Department of Mechanical Engineering

EMEC-4969-YC
Degree Project

Capstone Project Report

Design Optimization of a Repetitive Impact Testing Apparatus to Improve the Performance, Accuracy and Adaptability of Injury Biomechanics Simulations

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Abstract

The repetitive impact testing apparatus is actively utilized by the School of Kinesiology in injury biomechanics research. In order to improve the variety and accuracy of performed experiments, several components required a functional redesign. Difficulty fastening rubber cane tips onto an actuating cylinder prompted the development of a stepped shaft properly sized for each type of cane tip utilized in experiments. Improved rigidity of the force plate mechanism was implemented through the development of new support brackets, which allowed operation at standardized orientations. The need for improved maneuverability of the protective equipment during experimental procedures was resolved through the development of an adjustable impact rod. By mimicking the attachment mechanisms of the existing actuating cylinder, several additional means of adjustment were introduced to existing processes. The oscillation of the actuating cylinder during test operations was reduced through the development of an alternative mechanical connection method. Finally, additional machine improvements and repairs were proposed.

Acknowledgement

The entire group would like to express their gratitude to Dr. Meilan Liu for her supervision and support. The guidance we received has truly been invaluable and has improved the overall quality of the designs presented in this project report. It was through her teachings that a much clearer understanding of the intricacies of engineering design could be understood. In particular, the practical aspects and tools utilized in progressive design iterations.

We would also like to acknowledge the support of Morgan Ellis for his valuable input at various stages of design, and for assisting with the purchase of raw materials. Morgan was able to provide targeted feedback at critical stages of design, greatly enhancing the practicality of all final designs.

Finally, we would like to thank Dr. Carlos Zerpa and the School of Kinesiology for providing another opportunity for a capstone degree project. This project allowed us to apply the engineering theory we've developed at Lakehead University, and we're thrilled to utilize that knowledge to foster growth and collaboration within Lakehead University.

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Nomenclature

Symbol	Origin	Description	Units
A_c	Latin	Cross-sectional area	in^2, mm^2
d_i	Latin	Diameter of inner member	in, mm
d_o	Latin	Diameter of outer member	in, mm
D	Latin	Diameter	in, mm
E_i	Latin	Young's modulus of inner material	psi, Pa
E_o	Latin	Young's modulus of outer material	psi, Pa
F_{crit}	Latin	Critical load	lbf, N
l	Latin	Length	in, mm
M	Latin	Moment	$ft\text{-}lb, N\text{-}m$
n	Latin	Factor of safety	—
n	Latin	End-fixity conditions (J.B. Johnson buckling formula only)	—
p	Latin	Interference pressure	psi, Pa
S_e	Latin	Endurance limit	psi, Pa
t	Latin	Wall thickness	in, mm
δ	Greek	Diametral interference	in, mm
μ	Greek	Coefficient of friction	—
ν_i	Greek	Poisson's ratio of inner material	—
ν_o	Greek	Poisson's ratio of outer material	—
ρ	Greek	Density	kg/m^3
ρ	Greek	Radius of gyration (J.B. Johnson buckling formula only)	in, mm
σ_{yp}	Greek	Yield stress	psi, Pa
τ	Greek	Shear stress	psi, Pa

1.0 Introduction

This project's purpose is to improve and innovate on a Repetitive Impact Testing Apparatus (RITA) utilized by the School of Kinesiology at Lakehead University. The machine is utilized in injury biomechanics research to measure the force experienced by various objects due to repetitive impacts. Some of these applications include the analysis of forces on various walking cane tips, shoes, and protective sports equipment such as helmets. The current machine in Kinesiology has several shortcomings that our team believes that we can improve upon. Our team will improve the functionality of the machine by increasing the amount of compatible processes. This will be accomplished by designing intermediate adapters to mount different diameter cane tips and adding a range of motion to accommodate different angles of impact for protective equipment. The durability of the machine will also be improved by selecting material with characteristics more applicable to the tests performed, such as a higher yield point, higher strength, and corrosion resistance. These will be achieved by selecting four main parts for major redesign. These parts are the brackets of the force plate, the adjustment knuckle, the stepped shaft and the impact rod.

2.0 Problem Statement

The School of Kinesiology at Lakehead University provided an opportunity to redesign a Repetitive Impact Testing Apparatus (RITA). Dr. Zerpa indicated several issues with the current design, all of which limit the ability of the test apparatus to test the impact of various protective equipment. The apparatus will be redesigned to fix as many of these issues as feasible. Through the guidance of Dr. Liu, the new design will work to nullify the existing problems with the apparatus in order to satisfy the testing requirements of the School of Kinesiology Lakehead University.

3.0 Constraints and Criteria

The governing objectives and constrains utilized for major and minor component design are indicated below.

Design objectives:

- Design a rod attachment that can be adjusted to provide an impact on the tested object at any angle required.
- Design a replacement knuckle to reduce the damaged component, and reduce equipment oscillation.
- Redesign the force plate to be able to be struck at various angles during testing procedures.
- Design a means for securing rubber cane tips to the test apparatus more easily.
- Suggest improvements to the machine guarding elements.
- Reduce operating noise during testing procedures.

Design constraints:

- Machine adjustment must work with existing design.
- The new design must utilize the same actuating rod.
- Pneumatics and limit switches must stay the same.

