Final Report – Fall 2013

Team 2 – Biaxial Test Fixture

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Abstract

The point 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.

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.

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 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 more understanding and the possibility of broadening the types materials is the inspiration for this project.

Background

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.

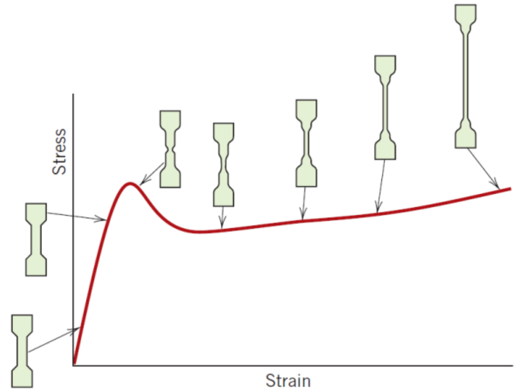


Figure 1: Tensile stress–strain curve for a semicrystalline polymer[1]

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].

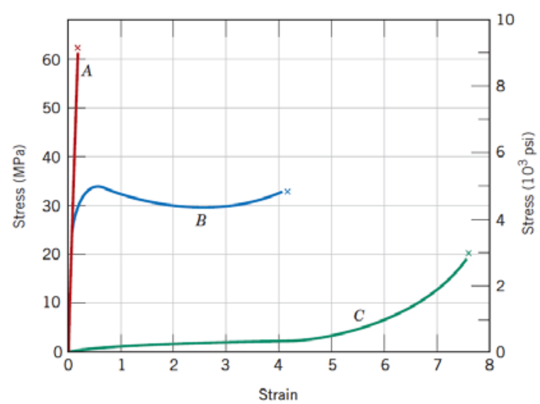


Figure 2: Stress-strain behavior for a polymer (A) brittle (B) plastic (C) elastomer[1]

Elastomers

The gasket material that will be tested in the biaxial test fixture is a nearly incompressible elastomer; that is capable

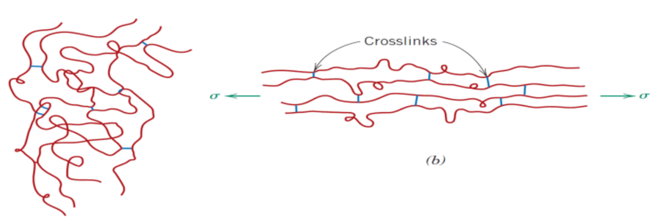


Figure 3: Representation of the crosslinked chains molecules found in polymers, and how they react with applied stress[1].

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.

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.

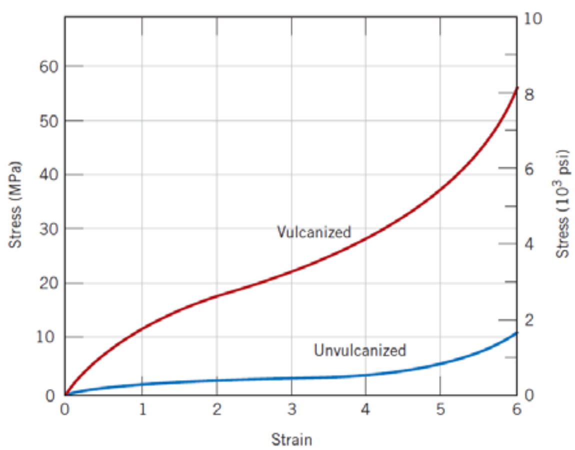


Figure 4: Stress-strain curve to 600% elongation for natural rubber both vulcanized and unvulcanized[1].

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].

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

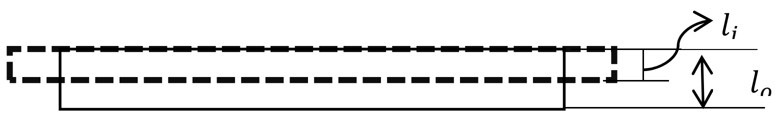


Figure 5 A schematic representation of the loading effects for an incompressible elastomer.

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].

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	

Existing Biaxial Machines

The existing biaxial machines were not ideal for the purpose of Cummins' gasket material testing. This can be attributed to several instances of cons in the design. Fig. 6 shows a biaxial machine pulling along two different axes, as the name suggests. The problem arises in the types of materials being tested. Polymers strained in this manner cause the material along the edge that is not gripped by the test rig to bow inwards while the gripped material is pulling outwards. This is a function of the material's elasticity and lack of stiffness.

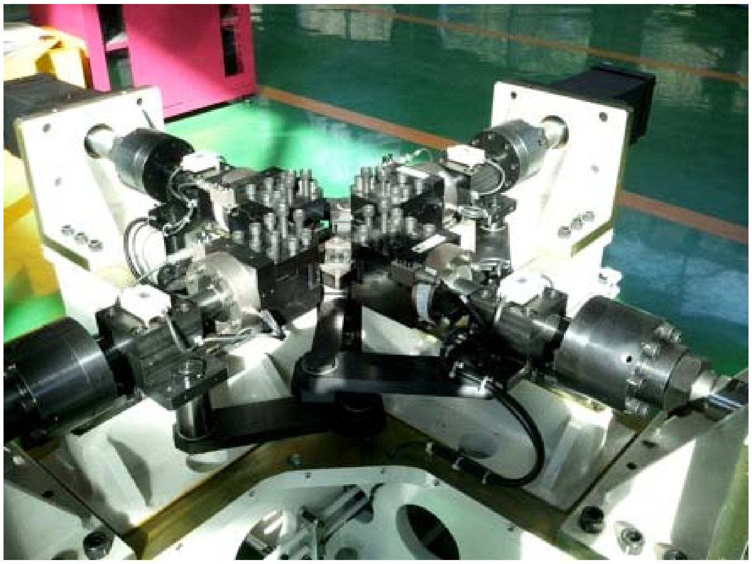


Figure 6: Tensile Machine Pulling Along Two Axes

Something a little closer to what we are searching for in our design can be seen in Fig 7. This is a multiaxial machine that will eliminate the unevenness that occurs while stretching the material. The problem with this particular design is that the material be testing will be a much larger specimen size then what is required in this fixture. Some of the provided gasket material is a polymer reinforced, fibrous paper that experience strain depending on the orientation of the fibers

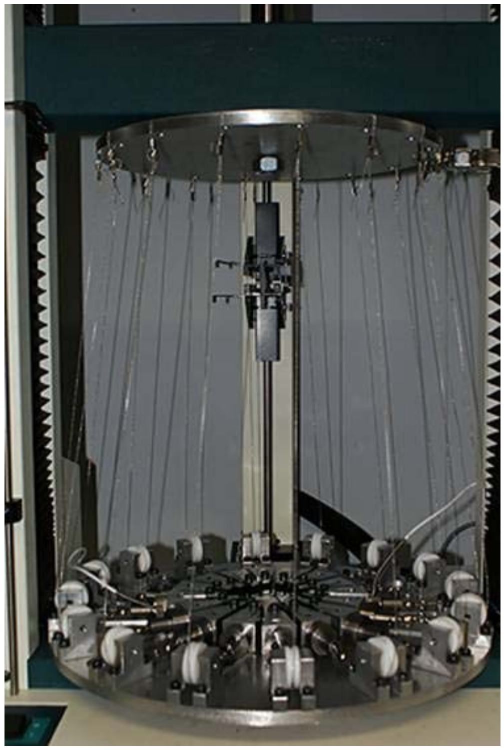


Figure 7: Multiaxial Test Machine

Another multiaxial design can be seen in Fig. 8. This is the closest to the intended direction our design. The main problem with this is that the force applied to the specimen is done by a deadweight type of technique. That means that they use variable weights added to each pulley individually. This is a problem because this really limits how much weight is added to each pulley. This is something that can easily be addressed in our design.

Figure 8: Another Multiaxial Test Rig

Gripping Techniques

The most important component of our device will be the grips. As the interface between the applied force and the sample there are some basic requirements for an accurate result to be achieved. The grips should minimize the deformation of the sample. The reason that the grips should minimize the deformation is that as the area of the sample is reduced, the stress is increased. A large deformation makes it more likely that the sample will fail at the grip. Ideally we would like to be able to have the failure occur in the center of the sample, and if it fails at a low enough stress near the grip then no useful data will be acquired.

Another important requirement of the grip is that it needs to maintain a planar alignment of the sample to prevent the introduction of shear forces or bending moments. In order to accomplish this, the grips will need to be precisely manufactured to the same size and mount to the carrier at exactly the same height.

The grips have to be able to hold the sample through the entire loading process without slipping. In order to accomplish this, the grips will need to create friction between themselves and the sample. We can create a raised geometry of horizontal lines or a pattern of spikes, this will increase the amount of surface in contact with the grips while minimizing the overall compression of the sample. An important part of the grips ability to hold the sample will be the manner in which they are tensioned. Screws or bolts would allow the user to vary the tension easily, and the use of a single bolt that draws both of the grips together at the same rate would likely be the best method to maintain planar alignment.

Finally, the grips and carrier will be custom made pieces, therefore it is important that they are a strong point in the design, it will much easier for the customer to repair a standard sized cable than a custom machined piece. Due to the cyclical loading cycles it will be very important to design a grip and carrier that can not only endure the forces placed on it but will not fatigue over time.

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.

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.

Design and Analysis

Biaxial Test Fixture Design

In producing a viable design, several different approaches were examined. There were two main theories used in designing, however. This is whether to have the design self-driven or to have it integrated into an existing MTS machine. Figure 10 shows a very simple schematic on how the design could look. This was ultimately ruled out due to the inability to alter the linkage

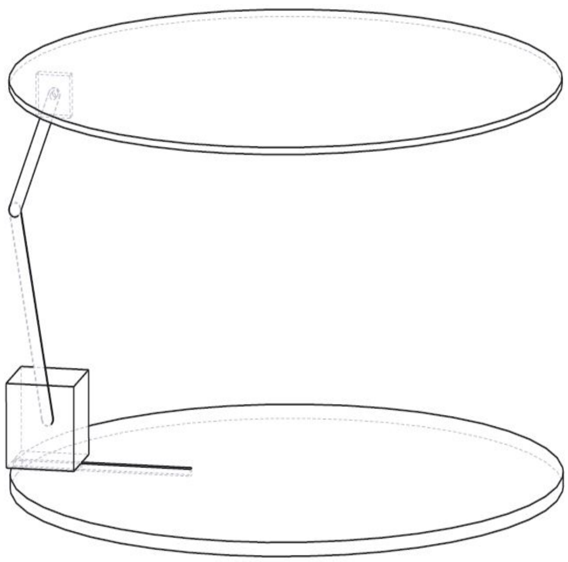


Figure 10: Linkage Design Simple Schematic with Only One Pulling Location

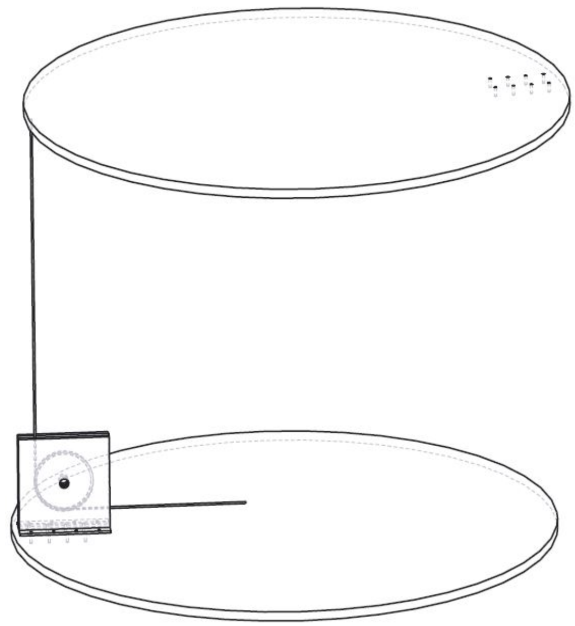


Figure 11: Pulley Design at the Baseplate with Only One Pulling Location

without difficulty and due to difficulty while working. 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. Figure 11 then shows another simple schematic of how the pulley design is to be accomplished at the baseplate. This design was much more desirable in the sense that cables are much easier to work around than linkages due to the flexibility and space around the baseplate. 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 was conducting a test with the device would also need to be training 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.

Figure 12 shows the final assembly of the baseplate with pulleys incorporated. Figure 13 then shows the full tiered design of the pulley system.

