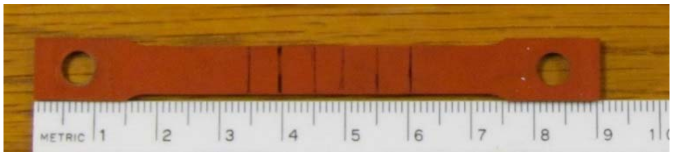


*Figure 7: Red Rubber in Dogbone Geometry*

*Figure 8: Black Rubber in Dumbbell Geometry*

In order to validate our test fixture, there was a need to obtain uniaxial test data from a tabletop MTS machine to compare to uniaxial data that we would obtain from the test fixture. This was done by laser cutting the rubber materials to a specific dog-bone shape developed from the ASTM standard for metals. A dumbbell sample geometry was later used because the dog-bone samples began fracturing in the grips instead of in the gage section. The differences can be seen in Figures 7 and 8.

The dumbbell shape was ideal due to the greater grip width to gage width ratio of 2.2 compared to that of 1.27 in the dogbone samples. These graphs are then compared to those that we obtain in the uniaxial testing done in our test fixture and compared.

There were several gasket materials tested as well. They were tested using a smaller dogbone punch developed for a previous senior design project. The geometry can be seen in Figure 9. The samples were cut at increments of 45 degrees because the gasket materials are notoriously anisotropic.

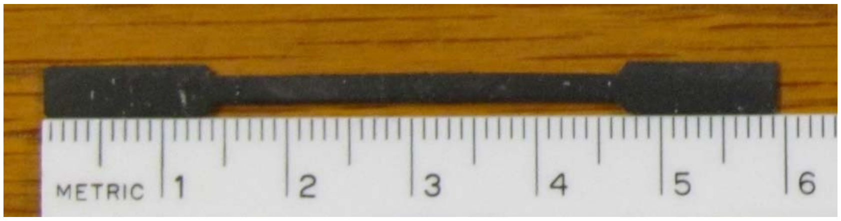



*Figure 9: Dogbone Geometry of C5270*

### 5.3.2 Uniaxial Proof Testing

In order to show that our device was functioning properly, and to calibrate it for friction, we ran several uniaxial tests with different materials. Our goal was to show that each of the carriers behaved in the same way and to correlate that data with the tabletop MTS uniaxial testing. To this end, we tested the samples and gathered Axial Force to compute stress, and used a microscope to record scaled videos and pictures in order to compute strain and stretch.

The microscope is called a Dino Scope. It utilizes a 0.3 megapixel camera, but it has a large magnification so that small number of megapixels can be utilized in a 3 inch by 3inch area, giving a fairly clear picture. The camera connects to a USB port and has associated software. The software allows you to take video and pictures. It also features a drawing capability that allows you to draw lines and shapes to scale on the pictures. The initial lines stay on the pictures, and you can draw more shapes later so that you can measure changes in the distance between gauge marks drawn on the actual sample. Figures 10-12 below show how the pictures appear in the program at 100%, 150% and 200% stretch.

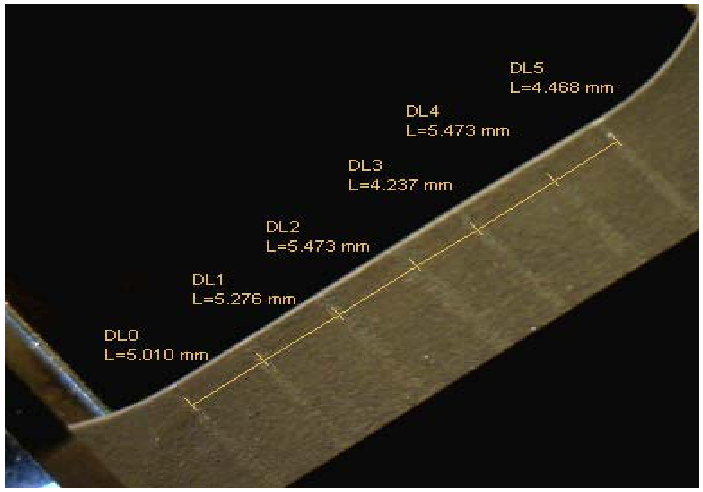



*Figure 10: Uniaxial Sample at 200%*

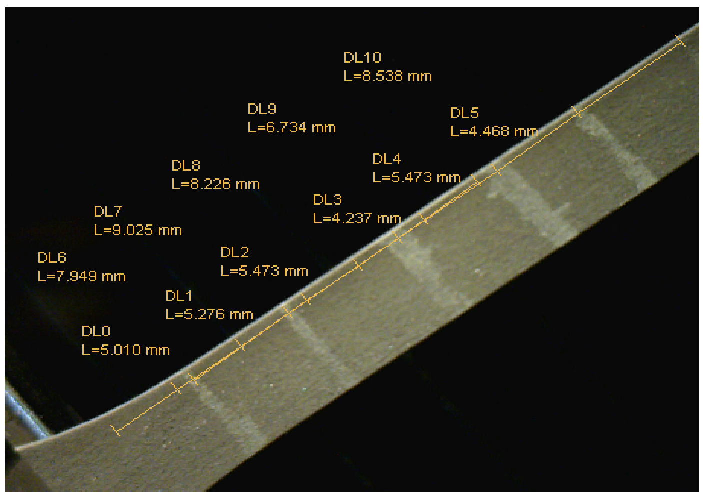



*Figure 11: Uniaxial Sample at 150%*

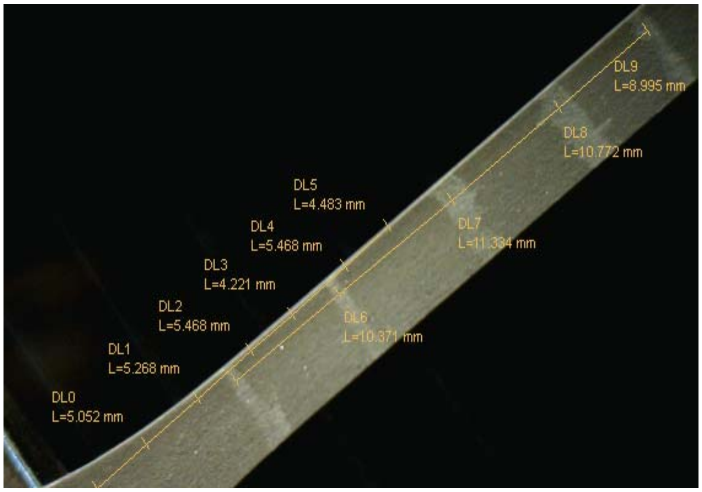



*Figure 12: Uniaxial Sample at 100%*

These measurements can then be manually input into an excel file. A graph of the results for the red rubber samples can be viewed in Section 5.4.2.

### 5.3.3 Biaxial Testing

The point of our device is to be able to pull equally in all directions. In order to show that our device was capable of achieving this we inserted laser cut sample specimens into the carriers and tested. We were able to pull some of the samples to failure. We were able to get screen captures at the correct points to show that they did indeed deform equally to over 200% stretch. Failure occurred sometimes across the reduced gauge sections of our sample (likely due to too high of a strain rate) and sometimes it failed at the relief geometry and across the center of the sample. Further FEA analysis of the sample geometry may result in even higher stretch percentages, but the sample design we employed went beyond the goal of our project.

## 5.4 Analysis

### 5.4.1 Uniaxial Tabletop Data Analysis

Testing all of the uniaxial samples in the tabletop MTS gave us a good idea of the stretch and load to anticipate during the calibration of our machine.

This was recorded in a load versus displacement curve. This was then converted over to a stress versus stretch plot. The stress was found using Equation 2 while the stretch was found using Equation 3.

$$\sigma = \frac{F}{A} \tag{2}$$

$$\lambda = \frac{l}{l_0} \tag{3}$$

The data, after being converted, can be seen in Figures 13 and 14. As it can be seen, this material is fairly isotropic. This is important because this is the only way our device can be validated uniaxially. We have no way of measuring load in different directions because we don't have a load cell at every carrier location. Figure 15 shows the graph of the uniaxial test for C5270 gasket. This is also reported in stress versus stretch due its composite nature. This graph proves that the gasket material is very anisotropic. The stress is more than double that in another direction, leading to the observation that the material properties differ in different directions.