3.1 Codes and Standards

Recommended force requirements from The Government of Canada Occupational Health and Safety were integrated into design calculations in order to ensure an appropriate metric for force was utilized. The minimum requirement for factor of safety of the stepped shaft is 1.[14] All final stepped shaft designs exceed this requirement, and most are within the ideal range of $2-4$.

For calculations performed on fasteners, some simplifications were required for analyses. The ultimate shear strength of a bolt is equal to 0.6 times the ultimate tensile strength.[6]

Additionally, for the impact rod yielding must be the governing mode of failure to ensure safety of the equipment's users. An ultimate fracture could create dangerous broken fragments during part failure. In order to ensure that components could be assembled without the use of power tools, a reference value of $50\ ft\text{-}lbs$ was utilized for fastener calculations.

4.0 Concept Generation

Several preliminary designs for a cane tip stepped shaft, support bracket, adjustment knuckle, and the impact rod were developed. The results of these preliminary designs are indicated below.

4.1 Cane Tip Stepped Shaft

For the cane tip stepped shaft, only one preliminary design was made. For the stepped shaft, the concept from all group members was essentially identical, as there are very limited ways to change the proposed design. Because of this Adam came up with the sole design, and it was utilized throughout all the stages of the design process. This design was required because the current method of rubber cane tip testing involved forcing a cane tip onto the end of the actuating cylinder, which had proven to be very difficult depending on the cane tip being tested. Our research had indicated that cane tips come in a standard size (with some exceptions), typically with an internal diameter of 3/4" and internal depth of 1-1/2". The narrow end of the actuating cylinder was measured to be approximately 0.94", approximately 25% larger than the internal diameter of the standard cane tip.

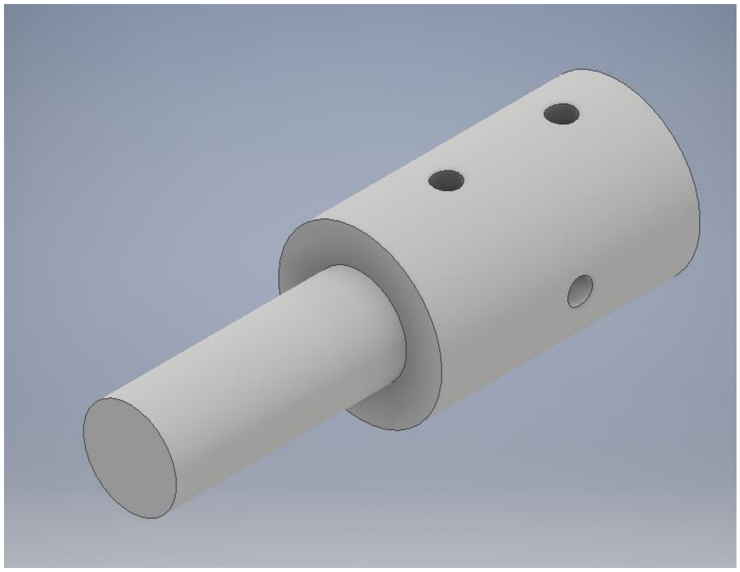


Figure 1: Cane tip stepped shaft preliminary design

As a group, we determined that a stepped shaft, illustrated above in Figure 1, that fits onto the end of the actuating cylinder would be the best way of incorporating the cane tip into testing procedures. No design tools were required for the development of this design, but an experimentally obtained modulus of elasticity for the cane tip was required.

4.2 Support Bracket

For the support bracket each group member came up with a design that they felt would be suitable to meet the design objectives. The existing setup is shown directly below in Figure 2, followed by each group members design in Figures 3 to 6. As can be seen, there were a lot of similarities between the designs proposed by the group members.