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 initial analysis performed to validate the selection of pulleys and cable selected for the project. This analysis can be seen in

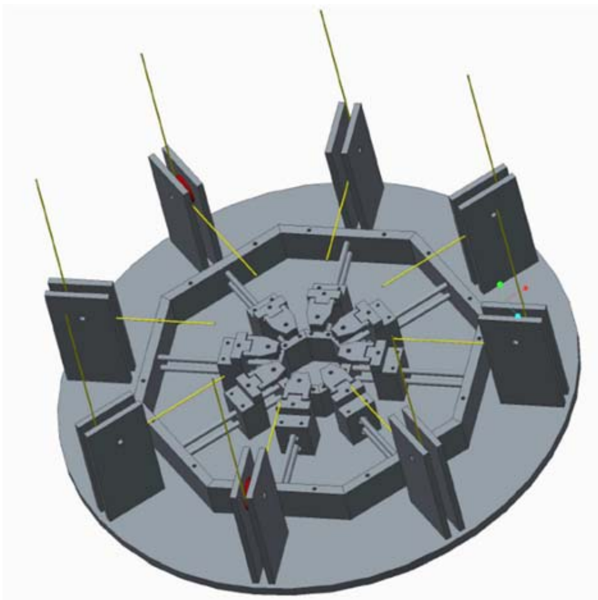


Figure 12: Pulley System at Baseplate

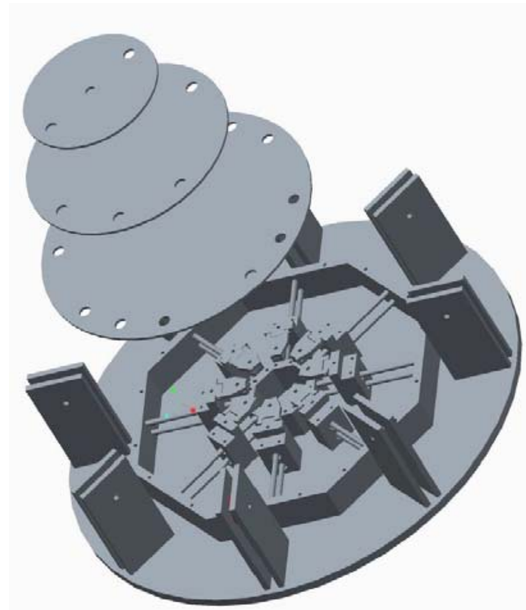


Figure13: Full View of Pulley System (Minus Cables)

Appendix A. The pulleys and cable selected were sufficient for the maximum amount of load determined. The cable had a safety factor of 2.35 while the pulley had a safety factor of 1.37.

Specimen Geometry

Biaxial tensile testing is a specialized type of analysis that is not necessary for most materials. Uniaxial tensile, pure shear, and uniaxial compression are the three standard types of mechanical testing that provided adequate data for accurate modeling. However, problems arise while testing materials that do not have linear elastic regions, such as elastomers. While testing elastomeric material in uniaxial compression the effects of the friction between the load plates and the specimen cannot be ignored. This is when a biaxial tensile test would be employed.

While researching existing biaxial machines, it was discovered that biaxial was not limited to 2 axes. Instead, we found that the ideal loading conditions for this type of testing would be to clamp and pull equally around the entirety of the diameter. This creates a state of equibiaxial tensile strain; which is equivalent to a uniaxial compressive strain. Understanding this concept, the goal of the specimen design is to achieve a uniform strain distribution throughout the sample. The question then became how many axes should our biaxial tensile fixture pull from in order to accomplish this.

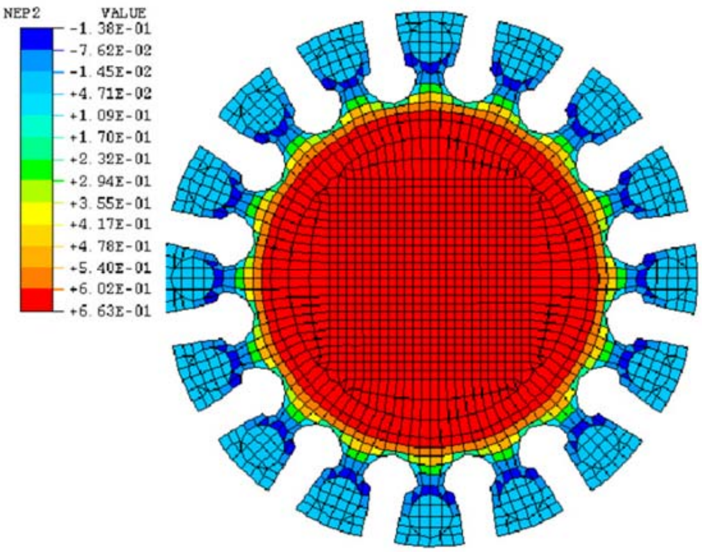


Figure 14: FEA Analysis of specimen geometry used by Axel Products, Inc. for biaxial testing

Figure 14 is the specimen geometry used by Axel Products, Inc. a material characterization company that specializes in the testing of nonlinear materials such as elastomers and plastics[1]. The concept of two concentric diameters with a reduced gauged section between them was used for the backbone of our specimen design. The radius of the gauged section remained constant while the number of axes to pull from was increased.

The material that will be tested in our biaxial fixture will range from paper to rubber gaskets. This is a large range for both the stress and the strain parameters during testing. To ensure this geometry would be applicable for the varieties of materials given some major assumptions were made. The specimen was modeled and stress analysis was done using Autodesk Inventor Professional 2014. The properties of natural rubber were used during simulations because this material would have the greatest amount of deformation during actual testing. Half order symmetry was employed to constrain the specimen. The load was applied radially and assumed to be perfectly symmetrically. The effects of the clamping that would occur in the grips during actual testing were also neglected.

As seen below in Figure 15 the stress concentrations at the gauged sections decreased as the number of axes was increased. However, the deformation along the griping area increased. This could cause slip to occur in the gripping region. The design then needed to be optimized for the uniformity of strain distribution to the amount of deformation during loading. Taking into consideration these parameters the specimen design which pulled along 4 axes was selected.

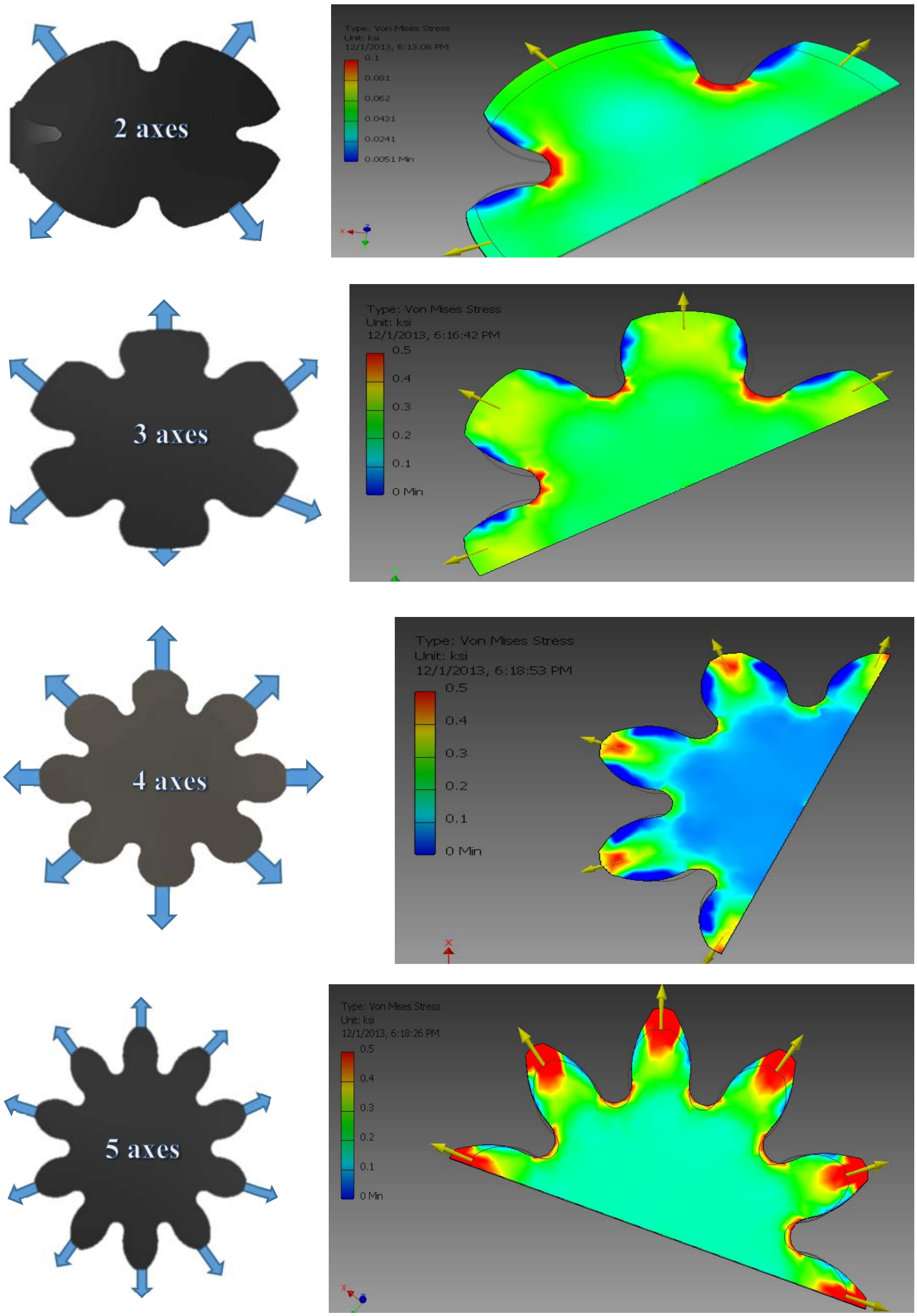
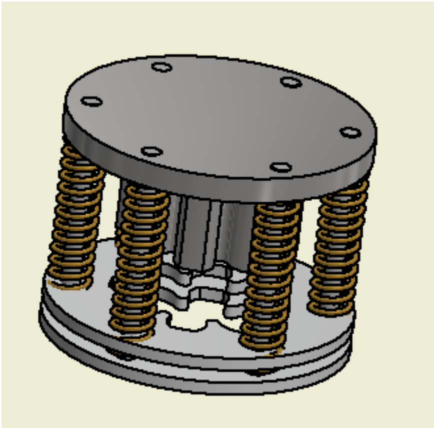


Figure 15: The Von Mises Stresses that occur during deformation of the specimen geometry.

To ensure accurate testing each sample must be machined to the same tolerances every time. In order to produce the selected design a punch needed to be constructed. As seen in Figure 16, the gasket material will be placed between the bottom plates. A press is then used to compress the punch, and the specimen design is cut out of the gasket.



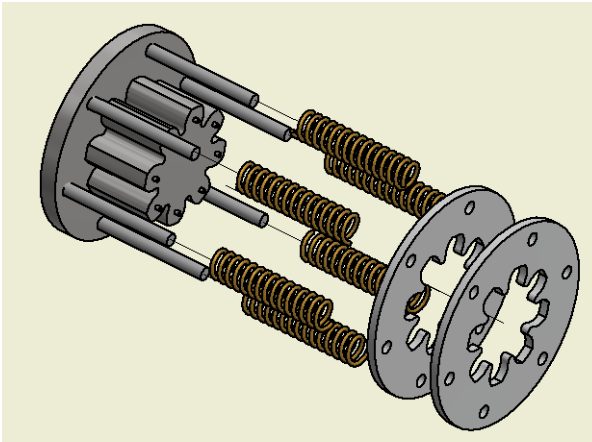


Figure 16: Punch design for specimen production

Risk and Reliability Assessment

Materials testing carries with it some inherent risk. The biaxial test rig designed here will be used on an MTS machine that is designed to compress or pull materials to failure. As such, all employees in materials testing labs are required to be knowledgeable in the best safe industry practices. To this end, they are required to attend regular safety meetings and maintain a safe work environment. The test rig itself does pose some extra risks that are not normally present. As such, technicians operating the rig will be informed that a maximum safe applied load of 4,000lbf should not be exceeded, this load significantly exceeds any load required to break the materials that this rig is designed for. Furthermore, technicians will be required to periodically inspect each component of the rig for any signs of excessive wear and tear such as crack propagation in high stress areas to be identified in the technician's manual. Another key component of the safety process will be to inspect each cable before and after each test for signs of fraying or excessive stretch that would indicate an impending failure. Once the device is constructed and tested a complete, detailed technicians manual will lay out a very specific inspection regimen to ensure the safety of all lab workers.

Communications

Our group has had a weekly telephone meeting with our client, Terry Shaw from Cummins, Inc., on every Monday since the third week of the semester. He has 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. Our Faculty Advisor, Dr. Oates, has proven invaluable to our understanding of the data required to be obtained and the manner in which it should be obtained to produce the desired result, a better Finite Element Analysis. 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.

Procurement

Currently, our team has yet to procure the raw materials necessary for our design. Some last minute improvements and cost saving measures have been made to improve our design after an enlightening meeting with a manufacturing engineer who provided some guidance in regards to questions we had about minimizing machining time. Within the next week we will have a finalized materials list for a design that maintains quality and reduces material and machining costs. We judged it to be more prudent to be sure that we have the best, most cost-efficient design possible and have our materials in the beginning of January, than to have had materials for a less effective design or to have purchased unnecessary materials for the end of December. For a complete breakdown of our latest, leanest and meanest materials list please see the budget list in Table 3.

