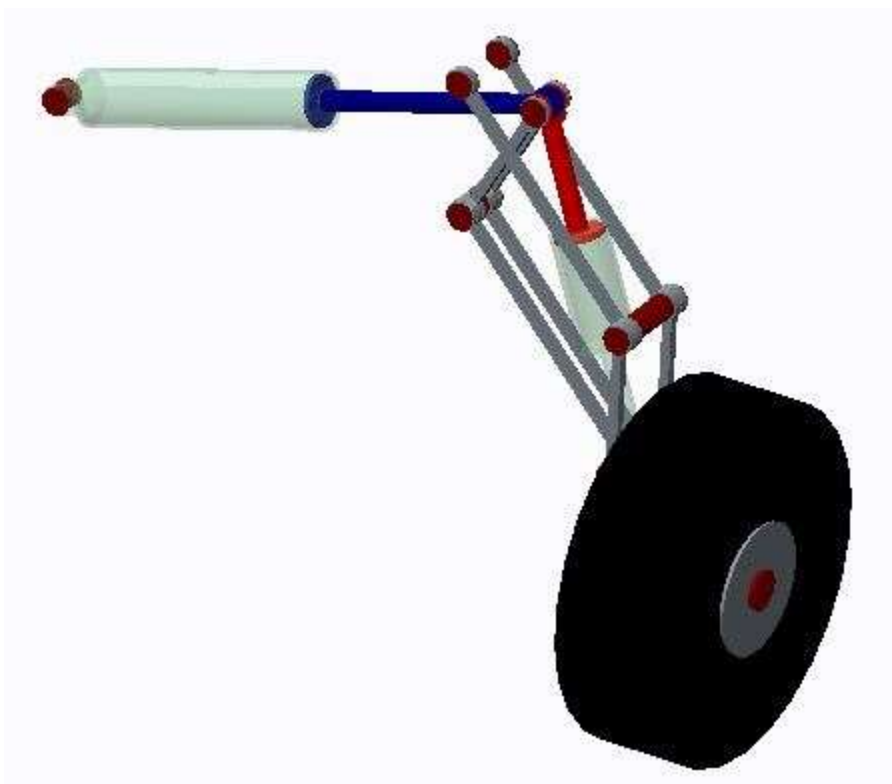
Project 3
Airplane Landing Gear



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ME381

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Executive Summary

The purpose of this project was to utilize Creo to redesign and analyze the properties of an airplane landing gear. Originally, the landing gear had several flaws and inconsistencies which had to be adjusted in order to meet the given specifications and requirements. Several parts had to be modified without affecting the specifications, thus engineering judgment was employed. Every component was carefully analyzed and revised many times to optimize the final assembly in Creo. Once the landing gear was redesigned and assembled, the reaction forces exerted on each pin from the respective links were obtained. In order to test out the overall safety factor, the largest force of each pin was chosen to analyze. Through the use of Creo's Finite Element Analysis, the von Mises stress and convergence graphs of each pin provided the information of where the maximum stresses were on each pin visually. The analysis and calculations made by Creo were verified via hand calculations in order to determine whether or not Creo was accurate in its calculations.

The initial drawing of the landing gear was drawn in a 2 dimensional plane, thus there were no specifications on the depth location of the parts for the assembly, leaving the 3 dimensional design up to good engineering judgment. This assembly was designed with multiple pairs of links on either side in order to maintain a statically equilibrium moment and to distribute the stress along the other pins. After the assembly was completed, a dynamic and static analysis was run to simulate the force on the pins of wheel retraction and the force on the pins from a normal force applied with a shock absorber. To run the analysis, the ends of the pin were constrained with a loading force that was applied through the middle. Through analysis with Creo, the maximum forces obtained for Pins A, B, C, D, E, and F were 6528.31 lbf, 6528.31 lbf, 48261.45 lbf, 67135.85 lbf, 67135.85 lbf, and 53497.79 lbf respectively while the von Mises stresses at the pins were 7.024 ksi, 11.823 ksi, 176.7 ksi, 91.37 ksi, 122.28 ksi, and 74.83 ksi respectively.

After running the analysis and obtaining results, Creo's calculations had to be validated through hand verification to check for accuracy. For the dynamic analysis, a 4 bar linkage problem was performed and for the static analysis, a cantilever beam was calculated to validate. From the hand verifications, there was a 12.1% error for the dynamic analysis and an 11.9% error for the static analysis.

Based on the results, the landing gear did not fail and met all specifications and requirements. Despite the fact that the results were successful, the deviation between Creo and the hand calculations were not trivial. Using the information and calculations, the areas with greater stresses can be analyzed and may be improved by changing materials, hollowing out parts, or dimension tweaking. This project proves the usefulness of the Finite Element Analysis function in Creo and its application for mechanism behavior.

Introduction

The landing gear is perhaps the most significant mechanism for airplanes. It plays a role in liftoff as well as landing. The component is to be designed with extreme calculations as well as engineering judgment to minimize the possibility of failure. Due to aerodynamic properties when flying, airplane landing gears need to have the ability to stow itself away during flight to minimize drag as shown in Figure 1.

It is important to note a few of the special parts of the mechanism such as the hydraulic piston and shock absorber. The hydraulic piston, parts 5 and 6, is the part which was where the retracting forces originates, allowing for the airplane to stow the wheels away. The shock absorber, part 2 would absorb the normal force exerted from the ground when landing.

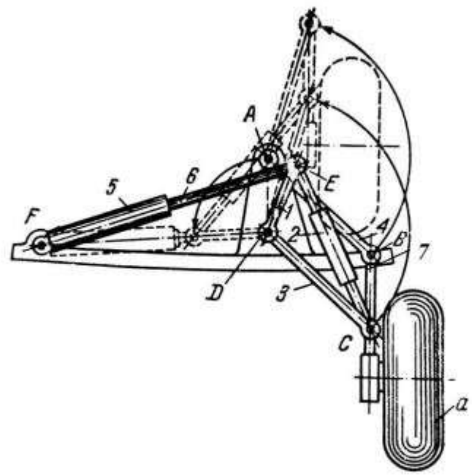


Figure 1: Landing gear general assembly and function. Note the retraction mechanism and fixed points (A, D, and F).

Several parts of the landing gear had dimensions which were open to interpretation, letting there be design freedom in this project. Some of the specifications in this project were the fixed points of Pins A, D, and F, the stretched length of the shock absorber; the landing gear would have to retract to such a position where the wheel would be 42 inches off of the ground.

When performing modifications to the parts, global interferences were to be taken into account to make sure that all the parts can move in synchronization. Based upon my final assembly design, good engineering judgment along with 20/20 hindsight was employed to decide to adjust Link 1, Link 7, Tire, and Pins.

Modifications

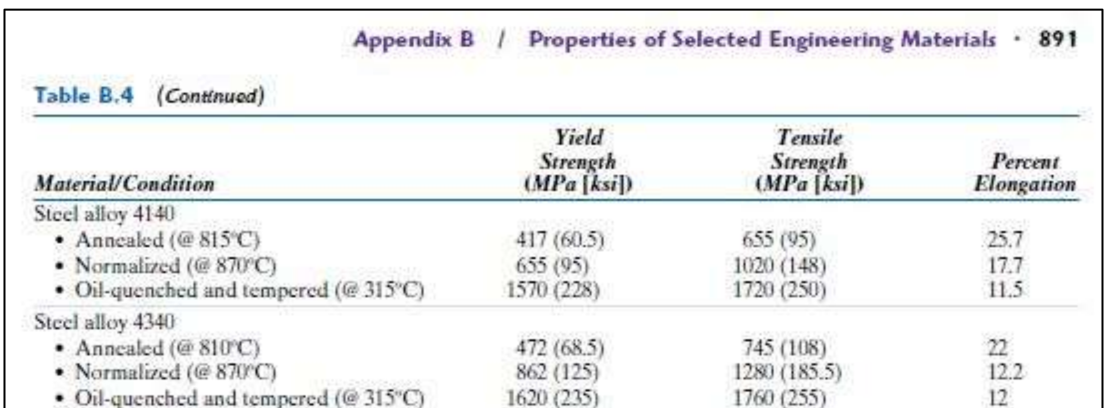
Due to the interference from the shock absorber into Link 1 when retracting, the top portion of Link 1 was trimmed to provide adequate room to accommodate the shock absorber's radial movement as noted in Figure 2. The top

Figure 2: Link 1

half of the link hole aligned with that of the shock absorbers was reduced by half its original thickness. Another adjustment was redesigning Link 7 to accommodate a design change to make the final assembly as even and symmetric as possibly along the front view to avoid a moment on one side as shown in Figure 3. Essentially, instead of placing the link and shock absorber on the same or opposite sides, the shock absorber was placed in between the twin head of the top of Link 7. Ultimately, given the initial restriction, and to eschew from having non-static moment equilibrium, this had to be done in order to not duplicate this link since the wheel is attached to the bottom. The tire was also made hollow, having walls that are 1 inch thick. This was done in order to reduce the weight of the tire for the wheel retraction and landing, thus simulating this as accurately as possible.

Figure 3: Link 7

The next modifications were to the pins and axle, which were adjusted to accommodate the large amount of force which would be exerted. Firstly, several different length pins were created to connect the different positions of the appropriate coaxial links. Then, the radius of the inner pin diameter was raised from 2 inches to 2.47 inches to minimize stress. Next, the materials of the pins, axle, and respective caps were changed from 1030 quenched and tempered steel to 4340 oil quenched and tempered steel. This was done since the densities for both are same thus not affecting the overall mass and the latter has a yield strength of 235 ksi as opposed to the 1030 steel, which has a yield strength of 64 ksi as noted in Figure 4. Also, the



Appendix B / Properties of Selected Engineering Materials • 891

Table B.4 (Continued)

Material/Condition	Yield Strength (MPa [ksi])	Tensile Strength (MPa [ksi])	Percent Elongation
Steel alloy 4140			
• Annealed (@ 815°C)	417 (60.5)	655 (95)	25.7
• Normalized (@ 870°C)	655 (95)	1020 (148)	17.7
• Oil-quenched and tempered (@ 315°C)	1570 (228)	1720 (250)	11.5
Steel alloy 4340			
• Annealed (@ 810°C)	472 (68.5)	745 (108)	22
• Normalized (@ 870°C)	862 (125)	1280 (185.5)	12.2
• Oil-quenched and tempered (@ 315°C)	1620 (235)	1760 (255)	12

Figure 4: Properties of 4340 Oil Quenched and Tempered Steel. From Materials Science and Engineering, An Introduction, 9th Ed by W. D. Callister Jr.

cost of this steel would not vary too much from that of the original since the industrial price of steel is based from the supply and demand of the readiness of the corresponding alloys, which typically do not exceed a difference of $100 per ton. Also spending money to optimize the safety factor will be small when compared to that of the cost of building the entire plane. Ultimately, changes applied were done in order to decrease the stress onto the pins exerted from their respective links to improve the safety factor.

Assembly

From the modifications and parts, the final assembly was put together through PTC Creo Parametric (referred to as Creo from now on). For this, only the pins on the fixed pins were constraint to be rigid, the hydraulic piston and shock absorber were assembled with slider connections, and the other pins and links were constraint via “pinning”. The rest of the materials were created with the materials stated from the assignment. After assembling all parts with no global interferences and with the ability to retract in the desired motion, the dynamic analysis and static analysis was run to simulate and determine the forces acting on each pin. The dynamic analysis had a motor placed inside of the hydraulic piston to imitate the action of airplane retraction. The static analysis had a spring placed in the shock absorber with a normal force of 26500 lbf applied from the bottom of the wheel, much like when an airplane is landing. After these simulations, the highest forces at each pin were used to find the von Mises stress criterion. The maximum forces ranged from 2.95 lbf to 67135.85 lbf. The maximum forces were then applied to the load in the Finite Element Analysis.

Since this project allowed for an open-ended interpretation of an airplane landing gear, modifications were made to the parts and design. The adjustments made were performed in order to get rid of global interferences, improve the equilibrium state of the assembly, and to optimize the safety factor of the pins to reduce failure. The calculated results from Creo are in the results section.

Results of Analysis

Exhibited below is the overall maximum resulting forces, yield stress, and safety factor (ratio of the material's yield stress to the maximum stress) taken from both the dynamic and static analysis on each pin and the axel.

Pin	Maximum Dynamic Force (lbf)	Maximum Static Force (lbf)	Overall Maximum Force (lbf)	Maximum Stress (ksi)	Material Yield Stress (ksi)	Safety Factor
Pin A	206.20	6528.21	6528.21	7.024	235	33.46
Pin B	200.14	6528.21	6528.21	11.823	235	19.88
Pin C	599.37	48261.45	48261.45	176.7	235	1.33
Pin D	924.55	67135.85	67135.85	91.37	235	2.57
Pin E	920.93	67135.85	67135.85	122.28	235	1.92
Pin F	665.11	53497.79	53497.79	74.83	235	3.14
Axle	271.97	26499.95	26499.95	28.237	235	8.32

Table 1: Maximum Forces, Stresses, and Safety Factors of all pins

Table 1 shows the tabulated values of the pin's overall maximum forces and stress from Creo, which is then compared with the values generated via hand calculations. Verification is imperative in order to know if the program is accurate and precise. According to the hand calculations located in Appendix II, at Pin A there was a von Mises Stress of 6.82 ksi, however Creo had calculated the stress to be 7.02 ksi, resulting in a percent error of 11.9%. Also from the kinematic analysis, which was verified from executing the 4 bar linkage involving Links 3, 4, and 7; the hand calculations had garnered a value of 773.89 m/s^2 for the acceleration about point C whereas Creo calculated a value of 690.25 m/s^2. This resulted in a percent error of 12.1%. These results are displayed below in Table 2.

Dynamic Analysis (4 bar linkage)				**Static Analysis (cantilever beam)**			
Variable	Creo Value	Hand Verified	Percent Error	Variable	Creo Value	Hand Verified	Percent Error
Stress	7.02 ksi	6.82 ksi	11.9%	Acceleration	690.25 m/s^2	773.89 m/s^2	12.1%
				k	40000 lbf/in	-	-
				x	1.22 in	-	-
				$F=kx$	48800 lbf	-	-

Table 2: Creo computed results compared with the hand calculated verification.

Discussion of Results

As mentioned earlier, a servo motor was utilized by placing it on the hydraulic piston. With the motor having a given velocity on the piston connection and it being able to generate acceleration, forces and constraints are put on the pins as well as other components of the landing gear. After running the motion and force analysis, Creo is able to create a Force vs Displacement graph by recording the forces that have an effect on the masses of the pins and comparing that to the piston displacement. The van Mises stress is calculated though finding the maximum forces that each pin will undergo in both the dynamic and static analysis and running it through the Finite Element Analysis to generated and display both a fringe graph along with a convergence graph. The maximum stress of each pin can be determined through comparison of the von Mises with the yield stress of that of the material. In this project, the materials for all the pins are 4340 Oil Quenched and Tempered steel with a yield stress of 235 ksi. The maximum stress compared to the yield stress can be used to find the Safety Factor.

Finite Element Analysis

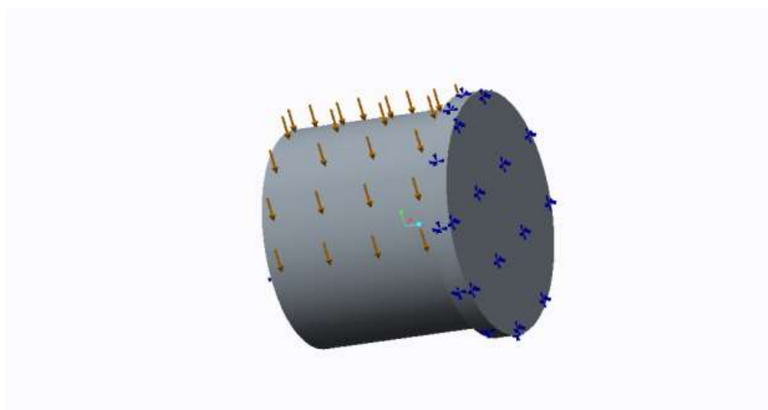


Figure 5: Pin F with force and constraints

Typically when running the finite element analysis, the applied load force and constraints for the pin look something along the lines of Figure 5. As noted, the blue crosshairs indicate constraints while the orange arrows pointing into the pin is the force. In this situation, the applied force was only from Link 3. The reason why the load only appears on one side is because when the link is rotating about the pin, the force is only where the two surfaces meet, thus leaving the other half alone. The force applied to Pin A is 6528.21 lbf, therefore producing a stress of 7.02 ksi. As previously mentioned, the safety factor of the pin is the material yield stress divided by the maximum stress of the assembly. Essentially, the higher the safety value, the less likely the rate of failure is for the pin since it would mean that the maximum stress is for the pin is less than the material yield stress, meaning that there is no deform. Higher stress is always where the force is greater over a small area. As shown in Appendix II the same procedure for the other pins is done the same method.

Kinematic and Static Analysis

The kinematic analysis was also performed on the landing gear in Appendix II. This is essentially a 4 bar linkage problem with the joints at Pin A, Pin B, Pin C, and Pin D. The ground was defined to be the axis from Pin A to Pin D. Since Creo had the ability to calculate the V_A , the V_B and A_B could be determined. The lengths of the bars are known. After performing the hand calculation, the value was compared to the one calculated by Creo. Overall, the percent error was 11.9%.

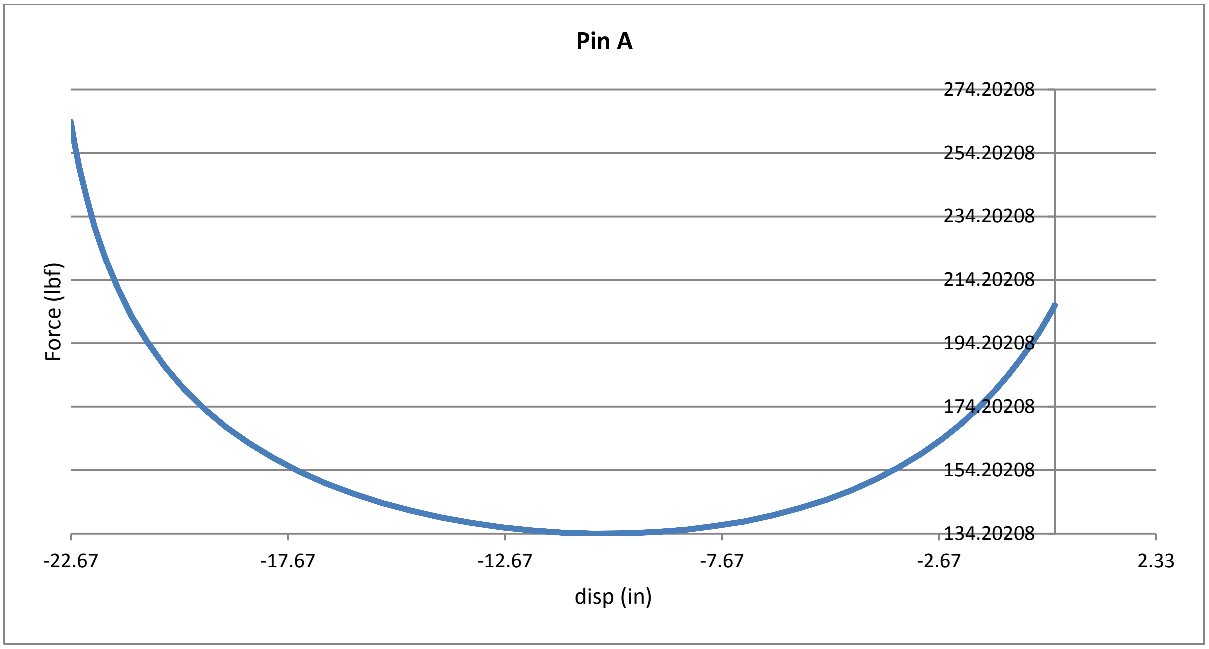
The static analysis was performed by inserting a spring within the shock absorber and a normal force of 26500 lbf from the ground to the wheel. The requirement for this situation was that with the given normal force, the wheel can be lifted at precisely 2 inches off of the ground. It was then determined that the shock absorber abides by Hooke's Law $F=kx$, with k being the spring constant determined from Creo which was seen to raise the wheel as close as possible to a height of 2 inches from the normal force. It was also determined that the spring displacement, x, is 1.22 inches, making the Creo value for force be 38800 lbf. In conclusion, with a roughly 10% accuracy, Creo is a tool that is relatively accurate.

Summary and Conclusion

From the results of both the analysis numerical values generated by Creo alongside with the hand verified results, it can be safe to say that the design of the landing gear is not only fully functional, but also devoid of any global interference. From both the kinematic and static analyses, it is evident that the landing gear assembly has the ability to withstand the force generated from a motor along with a 26500 lbf normal force given to adjust for the spring in the shock absorber. The original design was remodeled in order to make the landing gear more symmetric and not result in any non-equilibrium moments arise and adding multiple links allowed for the distribution of stress and force. By looking at the safety factors for this project, it can be determined that the safety factor of the overall design is a 1.33 due to the fact that it is the lowest safety factor. Some forces which were unaccounted for were the effect of external forces on the pins such as air resistance.

In order to improve this mechanism, several components may be adjusted such as hollowing out the pins and links to decrease the weight of the landing gear as well as slowing down the servo motor. Despite the slightly low safety factor, the overall design can both satisfy the specifications as well as carry out the given specific tasks of raising the wheel to 42 inches and having a spring which can raise the wheel 2 inches from adjusting the spring constant. With a longer time for testing, this design may be used in the future for practical purposes.