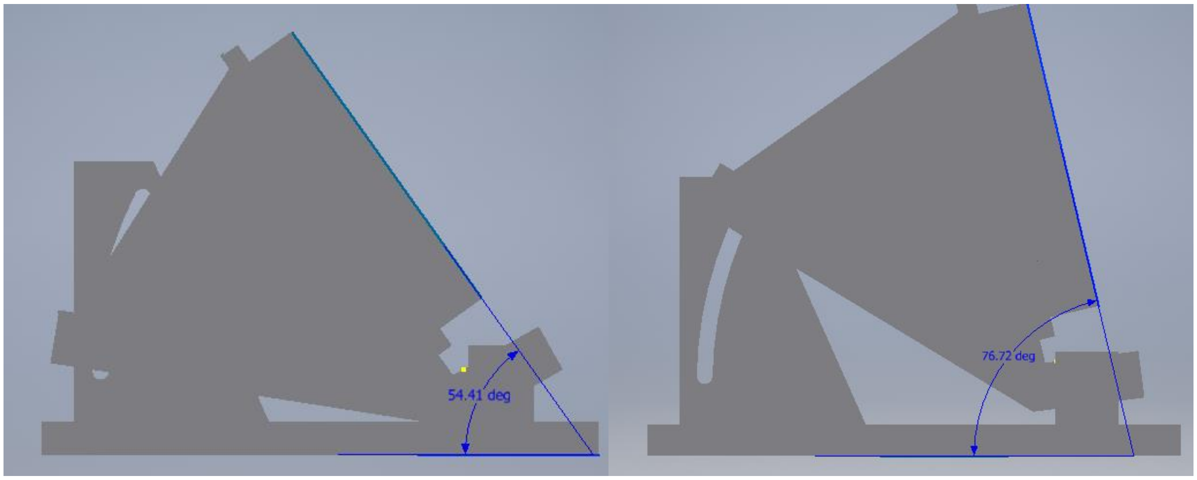


Figure 2: Current minimum (left) and maximum (right) force plate angles

The current force plate adjustment bracket does not allow the force plate to be set at a position that is perpendicular to the natural line of action of the actuating rod. As illustrated in Figure 10 above, the current setup has a minimum testing angle of $54.41°$ (due to machine interference) and a maximum testing angle of $76.22°$. In order for this to be remedied, both force plate adjustment brackets would have to be removed and replaced with resized brackets that have been explicitly designed for this purpose.

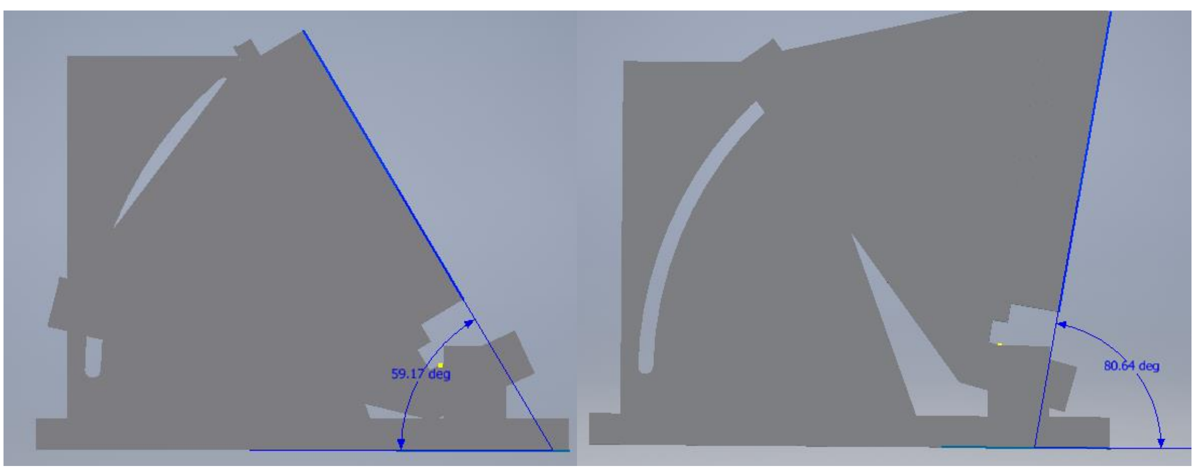


Figure 3: Adam's design for the support bracket

Inclusion of the proposed bracket would allow the force plate to be adjusted at a maximum value that exceeds $90°$. 3D simulations (Figure 3 above) indicate a new minimum angle of $59.17°$ (due to new machine interference) and a maximum angle of $99.36°$.