Detailed Design and Design for Manufacturing

For our design, we have chosen to integrate with an MTS machine because the already included load cell and actuator would have caused us to go over budget on a standalone device. Our device will consist of grips constrained on linear motion shafts pulled by steel wire ropes. The rate of strain will be governed by the MTS machine and the strain measured using video extensometry. The total stress will be measured with the MTS' load cell.

Our device will consist of two parallel aluminum plates. The lower plate will be made of ¾” 6061 aluminum alloy, it will attach to the MTS machine’s lower grip and will hold the wire rope (see Appendix B). The upper plate will be positioned slightly above the lower plate. In the center will be an octagon shaped piece of 6061 aluminum that will have two holes on each face to constrain the hardened steel linear motion rods, it will be attached with two bolts that go through the bottom of the plate and into the support (see Appendix B). The two rods coming out of the each face of the center support will be 8 ¾” long and 12mm in diameter (see Appendix B). The reason for the mixing of SAE and metric units is the extreme cost difference in the linear bearings. The rods will support carriers that are driven by the attached wire ropes and hold the specimen with two alignment pins and a second piece of metal that is clamped down with screws, the carriers contain two linear ball bearings inserted into them which allow a very smooth and linear path of travel (see Appendix B). The outside of the hardened rods are constrained in a block of 6061 aluminum, additionally this block supports the pulley that transfers the motion of the wire rope from vertical to horizontal. The pulley is attached to a shaft which is supported on each side by roller bearings (see Appendix B).

Each of our components has been designed to minimize the amount of machining necessary to create them. The Bottom base plate will need to be cut round and have nine holes cut through it, one in the center for attachment to the MTS, and nine around the outside for the attachment of the wire ropes. The top base plate will also need to be cut round and it will need 19 holes cut through it: one for the center support, two for threaded rods that will attach it to the crosshead at the top of the MTS, and two at each of the eight exterior supports. The center support will need to be cut into an octagonal shape, and have 17 holes drilled: one in the center, and two on each of the eight faces. The carrier will need to be cut on two sides and the top and have nine holes: two for the bearings, two for set screws for the bearings, one for wire rope, two for the guide pins and two for the grip plate. The grip plate will have to be cut to size and have two holes cut through it for attachment screws. (173 total holes; 48 tapped, 2 round cuts, 6 sides of the octagon, 16 sides of the carriers).

Conclusion

Understanding how biaxial tension achieves a compressive strain took the greater part of the semester to fully grasp. The background research was paramount in being able to achieve a working product for our consumer. Gaskets are formed into complex geometric designs, and the need to properly model its behavior was the driving force to the funding of our project. The nonlinear behavior of the elastomeric material coupled with its incompressibility led to the need for a fixture such as this.

The design and concept development of our fixture is still in progress. The fundamentals of how we are going to achieve the motion of biaxial extension are sound, but there is a need to further simplify the design due to budget constraints. The decision to use the MTS machine instead of a standalone system was dictated by a couple factors. The most important factor being that MTS machines have integrated load cells and data acquisition systems. The use of pulleys and cabling to achieve the radial motion reduces the amount of material to be machined. However, it opens another potential problem of calibration. Further testing of cabling will be done to ensure that a symmetric radial load is applied. As mentioned before, the force applied to the sample will be measured by the integrated load cell and distributed equally through the eight pulleys. The strain will be measured by applying a light overspray of paint and using a video extensometer to measure the point to point displacement. Both of these systems are available for our team to operate.

Environmental and Safety Issues 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.

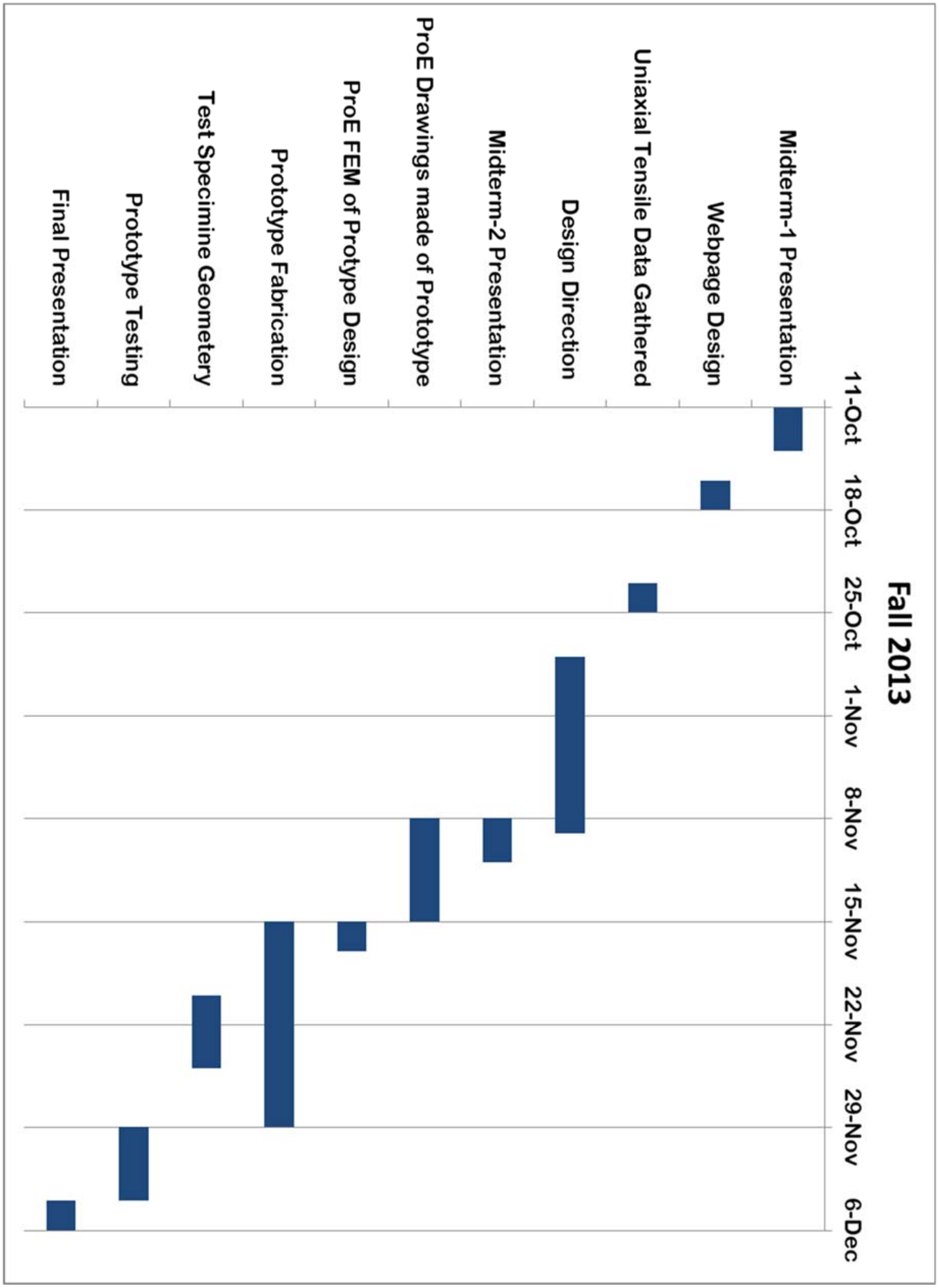
There are no ethical constraints while performing material testing of gaskets.

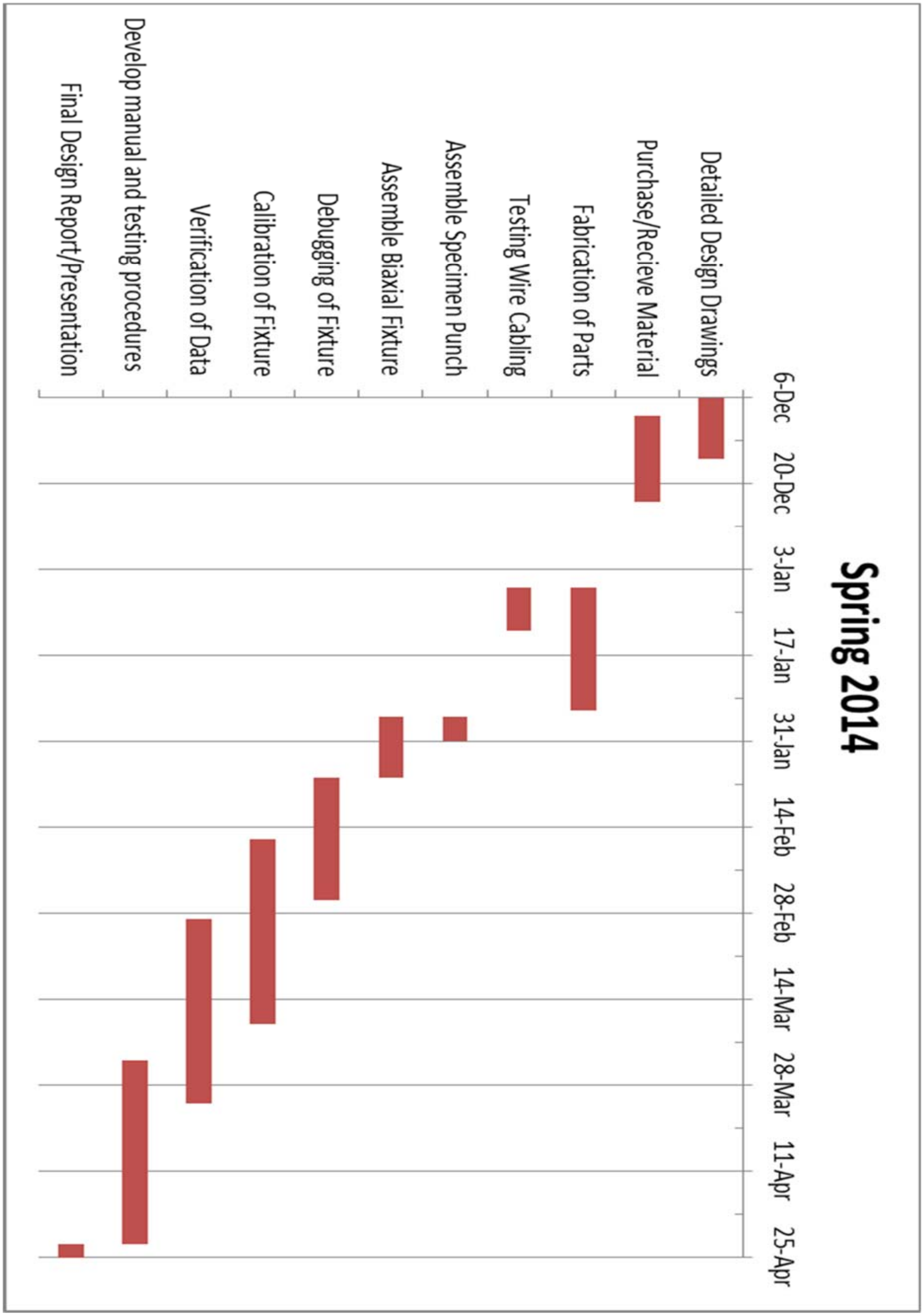
Future plans for prototype and others

The initial cost for the raw materials for our prototype came close to exceeding the given budget. To reduce cost, meetings with the PE Design Engineer at the National High Magnetic Field Laboratory will take place to streamline the fixture. This will minimize the need for machining and the amount of raw material necessary. After the simplification of the design has been finalized, the ordering of material and hardware will occur.

Once the raw material is here the machining and assembly of the fixture will begin. After assembly the machine will be debugged and the cabling calibrated. There will be a substantial amount of time spent during this stage of the fixture development. Once we are satisfied that each cable is pulling at the same rate, the verification of results will begin. This means intentionally testing defected specimens and ascertaining if the data shows any discrepancy. After the data has been verified, a manual will be developed for safe and complete instructions of operation.