APPENDIX I – GRAPHS/CALCULATIONS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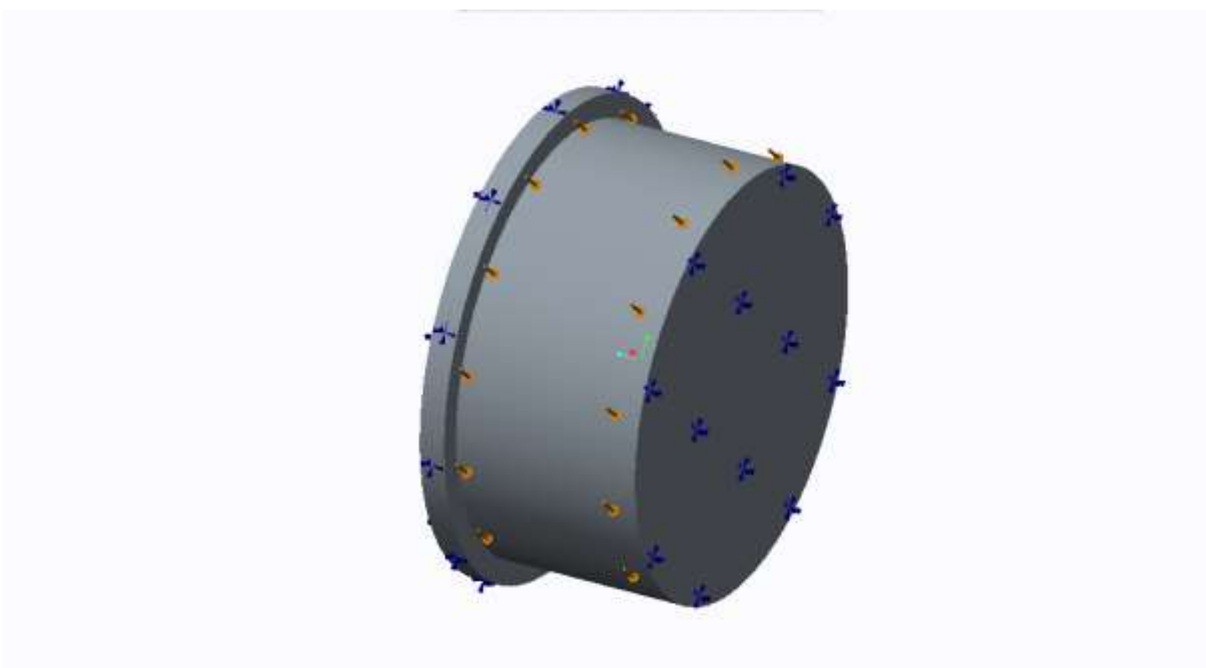
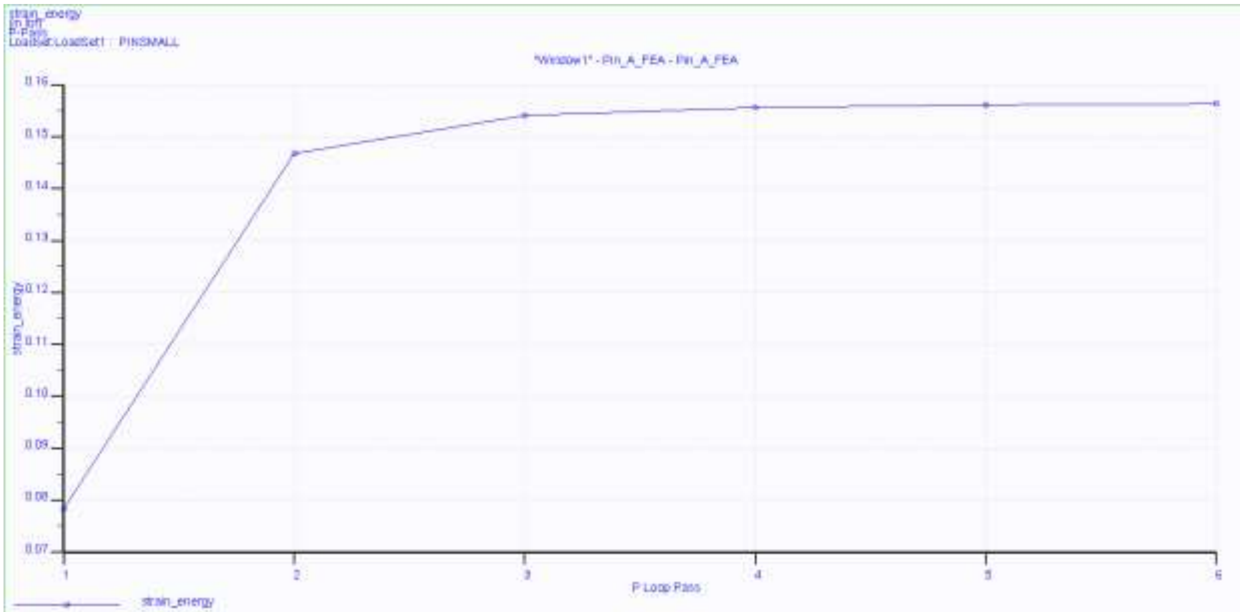
274.20208
254.20208
234.20208
214.20208
194.20208
174.20208
154.20208
134.20208
Force (lbf)
-22.67
-17.67
-12.67
-7.67
-2.67
2.33
disp (in)





strain_energy
in lbf
P-Pass
Loadset:LoadSet1 : PINSMALL
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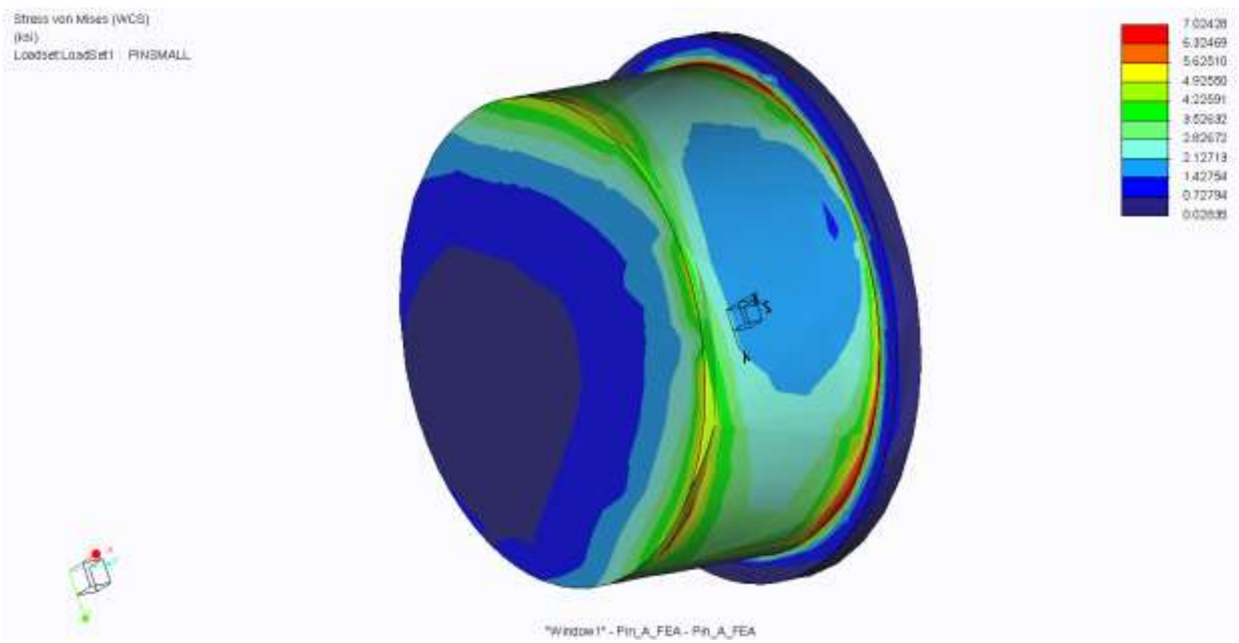
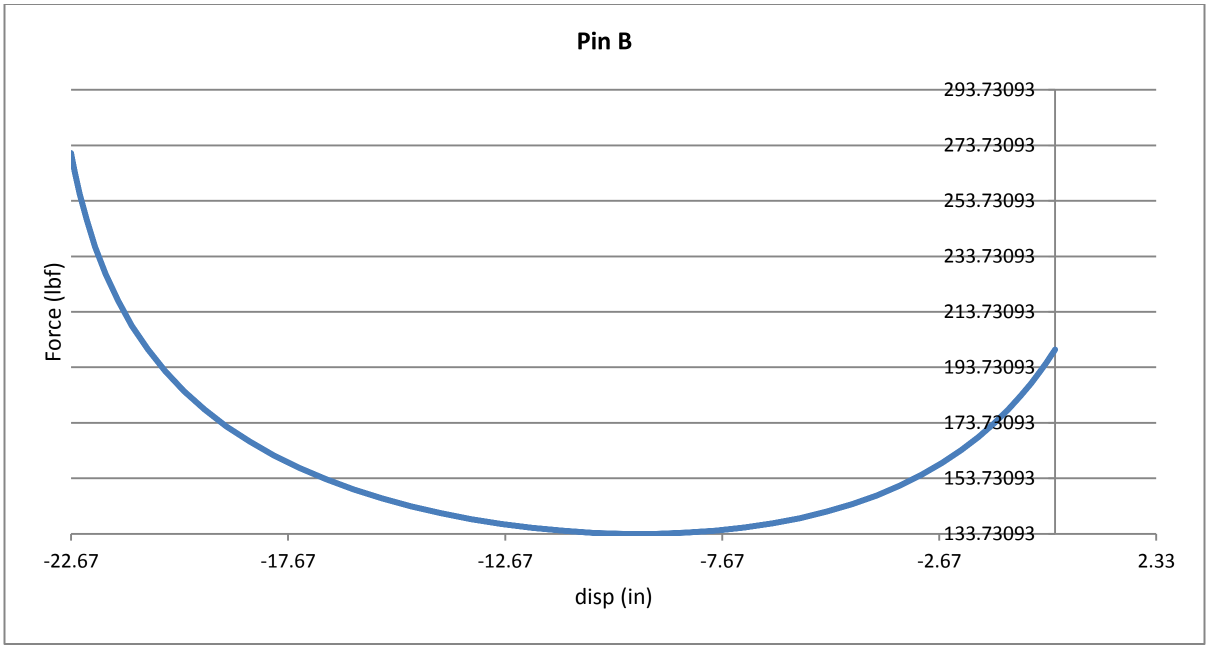
0.16
0.15
0.14
0.13
0.12
0.11
0.10
0.09
0.08
0.07
strain_energy
1
2
3
4
5
6
P Loop Pass
strain_energy



Stress von Mises (WCS)
(ksi)
Loadset:LoadSet1 : PINSMALL
7.02428
6.32469
5.62510
4.92550
4.22591
3.52632
2.82672
2.12713
1.42754
0.72794
"Window1" - Pin_A_FEA - Pin_A_FEA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233.73093
213.73093
193.73093
173.73093
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Force (lbf)
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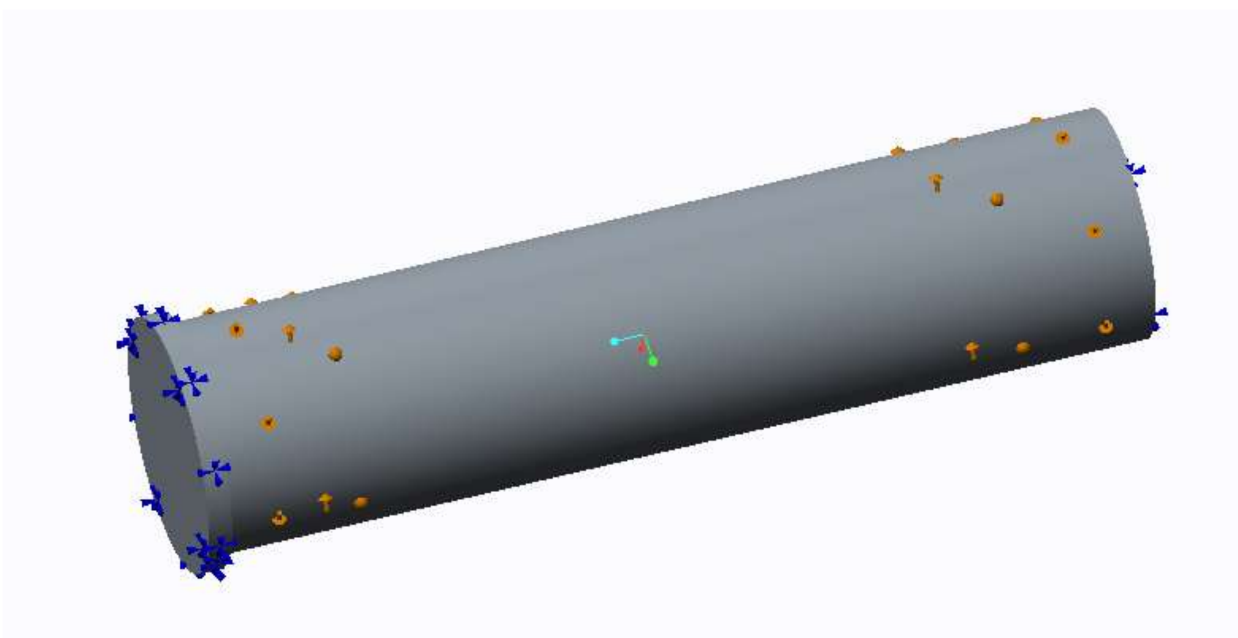
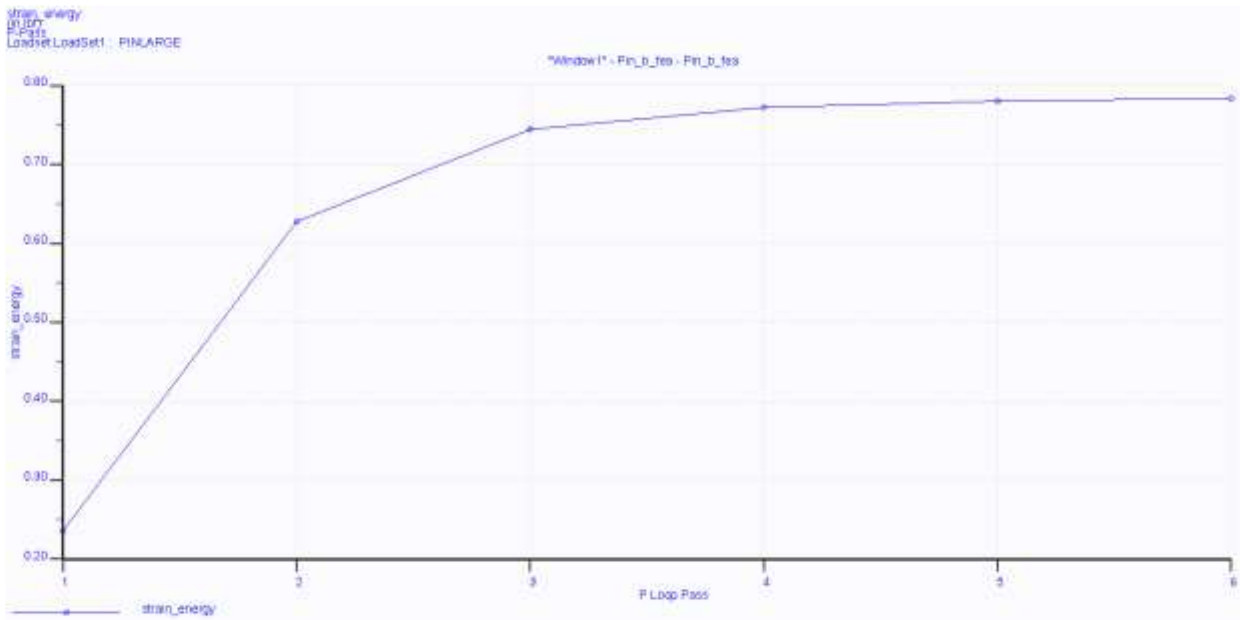
-17.67
-12.67
-7.67
-2.67
2.33
disp (in)

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P-Pass
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0.80
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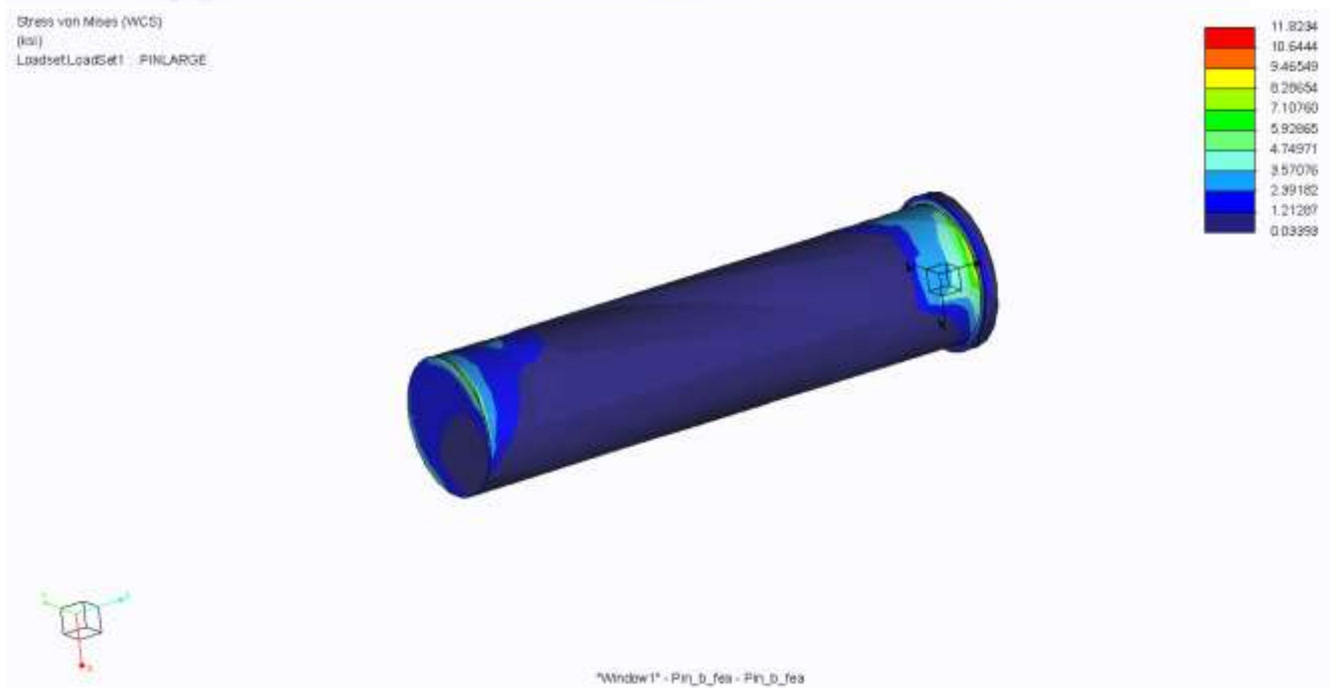
0.60
0.50
0.40
0.30
0.20
strain_energy
1
2
3
4
5
6
P Loop Pass
strain_energy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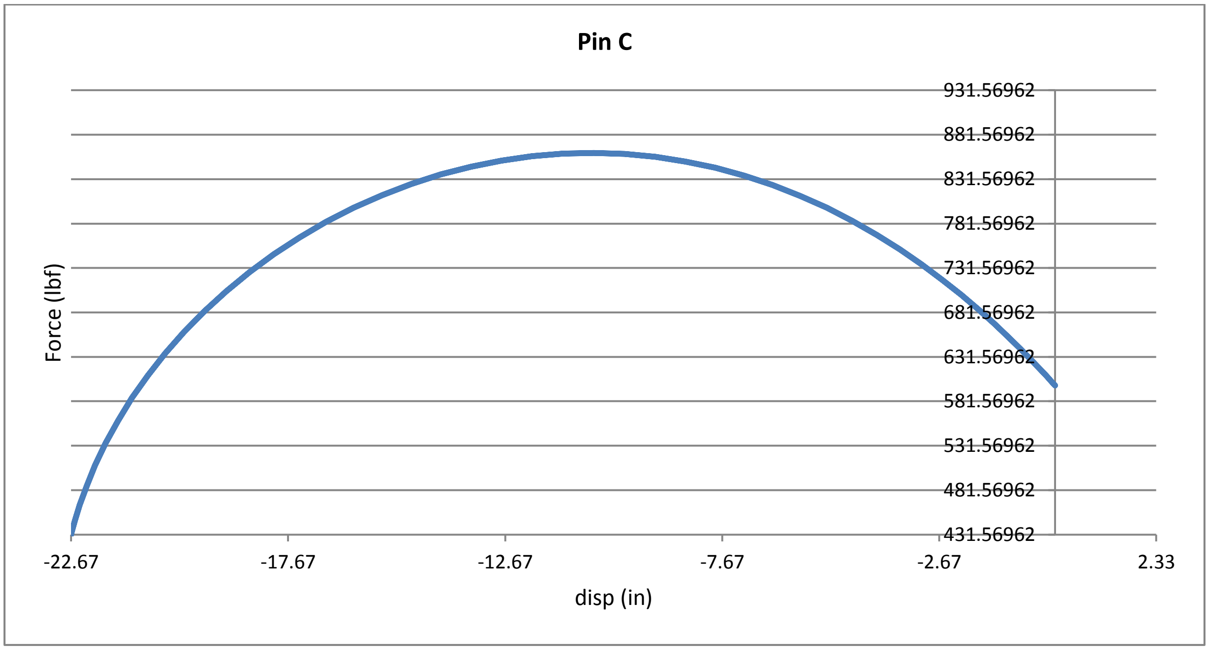
(ksi)
LoadsetLoadSet1 : PINLARGE
11.8234
10.6444
9.46549
8.28654
7.10760
5.92865
4.74971
3.57076
2.39182
1.21287
0.03393
"Window1" - Pin_b_fea - Pin_b_fea