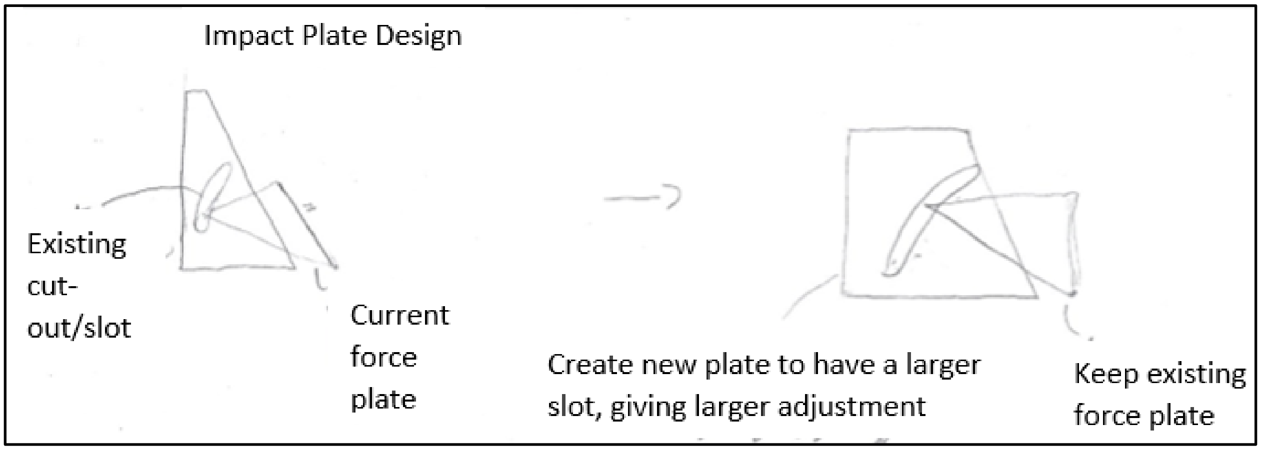


Figure 4: Dale's proposed impact plate modification

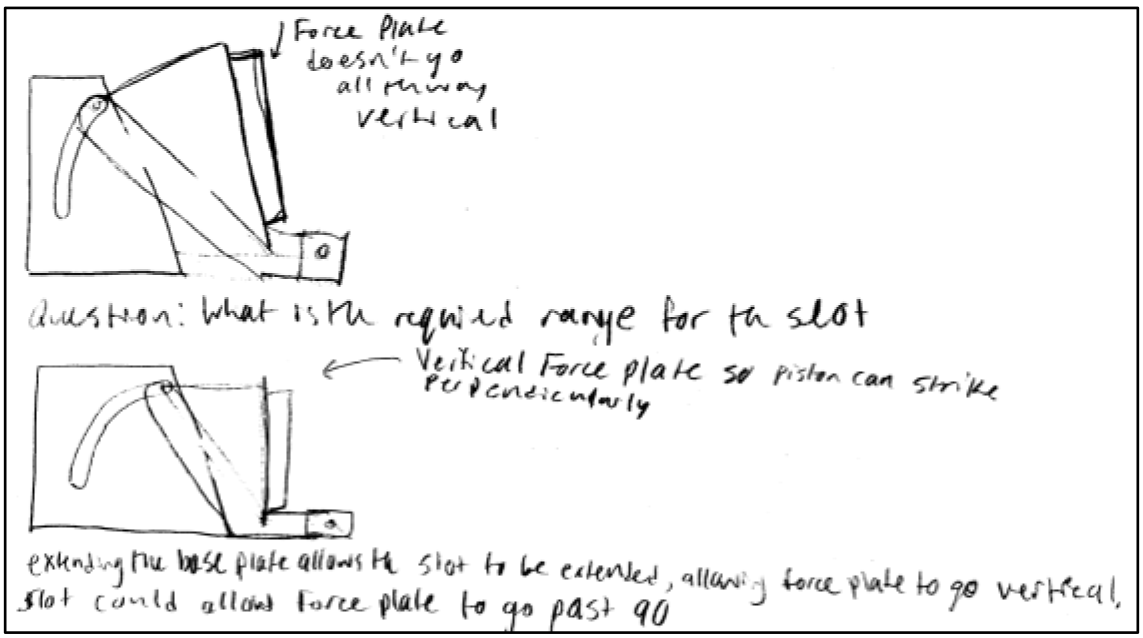


Figure 5: Courtis' current (top) and proposed (bottom) base for force plate

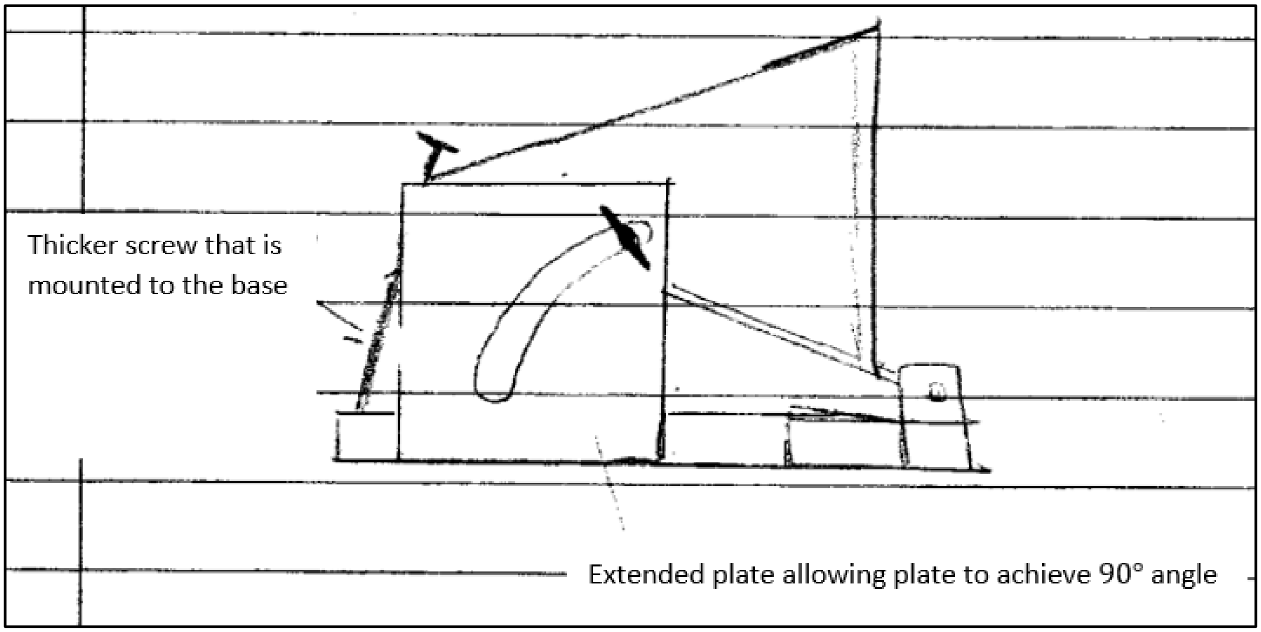


Figure 6: Mat's design for the support bracket

Through group discussion other design alternatives were eliminated, and Courtis' preliminary design was decided to be utilized for the next stages of the design process.

4.3 Adjustment Knuckle

Similarly, for the adjustment knuckle each group member came up with a design that they felt would be suitable to meet the design objectives. The existing setup is shown directly below in Figure 7, followed by each group members design in Figures 8 to 11. As can be seen, each group member's design was very similar to one another.