Gantt Chart, Resources, Budget





Spring 2014
Detailed Design Drawings
Purchase/Recieve Material
Fabrication of Parts
Testing Wire Cabling
Assemble Specimen Punch
Assemble Biaxial Fixture
Debugging of Fixture
Calibration of Fixture
Verification of Data
Develop manual and testing procedures
Final Design Report/Presentation
6-Dec
20-Dec
3-Jan
17-Jan
31-Jan
14-Feb
28-Feb
14-Mar
28-Mar
11-Apr
25-Apr

Table 3: Itemized Material Budget

Metal Stock Costs	32” x 32” Al tool steel	$	235.55
	42” x 40” 6061 Al	$	447.69
	2” x 2”x 30” 6061 Al	$	46.41
	Shipping	$	333.90
	Total	**$1,063.55**	
Base Plate Hardware Costs	12mm Linear Bearings	$	20.22
	12mm Hardened Steel Rods	$	134.78
	50) ¼”x 4 ½” Bolts	$	7.49
	(100) ¼” Washers	$	3.29
	(50) ¼” Nuts	$	4.74
	(50) ¼-20 1.5” Screws	$	8.95
	(50) 5-40 x 5/8” Screws	$	5.99
	Total	**$**	**185.46**
Cable and Pulley System Costs	(8) Stainless Steel Pulley	$	56.72
	(8) Galvanized Steel Eyebolt	$	36.32
	(16) Steel Ball Bearings	$	113.44
	(20) End-Fitting for Wire	$	261.66
	Aluminum Stop Compression Sleeve	$	7.97
	Total	**$**	**476.11**

:

Table 4: Total Budget

Budget	**$2,000.00**	
Metal Stock	$	1,063.55
Base Plate Hardware	$	185.46
Cable and Pulley System	$	476.11
Total Cost	**$**	**1,725.12**
Money Remaining	$	274.88

Appendix A

2 Axes Simulation

General objective and settings:

Design Objective	Single Point
Simulation Type	Static Analysis
Last Modification Date	12/6/2013, 5:16 AM
Detect and Eliminate Rigid Body Modes	No

Mesh settings:

Avg. Element Size (fraction of model diameter)	0.1
Min. Element Size (fraction of avg. size)	0.2
Grading Factor	1.5
Max. Turn Angle	60 deg
Create Curved Mesh Elements	Yes

Material(s)

Name	Rubber, Silicone	
General	Mass Density	0.0451591 lbmass/in^3
	Yield Strength	1500.73 psi
	Ultimate Tensile Strength	943.396 psi
Stress	Young's Modulus	0.435414 ksi
	Poisson's Ratio	0.49 ul
	Shear Modulus	0.146112 ksi
Part Name(s)	Specimen_4	

Operating conditions

Force: 1

Load Type	Force
Magnitude	4.000 lbforce
Vector X	2.777 lbforce
Vector Y	-2.879 lbforce
Vector Z	0.000 lbforce

Force: 2

Load Type	Force

Magnitude	2.000 lbforce
Vector X	-0.980 lbforce
Vector Y	-1.743 lbforce
Vector Z	0.000 lbforce

Force: 3

Load Type	Force
Magnitude	2.000 lbforce
Vector X	1.735 lbforce
Vector Y	0.996 lbforce
Vector Z	0.000 lbforce

Frictionless Constraint: 1

Constraint Type	Frictionless Constraint

Fixed Constraint: 1

Constraint Type	Fixed Constraint

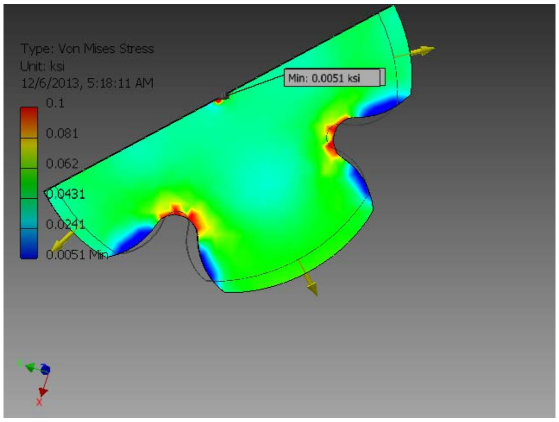
Results

Result Summary

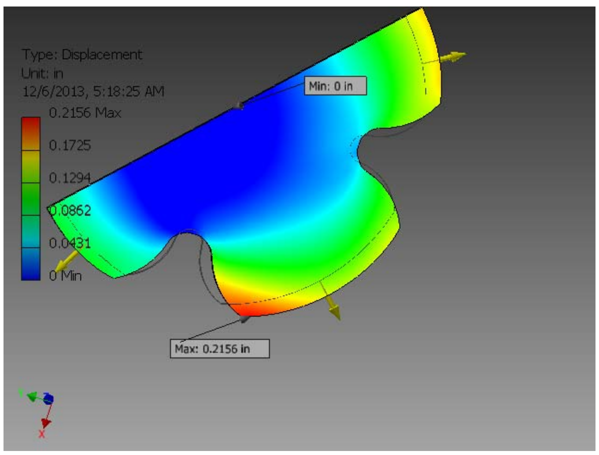
Name	Minimum	Maximum
Volume	0.16779 in^3	
Mass	0.00757726 lbmass	
Von Mises Stress	0.00509978 ksi	0.472327 ksi
1st Principal Stress	-3.25269 ksi	2.88354 ksi
Displacement	0 in	0.215601 in
Z Displacement	-0.0111223 in	0.000313027 in
Strain ZZ	-0.320442 ul	0.115712 ul

Figures

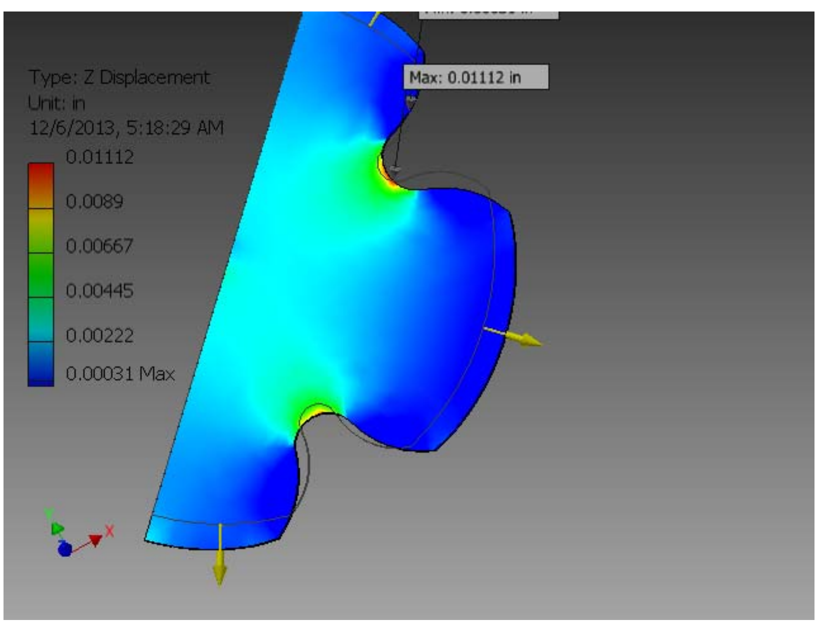
Von Mises Stress



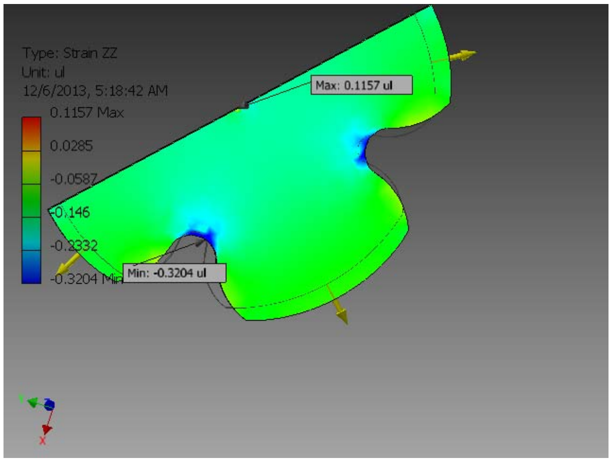
Displacement



Z Displacement



⊐ Strain ZZ



3 Axes Simulation

General objective and settings:

Design Objective	Single Point
Simulation Type	Static Analysis
Last Modification Date	12/1/2013, 5:31 PM
Detect and Eliminate Rigid Body Modes	No

Mesh settings:

Avg. Element Size (fraction of model diameter)	0.1
Min. Element Size (fraction of avg. size)	0.2
Grading Factor	1.5
Max. Turn Angle	60 deg
Create Curved Mesh Elements	Yes

⊐ Material(s)

Name	Rubber, Silicone	
General	Mass Density	0.0451591 lbmass/in^3
	Yield Strength	1500.73 psi
	Ultimate Tensile Strength	943.396 psi
Stress	Young's Modulus	0.435414 ksi
	Poisson's Ratio	0.49 ul
	Shear Modulus	0.146112 ksi
Part Name(s)	Specimen_6	

⊐ Operating conditions

⊟ Force: 1

Load Type	Force
Magnitude	6.000 lbforce
Vector X	-4.978 lbforce
Vector Y	3.350 lbforce
Vector Z	0.000 lbforce

⊟ Force: 2

Load Type	Force
Magnitude	12.000 lbforce
Vector X	0.216 lbforce
Vector Y	11.998 lbforce
Vector Z	0.000 lbforce

⊟ Force: 3

Load Type	Force
Magnitude	12.000 lbforce
Vector X	10.244 lbforce
Vector Y	6.250 lbforce
Vector Z	0.000 lbforce

⊟ Force: 4

Load Type	Force
Magnitude	6.000 lbforce
Vector X	5.421 lbforce
Vector Y	-2.572 lbforce
Vector Z	0.000 lbforce

⊟ Fixed Constraint: 1

Constraint Type	Fixed Constraint

⊟ Frictionless Constraint: 1

Constraint Type	Frictionless Constraint

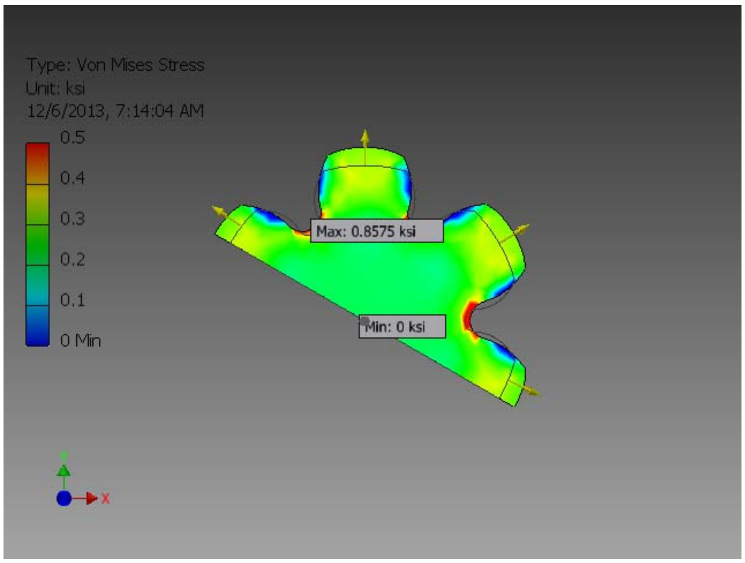
⊟ Result Summary

Name	Minimum	Maximum
Volume	0.141687 in^3	

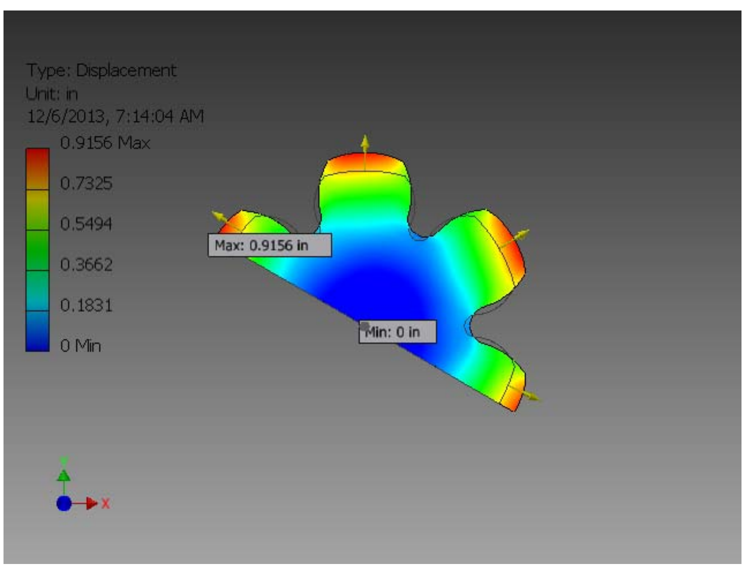
Mass	0.00639844 lbmass	
Von Mises Stress	0 ksi	0.85694 ksi
Displacement	0 in	0.915624 in
Z Displacement	-0.0399947 in	0.0000118813 in
Strain ZZ	-1.17209 ul	0.0119115 ul