Pin C
Force (lbf)
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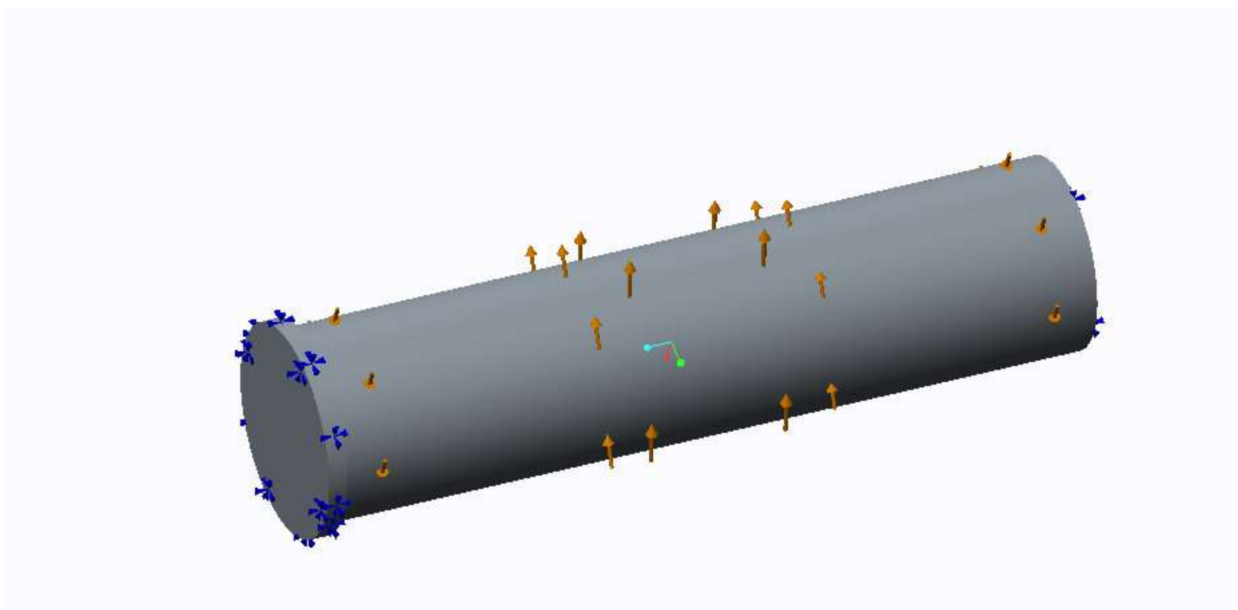
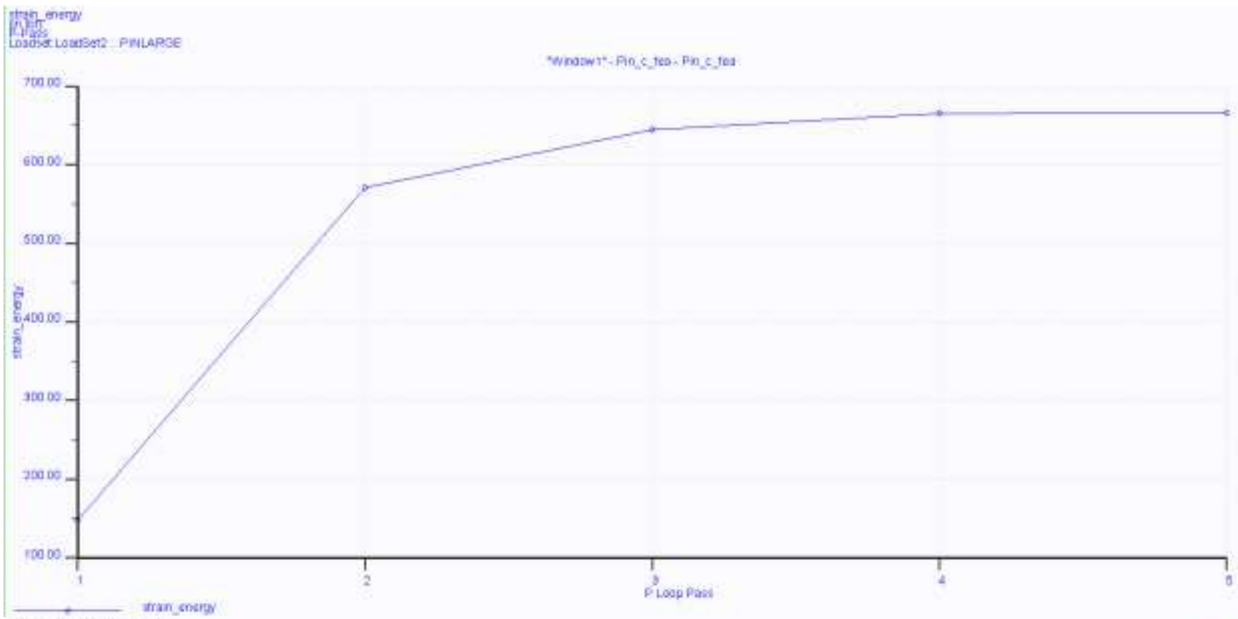
881.56962
831.56962
781.56962
731.56962
681.56962
631.56962
581.56962
531.56962
481.56962
431.56962
-22.67
-17.67
-12.67
-7.67
-2.67
2.33
disp (in)

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300.00
200.00
100.00
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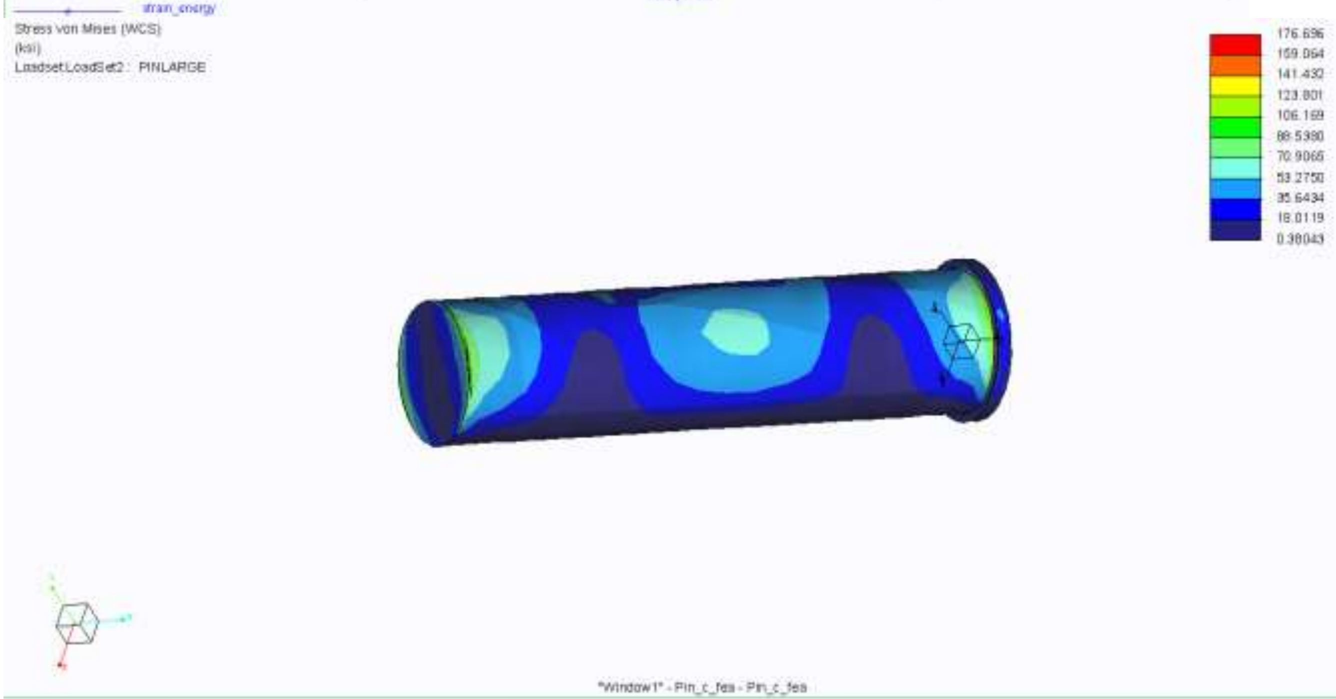
1
2
3
4
5
P Loop Pass
strain_energy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35.6434
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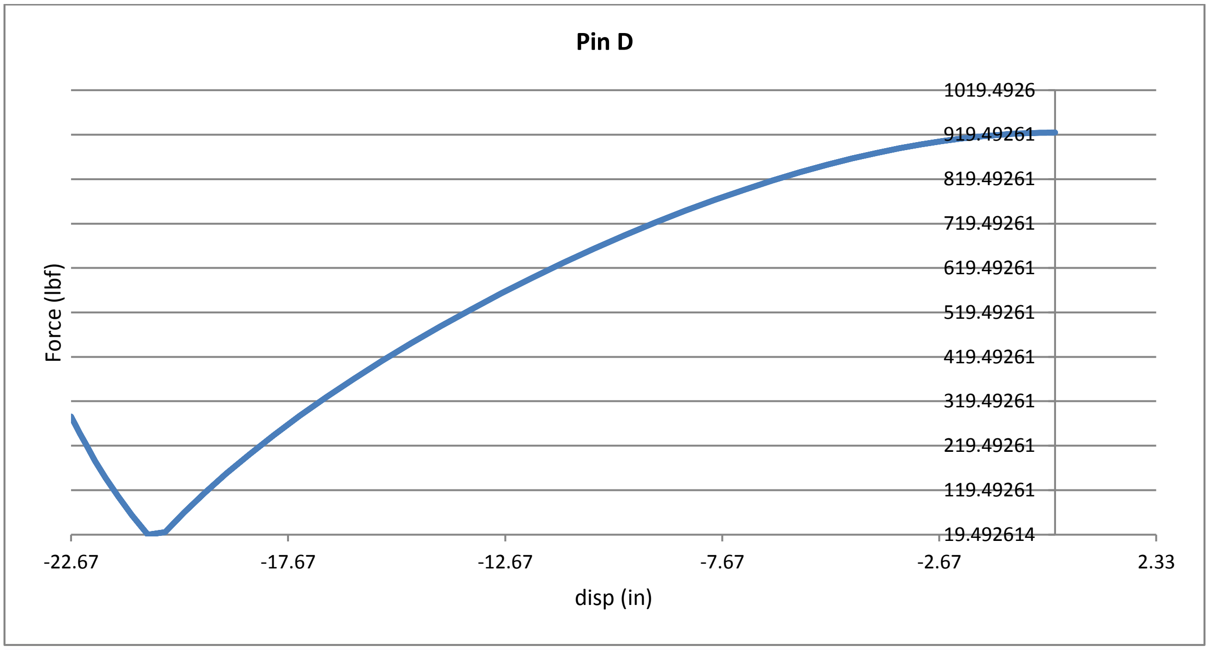
0.38043
"Window1" - Pin_c_fea - Pin_c_fea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919.49261
819.49261
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619.49261
519.49261
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319.49261
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Force (lbf)
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-7.67
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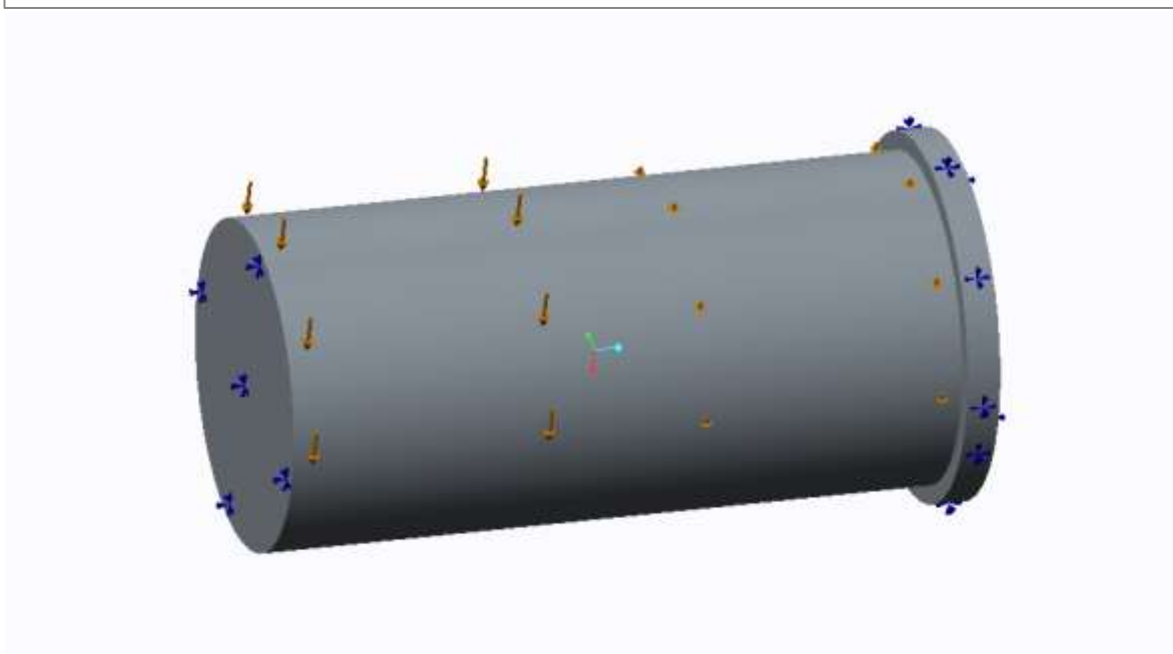
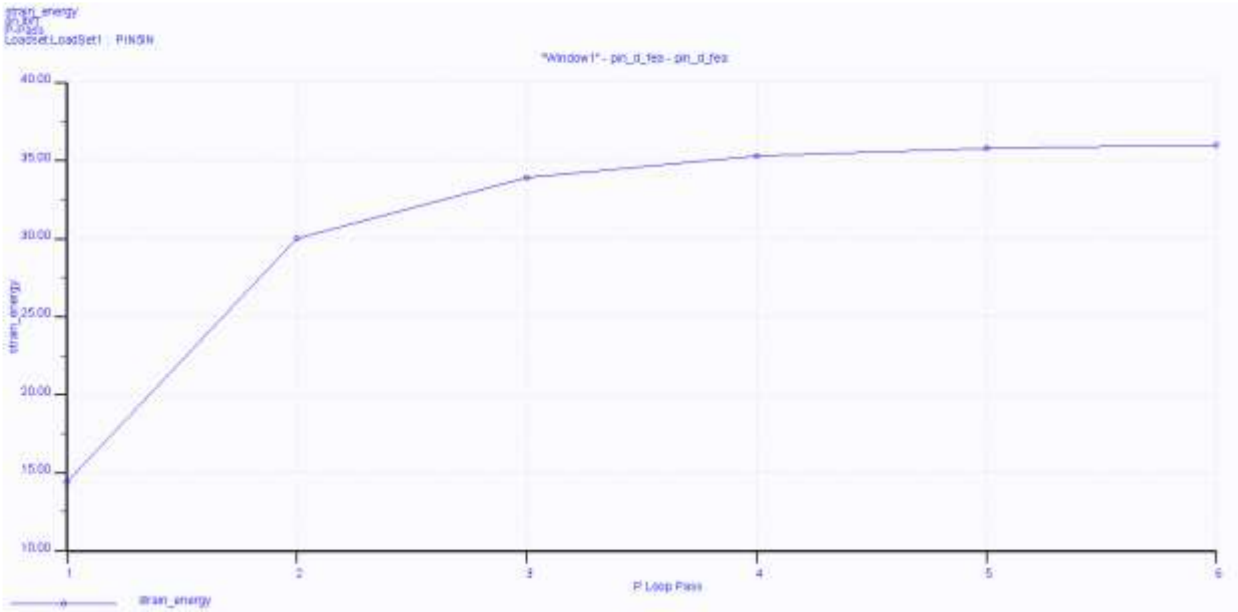
disp (in)

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10.00
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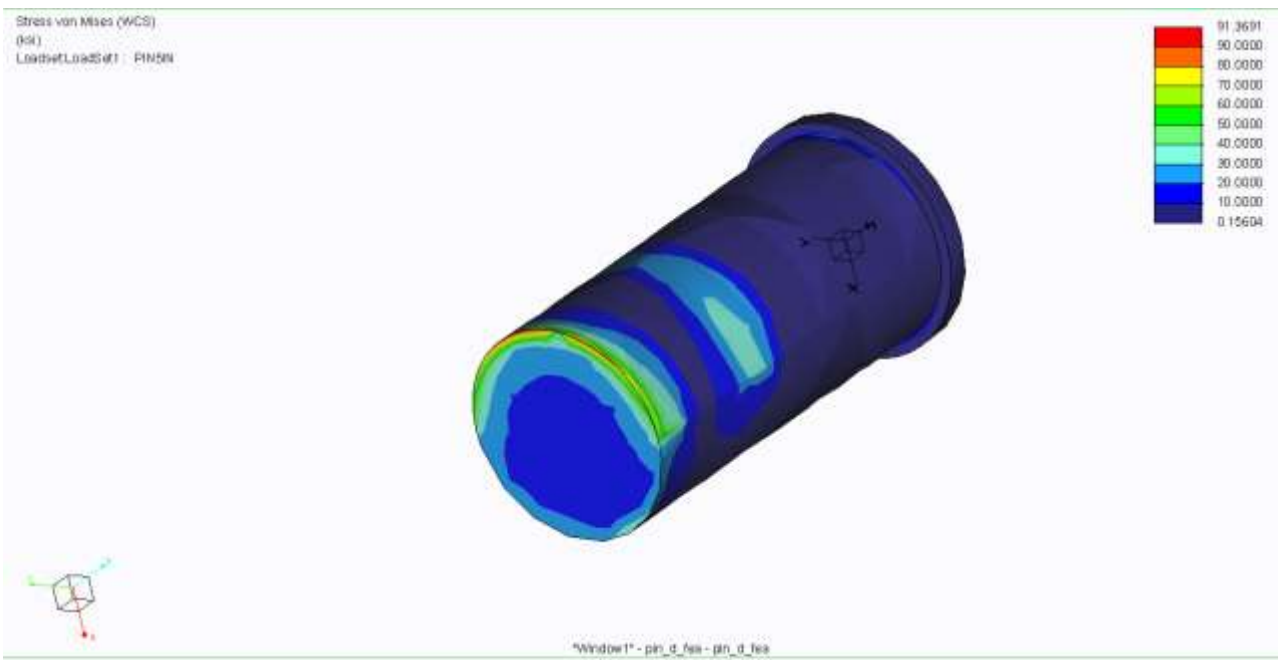
1
2
3
4
5
6
P Loop Pass
strain_energy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(ksi)
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60.0000
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20.0000
10.0000
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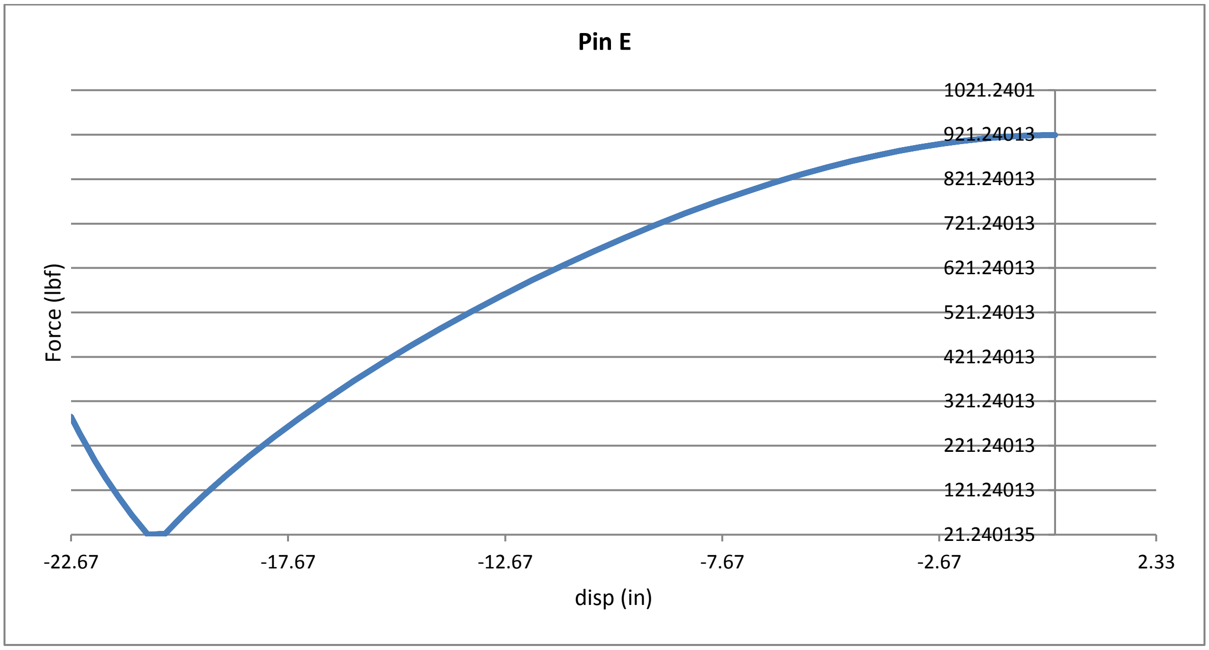
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821.24013
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621.24013
521.24013
421.24013
321.24013
221.24013
121.24013
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Force (lbf)
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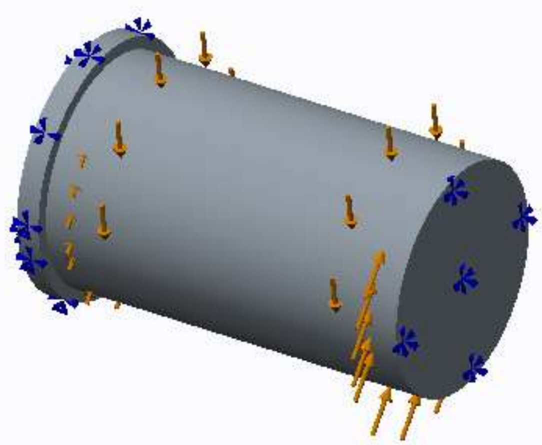
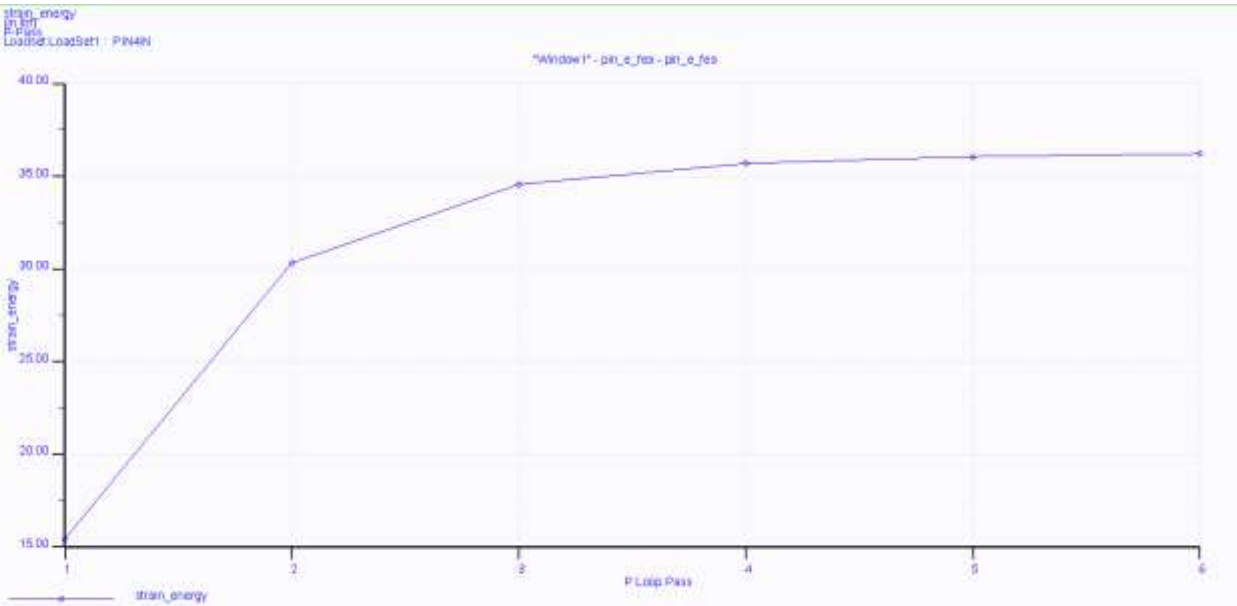
-17.67
-12.67
-7.67
-2.67
2.33
disp (in)