Figure 7: Existing knuckle design

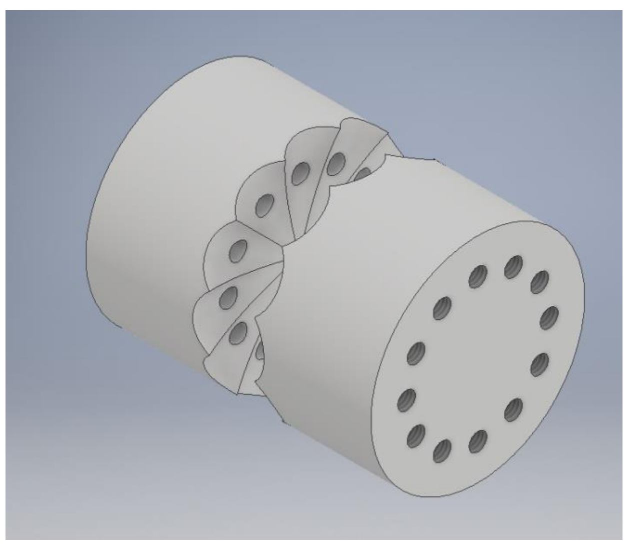


Figure 8: Adam's proposed knuckle design

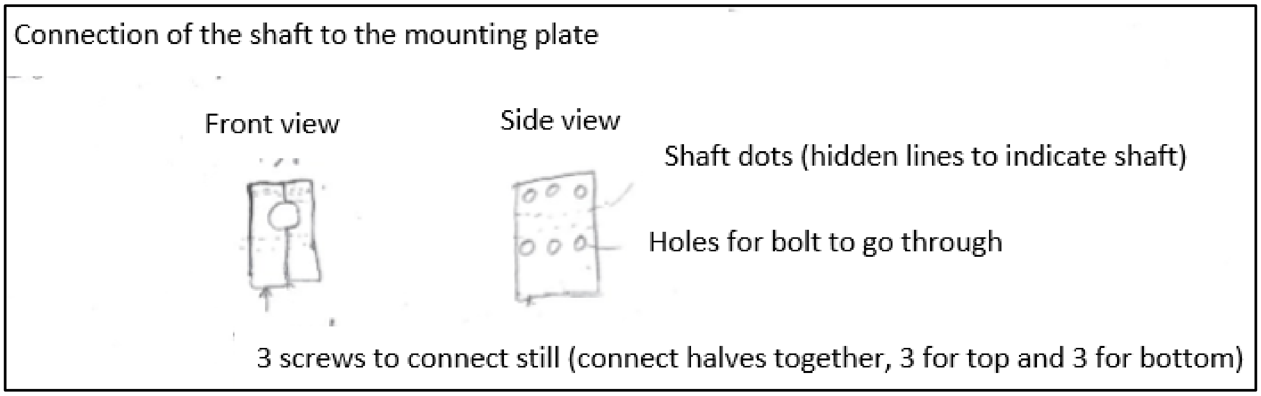


Figure 9: Dale's proposed knuckle design

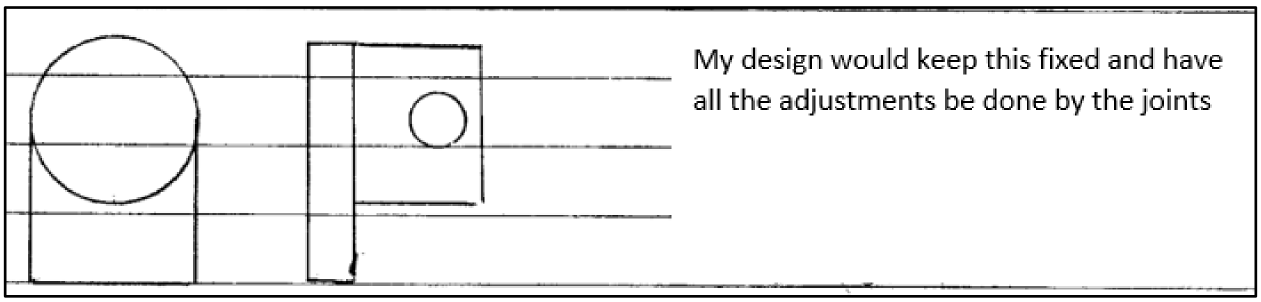


Figure 10: Mat's proposed knuckle design

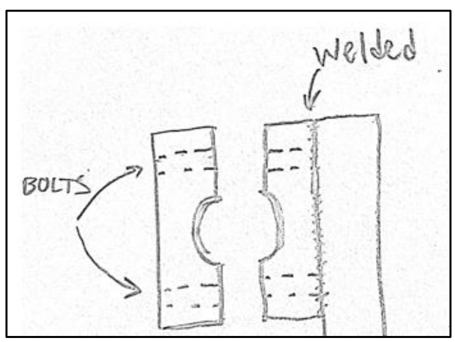


Figure 11: Courtis' proposed knuckle design

Through group discussion other design alternatives were eliminated, and Mat's proposed design was selected for the knuckle to move forwards to the next stages of the design process.