Figures

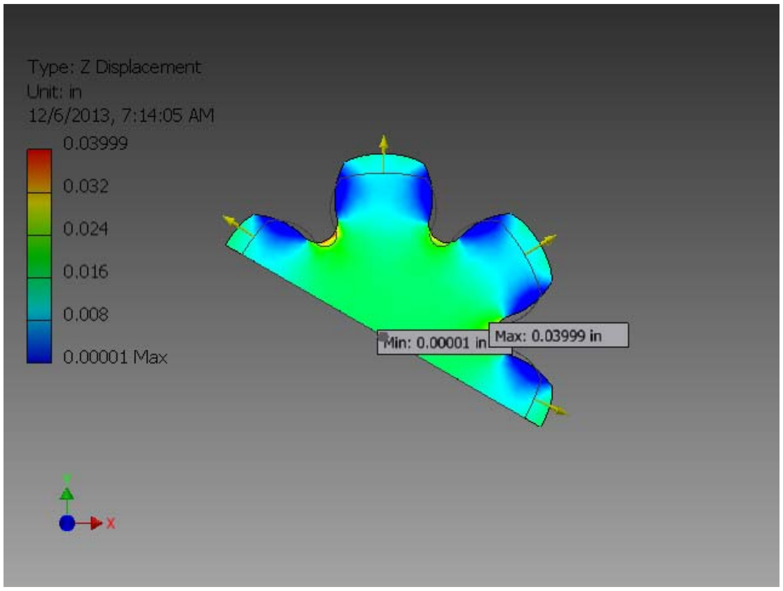
Von Mises Stress



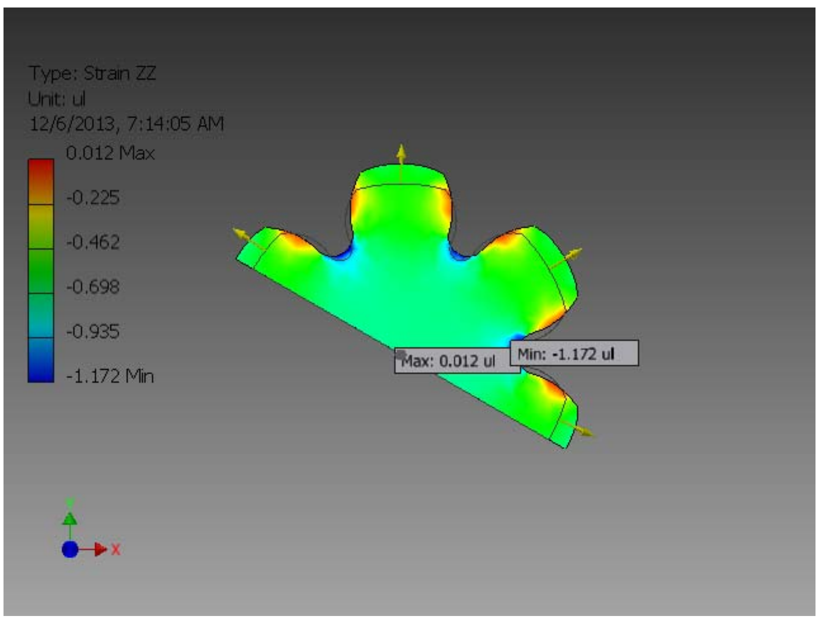
Displacement



Z Displacement



Strain ZZ



4 Axes Simulation

General objective and settings:

Design Objective	Single Point
Simulation Type	Static Analysis
Last Modification Date	12/1/2013, 6:04 PM
Detect and Eliminate Rigid Body Modes	No

Mesh settings:

Avg. Element Size (fraction of model diameter)	0.1

Min. Element Size (fraction of avg. size)	0.2
Grading Factor	1.5
Max. Turn Angle	60 deg
Create Curved Mesh Elements	Yes

⊟ Material(s)

Name	Rubber, Silicone	
General	Mass Density	0.0451591 lbmass/in^3
	Yield Strength	1500.73 psi
	Ultimate Tensile Strength	1666.15 psi
Stress	Young's Modulus	60.3917 ksi
	Poisson's Ratio	0.49 ul
	Shear Modulus	20.2657 ksi
Part Name(s)	Specimen_8	

⊟ Operating conditions

⊟ Force: 1

Load Type	Force
Magnitude	6.000 lbforce
Vector X	5.580 lbforce
Vector Y	-2.205 lbforce
Vector Z	0.000 lbforce

⊟ Force: 2

Load Type	Force
Magnitude	6.000 lbforce
Vector X	2.225 lbforce
Vector Y	-5.572 lbforce
Vector Z	0.000 lbforce

⊟ Force: 3

Load Type	Force
Magnitude	6.000 lbforce
Vector X	-2.207 lbforce
Vector Y	-5.579 lbforce
Vector Z	0.000 lbforce

⊟ Force: 4

Load Type	Force
Magnitude	3.000 lbforce
Vector X	2.843 lbforce
Vector Y	0.957 lbforce
Vector Z	0.000 lbforce

⊟ Force: 5

Load Type	Force
Magnitude	3.000 lbforce
Vector X	-2.734 lbforce
Vector Y	-1.235 lbforce
Vector Z	0.000 lbforce

⊟ Fixed Constraint: 1

Constraint Type	Fixed Constraint

⊟ Frictionless Constraint: 1

Constraint Type	Frictionless Constraint

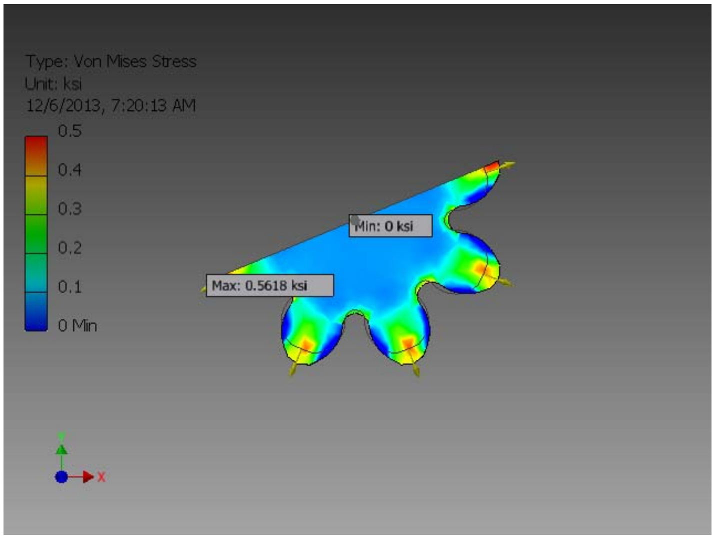
⊟ Results

⊟ Result Summary

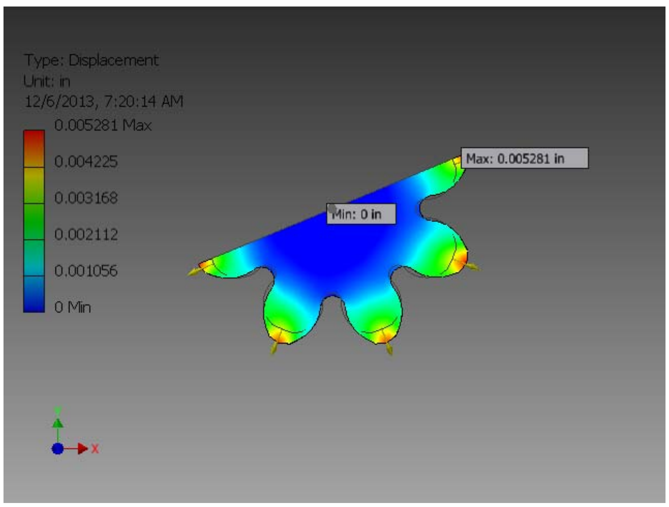
Name	**Minimum**	**Maximum**
Volume	0.147878 in^3	
Mass	0.00667803 lbmass	
Von Mises Stress	0 ksi	0.561411 ksi
Displacement	0 in	0.0052807 in
Z Displacement	-0.000305038 in	0.0000104215 in
Strain ZZ	-0.0076974 ul	0.000941554 ul

⊟ Figures

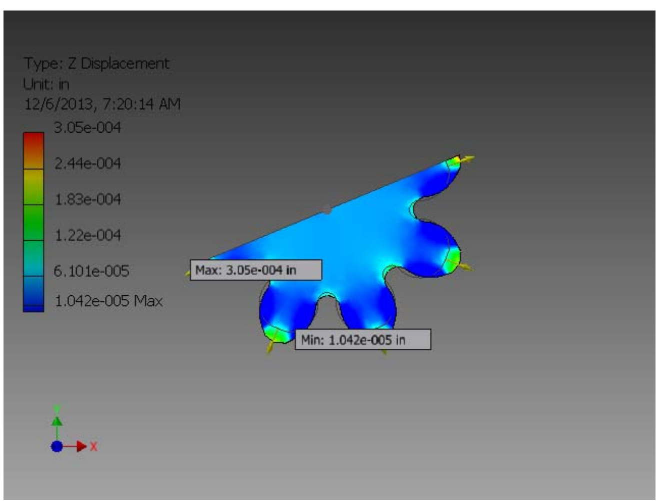
⊟ Von Mises Stress



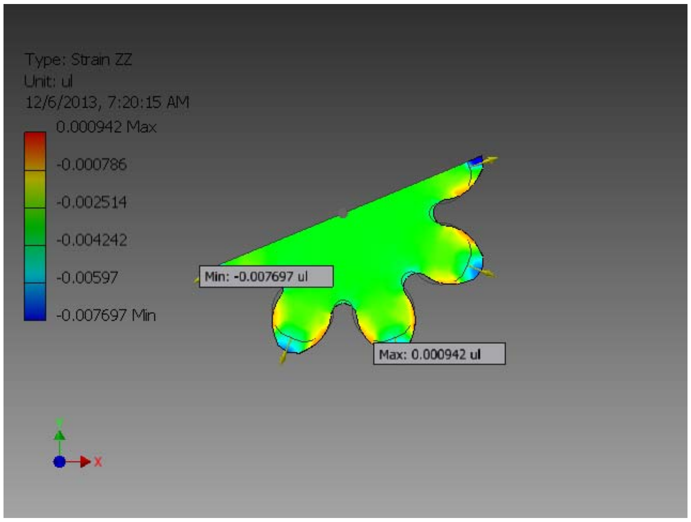
Displacement



Z Displacement



Strain ZZ



5 Axes Simulation

General objective and settings:

Design Objective	Single Point
Simulation Type	Static Analysis
Last Modification Date	12/1/2013, 5:47 PM
Detect and Eliminate Rigid Body Modes	No

Mesh settings:

Avg. Element Size (fraction of model diameter)	0.1
Min. Element Size (fraction of avg. size)	0.2
Grading Factor	1.5
Max. Turn Angle	60 deg
Create Curved Mesh Elements	Yes

Material(s)

Name	Rubber, Silicone	
General	Mass Density	0.0451591 lbmass/in^3
	Yield Strength	1500.73 psi
	Ultimate Tensile Strength	943.396 psi
Stress	Young's Modulus	0.435414 ksi
	Poisson's Ratio	0.49 ul
	Shear Modulus	0.146112 ksi
Part Name(s)	Specimen_10	

Operating conditions

Force: 1

Load Type	Force
Magnitude	3.000 lbforce
Vector X	-2.840 lbforce
Vector Y	-0.967 lbforce
Vector Z	0.000 lbforce

Force: 2

Load Type	Force
Magnitude	6.000 lbforce
Vector X	-3.513 lbforce
Vector Y	-4.864 lbforce
Vector Z	0.000 lbforce

Force: 3

Load Type	Force
Magnitude	6.000 lbforce
Vector X	0.033 lbforce
Vector Y	-6.000 lbforce
Vector Z	0.000 lbforce

Force: 4

Load Type	Force
Magnitude	6.000 lbforce
Vector X	3.470 lbforce
Vector Y	-4.894 lbforce
Vector Z	0.000 lbforce

Force: 5

Load Type	Force
Magnitude	6.000 lbforce
Vector X	5.697 lbforce
Vector Y	-1.881 lbforce
Vector Z	0.000 lbforce

Force: 6

Load Type	Force
Magnitude	3.000 lbforce
Vector X	2.886 lbforce
Vector Y	0.818 lbforce
Vector Z	0.000 lbforce