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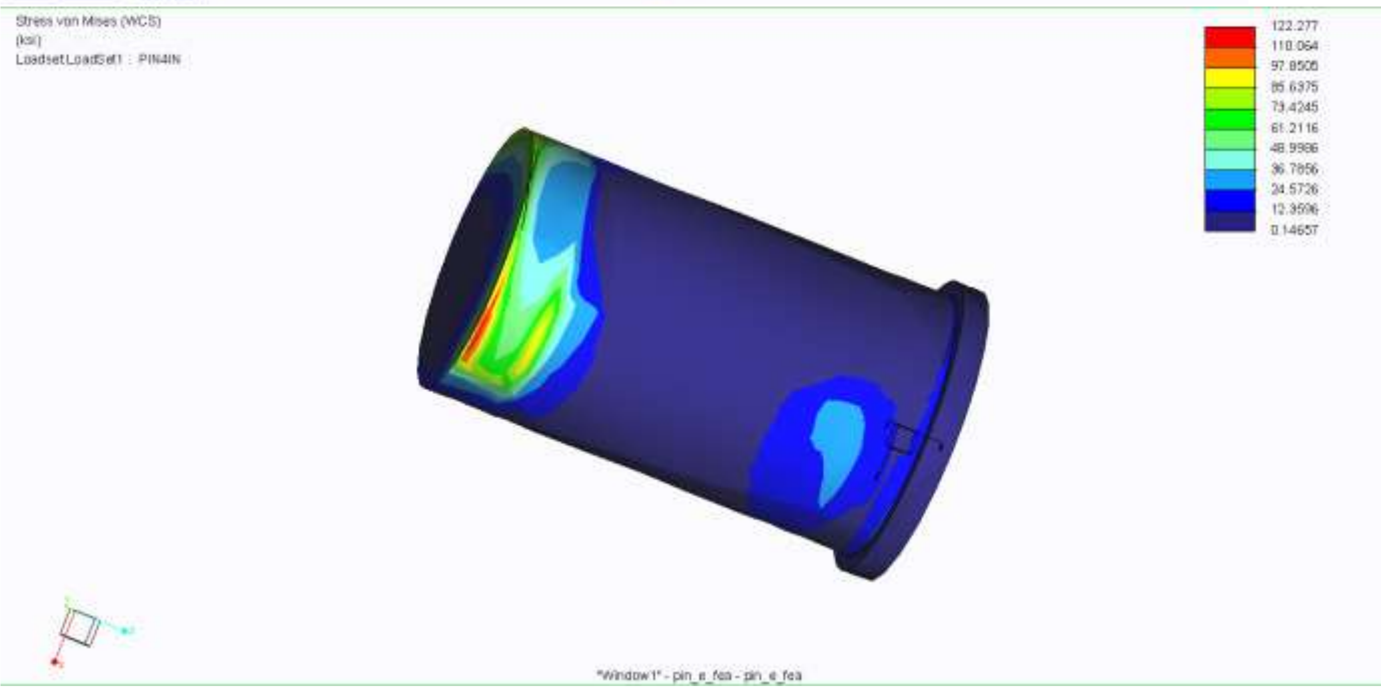
30.00
25.00
20.00
15.00
strain_energy
1
2
3
4
5
6
P Loop Pass
strain_energy


Stress von Mises (WCS)
(ksi)
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110.064
97.8505
85.6375
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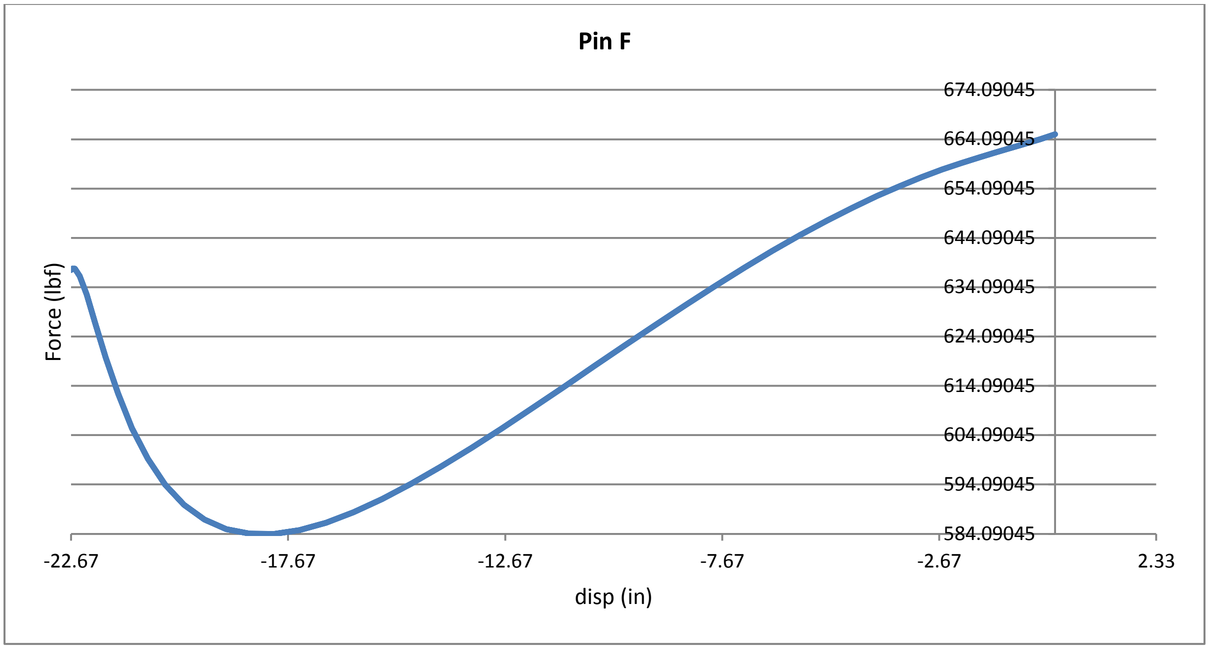
Pin F
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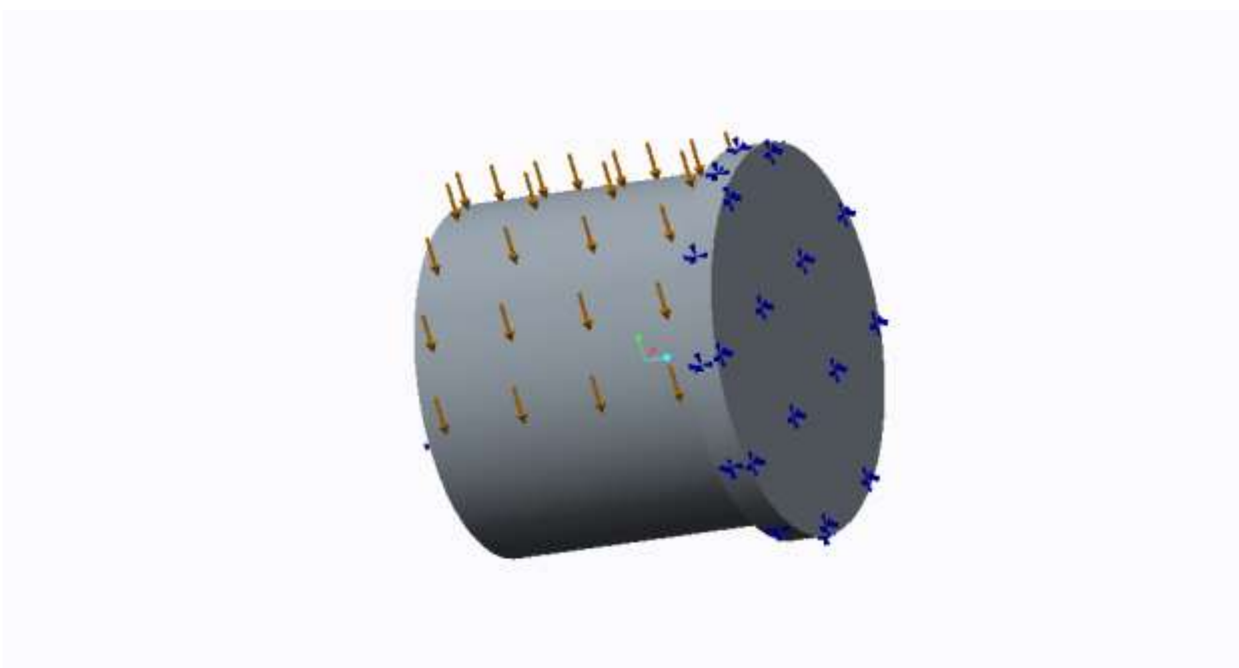
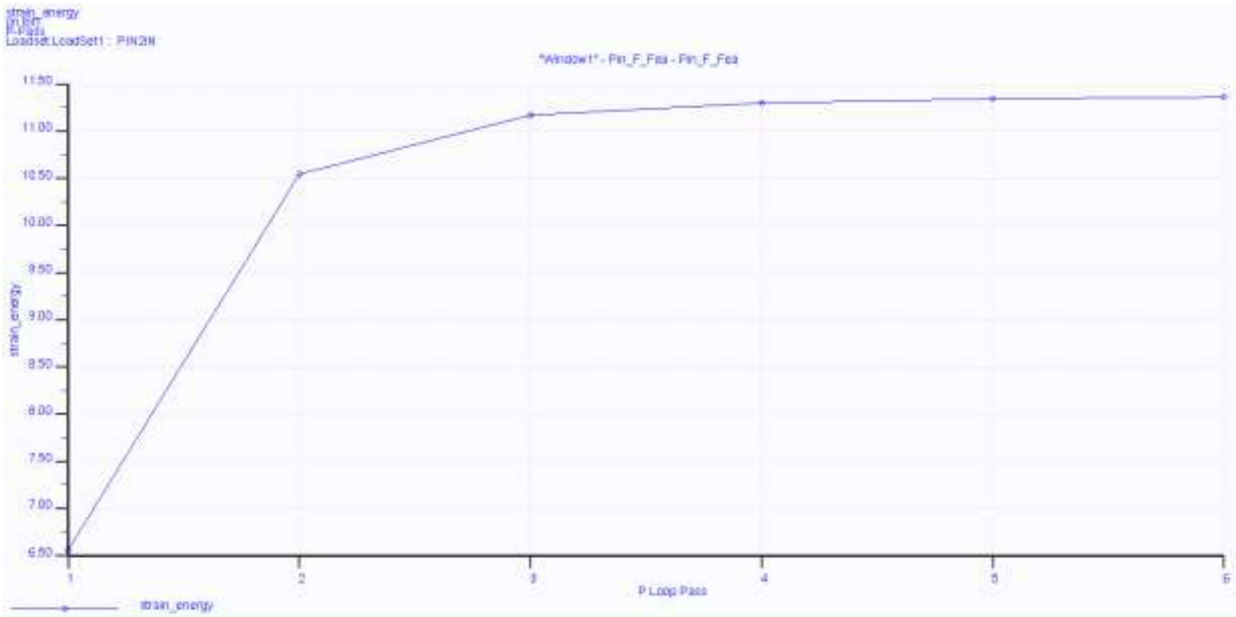
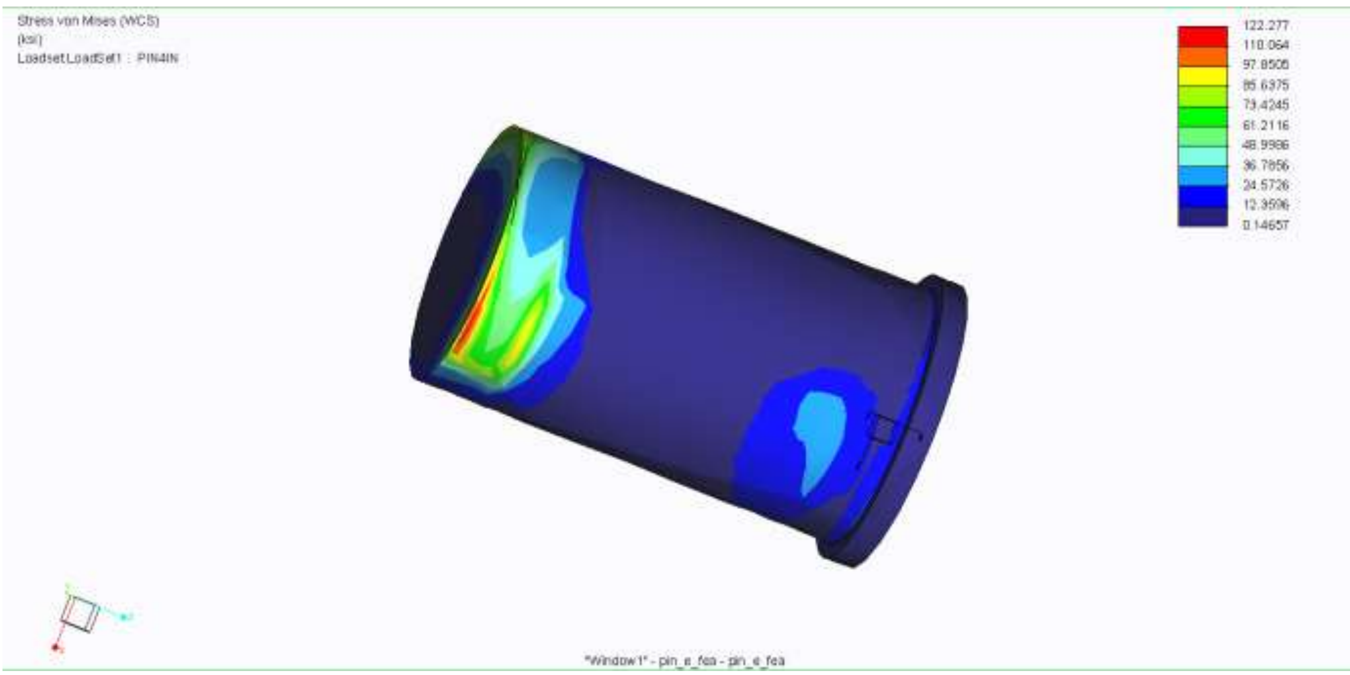
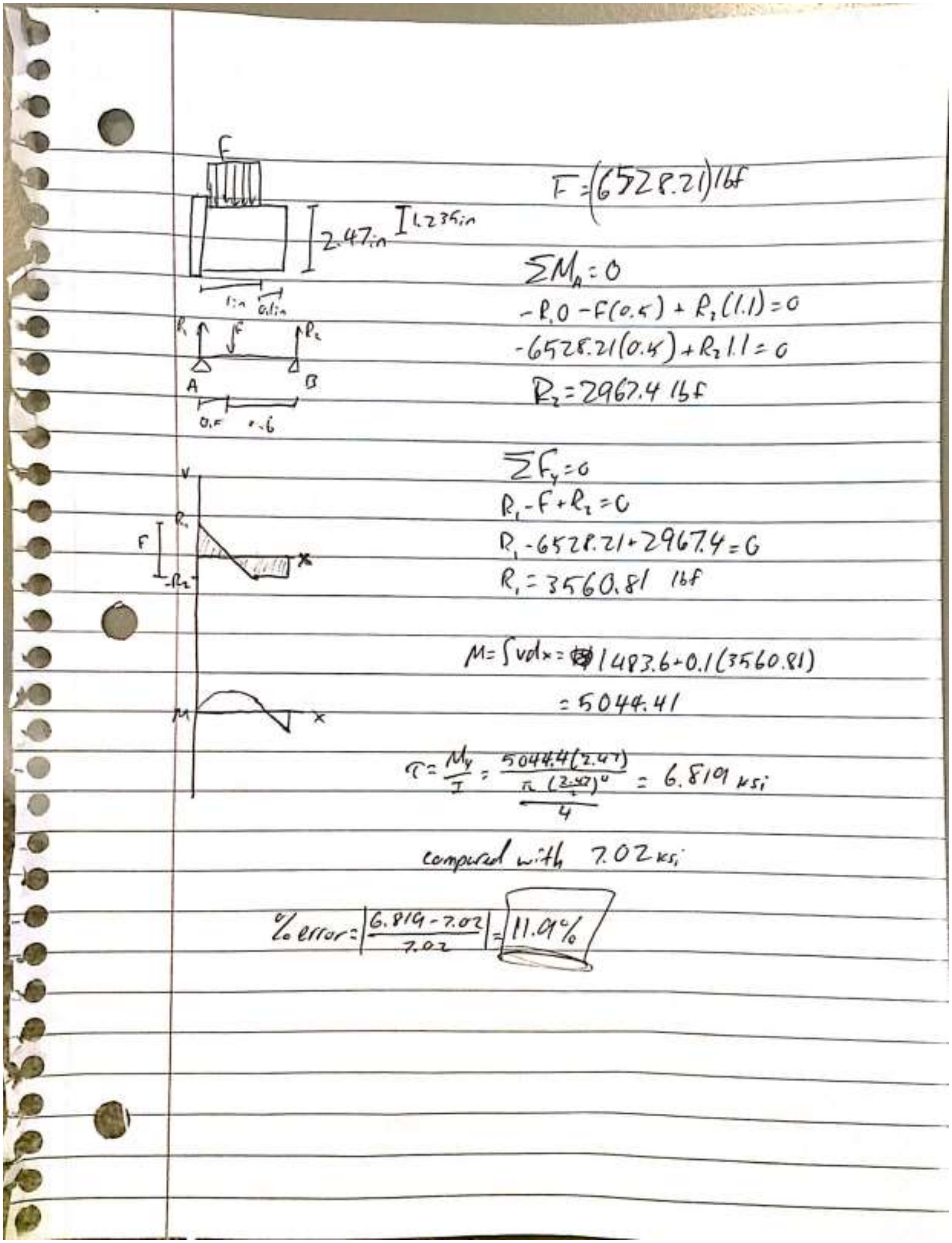
594.09045
584.09045
Force (lbf)
-22.67
-17.67
-12.67
-7.67
-2.67
2.33
disp (in)




strain_energy
Loadset:LoadSet1 : PIN2IN
"Window1" - Pin_F_Fea - Pin_F_Fea
11.50
11.00
10.50
10.00
9.50
9.00
8.50
8.00
7.50
7.00
6.50
strain_energy
1
2
3
4
5
6
P Loop Pass
strain_energy


Stress von Mises (WCS)
(ksi)
Loadset:LoadSet1 : PIN4IN
122.277
110.064
97.8505
85.6375
73.4245
61.2116
48.9986
36.7856
24.5726
12.3596
0.14657
"Window1" - pin_e_fea - pin_e_fea

Pin Stress Analysis


F
2.47in
1.235in
1in
0.1in
R₁
F
R₂
A
B
0.5
0.6
V
R₁
F
-R₂
x
M
x

$F = (6528.21)$ lbf

$\Sigma M_A = 0$

$-R_1 0 - F(0.5) + R_2(1.1) = 0$

$-6528.21(0.5) + R_2 1.1 = 0$

$R_2 = 2967.4$ lbf

$\Sigma F_y = 0$

$R_1 - F + R_2 = 0$

$R_1 - 6528.21 + 2967.4 = 0$

$R_1 = 3560.81$ lbf

$M = \int v\,dx = 1483.6 + 0.1(3560.81)$

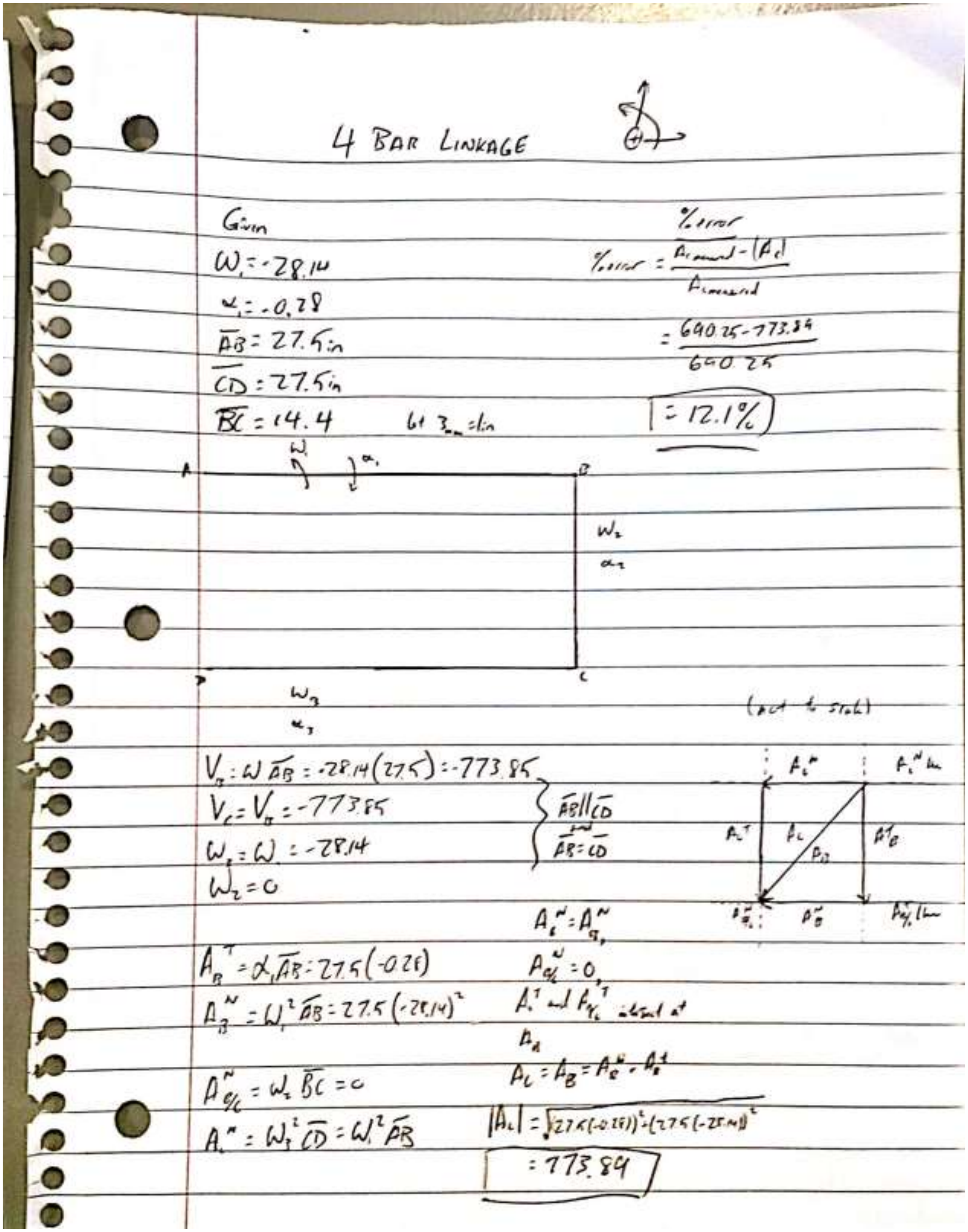
$= 5044.41$

$\tau = \frac{My}{I} = \frac{5044.4(2.47)}{\frac{\pi(\frac{2.47}{2})^4}{4}} = 6.819$ ksi