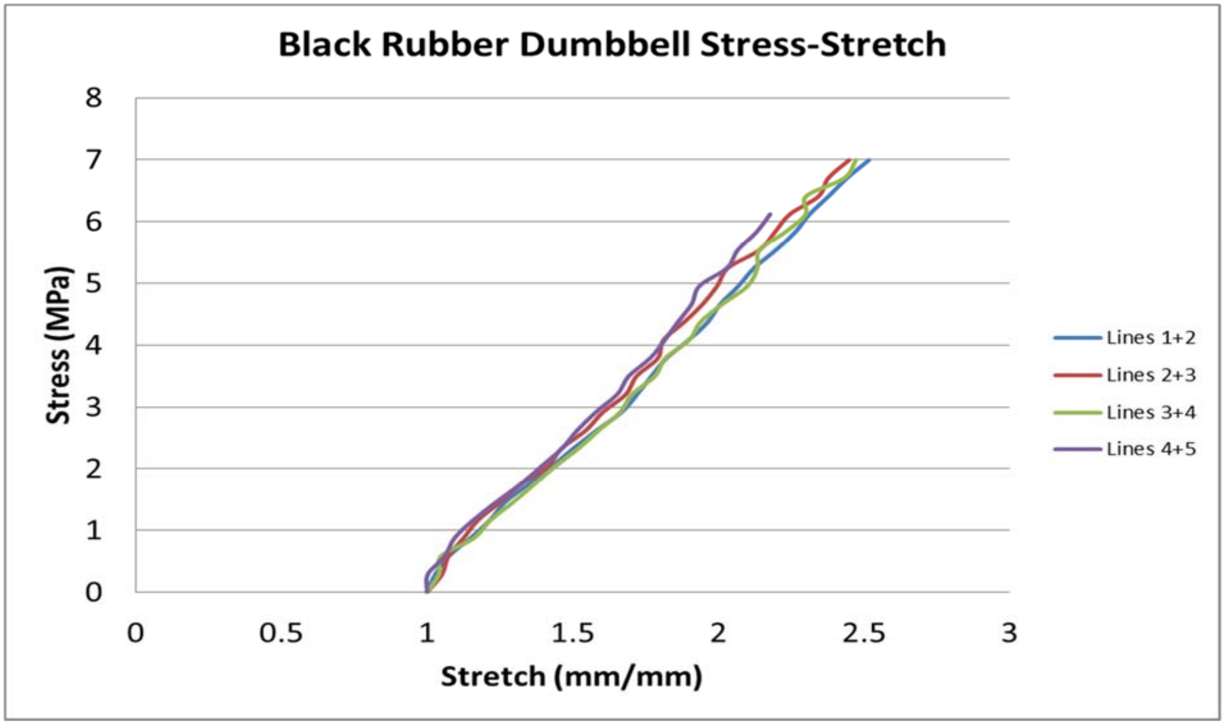



*Figure 13: Stress-Stretch Plot of Black Rubber*

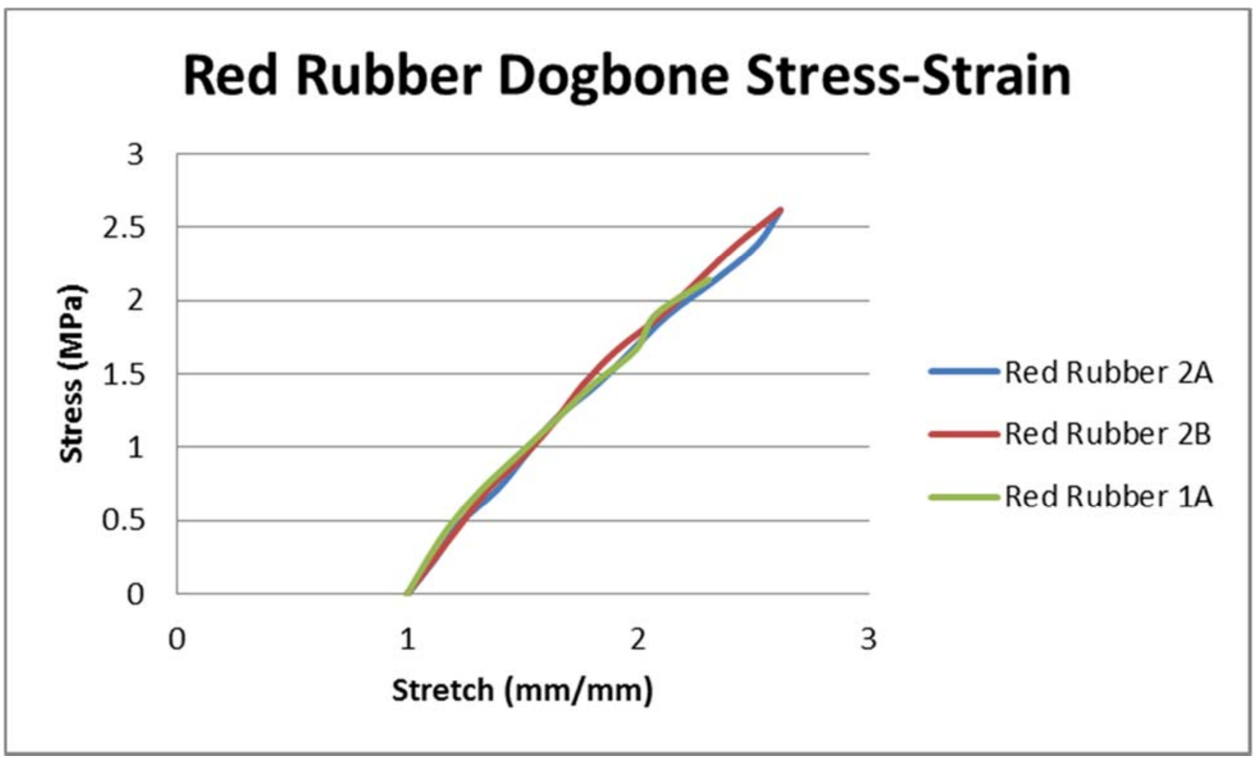



*Figure 11: Stress-Stretch Curve of C5270*

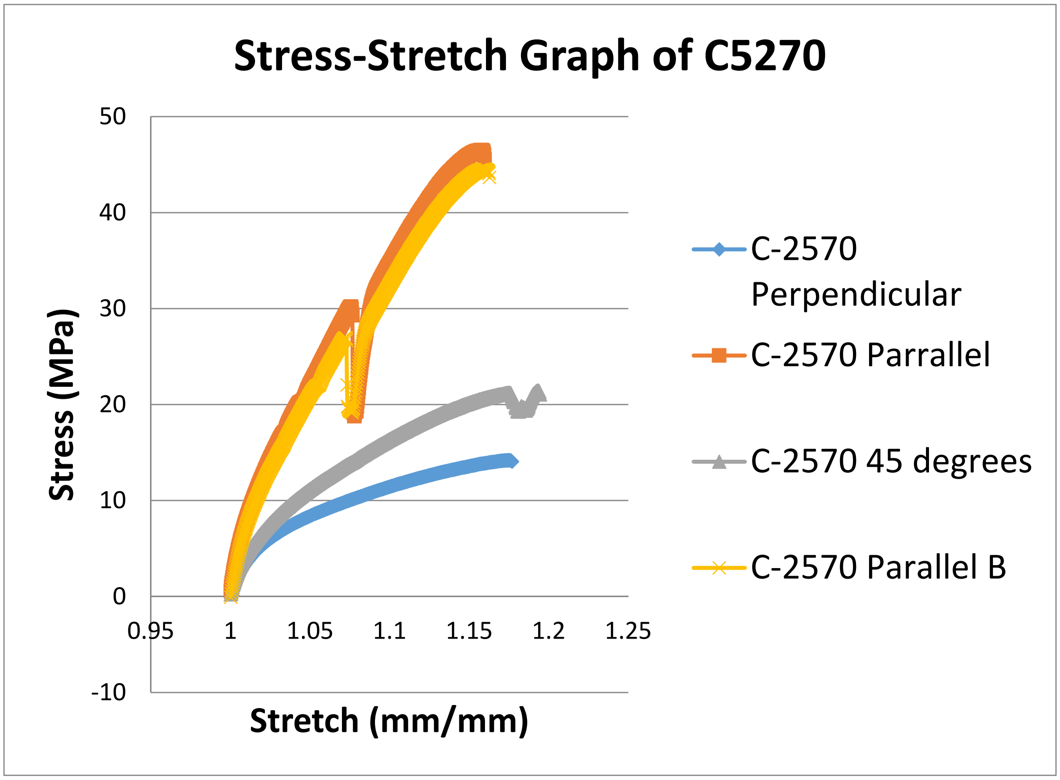



*Figure 15: Stress-Stretch Plot of Red Rubber*

### 5.4.2 Uniaxial Proof Testing Analysis

After testing the uniaxial samples we analyzed the data in excel in the same way as with the tabletop data, except using the dino-scope measurements to calculate the stress-stretch relationship of the red rubber on each of the carrier pairs. As you can see below in Figure 16 we achieved excellent agreement between the carrier pairs with a small margin of error likely due mostly to errors in the strain measurements with the dino scope.