Fixed Constraint:1

Constraint Type	Fixed Constraint

Frictionless Constraint:1

Constraint Type	Frictionless Constraint

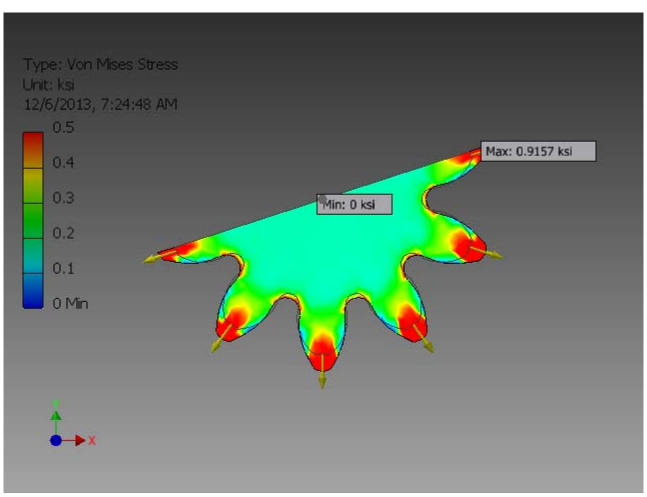
Results

Result Summary

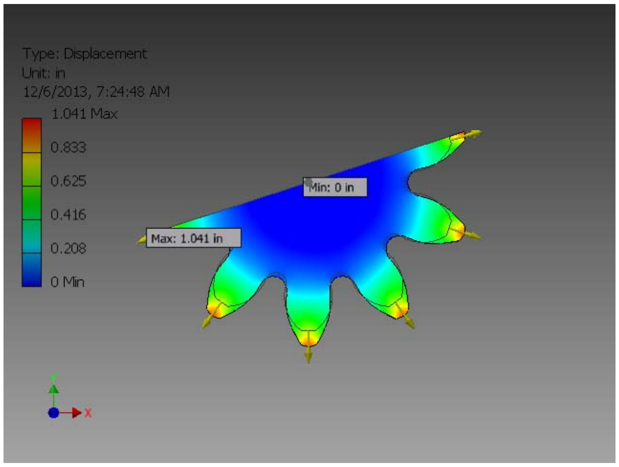
Name	Minimum	Maximum
Volume	0.123918 in^3	
Mass	0.00559601 lbmass	
Von Mises Stress	0 ksi	0.915045 ksi
Displacement	0 in	1.04122 in
Z Displacement	-0.0599227 in	0.0000688221 in
Strain ZZ	-1.69143 ul	0.0457261 ul

Figures

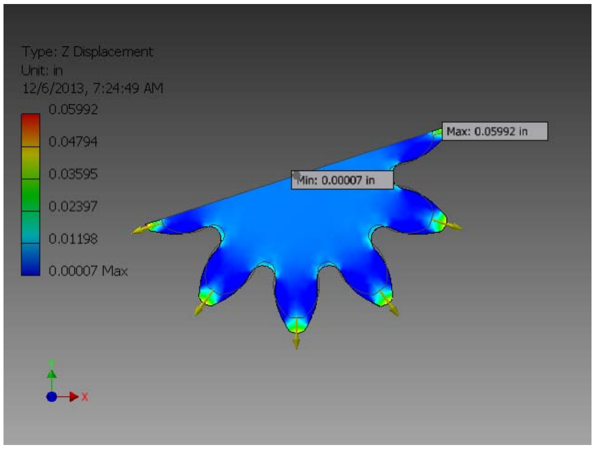
Von Mises Stress



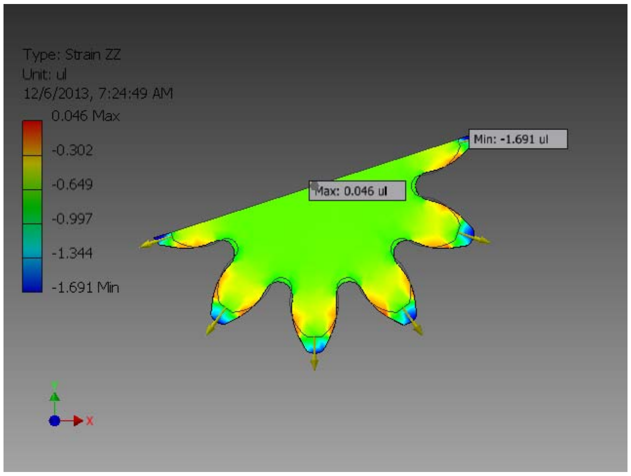
Displacement



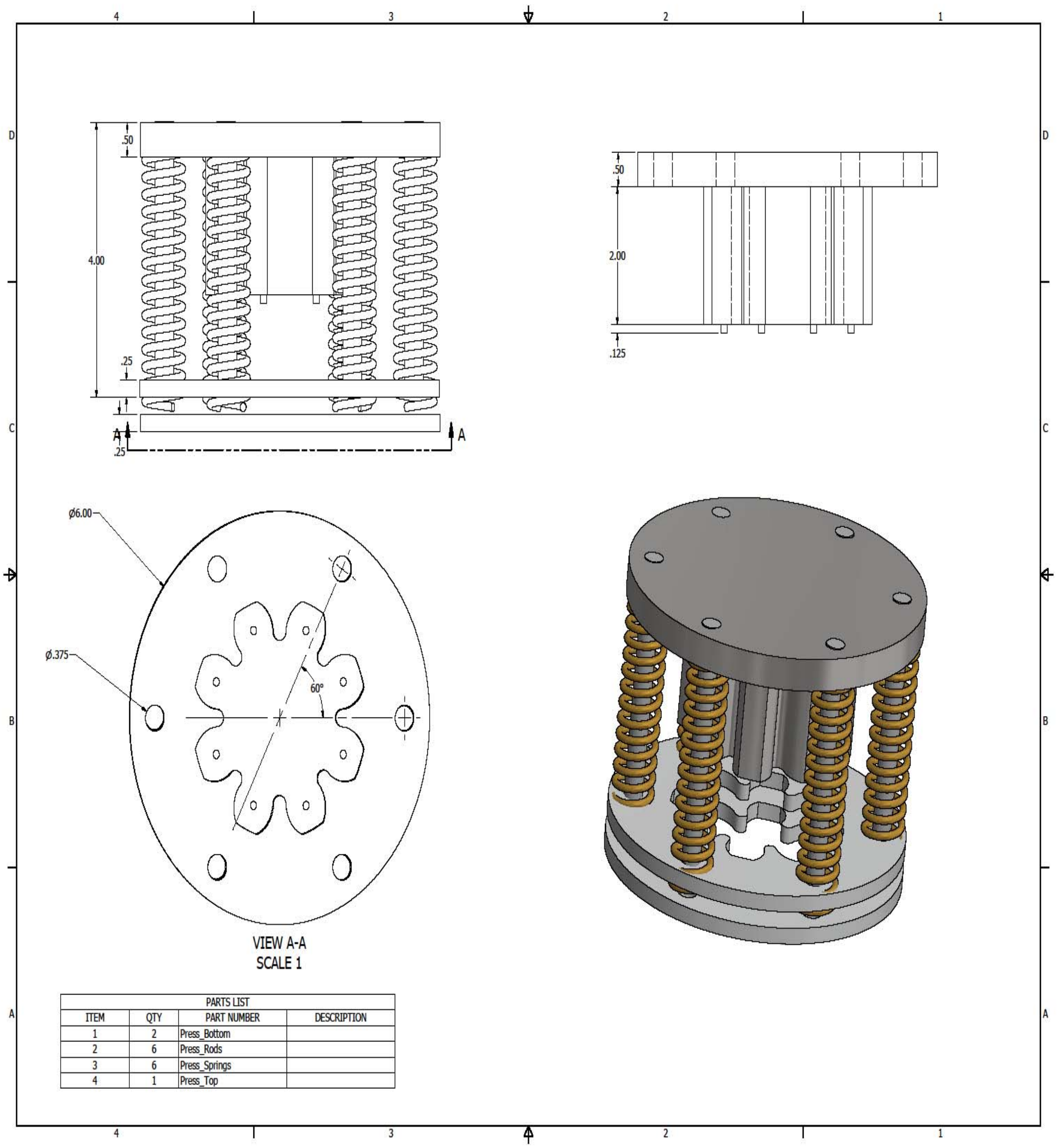
⊟ **Z Displacement**



⊟ **Strain ZZ**



Appendix B



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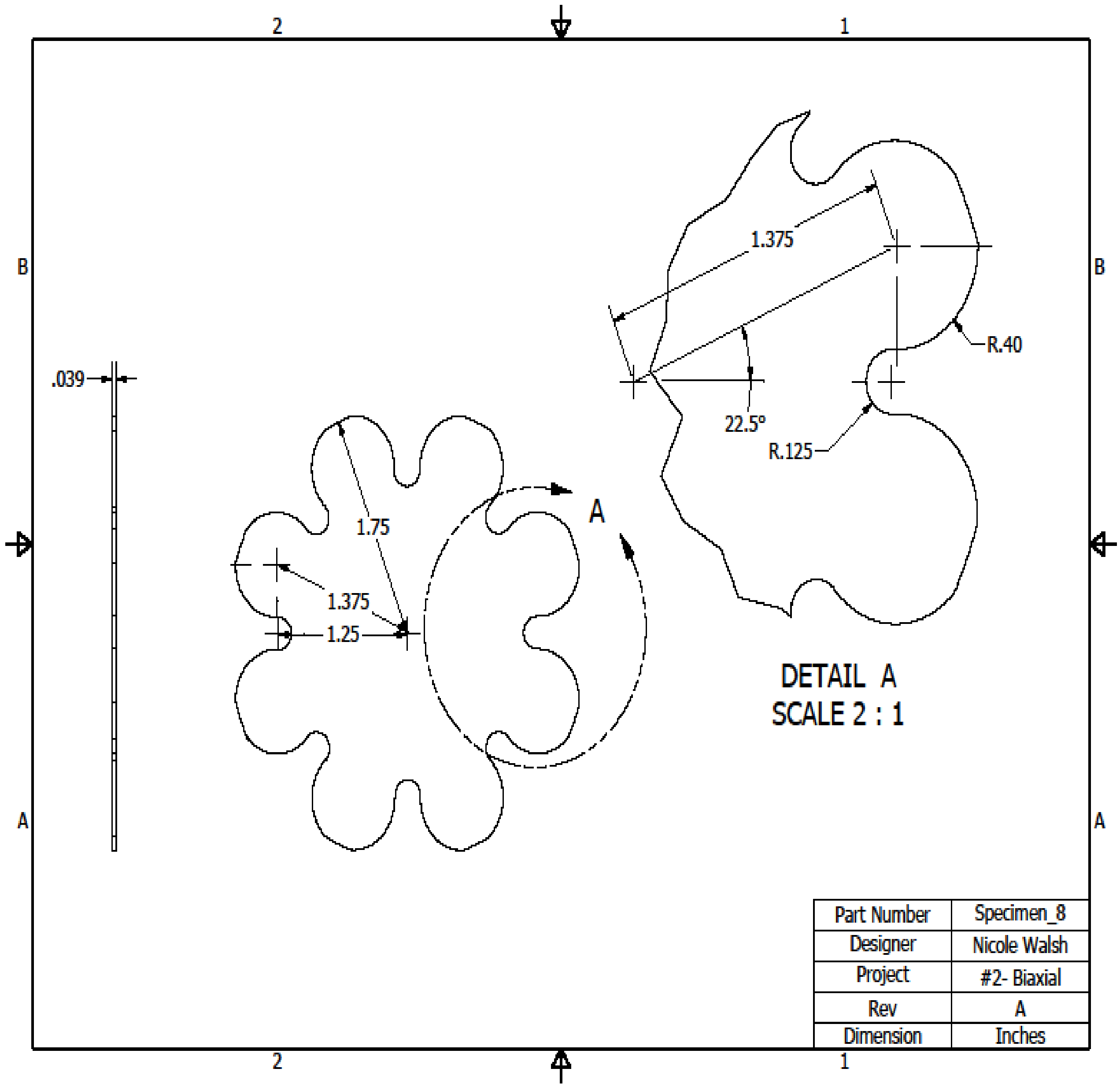
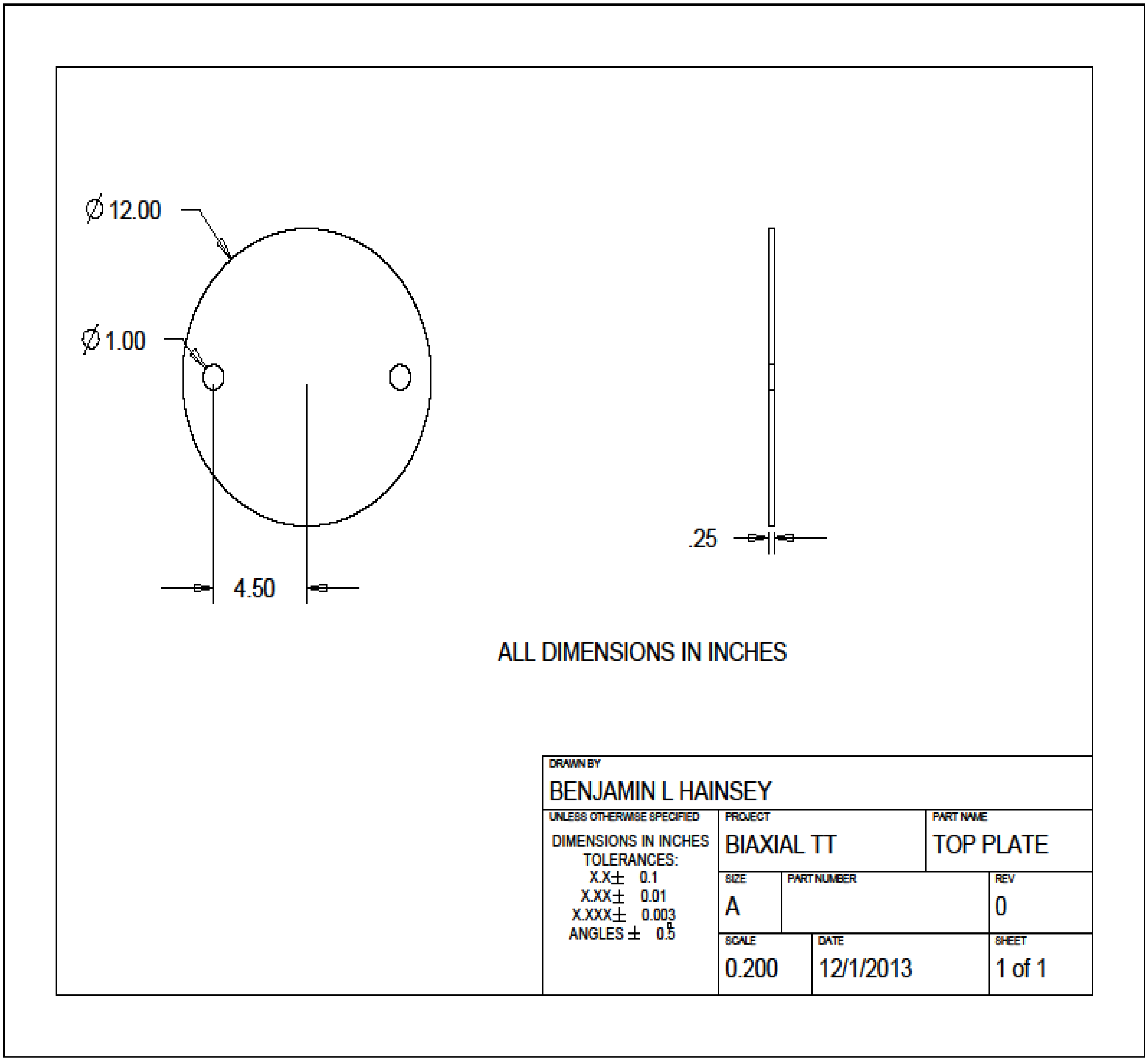
ITEM	QTY	PART NUMBER	DESCRIPTION
1	2	Press_Bottom	
2	6	Press_Rods	
3	6	Press_Springs	
4	1	Press_Top	