compared with 7.02 ksi

$\% \text{error} = \left| \frac{6.819 - 7.02}{7.02} \right| = 11.9\%$

Kinematic Analysis



4 BAR LINKAGE

Given

$\omega_1 = -28.14$

$\alpha_1 = -0.28$

$\overline{AB} = 27.5\text{in}$

$\overline{CD} = 27.5\text{in}$

$\overline{BC} = 14.4$

%error

$\%error = \frac{A_{measured} - |A_c|}{A_{measured}}$

$= \frac{690.25 - 773.84}{690.25}$

$= 12.1\%$

$V_B = \omega \overline{AB} = -28.14(27.5) = -773.85$

$V_C = V_B = -773.85$

$\omega_3 = \omega_1 = -28.14$

$\omega_2 = 0$

$\overline{AB} \parallel \overline{CD}$, $\overline{AB} = \overline{CD}$

$A_B^T = \alpha_1 \overline{AB} = 27.5(-0.28)$

$A_B^N = \omega_1^2 \overline{AB} = 27.5(-28.14)^2$

$A_{C/B}^N = \omega_2 \overline{BC} = 0$

$A_C^N = \omega_3^2 \overline{CD} = \omega_1^2 \overline{AB}$

$A_C^N = A_B^N$

$A_{C/B}^N = 0$

A_C^T and $A_{C/B}^T$ intersect at A_d

$A_C = A_B = A_B^N + A_B^T$

$|A_C| = \sqrt{(27.5(-0.28))^2 + (27.5(-28.14))^2}$

$= 773.84$

APPENDIX II – ASSEMBLY/DETAIL DRAWINGS

ASSEMBLIES

- **Hydraulic Assembly**
- **Shock Absorber**
- **Wheel Tire Assembly**

LINKS

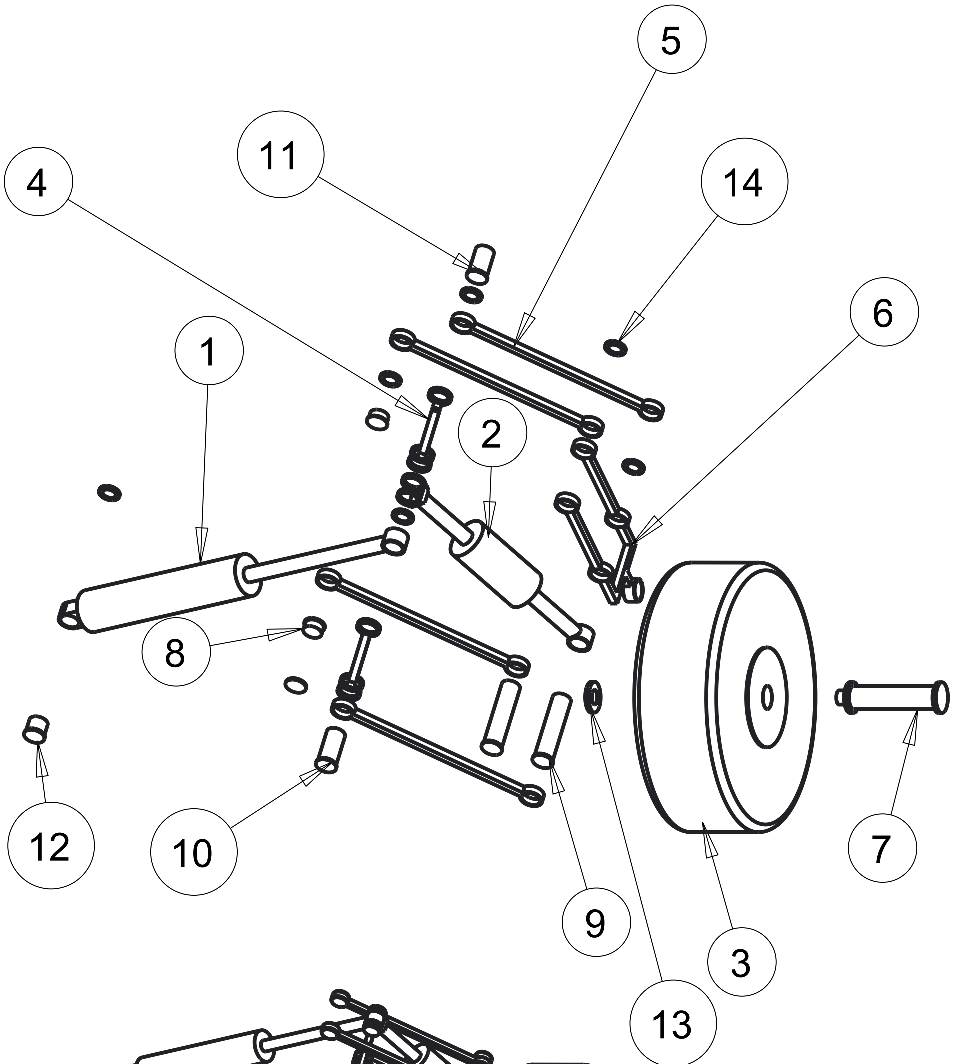
- **Link 1**
- **Link 3**
- **Link 7**

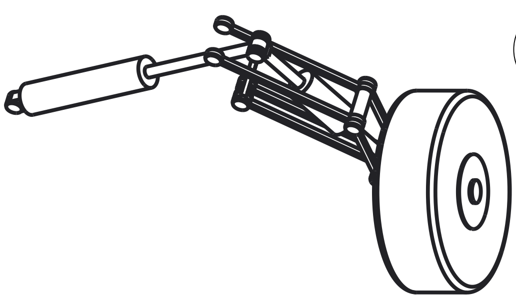
PINS

- **Wheel Axle**
- **Pin A**
- **Pin B**
- **Pin D**
- **Pin E**
- **Pin F**

CAPS

- **Axle Cap**
- **Pin Cap**





SCALE 0.030

ITEM	DESCRIPTION	QTY	MATERIAL
1	HYDRAULIC ASSEMBLY	1	Various
2	SHOCK ABSORBER	1	Various
3	WHEEL TIRE ASSEMBLY	1	Various
4	LINK 1	2	4130 Q&T
5	LINK 3	4	4130 Q&T
6	LINK 7	1	4130 Q&T
7	WHEEL AXEL	1	4340 Oil Q&T
8	PIN A	2	4340 Oil Q&T
9	PIN B	2	4340 Oil Q&T
10	PIN D	1	4340 Oil Q&T
11	PIN E	1	4340 Oil Q&T
12	PIN F	1	4340 Oil Q&T
13	AXEL CAP	1	4340 Oil Q&T
14	PIN CAP	7	4340 Oil Q&T

Unless otherwise shown: All tolerances ±.01 of last decimal given

BINGHAMTON UNIVERSITY
MECHANICAL ENGINEERING DEPARTMENT

LANDING GEAR

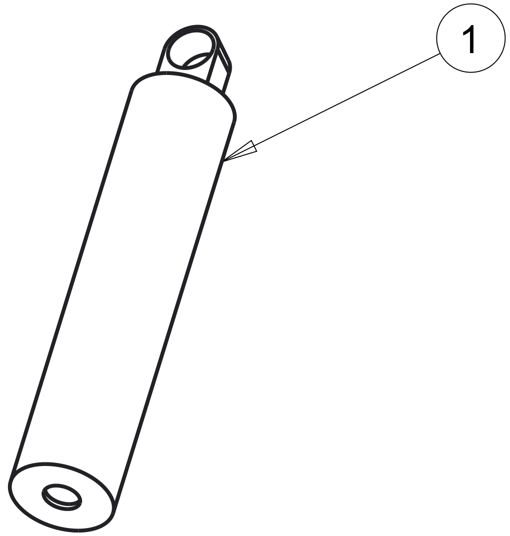
SCALE: 0.040

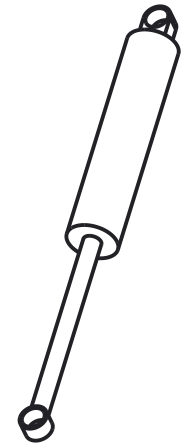
NO. 1

All dimensions in inches

DRWN BY Tyler Wei

DATE: 12/17/2014





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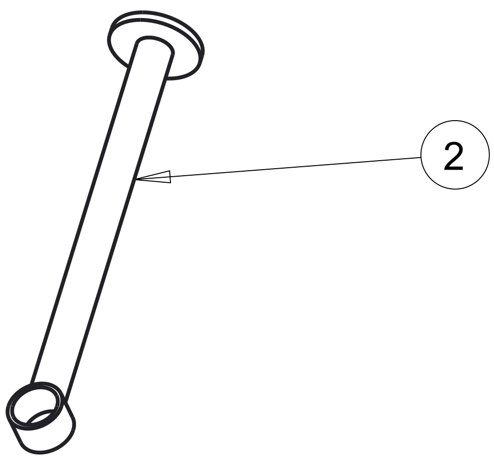

ITEM	DESCRIPTION	QTY	MATERIAL
1	CYLINDER	1	7075-T6
2	PISTON	2	1030 Normalized

Unless otherwise shown: All tolerances ±.01 of last decimal given

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MECHANICAL ENGINEERING DEPARTMENT

HYDRAULIC ASSEMBLY

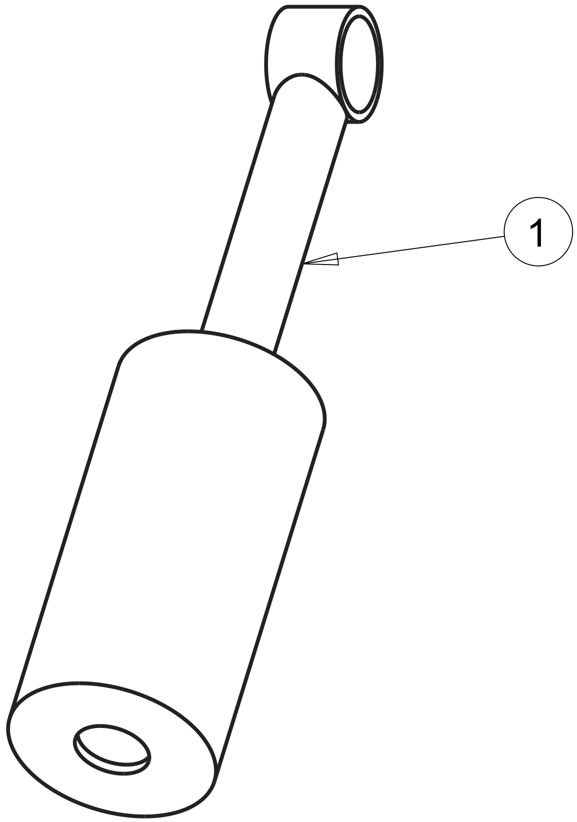
SCALE: 0.100

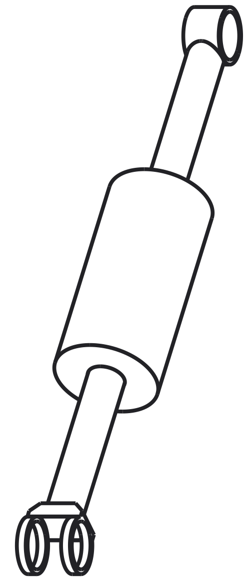
NO. 2

All dimensions in inches

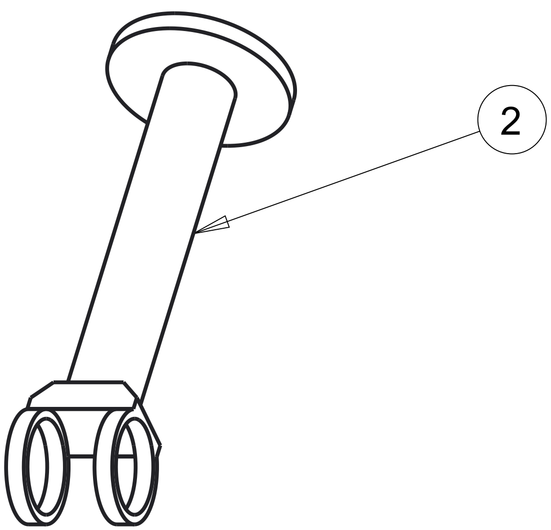
DRWN BY Tyler Wei

DATE: 12/17/2014



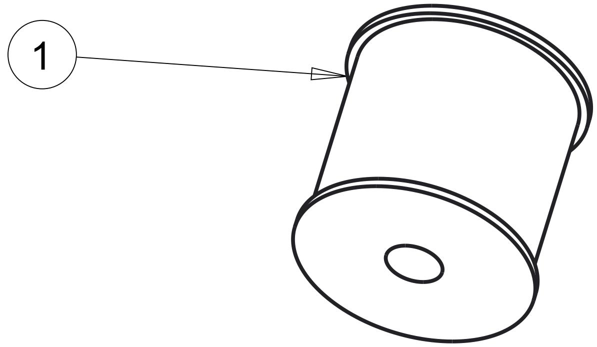


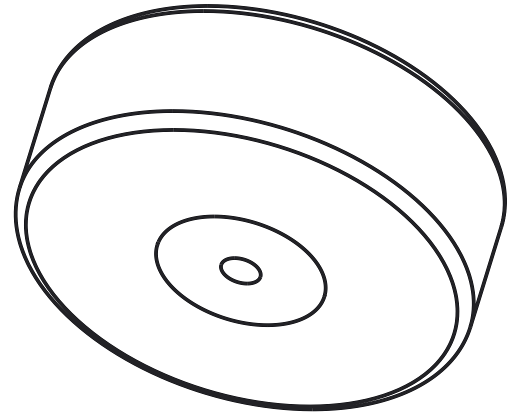
SCALE 0.100



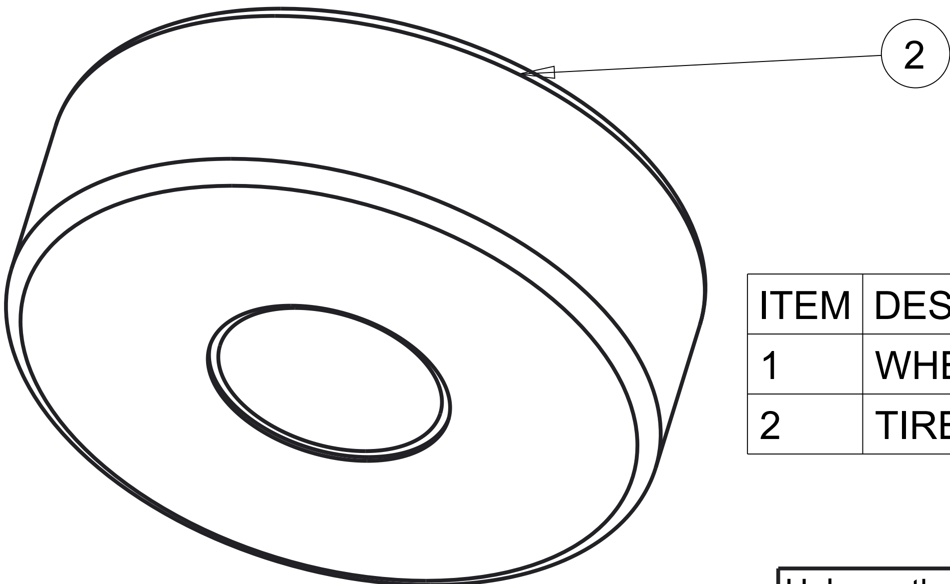
ITEM	DESCRIPTION	QTY	MATERIAL
1	SHOCK ABSORBER CYLINDER	1	7075-T6
2	SHOCK ABSORBER PISTON	1	1030 Normalized

Unless otherwise shown: All tolerances ±.01 of last decimal given	BINGHAMTON UNIVERSITY MECHANICAL ENGINEERING DEPARTMENT		
	SHOCK ABSORBER		
SCALE: 0.200	NO.	3	
All dimensions in inches	DRWN BY	Tyler Wei	DATE: 12/17/2014



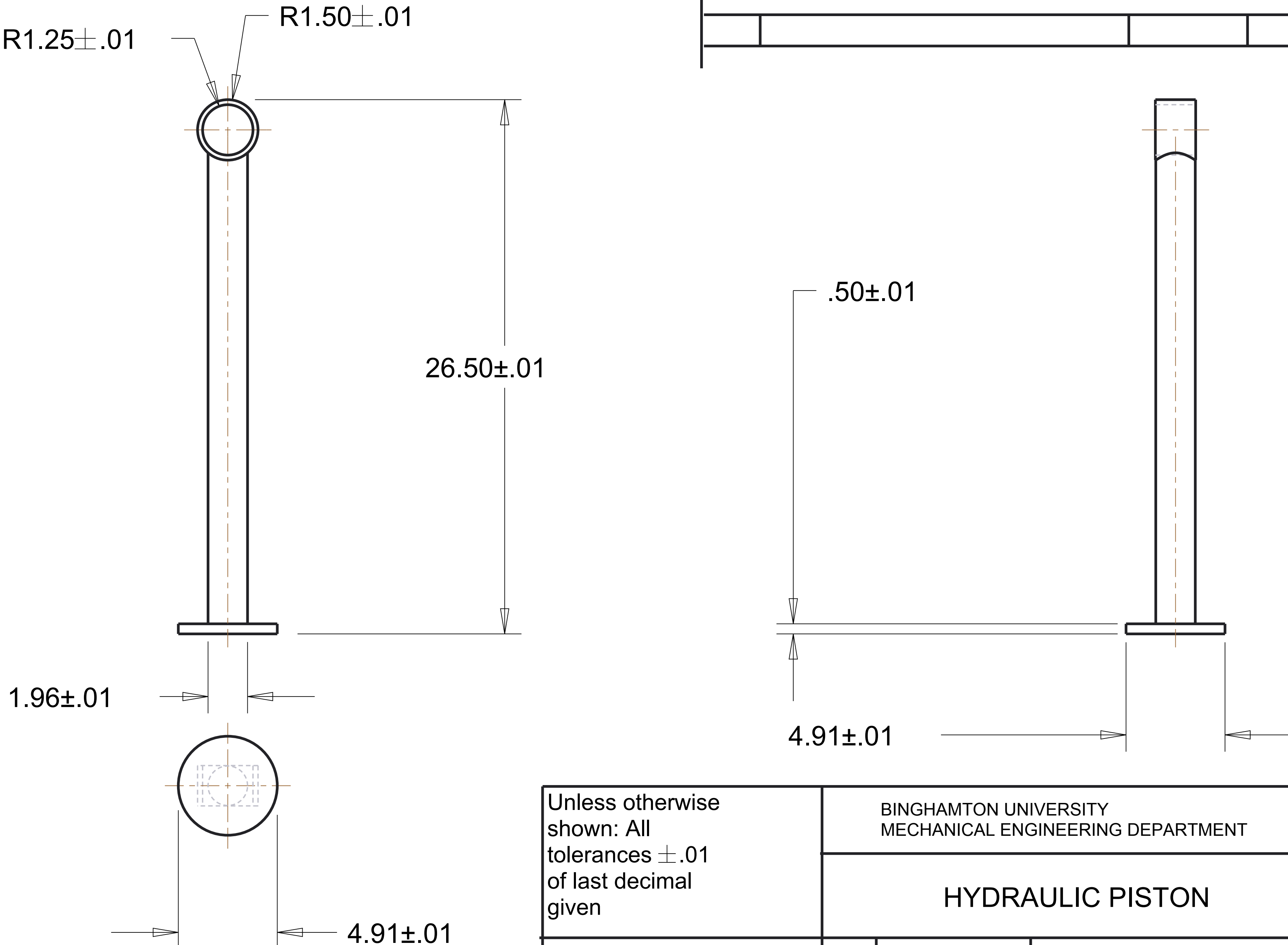


SCALE 0.070



ITEM	DESCRIPTION	QTY	MATERIAL
1	WHEEL	1	7075-T6
2	TIRE	1	Solution Styrene-Butadiene

Unless otherwise shown: All tolerances $\pm$.01 of last decimal given	BINGHAMTON UNIVERSITY MECHANICAL ENGINEERING DEPARTMENT		
	WHEEL TIRE ASSEMBLY		
SCALE: 0.100	NO.	4	
All dimensions in inches	DRWN BY	Tyler Wei	DATE: 12/17/2014



Unless otherwise shown: All tolerances ±.01 of last decimal given

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MECHANICAL ENGINEERING DEPARTMENT