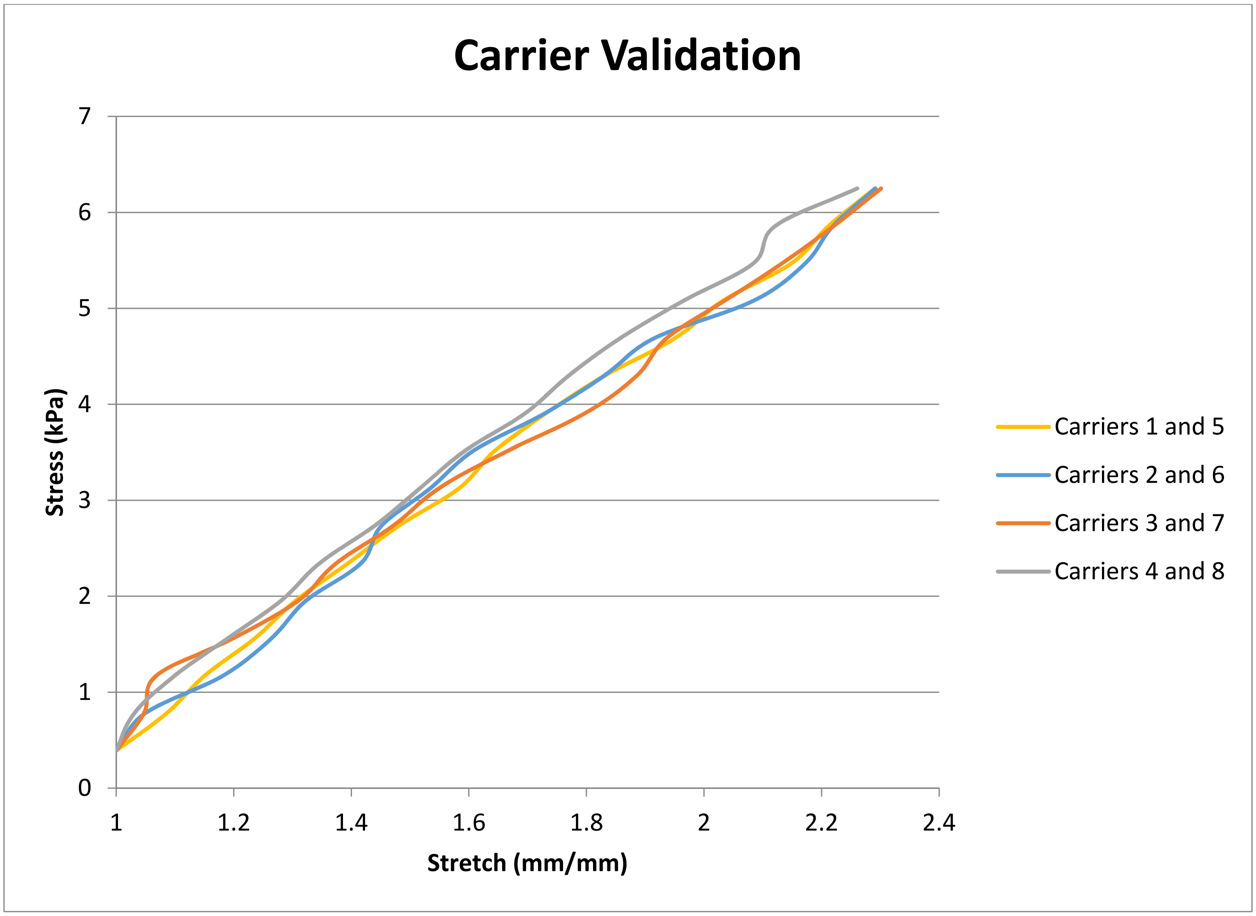



*Figure 16: Stress-Stretch Plot of Red Rubber on carrier pairs*

### 5.4.3 Biaxial Testing Analysis

Using the pictures from the dino scope we were able to show that the sample was stretched beyond our goal of 100% strain. The specimen also stretched equally in the center, creating a region of pure equibiaxial strain. When measured with a better strain measurement device, like digital image correlation, will provide accurate data to produce a stress-strain relationship for compression.

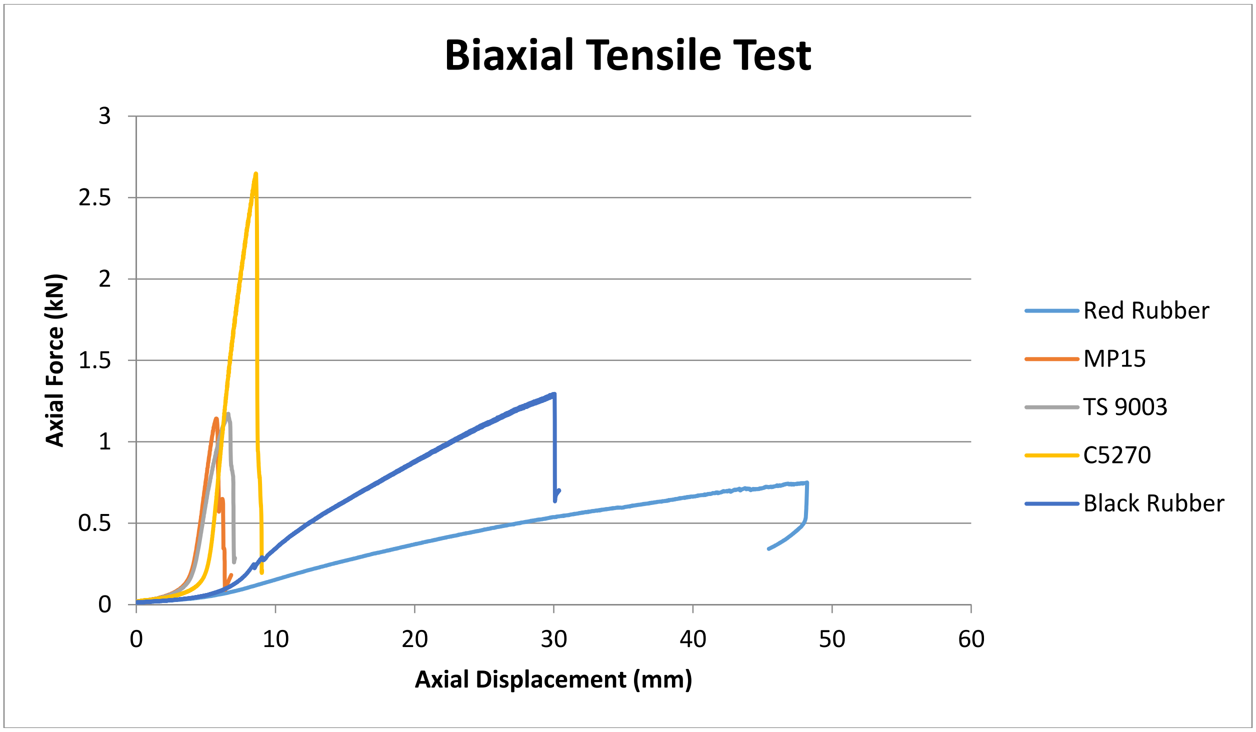



*Figure 17: Force-Displacement relationship of several materials in biaxial tension*