4.4 Impact Rod

For the impact rod each group member once again came up with a design that they felt would be suitable to meet the design objectives. There was no existing impact rod design, so each member's design was made entirely from scratch. As can be seen, each group member's design varied significantly from one another.

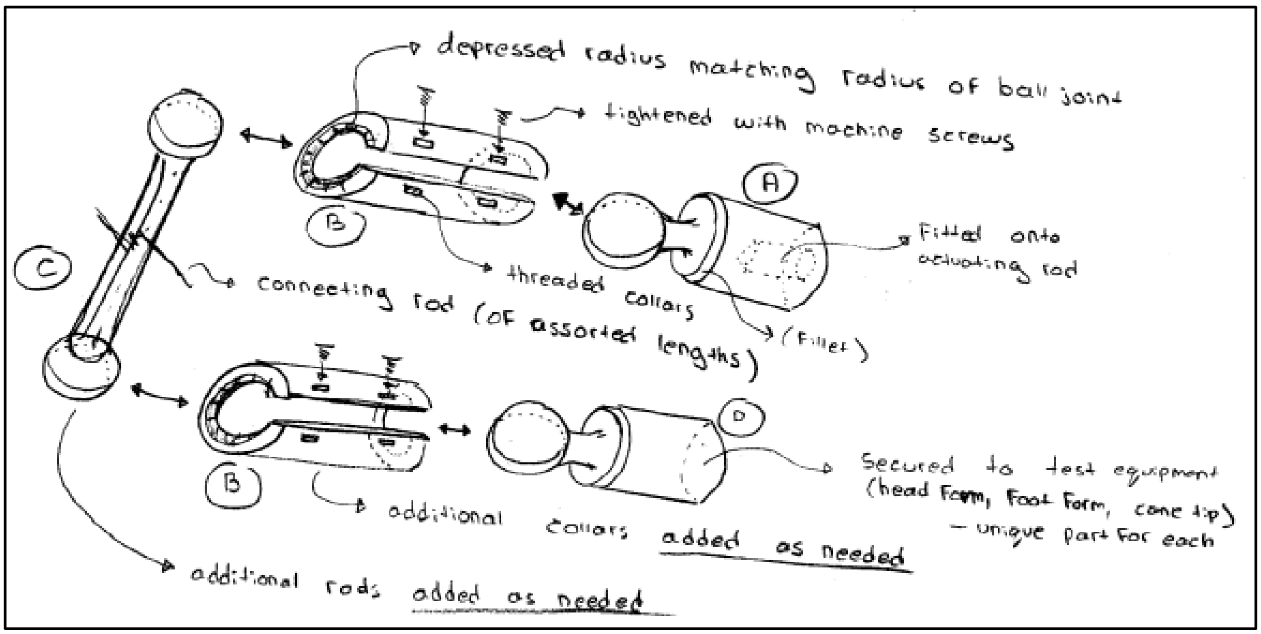


Figure 12: Adam's impact rod proposed design

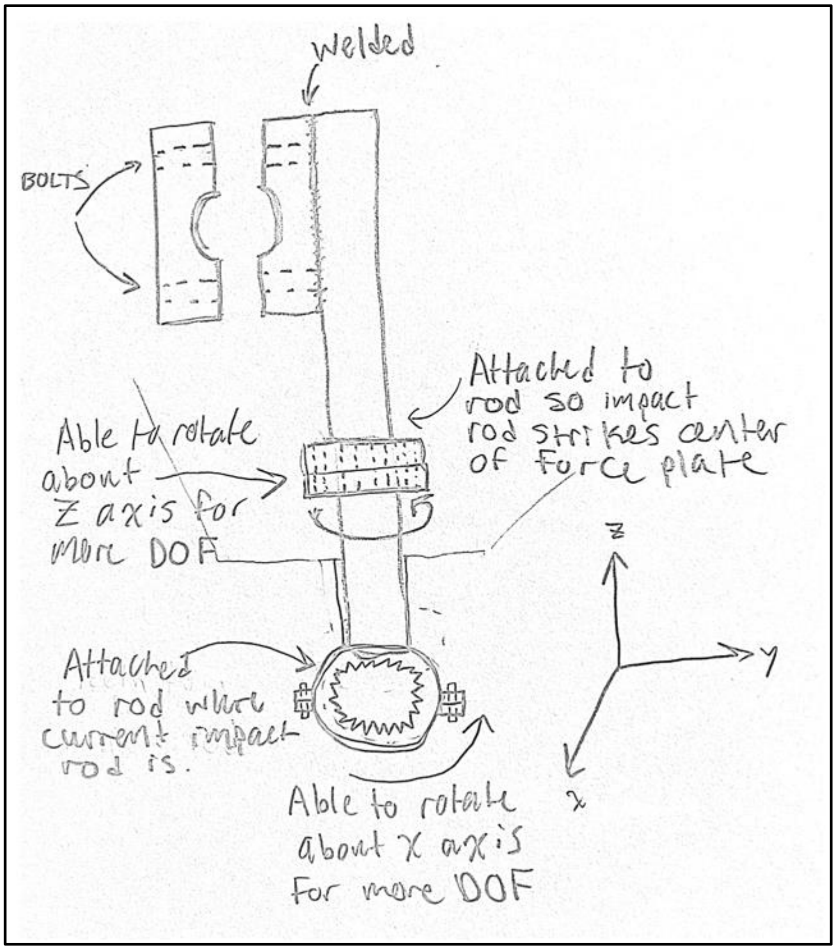


Figure 13: Courtis' Impact rod proposed design

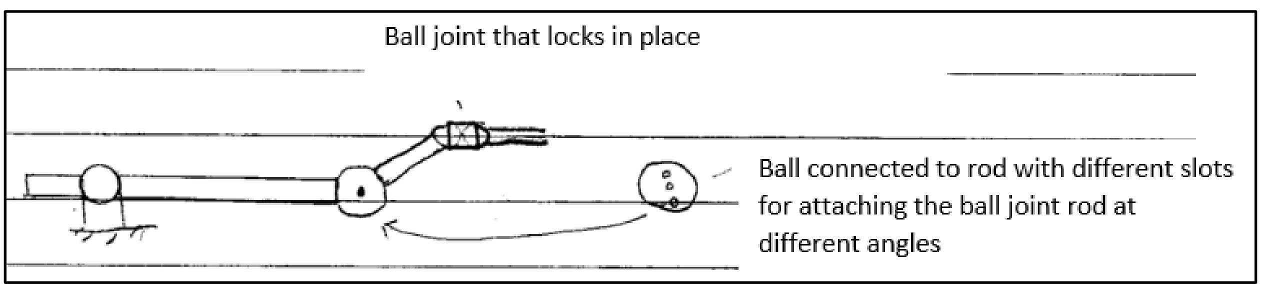


Figure 14: Mat's proposed impact rod design

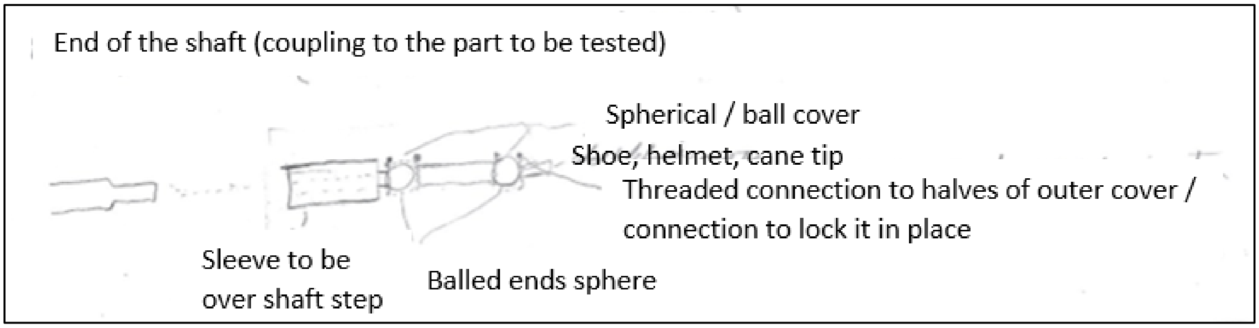


Figure 15: Dale's proposed impact rod design

Through the use of a pairwise comparison chart other design alternatives were eliminated, and Dale's proposed impact rod design was selected to use for the next stages of the design process. The reasons for this selection are highlighted within the next section.

5.0 Preliminary Design Layouts and Analyses

Each group member prepared their own pairwise comparison chart to rank the relative importance of identified secondary design objectives against each other (Tables 1 to 4). The resulting scores were then accumulated in an aggregate pairwise comparison chart (Table 5) to rank the order of importance of each design objective. Using