2
1
B
A
.039
1.75
1.375
1.25
A
1.375
22.5°
R.40
R.125
DETAIL A
SCALE 2 : 1
Part Number
Specimen_8
Designer
Nicole Walsh
Project
#2- Biaxial
Rev
A
Dimension
Inches



Ø 12.00
Ø 1.00
4.50
.25
ALL DIMENSIONS IN INCHES
DRAWN BY
BENJAMIN L HAINSEY
UNLESS OTHERWISE SPECIFIED
DIMENSIONS IN INCHES
TOLERANCES:
X.X± 0.1
X.XX± 0.01
X.XXX± 0.003
ANGLES ± 0.5°
PROJECT
BIAXIAL TT
PART NAME
TOP PLATE
SIZE
A
PART NUMBER
REV
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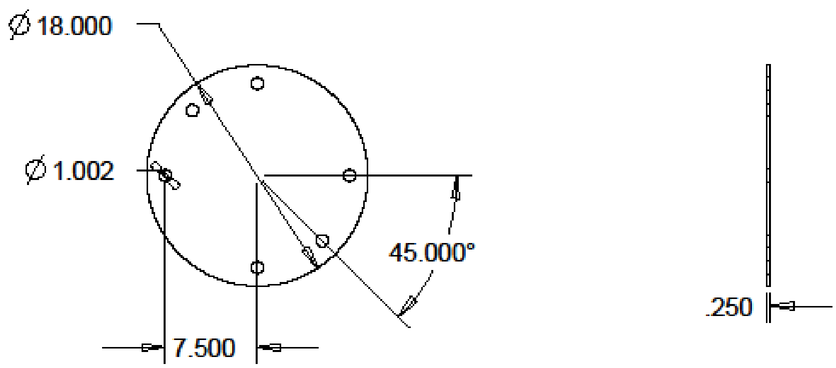
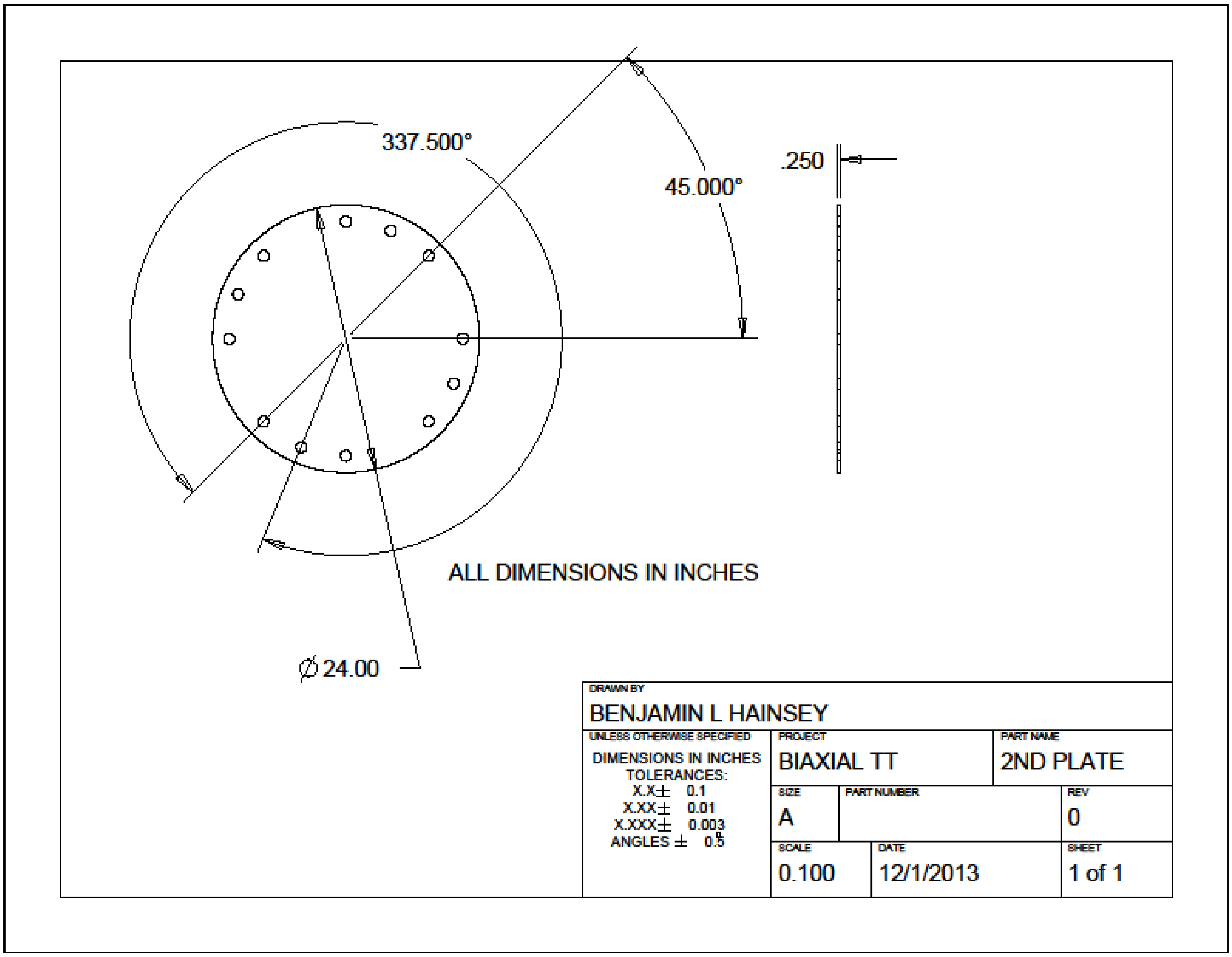
SCALE
0.200
DATE
12/1/2013
SHEET
1 of 1



ALL DIMENSIONS IN INCHES

DRAWN BY BENJAMIN L HAINSEY				
UNLESS OTHERWISE SPECIFIED DIMENSIONS IN INCHES TOLERANCES: X.X± 0.1 X.XX± 0.01 X.XXX± 0.003 ANGLES ± 0.5	PROJECT BIAXIAL TT		PART NAME 3RD PLATE	
	SIZE A	PART NUMBER		REV 0
	SCALE 0.100	DATE 12/1/2013		SHEET 1 of 1


337.500°
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∅24.00
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UNLESS OTHERWISE SPECIFIED
DIMENSIONS IN INCHES
TOLERANCES:
X.X± 0.1
X.XX± 0.01
X.XXX± 0.003
ANGLES ± 0.5°
PROJECT
BIAXIAL TT
PART NAME
2ND PLATE
SIZE
A
PART NUMBER
REV
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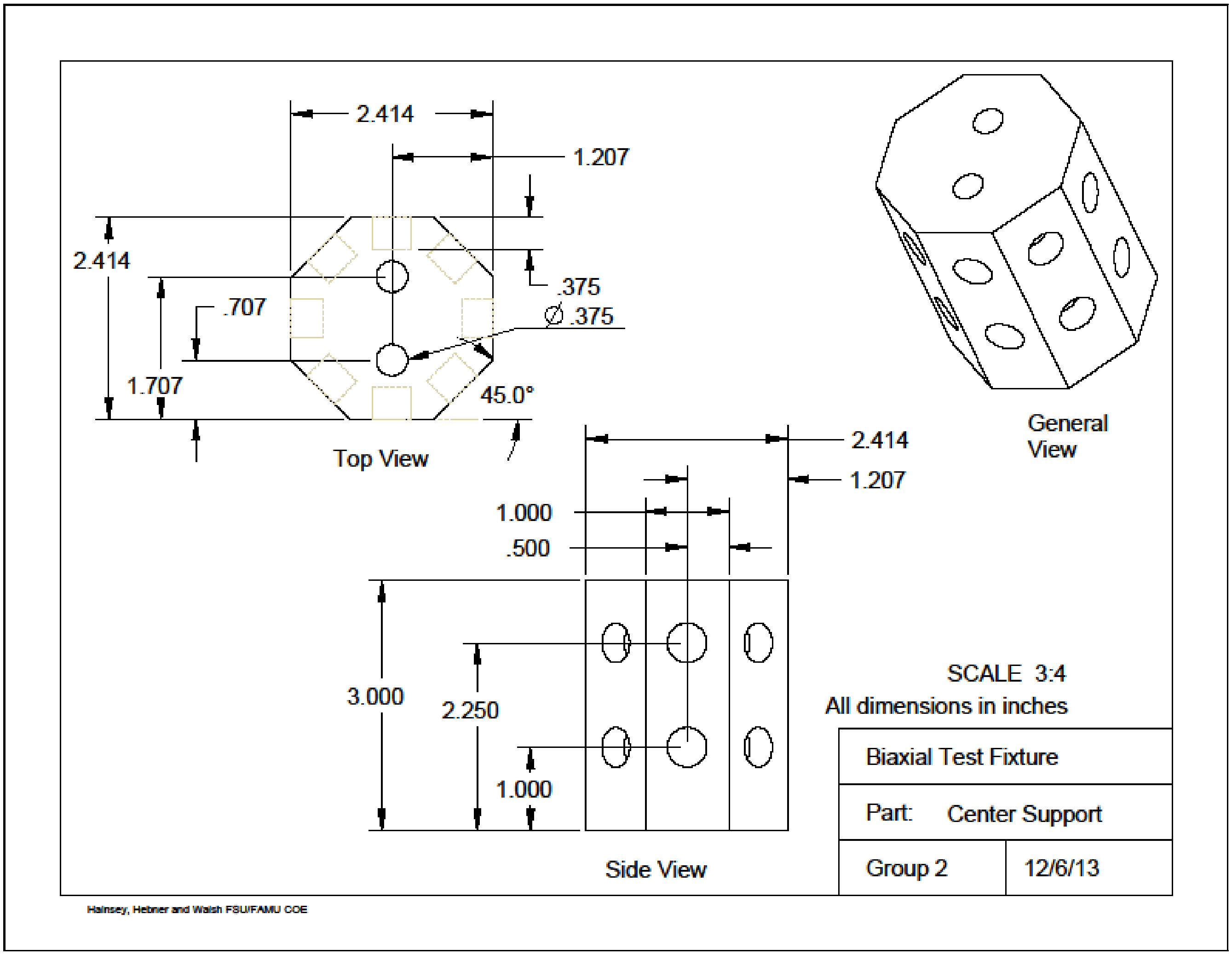
DATE
12/1/2013
SHEET
1 of 1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2.414
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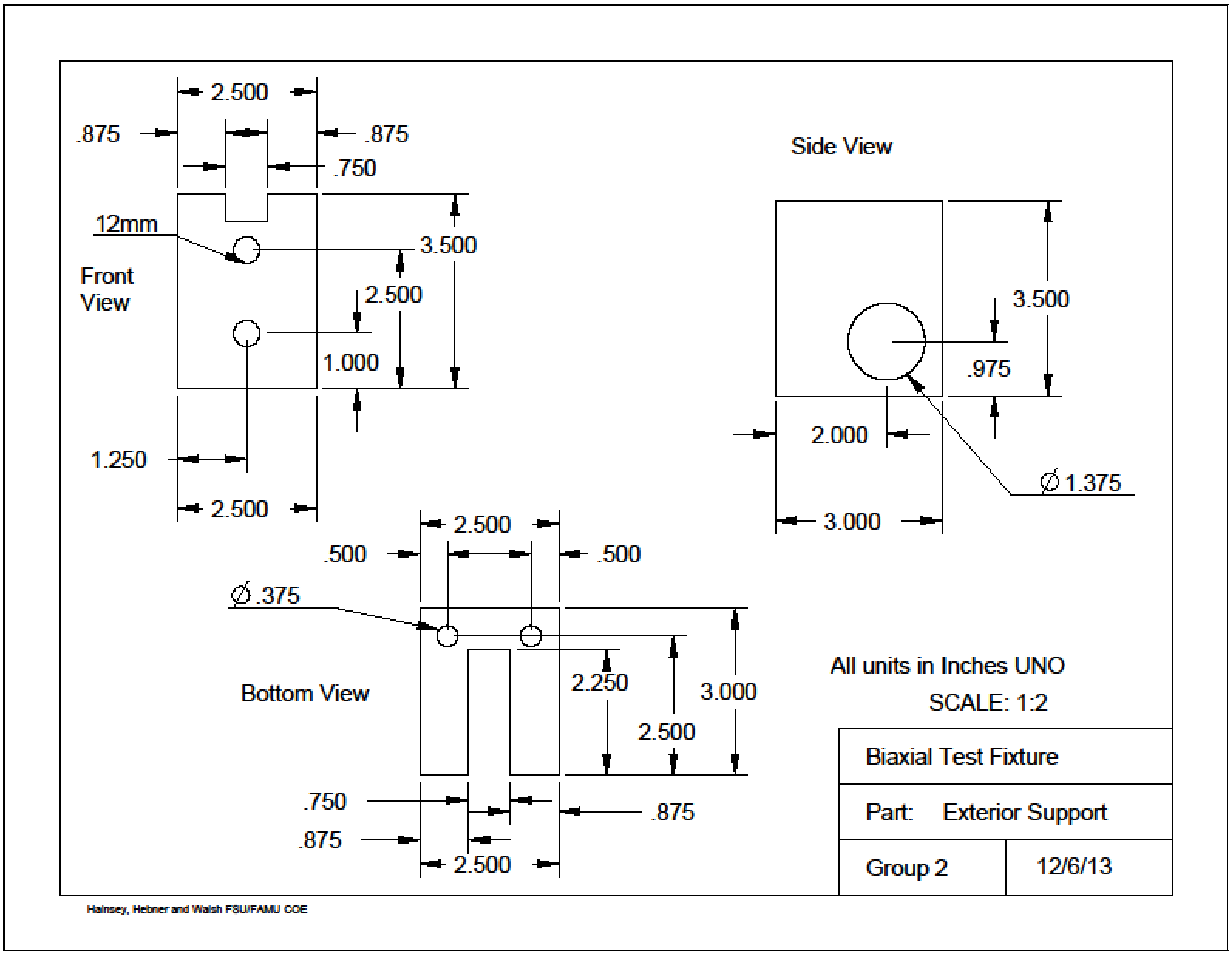
45.0°
Top View
General
View
2.414
1.207
1.000
.500
3.000
2.250
1.000
Side View
SCALE 3:4
All dimensions in inches
Biaxial Test Fixture
Part: Center Support
Group 2
12/6/13
Hainsey, Hebner and Walsh FSU/FAMU COE