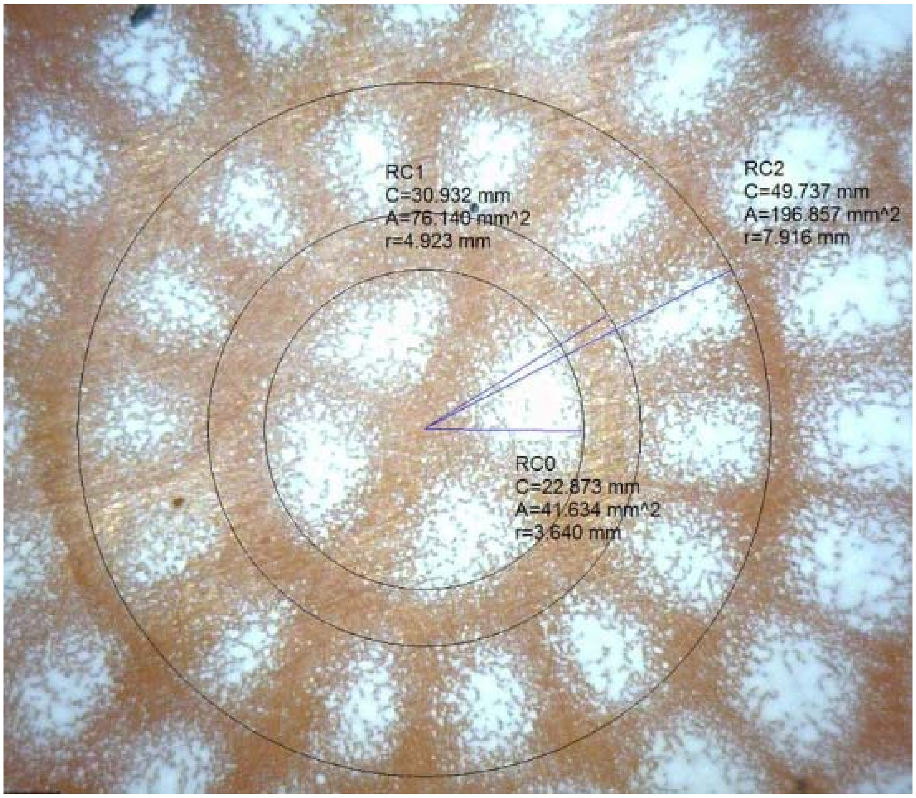



*Figure 18: Measurements of stencil locations with no force applied.*

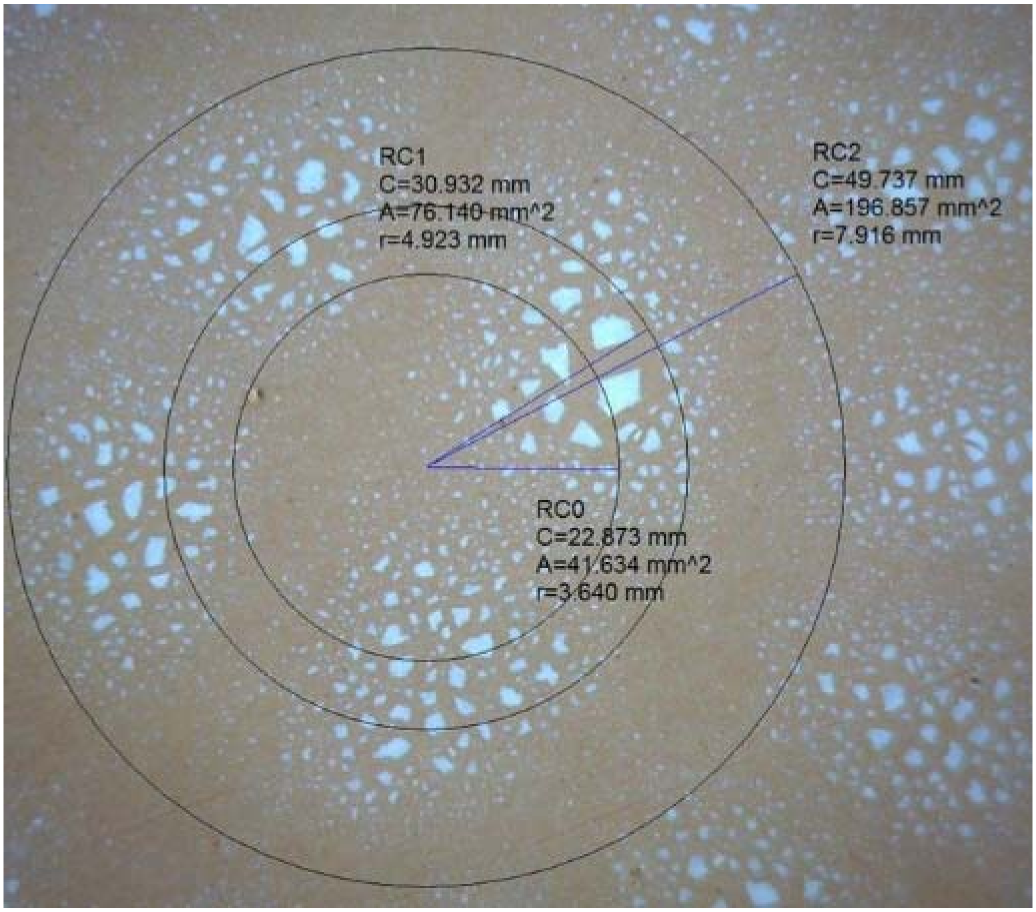



*Figure 19: View of the sample near failure, the circles are in the same locations as figure 6, it is the sample which has deformed to well beyond 200% stretch.*

Using Equation 3:
From Figure 18 Ao = 41.63 mm^2 and 76.14 mm^2
From Figure 19 A= 143.14 mm^2 and 213.82 mm^2
Stretch for inner 344%
Stretch for outer 281%

Given the nature of the stencil being difficult to measure at this second extension, the error is relatively minor and we have proven that our device can go beyond its goal of 200% stretch and reach 300%.

# 6.0 Design for Manufacturing, Reliability, and Cost

## 6.1 Design for Manufacturing

In the designing of the parts for our device, the machine shop was consulted for guidance on what should be done to ease the manufacturing process and get our parts done as quickly as possible. The main hurdle was how to get the complex parts completed when we had eight times the machining. This was done on the CNC machine. This cut down the machining time drastically and relied solely on the parts being done exactly how we wanted them to be done on Creo Parametric 2.0.

Along with the CNC machine, a water jet was used for a large portion of our parts. The parts were designed with the water jet in mind. This eliminated the need for a person to machine by hand and relied on the tolerances of a calibrated machine. This greatly reduced the amount of time needed to machine those parts because eight of each part could be water jetted in just a couple hours when it could take a person over a day to be able to accomplish the same task at equal tolerances.

The carrier was later modified to fit the end of the cable through it. The machine shop was again consulted to determine what the easiest method would be. We came to the conclusion that a simple hole through the top of the carrier would be the simplest for them while also accomplishing what we wanted out of the part. This drawing, as well as all the others, can be found in Appendix A. Along with the drawings, the bill of materials can also be found in Appendix A.

The materials selected were solely based off of cost and intuition. There wasn't a very ductile or very hard material needed because of the nature of the materials to be tested. With this in mind, we decided to go with aluminum 6061 for all of the parts except for the steel rods and the baseplate. The baseplate was made out of cast aluminum to reduce the amount of deflection seen while testing. This wasn't necessary, but it just increased the safety factor of the machine as a whole. The hardened steel rods were chosen so that no deformation was seen in them over time of testing samples. We wanted to build this device to be able to withstand years and thousands of samples all while just requiring an occasional recalibration.

## 6.2 Design for Reliability

### 6.2.1 Overview

Our machine was designed for the testing of incompressible (non-metallic) gasket material. We found that the strongest gaskets made for any application have a tensile strength that would equate to a little under 500lbf in each of the eight cables, however, the examples of gaskets that we were provided did not exceed 150lbf in each cable.

Our machine's design incorporates extremely robust permanent components designed so that not only can they withstand the loads that they are designed for, but they can do so with a deflection of less than .005".

The weakest components of our design are the cables and pulleys. Each pulley is rated for 700lbf, which is well in excess of any material that it is designed to test, and is also above the 650lbf that the bike cables were tested to. The bike cables broke at 650lbf, but they exited the elastic region and entered a region of plastic deformation at 475lbf. The final weak point is the cable ends, they are rated for 550lbf.

If desired the pulleys, cables and end ties could be upgraded, but stronger cables would be less ductile and require a larger preload.

### 6.2.2 Maintenance Items

Only three components of the machine are candidates for failure in any reasonable time frame.

The linear bushings and pulleys could be subject to wear from contaminants. Storing the machine in a clean, dry area and wiping the rods down before initial use after sitting for a long time should mitigate any potential long term issues. The bushings and pulleys should be inspected for excessive wear whenever a cable is changes.

The component of most concern with regard to maintenance and failure is the cables. Before each run, both ends and the length of the cables should be inspected for fraying, broken strands and unwinding, special attention should be paid to the end ties. If a cable needs to be replaced, it should be proof tested to at least 1kN. Special attention should be paid to every test for possible deficiencies in the cables' elasticity. If a deficiency is suspected, the suspect cable and at least two other cables should have their elastic modulus determined using a uniaxial tension test, if any cable is found to be defective than all cables should be tested.