a selection matrix, the designs developed in the first progress report were then ranked out of 5 on their ability to adequately meet the design objectives. This value was then multiplied by the score value determined in the aggregate pairwise comparison chart. Ultimately, the design with the highest total score would then be selected as the basis for the next iteration of the design, which could then be further developed with the objectives in the order of identified importance.

Table 1: Adam's pairwise comparison chart

Objectives	Safety	Cost	Durability	Ease of Use	Simplicity	Reliability	Score
Safety	-	1	1	1	1	1	5
Cost	0	-	0	0.5	0.5	0	1
Durability	0	1	-	1	1	1	4
Ease of Use	0	0.5	0	-	0.5	0.5	1.5
Simplicity	0	0.5	0	0.5	-	0	1
Reliability	0	1	0	0.5	1	-	2.5

Table 2: Dale's pairwise comparison chart

Objectives	Safety	Cost	Durability	Ease of Use	Simplicity	Reliability	Score
Safety	-	1	1	1	1	1	5
Cost	0	-	0	0.5	0.5	0	1
Durability	0	1	-	1	1	0.5	3.5
Ease of Use	0	0.5	0	-	0.5	0	1
Simplicity	0	0.5	0	0.5	-	0	1
Reliability	0	1	0.5	1	1	-	3.5

Table 3: Mat's pairwise comparison chart

Objectives	Safety	Cost	Durability	Ease of Use	Simplicity	Reliability	Score
Safety	-	1	1	1	1	1	5
Cost	0	-	0	0	0	0	0
Durability	0	1	-	1	1	0.5	3.5
Ease of Use	0	1	0	-	0	0	1
Simplicity	0	1	0	1	-	0	2
Reliability	0	1	0.5	1	1	-	3.5