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.875
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3.500
2.500
1.000
1.250
2.500
Side View
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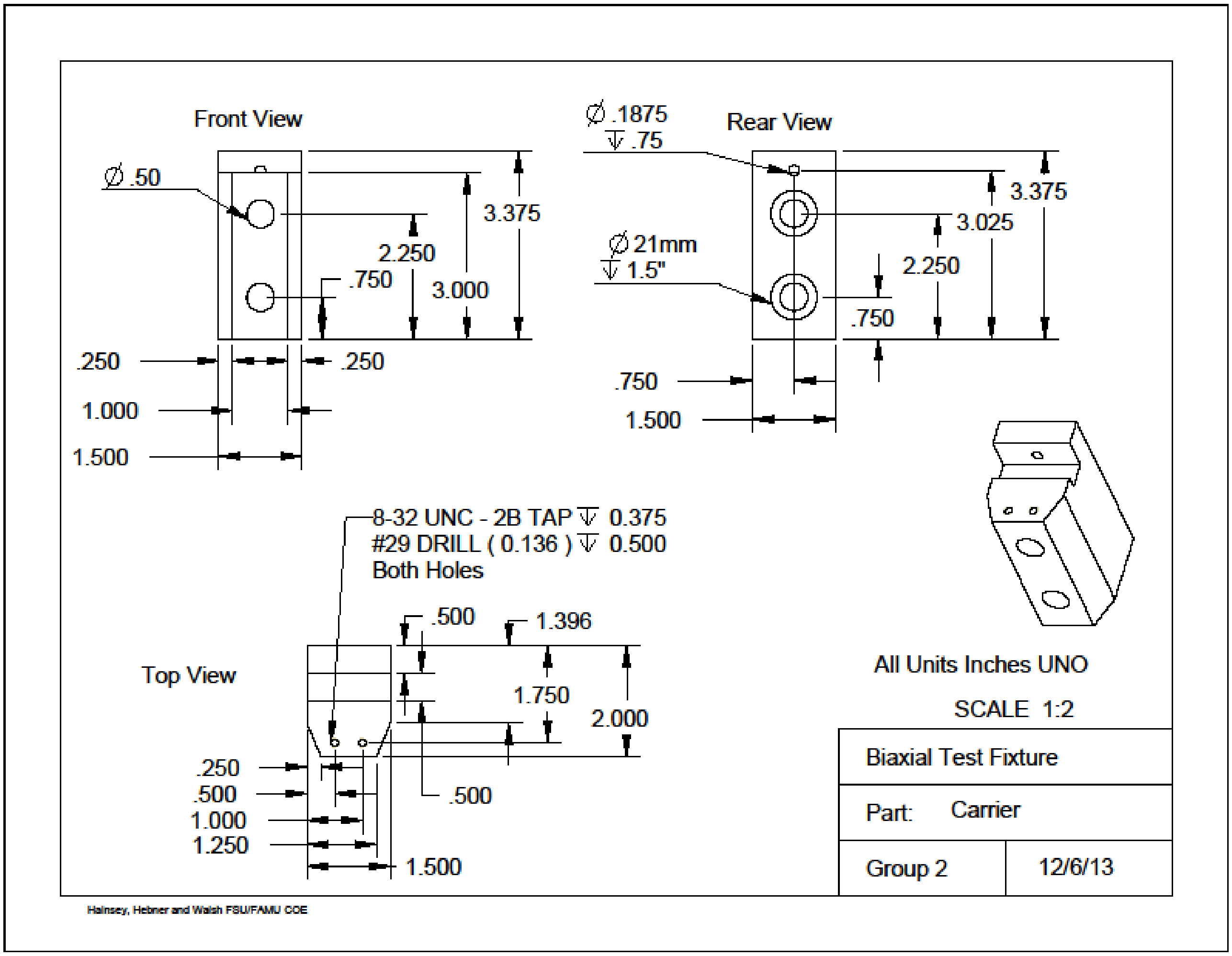
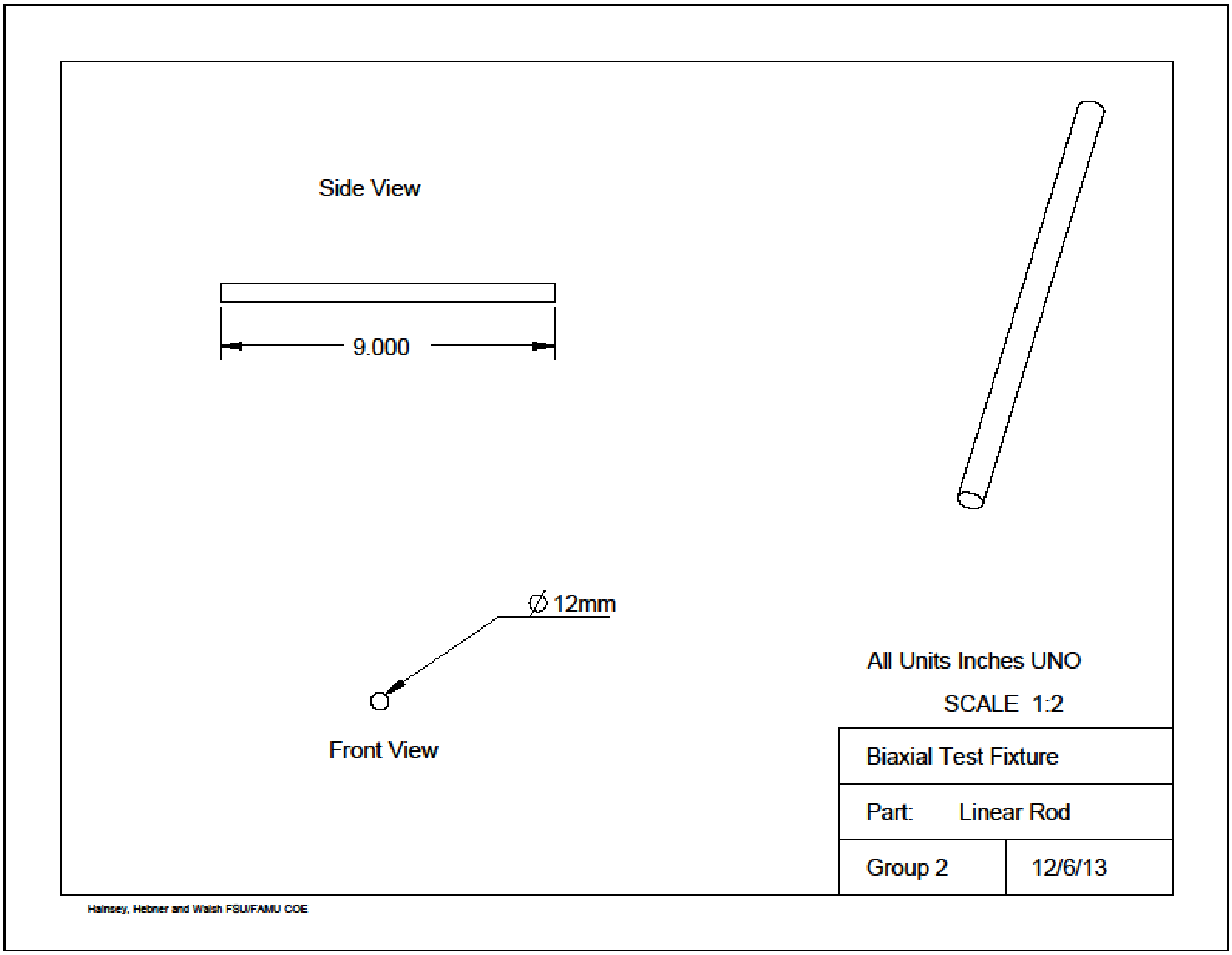
.975
2.000
Ø 1.375
3.000
Bottom View
2.500
.500
.500
Ø .375
2.250
3.000
2.500
.750
.875
.875
2.500
All units in Inches UNO
SCALE: 1:2
Biaxial Test Fixture
Part: Exterior Support
Group 2
12/6/13
Hainsey, Hebner and Walsh FSU/FAMU COE



Front View
Ø .50
3.375
2.250
.750
3.000
.250
.250
1.000
1.500
Ø .1875
↧ .75
Rear View
3.375
3.025
Ø 21mm
↧ 1.5"
2.250
.750
.750
1.500
8-32 UNC - 2B TAP ↧ 0.375
#29 DRILL (0.136) ↧ 0.500
Both Holes
.500
1.396
Top View
1.750
2.000
.250
.500
1.000
1.250
.500
1.500
All Units Inches UNO
SCALE 1:2
Biaxial Test Fixture
Part: Carrier
Group 2
12/6/13
Halnsey, Hebner and Walsh FSU/FAMU COE


Side View
9.000
Ø 12mm
Front View
All Units Inches UNO
SCALE 1:2
Biaxial Test Fixture
Part: Linear Rod
Group 2
12/6/13
Hainsey, Hebner and Walsh FSU/FAMU COE