### 6.2.3 Possible Failures and Associated Risks

If operated at a force of less than 500lbf per cable, and if the linear bushings and pulleys are occasionally inspected than there is only one mode of failure, the cable. Failure of the cable does not impose any risk of damage to the operator, the machine itself or the MTS machine that it is installed in if the operator insures that no electrical connections or hydraulic lines are within one foot of the device and the operator themselves must maintain a one foot distance during operation. Most MTS machines have plexi-glass guards that ensure this safe distance. If an operator was within one foot and the cable had a catastrophic failure at a high load than the operator could be injured by the broken cable. Likewise the cable could possibly damage wiring or hydraulic lines within one foot of the machine.

## 6.3 Design for Economics

We feel, after going through this experience, that the entire bill of materials can be machined with the water jet and CNC. This both reduces labor in the machining process but also strays away from human error. The only error comes from the tolerance of the manufacturing machines. The one material change we would make is to make the baseplate out of aluminum 6061, like the rest of the larger parts. This is much easier for the water jet to cut through and also much cheaper. It also satisfies all specifications needed in the design.

Assembly is all manual labor. There is unfortunately no way for a machine to put this together without a very large initial investment. The plus to the design, however, is that the prototype testing cost is next to none. The specimens to test come from the scrap materials of the gaskets. This means that there basically no cost to Cummins because the materials are recycled. The only cost to the company is manual labor to run the tests.

## 7.0 Considerations for Environment, Safety, and Ethics

With any machine, there runs a risk for safety without the proper training. However, the risk is elevated when performing material testing. Forces are being applied to material until they fail. Although anticipated, it is hard to account for all of the possible modes of fracture that could occur. This machine will not be any more danger than operating a typical MTS machine. Taking steps to use personal protective equipment such as safety glasses, and ensuring area is clear of any foreseeable problems will decrease the chance of incident. In order to further mitigate those risks, a manual with proper test procedures will be developed.

The global scope of the project is to help Cummins acquire better data for the modeling of their gaskets materials. This project could potentially increase the performance of gaskets and have the benefit of reducing engine leaks into the environment. The tensile specimen will also be produced out of scrap material from the gaskets made in the Cummins plant. This means that our test will be performed with recycled material. There will be no environmental safety concerns because all testing will be done in a lab with the gasket materials. All scraps will be disposed of according to Cummins protocol.

## 8.0 Communications

Our group has had a weekly telephone meeting with our client and faculty advisor, Terry Shaw from Cummins, Inc. and Dr. William Oates, on every Monday since the third week of the semester. They have provided us with ideas to spark innovation and challenged us to deepen our knowledge of biaxial tension testing. We have had several meetings as needed with our graduate advisor, Parker Harwood. His knowledge of the testing practices, requirements of the device's expected performance, and experience with the specimen materials have guided us well. Within the group all members are respectful of each other's opinions, are unafraid to challenge each other with new ideas, and are always available when called upon.

## 9.0 Conclusions

Although the biaxial test fixture met all of the necessary objectives, it is not quite up to par with where we wanted to be. We were able to prove that our device pulls equally in all eight carriers while still being able to replicate data. This is the main objective because it proves that Cummins can use this device for equal biaxial tension. It will be even better if the necessary changes discussed in the next chapter of this report are made.

On the whole, we are very pleased with the final product that was produced. We were able to come together as a team, discover each other's strengths and capitalize on that to get a great amount of work done in designing the specimen geometry, learning about gasket materials, learning about biaxial tension, designing a test fixture, building the test fixture, and test samples even the completed prototype.

# 10.0 Recommendations for Future Work

While we feel very satisfied with the outcome of our design, there are a few things that can be done to really refine the design and provide better results. The first of which would be obtaining some sort of mechanism to cut out the specimens. We were in contact with a company that was able to make a rule die for this very process for only $200. This was unfortunately not in the budget for us until it was too late because we didn’t know if there were to be failures to occur which would require replacement of parts.

Another addition that is very important is the addition of guide pins to limit the extension on the device. This is merely for safety concerns because this sets a limit on where the plates can move. Right now, this is being accomplished by placing two wooden blocks under the device.

As mentioned, slip was occurring in our specimens during testing. This raises the need for either a different gripping method or just a different material touching the specimen that would have a higher coefficient of friction. The emery board that is being used now greatly increased the friction, but the specimen still slipped at some point. Another idea for the carriers is to somehow fix a threaded 10-32 rod in the tapped hole where we have been screwing into. This allows the versatility of lining everything on the rod before tightening down. You would then just need washers and a nut to tighten the upper grip to the carrier. One way to accomplish this is by using an epoxy to fix it in the hole.

There is also a need for changes to the wire rope. Ideally, there should be a load cell attached to each wire to allow a more accurate reading of the load in each eighth of the sample. There is also an option of attaching an extensometer to each wire rope to aid in the calibration process. This helps by showing how much each individual wire will stretch on every run because it will change over time.

Although the microscope is working very well for what we want to do, a more accurate form of measurement is needed to be able to validate the results and claim that the data is reliable. This can be done with digital image correlation, which Cummins actually has access to. This is achieved by speckling the specimen with paint and tracking individual speckles with a high-resolution camera.

The final additions needed to really polish this device are to perhaps add better bushings or bearings to the carriers. The acetyl bushings that were made for our design were very good, considering the price, but they still had some slip-stick during testing. There are bearings solely designing for testing equipment called miniature metric ball bushing bearings. These limit the coefficient of friction to 0.0001 for distances up to 100 km. A more compliant cable might also be needed in the future. A cable that has less stretch will give much more accurate curves because the stretch of the cable will not be reflected in the data of the test. Until then, the data just needs to be adjusted to account for the stretch.

## 11.0 Schedule, Resources, and Budget

The Gantt chart for our project can be viewed in Appendix B. The final budget is show in Table 3. Most items were either ordered from Grainger or McMaster-Carr, aside from the stock material for machining. Table 4 shows the budget for a prospective company if they were to want a machine for themselves.

*Table 3: Budget Analysis*

| Suppliers | Item Type | Cost |
|---|---|---|
| McMaster-Carr | Various Items | $733.80 |
| Midwest Steel | Stock Materials | $598.00 |
| Ebay/Folger | Linear Bearings | $28.56 |
| West Marine | Dyneema Rope | $17.74 |
| Grainger | Wire Rope | $56.93 |
| | Total: | $1,435.03 |

*Table 4: Budget for Completed Design*

| Item Cost | Cost |
|---|---|
| Original Design | $1,435.03 |
| Load Cells | ~ $6,400.00 |
| Guide Pins | ~ $200.00 |
| Ball Bearings | ~ $960.00 |
| Labor | ~ $37,800.00 |
| Total: | ~ $45,795.03 |