HYDRAULIC PISTON

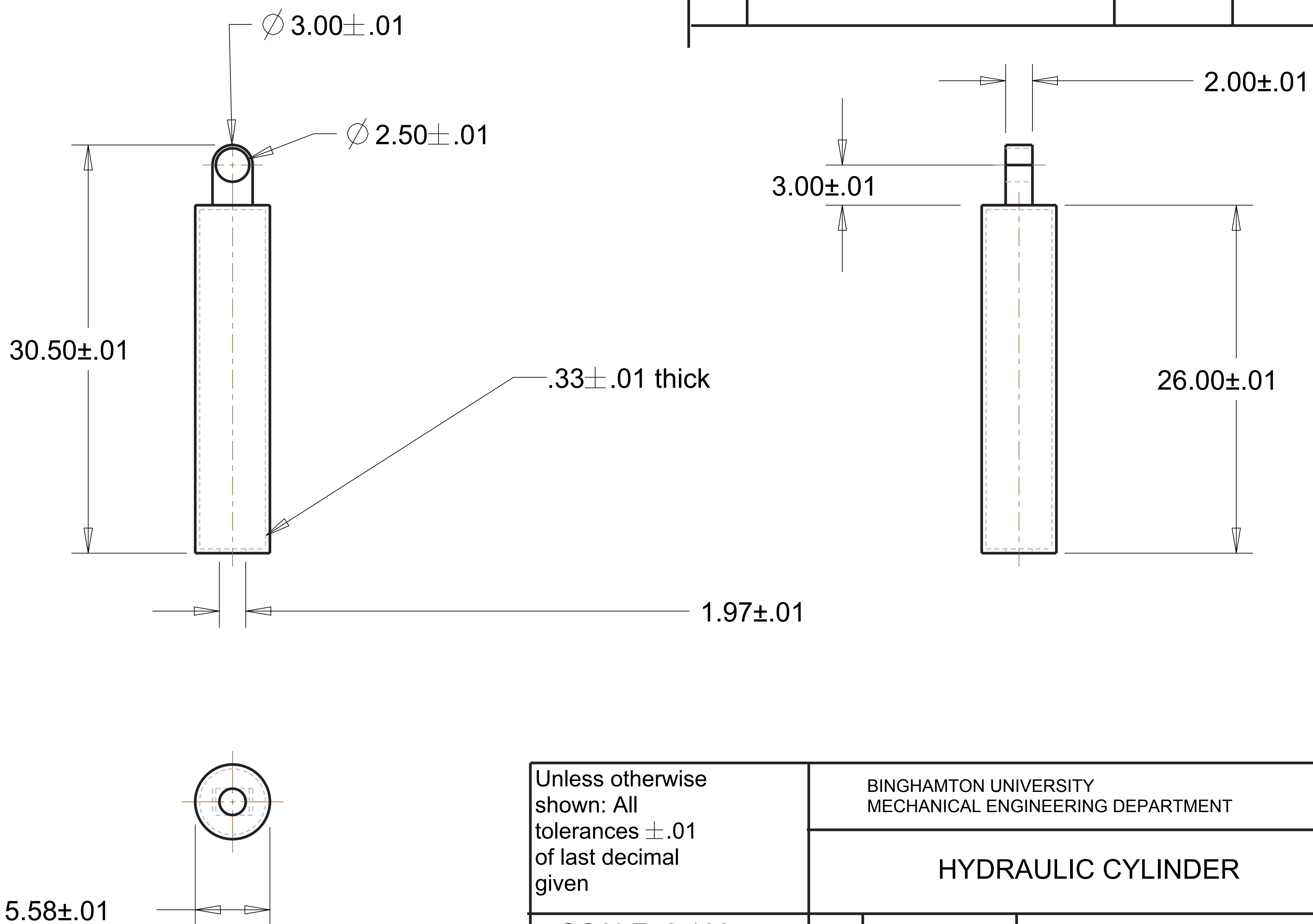
SCALE: 0.150

NO. 5

All dimensions in inches

DRWN BY Tyler Wei

DATE: 12/17/2014



Unless otherwise shown: All tolerances ±.01 of last decimal given

BINGHAMTON UNIVERSITY
MECHANICAL ENGINEERING DEPARTMENT

HYDRAULIC CYLINDER

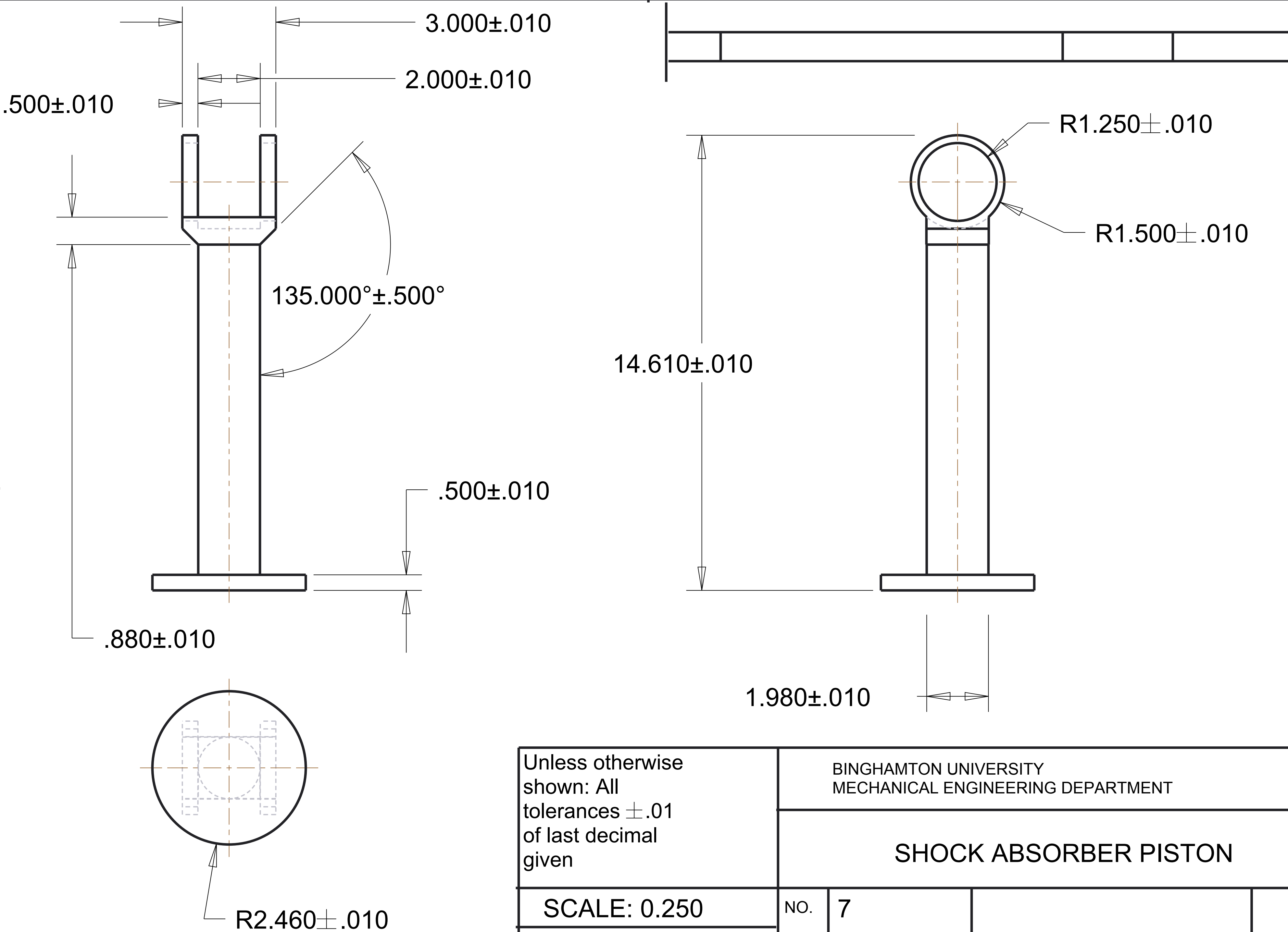
SCALE: 0.100

NO. 2

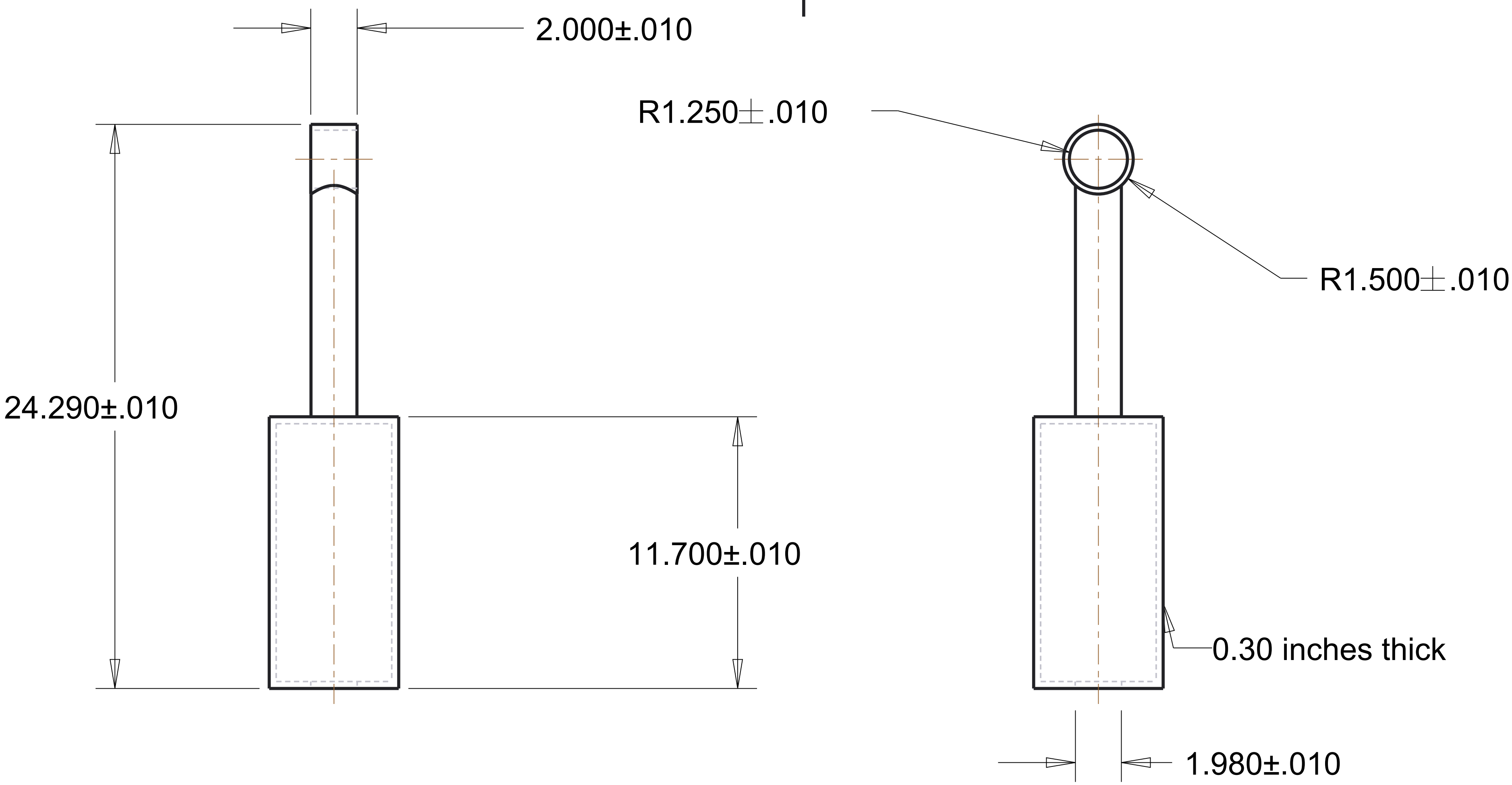
All dimensions in inches

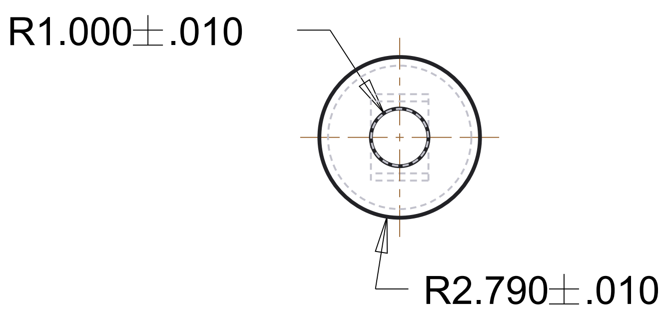
DRWN BY Tyler Wei

DATE: 12/17/2014



Unless otherwise shown: All tolerances ±.01 of last decimal given	BINGHAMTON UNIVERSITY MECHANICAL ENGINEERING DEPARTMENT
	SHOCK ABSORBER PISTON
SCALE: 0.250	NO. 7
All dimensions in inches	DRWN BY Tyler Wei — DATE: 12/17/2014





Unless otherwise shown: All tolerances ±.01 of last decimal given

BINGHAMTON UNIVERSITY
MECHANICAL ENGINEERING DEPARTMENT

SHOCK ABSORBER CYLINDER

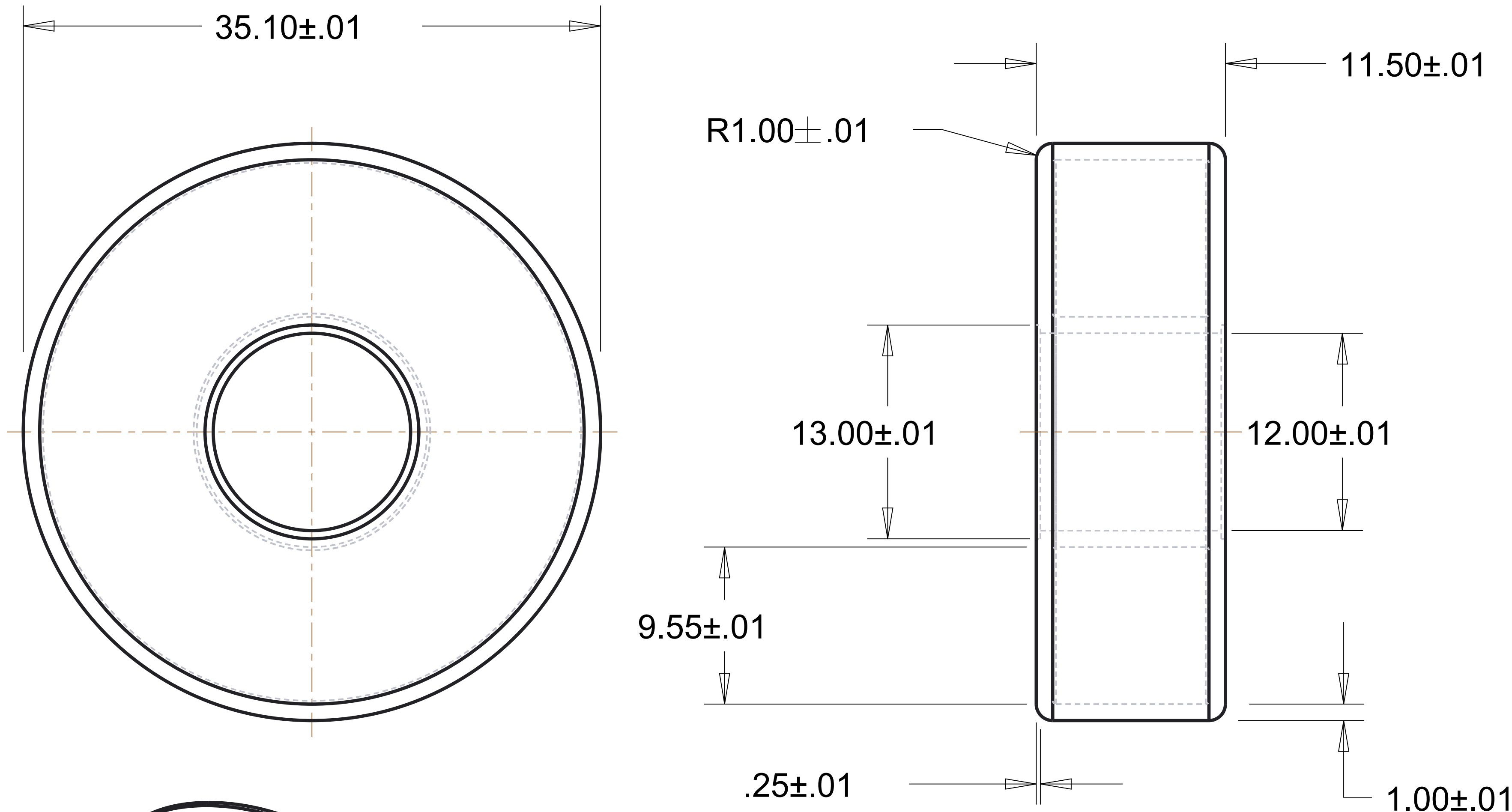
SCALE: 0.150

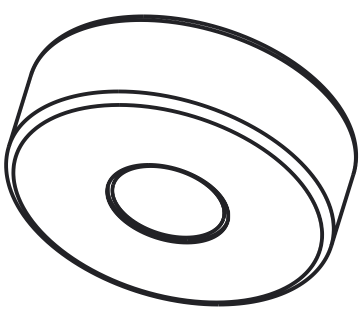
All dimensions in inches

NO. 8

DRWN BY Tyler Wei

DATE: 12/17/2014





SCALE 0.050

Unless otherwise shown: All tolerances ±.01 of last decimal given

SCALE: 0.100

All dimensions in inches

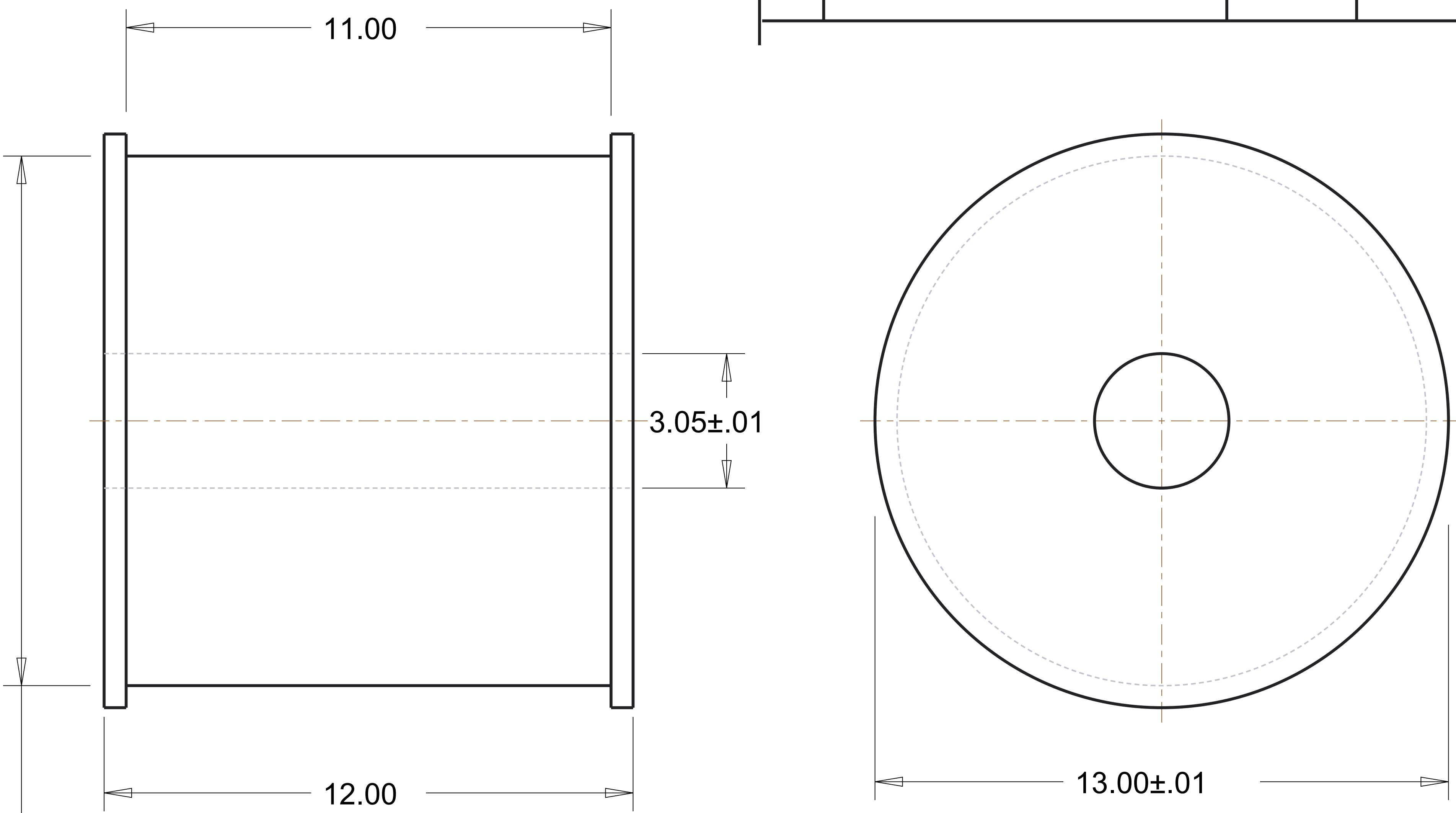
BINGHAMTON UNIVERSITY
MECHANICAL ENGINEERING DEPARTMENT

TIRE

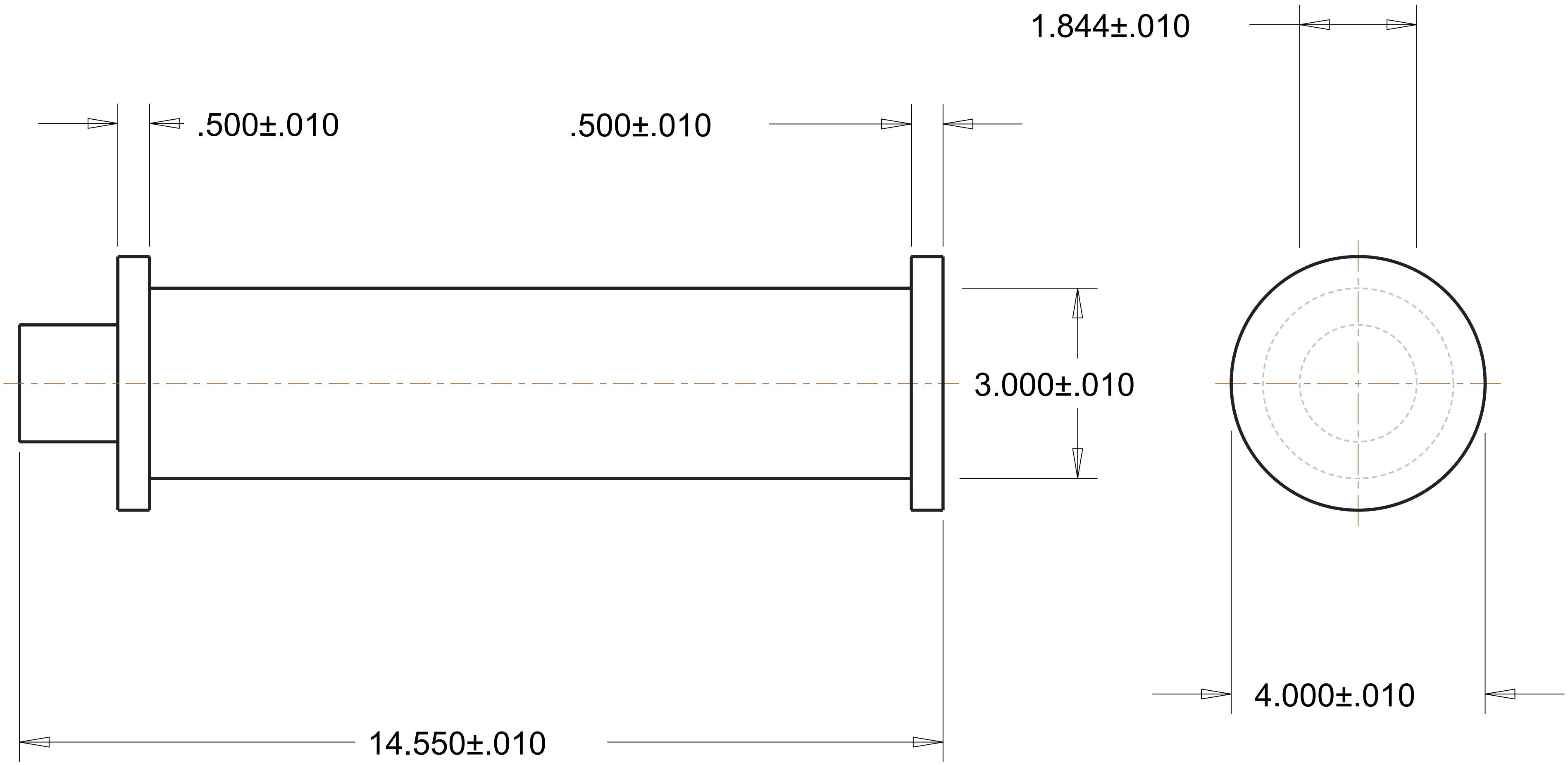
NO. 9

DRWN BY Tyler Wei

DATE: 12/17/2014



Unless otherwise shown: All tolerances ±.01 of last decimal given	BINGHAMTON UNIVERSITY MECHANICAL ENGINEERING DEPARTMENT WHEEL
SCALE 0.300	NO. 10
All dimensions in inches	DRWN BY Tyler Wei — DATE: 12/17/2014