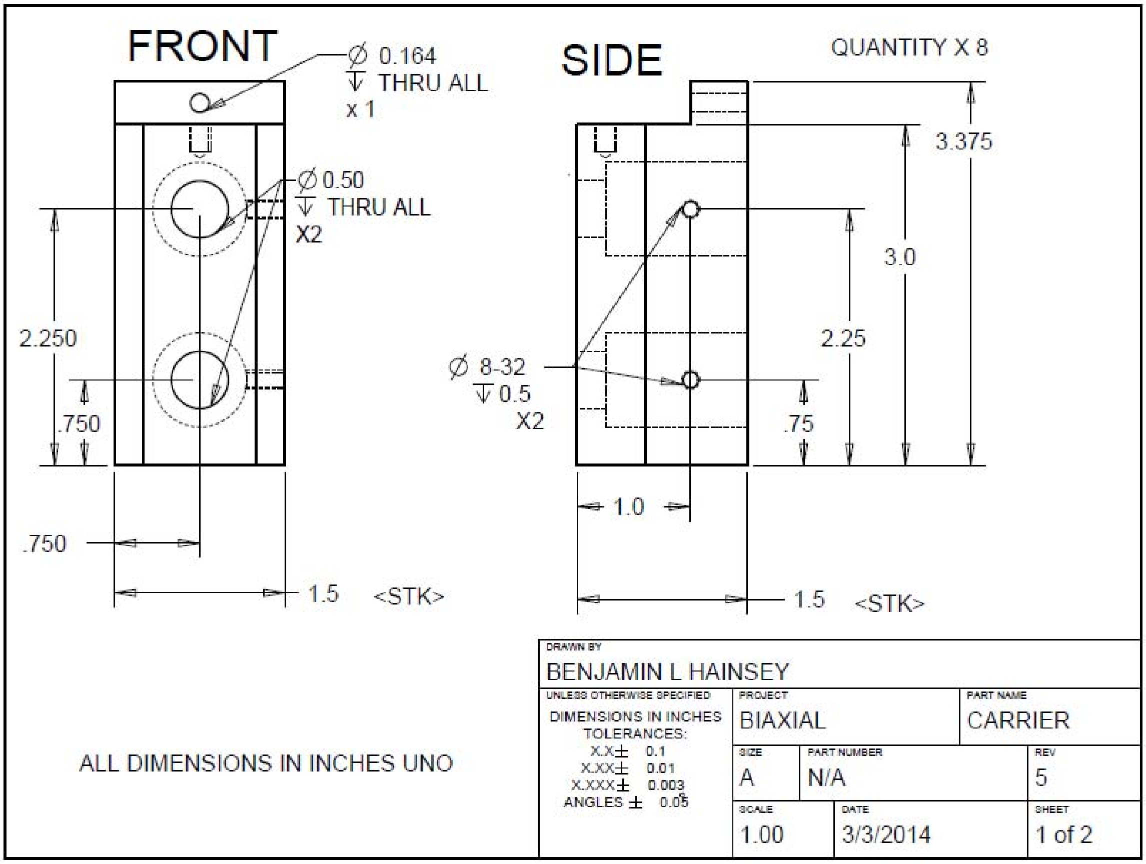
We had tremendous resources both at the NHMFL and the College of Engineering. The NHMFL provided us with many scrap materials to proof test some of our ideas before solidifying them in our design. It is also were most of the water jetting took place because it has a water jet larger than the College of Engineering. It is also where we had access to the microscope that we used to record the stretch of our samples. The College of Engineering provided us with the machine shop where seventeen of our parts were made. This was extremely beneficial because the cost of machining was free to us, so we were able to save a large amount of money by not outsourcing to another machine shop.

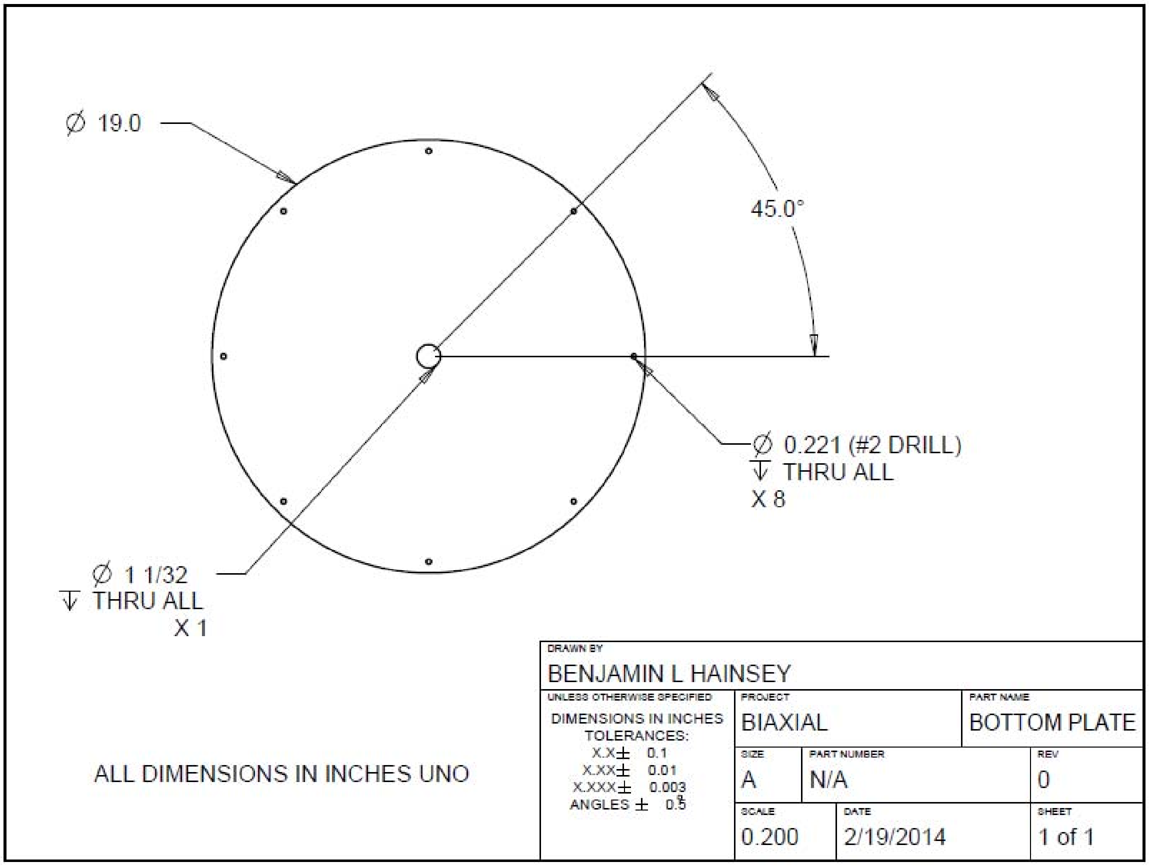
## 12.0 References

1. Callister, W.D. (2007). *Material Science and Engineering, An Introduction;* 7th ED. York, PA: John Wiley & Sons, Inc.

# 13.0 Appendices

## 13.1 Appendix A – Machine Shop Drawings

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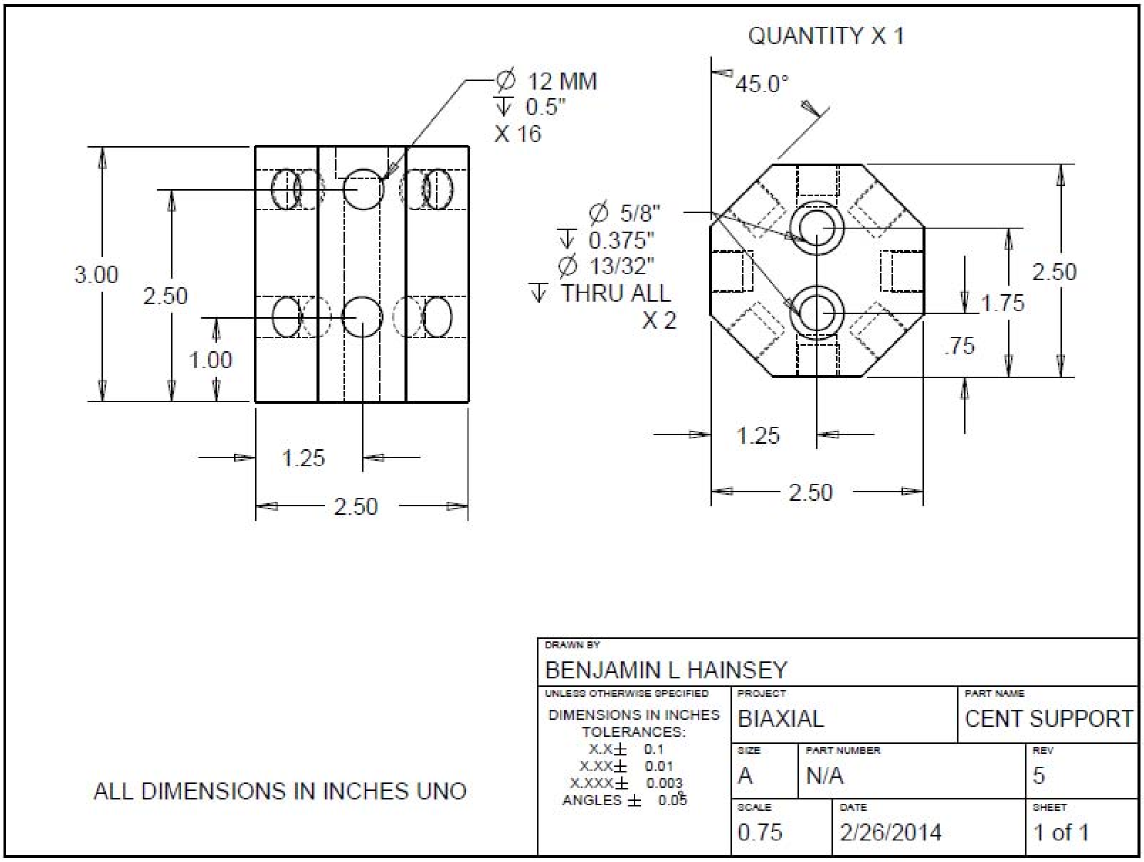
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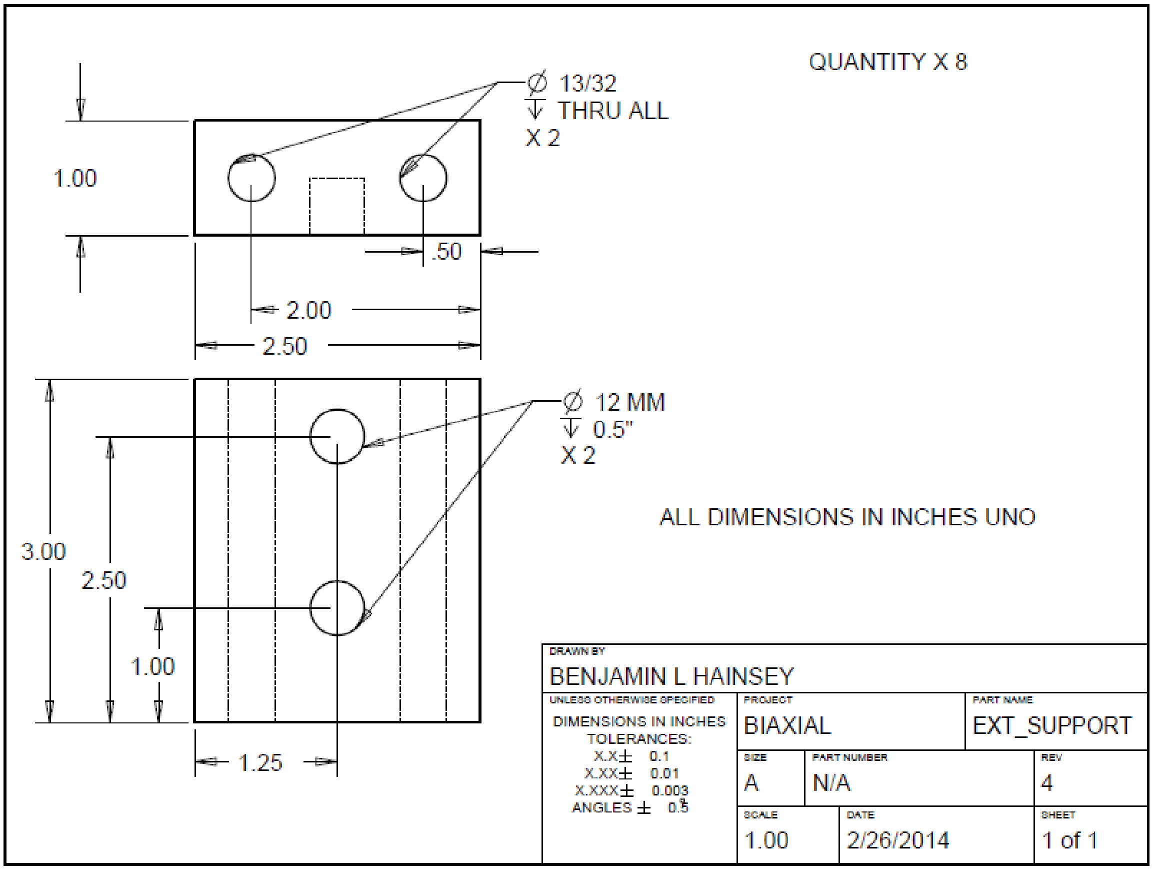
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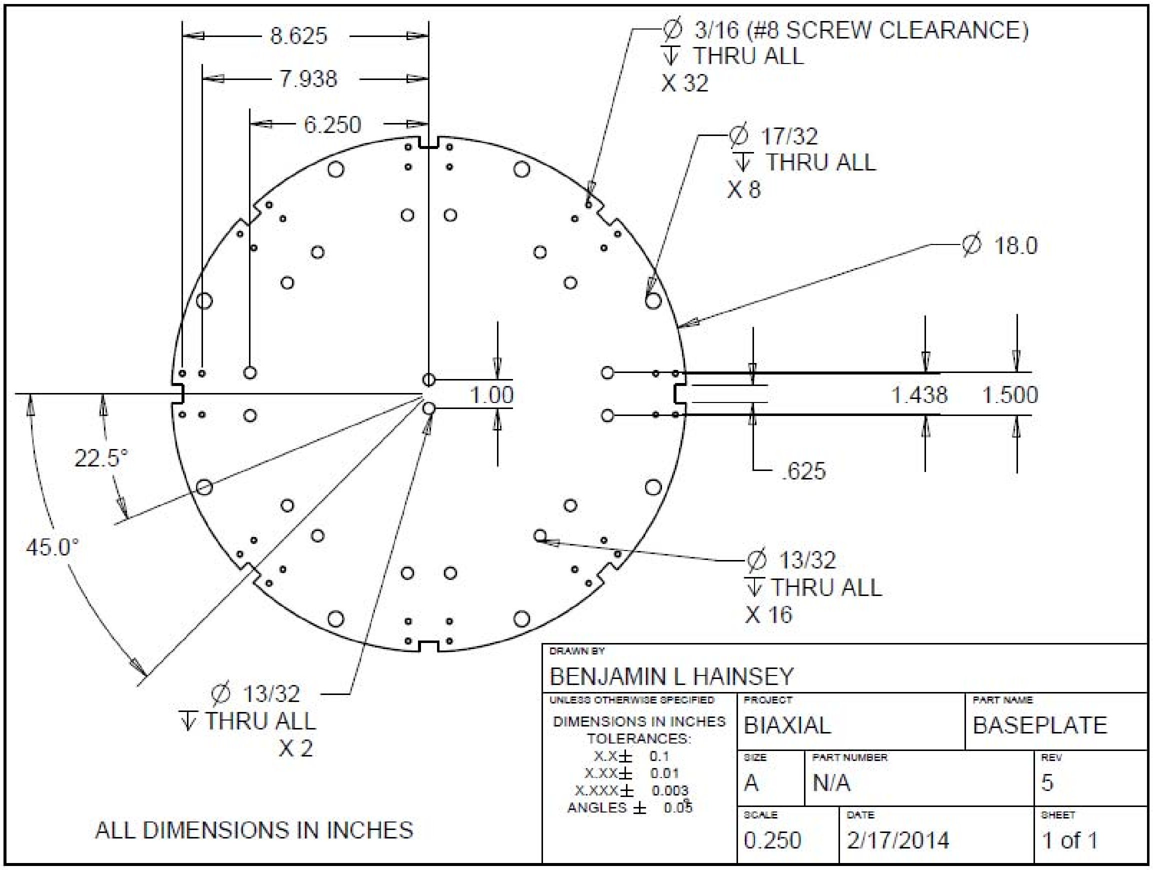
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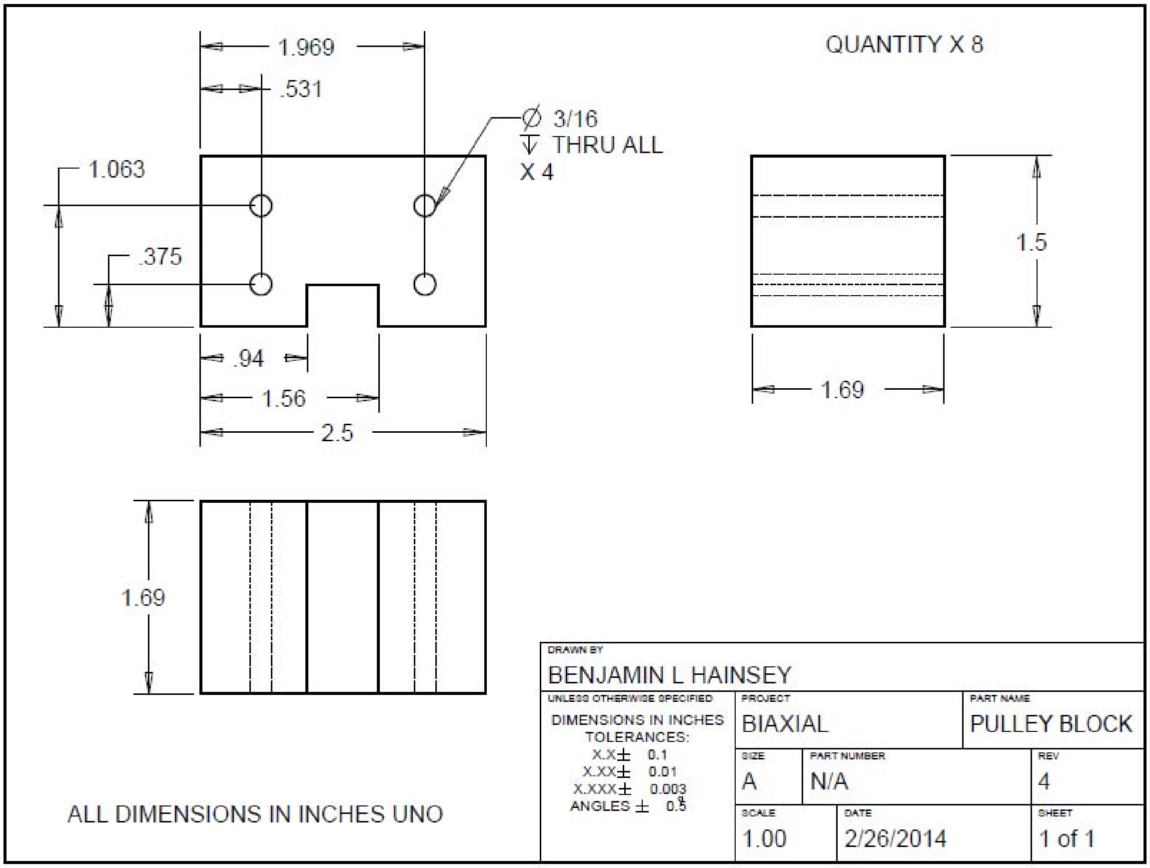