Unless otherwise shown: All tolerances ±.01 of last decimal given

BINGHAMTON UNIVERSITY
MECHANICAL ENGINEERING DEPARTMENT

WHEEL AXLE

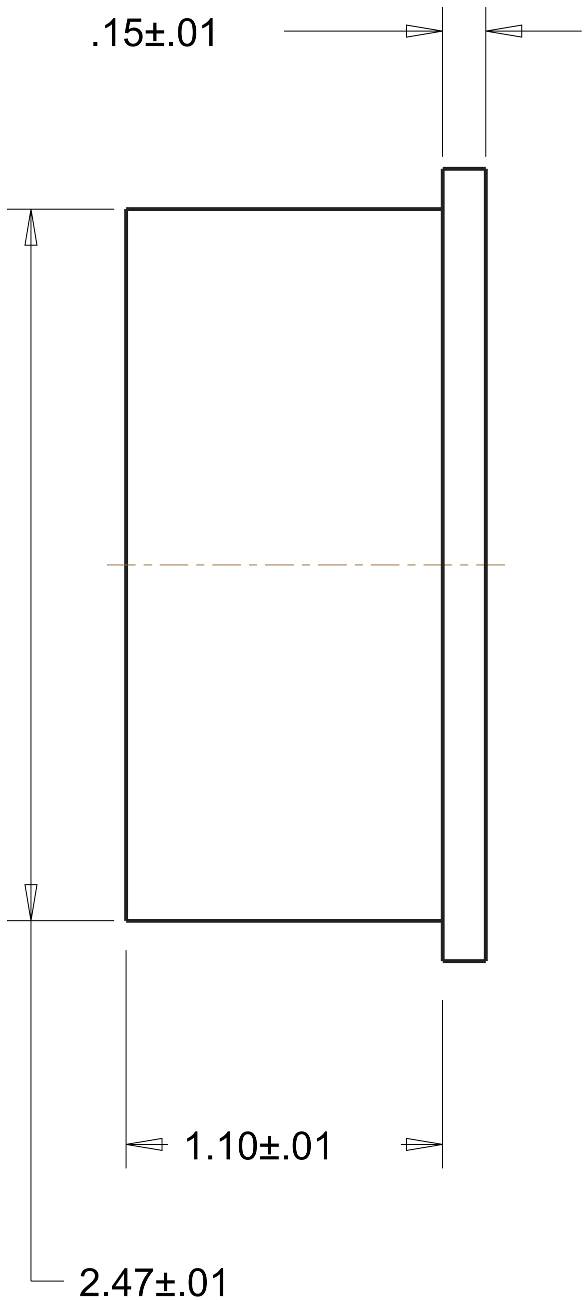
SCALE: 0.400

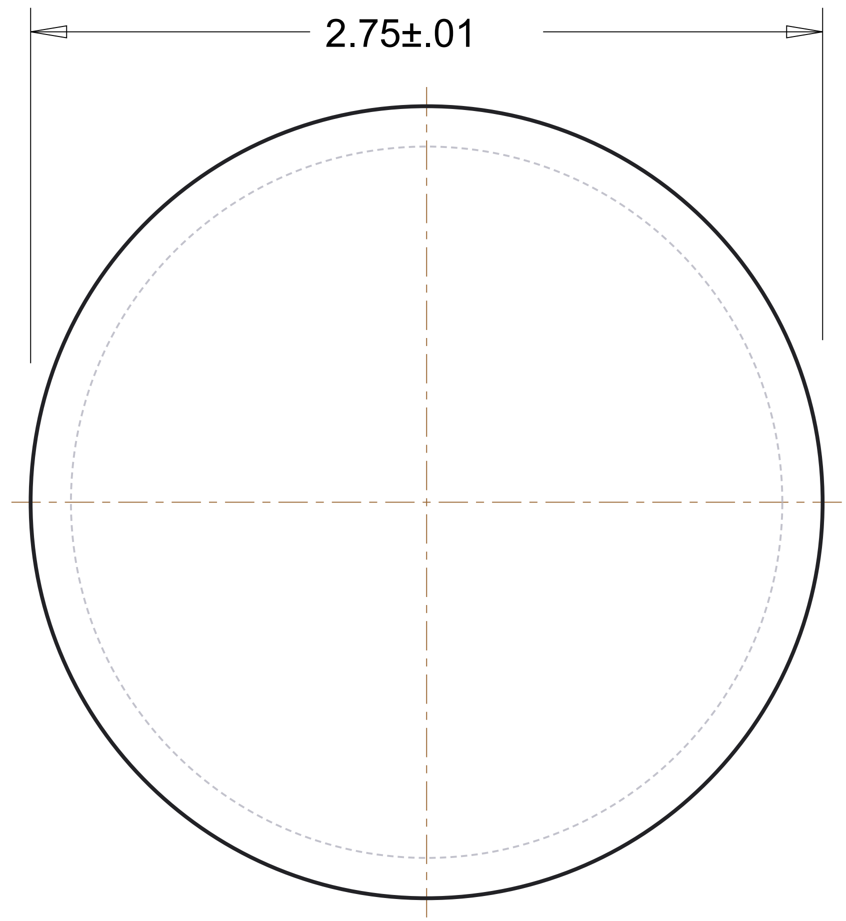
All dimensions in inches

NO. 11

DRWN BY Tyler Wei

DATE: 12/17/2014





Unless otherwise shown: All tolerances ±.01 of last decimal given

BINGHAMTON UNIVERSITY
MECHANICAL ENGINEERING DEPARTMENT

PIN A

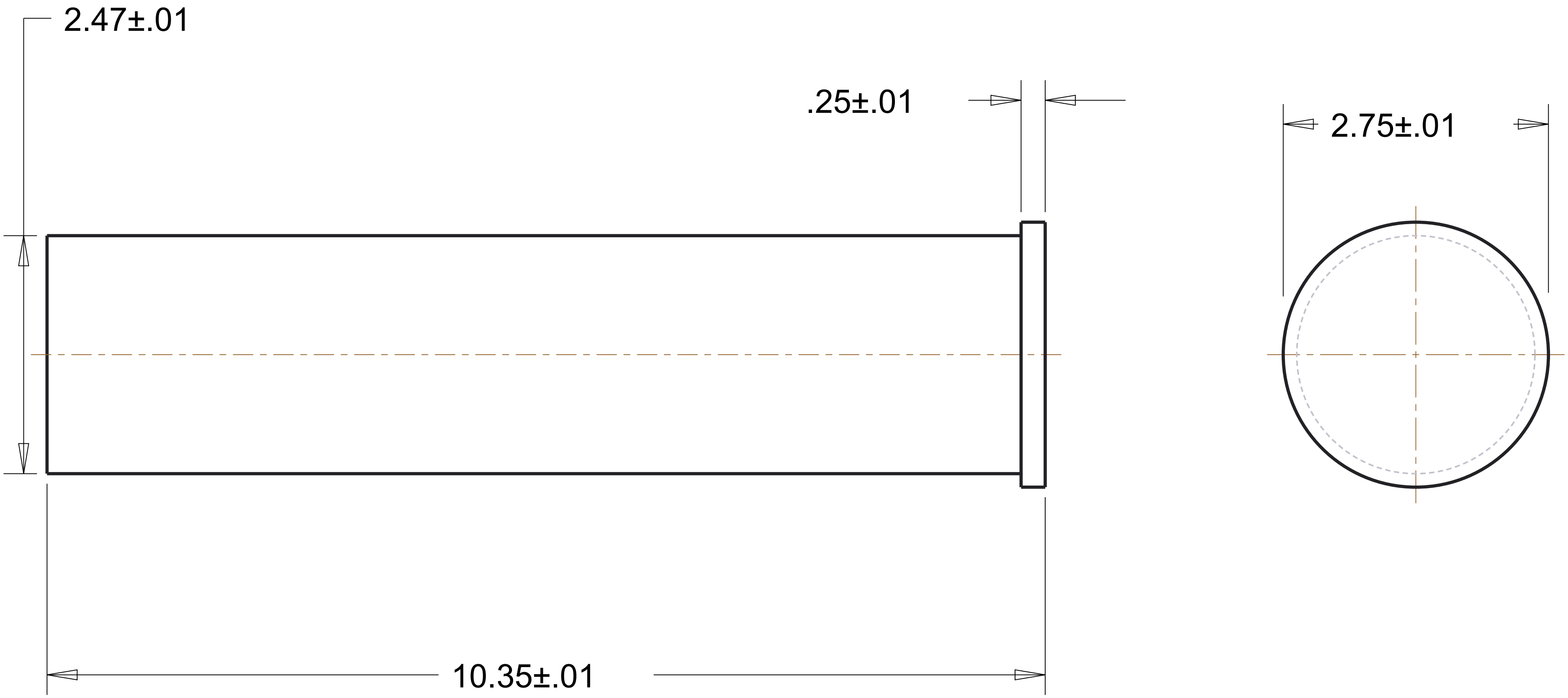
SCALE: 1.500

NO. 12

All dimensions in inches

DRWN BY Tyler Wei

DATE: 12/17/2014



Unless otherwise shown: All tolerances ±.01 of last decimal given

SCALE: 0.500

All dimensions in inches

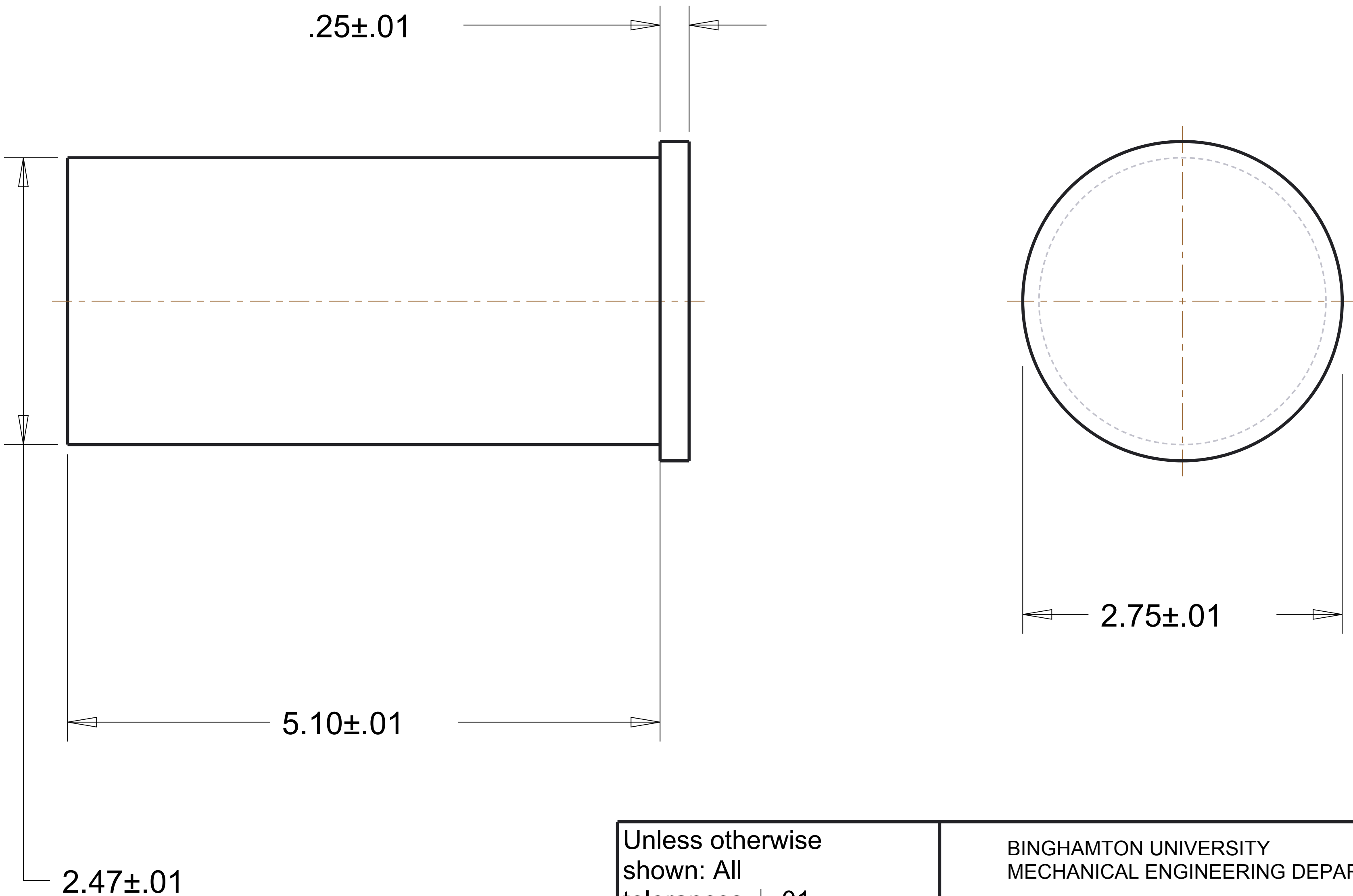
BINGHAMTON UNIVERSITY
MECHANICAL ENGINEERING DEPARTMENT

PIN B

NO. 13

DRWN BY Tyler Wei

DATE: 12/17/2014



Unless otherwise shown: All tolerances ±.01 of last decimal given

SCALE: 0.750

All dimensions in inches

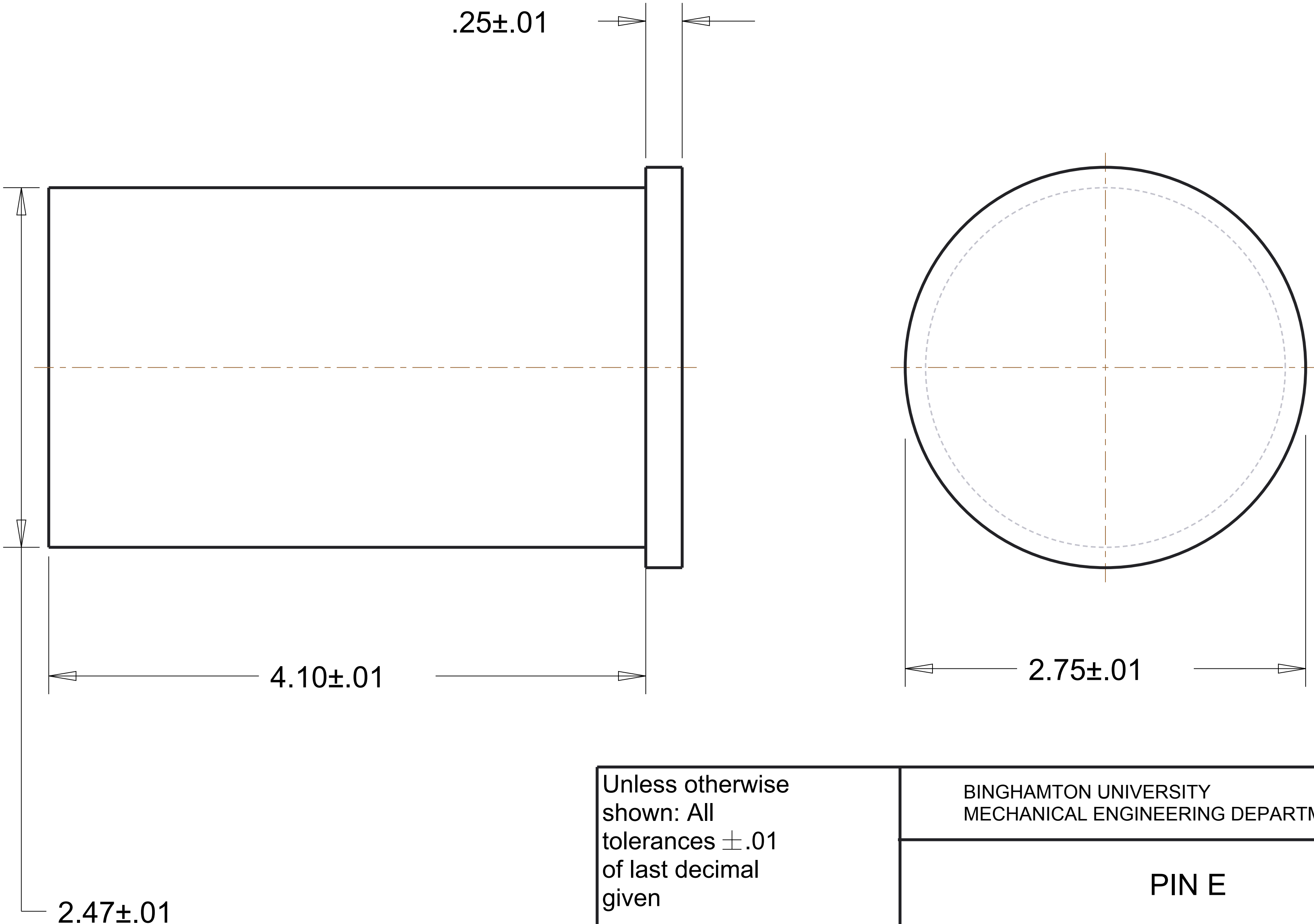
BINGHAMTON UNIVERSITY
MECHANICAL ENGINEERING DEPARTMENT

PIN D

NO. 14

DRWN BY Tyler Wei

DATE: 12/17/2014



Unless otherwise shown: All tolerances ±.01 of last decimal given

SCALE: 1.000

All dimensions in inches

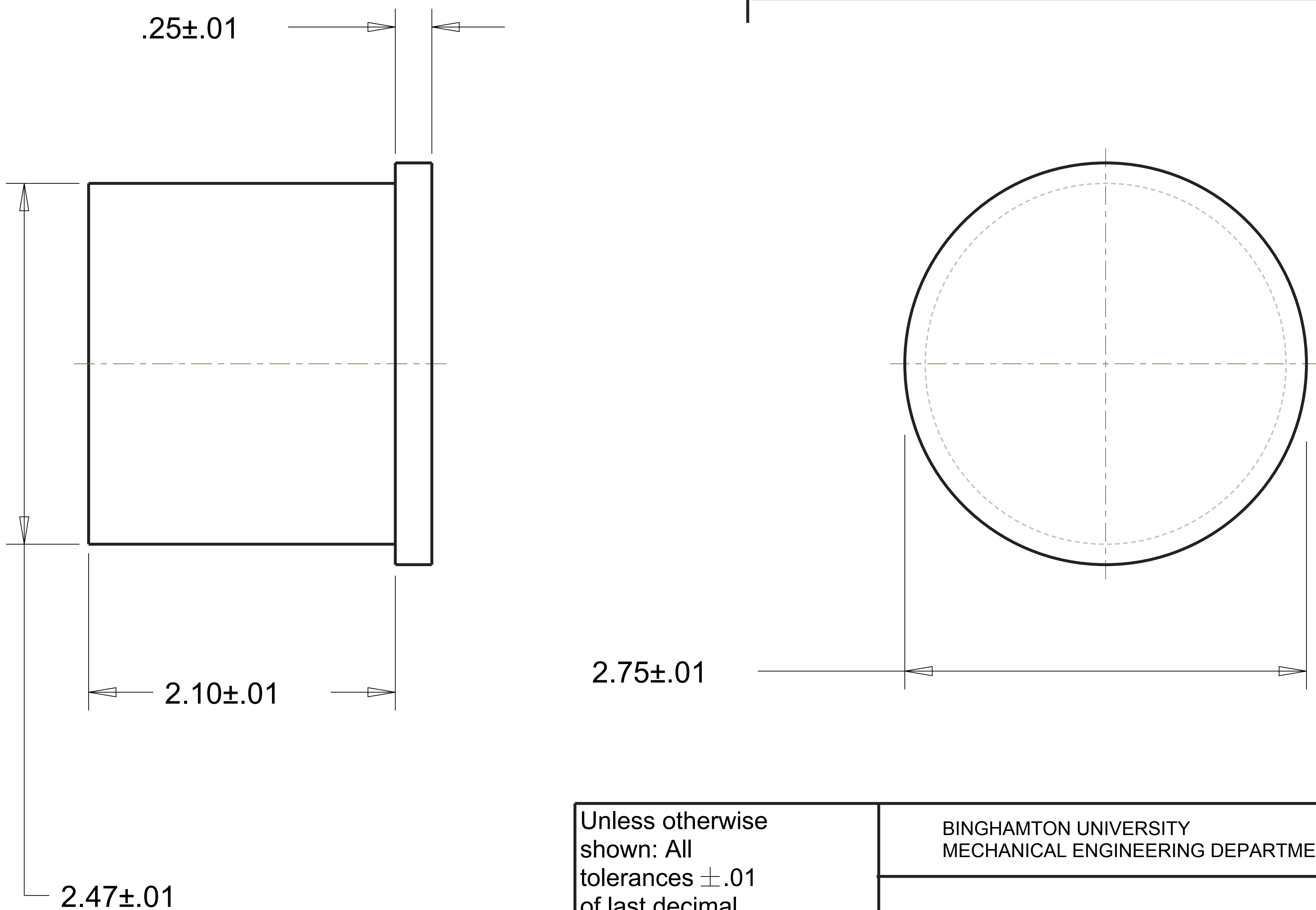
BINGHAMTON UNIVERSITY
MECHANICAL ENGINEERING DEPARTMENT

PIN E

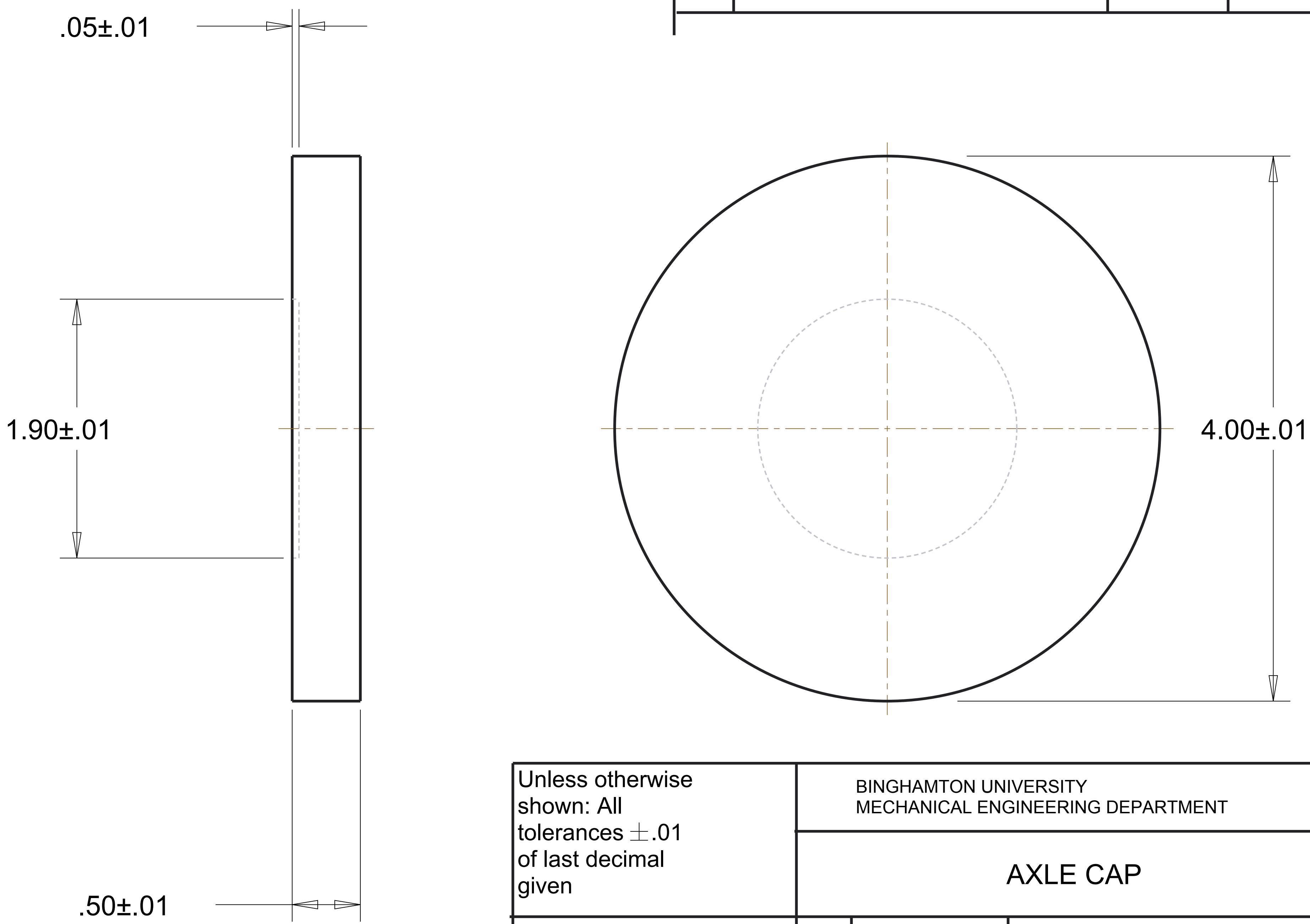
NO. 15

DRWN BY Tyler Wei

DATE: 12/17/2014



Unless otherwise shown: All tolerances ±.01 of last decimal given	BINGHAMTON UNIVERSITY MECHANICAL ENGINEERING DEPARTMENT		
	PIN F		
SCALE: 1.000	NO.	16	
All dimensions in inches	DRWN BY	Tyler Wei	DATE: 12/17/2014



Unless otherwise shown: All tolerances ±.01 of last decimal given

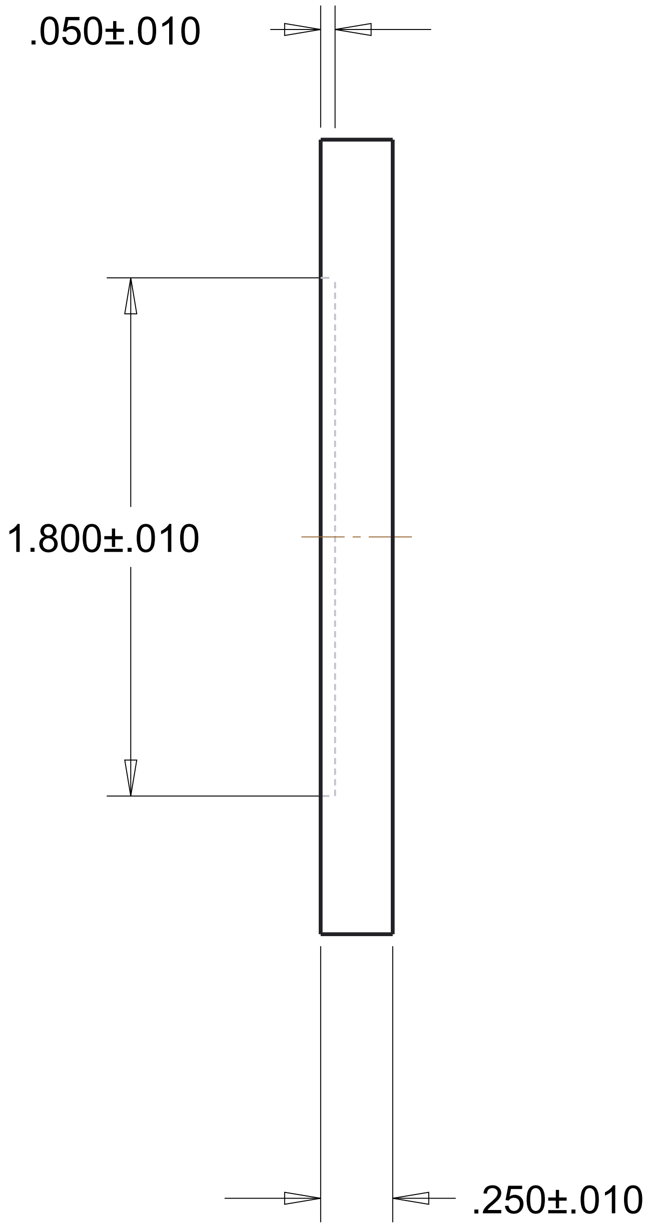
SCALE: 1.000

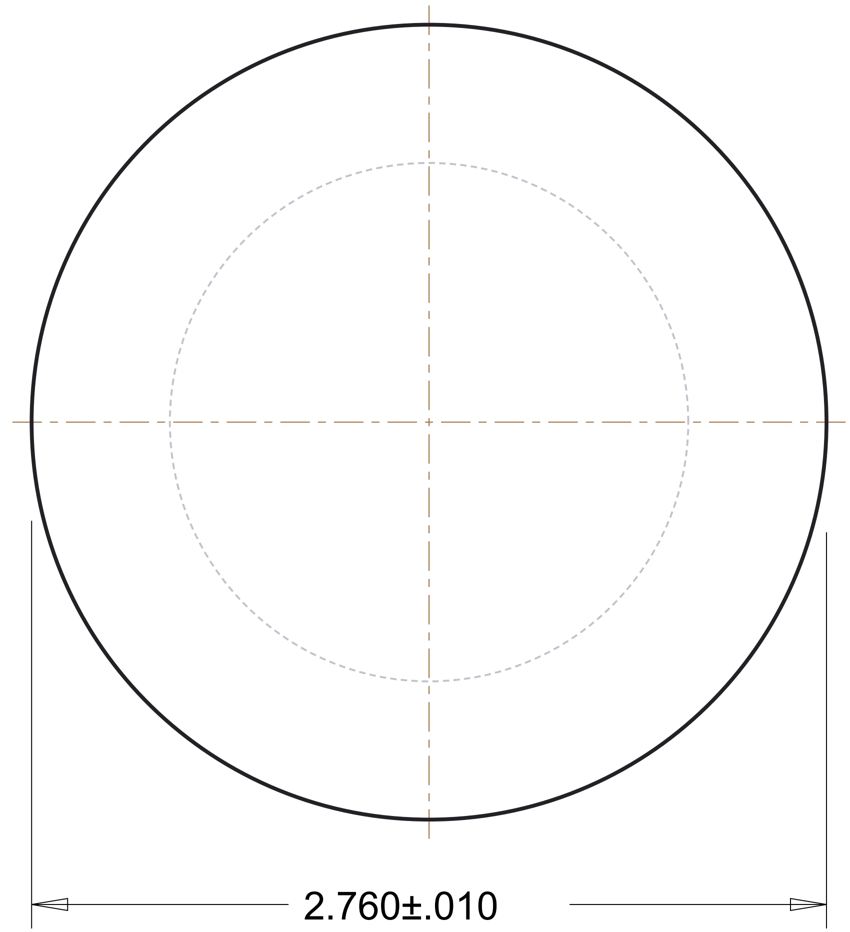
All dimensions in inches

BINGHAMTON UNIVERSITY
MECHANICAL ENGINEERING DEPARTMENT

AXLE CAP

NO.	17		
DRWN BY	Tyler Wei	DATE: 12/17/2014	





Unless otherwise shown: All tolerances ±.01 of last decimal given

BINGHAMTON UNIVERSITY
MECHANICAL ENGINEERING DEPARTMENT

PIN CAP

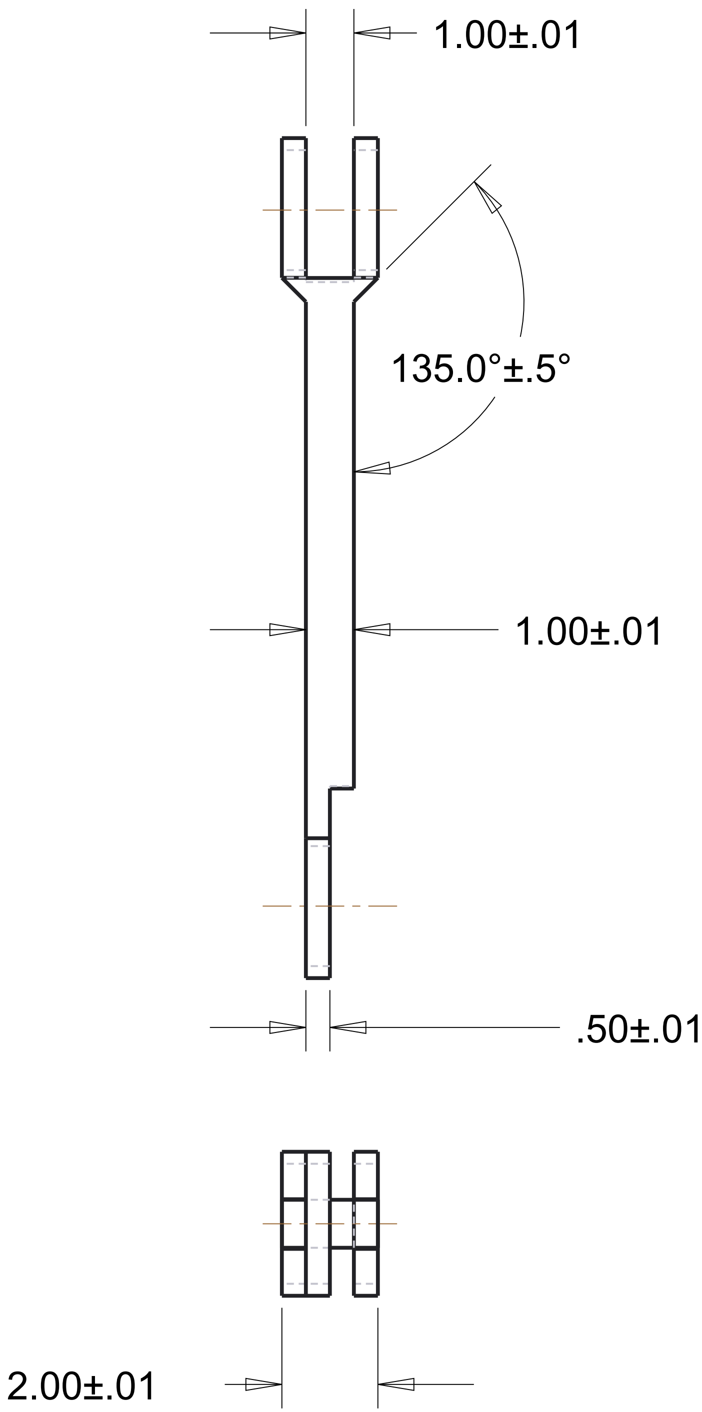
SCALE: 1.500

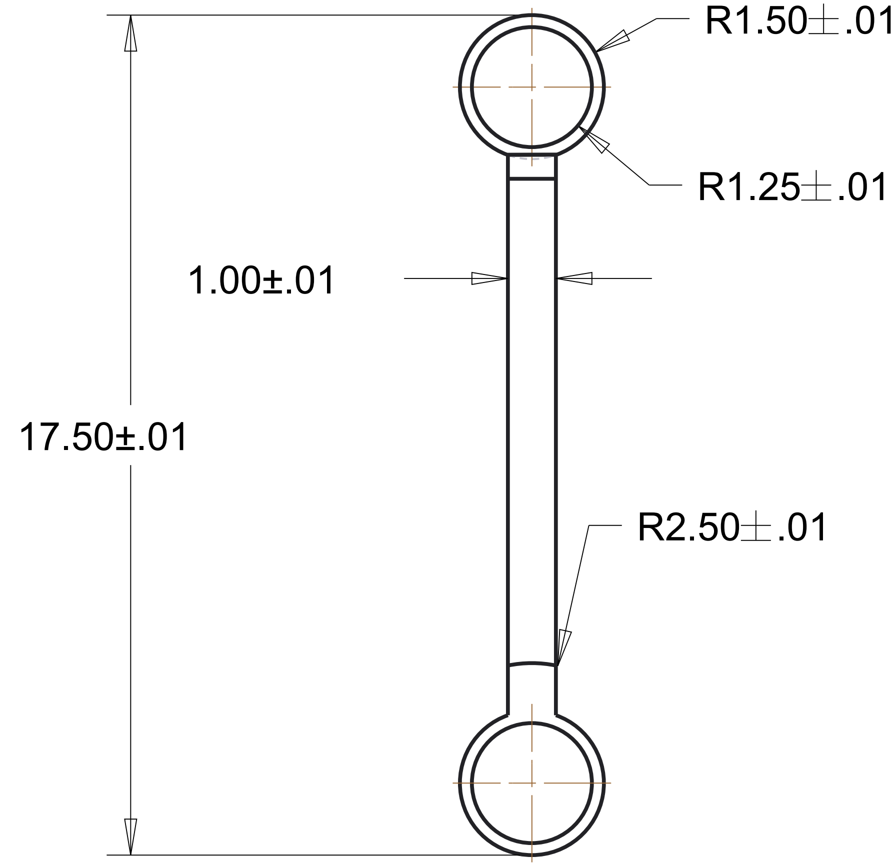
NO. 18

All dimensions in inches

DRWN BY Tyler Wei

DATE: 12/17/2014





Unless otherwise shown: All tolerances ±.01 of last decimal given

BINGHAMTON UNIVERSITY
MECHANICAL ENGINEERING DEPARTMENT

LINK 1

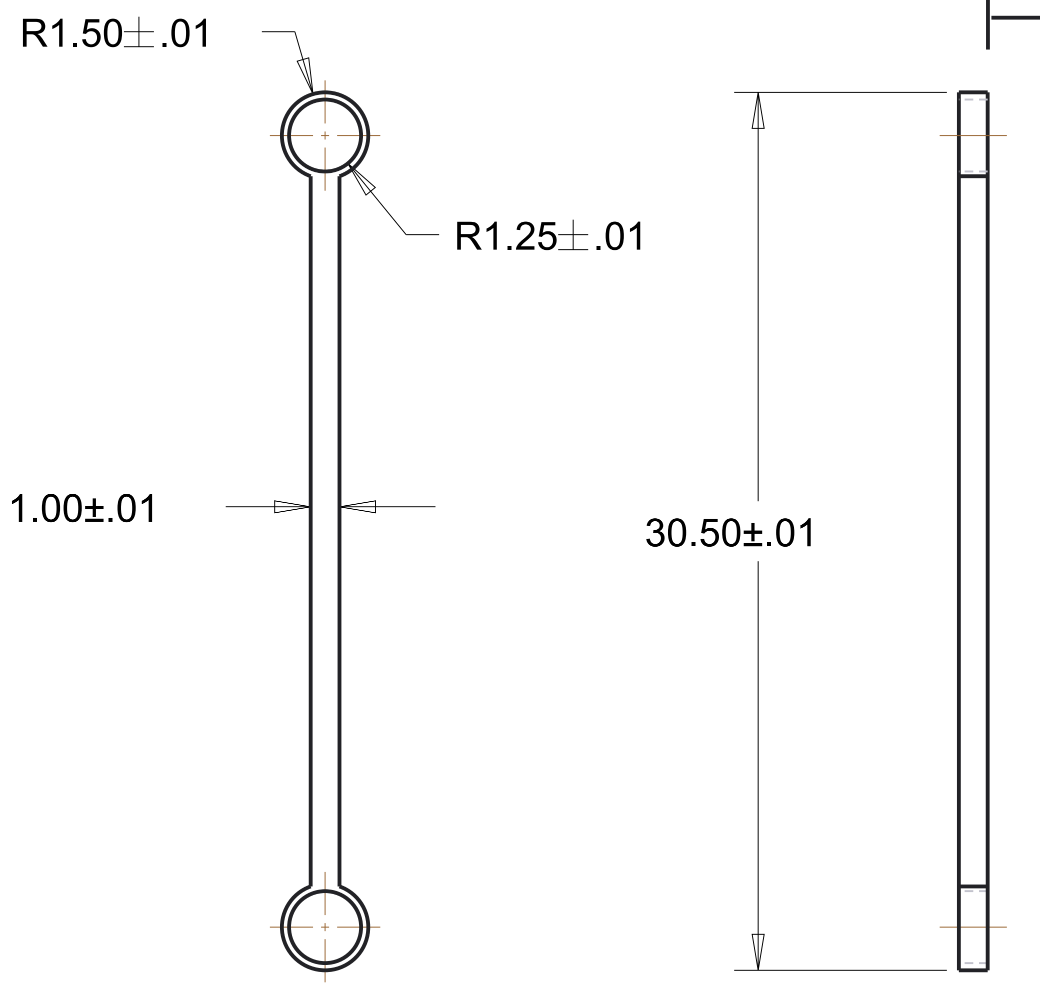
SCALE: 0.250

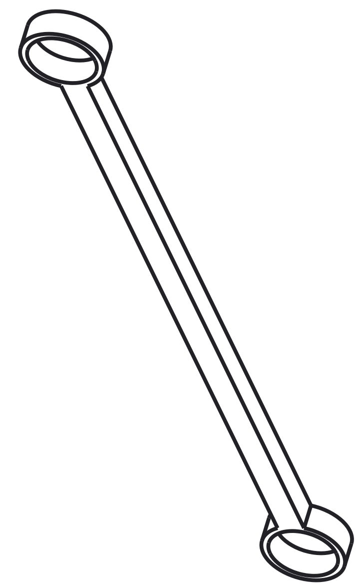
All dimensions in inches

NO. 19

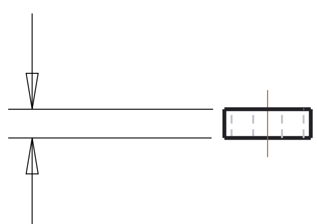
DRWN BY Tyler Wei

DATE: 12/17/2014





SCALE 0.150



Unless otherwise shown: All tolerances ±.01 of last decimal given

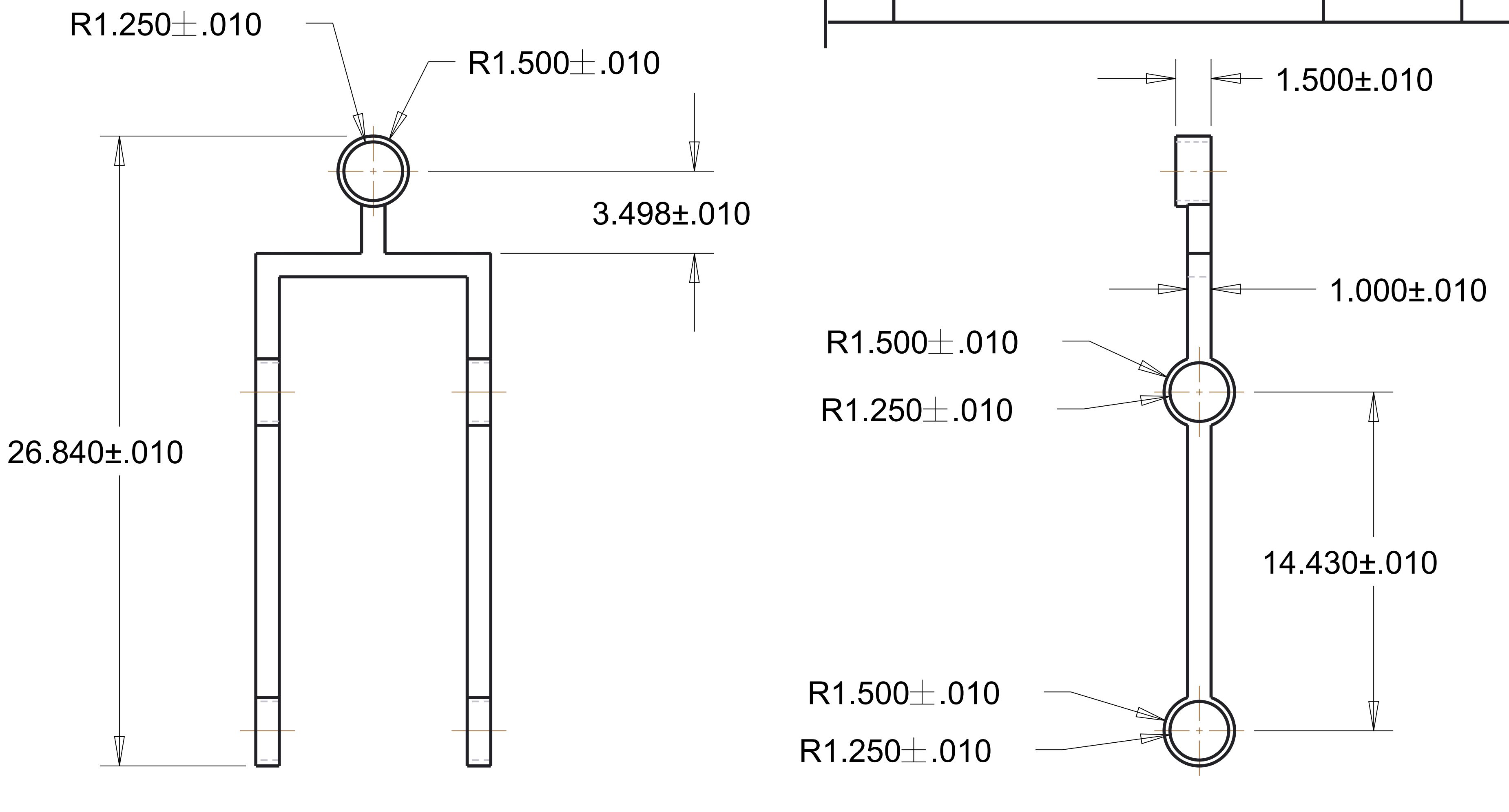
SCALE: 0.150

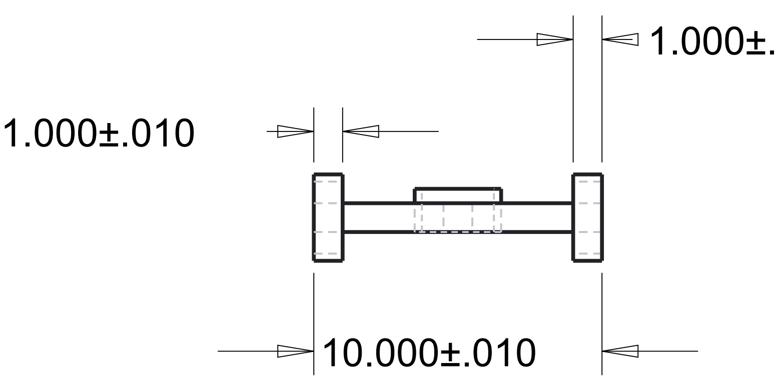
All dimensions in inches

BINGHAMTON UNIVERSITY
MECHANICAL ENGINEERING DEPARTMENT

LINK 3

NO.	20		
DRWN BY	Tyler Wei	DATE: 12/17/2014	





Unless otherwise shown: All tolerances ±.01 of last decimal given	BINGHAMTON UNIVERSITY MECHANICAL ENGINEERING DEPARTMENT		
	LINK 7		
SCALE: 0.150	NO.	21	
All dimensions in inches	DRWN BY	Tyler Wei	DATE: 12/17/2014