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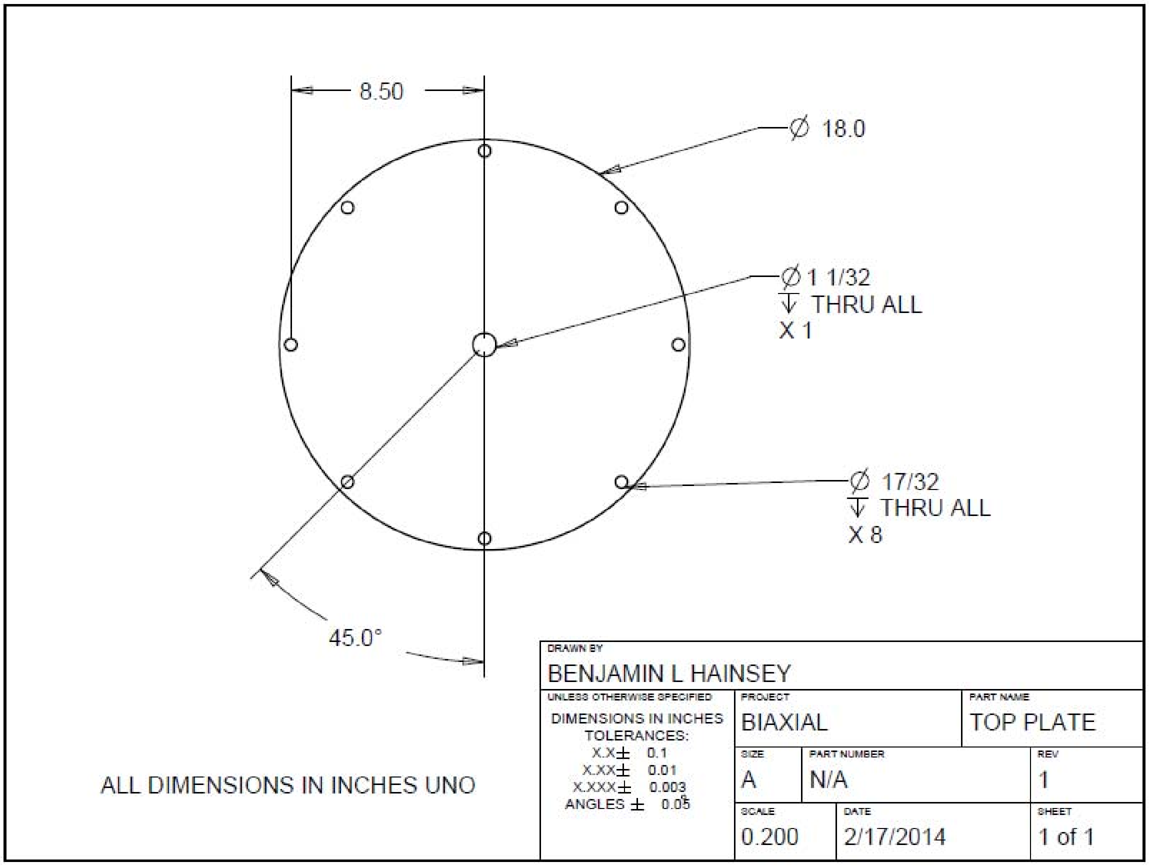
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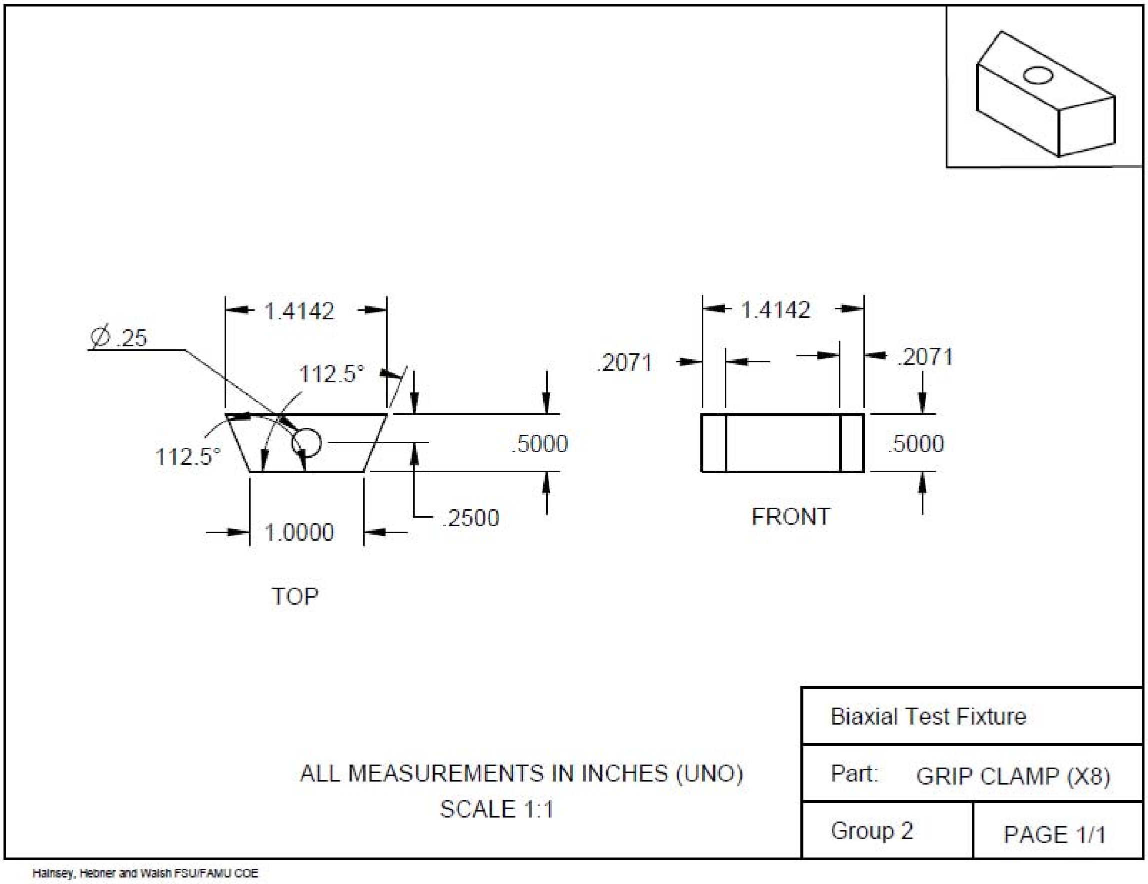


| Bill of Materials | |
|---|---|
| **Part Name** | **Quantity** |
| Carrier | 8 |
| Baseplate | 1 |
| Top Plate | 1 |
| Bottom Plate | 1 |
| Grip Clamp | 8 |
| Pulley Block | 8 |
| External Support | 8 |
| Center Support | 1 |

## 13.2 Appendix B