Table 4: Courtis' pairwise comparison chart

Objectives	Safety	Cost	Durability	Ease of Use	Simplicity	Reliability	Score
Safety	-	1	1	1	1	1	5
Cost	0	-	0.5	1	1	0.5	3
Durability	0	0.5	-	1	1	1	3.5
Ease of Use	0	0	0	-	0	0	0
Simplicity	0	0	0	1	-	0	1
Reliability	0	0.5	0	1	1	-	2.5

Table 5: Aggregate pairwise comparison chart

Objectives	Members				Weighted Value
	Adam	Dale	Mat	Courtis	
Safety	5	5	5	5	20
Cost	1	1	0	3	5
Durability	4	3.5	3.5	3.5	14.5
Ease of Use	1.5	1	1	0	3.5
Simplicity	1	1	2	1	5
Reliability	2.5	3.5	3.5	2.5	12

As indicated below in Table 6, for the impact rod, Dale's preliminary design was chosen as the basis for the next iteration of the design process.

Table 6: Selection matrix

Design Constraints	Adam's Preliminary Design	Dale's Preliminary Design	Mat's Preliminary Design	Courtis' Preliminary Design
Prev. Design Compatibility	✔	✔	✔	✔
Same Rod Attachments	✔	✔	✔	✔
Pneumatic devices stay the same	✔	✔	✔	✔
Design Objectives	Score	Score	Score	Score
Safety	3	4	2	3
Cost	2	3	3	3
Durability	3	4	3	3
Ease of Use	5	4	3	3
Simplicity	2	2	3	3
Reliability	4	5	2	2
Total Score (Score · Weighted Value)	380	440	320	340

As can be seen, an aggregate pairwise comparison chart was only used for the design selection of the impact rod. This was done for multiple reasons. Section 4, concept generation, shows that for the adjustment knuckle and support bracket all of the group members had rather similar ideas. With the high degree in similarity of the designs comparing them was not as highly important as that of the impact rod. Additionally, the impact rod was substantially more complex than other design components, this rod experiences a large amount of stress, and plays a crucial role in meeting the design objectives. Due to these facts, it stood that a higher degree of selection scrutiny should have been placed on the impact rod, so it was done. Through a group discussion with regards to our personal pairwise comparison charts the knuckle moved forward with Mat's preliminary design, and the support bracket moved forward with Courtis' preliminary design. Adam's concept was then used to move forward with the stepped shaft.

5.1 Mid-semester Design Presentation

On February 10th, 2020 the group conducted an informal design presentation with Dr. Liu, Dr. Zerpa, and Morgan Ellis to present major design progression. Through group collaboration, several design adjustments were identified to be integrated into each major final design. These changes are noted for each major design in the subsections below.

5.1.1 Cane Tip Stepped Shaft

Feedback verified that the most prevalent cane tip internal diameters were 3/4-inch, 7/8-inch, 1-inch. As a result, cane tip stepped shafts for these internal diameters will be developed.

5.1.2 Support Bracket

Since the force plate portion of the machine is very heavy, there is a concern for operator safety while adjustment to testing positions is being altered on the proposed design. A fail-safe mechanism was suggested, in the form of an additional bolt, that would catch the hinged force plate section during these scenarios. The additional bolt would be welded to the support bracket after installations, and would remain in place for all future equipment tests.

5.1.3 Adjustment Knuckle

The proposed design recreated the existing adjustment knuckle, but in fewer segmented components. To simplify the manufacturing process, the knuckle is no longer milled into a round shape, and will be cut from available square stock. The interfacing section between the adjustment knuckle and the actuating rod was suggested to be increased, and additional bolts utilized to secure the fit between the two sections.

5.1.4 Impact Rod

During the mid semester design presentation with the four group members, Dr. Liu, Dr. Zerpa, and Mr. Morgan Ellis important positive feedback was received, along with much appreciated ideas for improvement. Most important for the impact rod was Dr. Zerpa's approval of the design. Additional input from Mr. Ellis allowed for a few minor, but key changes to the cover's design that allowed for increased rigidity, strength, safety, and ease of use. Shown below in Appendix K, the first design represents the model before the changes and the second design is the model after the changes had been made. The first drawing does not show dimensions, as it was not needed as it would not be utilized for manufacturing purposes. As can be seen, the changes consisted of adding threads to one side of the cover, rather than using a nut on the back side.

This change allows for a reduction in bolt length required, and a second wrench will no longer be required to tighten or loosen the cover. The other change to the design is the addition of a recessed hole, which when used in part with a socket head bolt allows for the bolt face to be covered by the outer cover. This helps increase safety, as the bolt face will no longer be sticking out during operation of the RITA. Getting input from Mr. Ellis towards the design was very beneficial, as it was extremely helpful to obtain input from someone with significant manufacturing and hands-on experience relating to this.

6.0 Final Design and Analyses

Several analyses were performed to calculate critical components of major design aspects and verify their safe use for the conditions prescribed previously. The four major design components analyzed in this section are the Cane Tip Stepped Shaft, Support Bracket, Adjustment Knuckle, and the Impact Rod.

6.1 Cane Tip Stepped Shaft

One major design aspect that was addressed during equipment redesign was the creation of a stepped shaft that allowed cane tips to attach onto the actuating shaft more easily. In previous test iterations, rubber cane tips of differing internal diameters were stretched onto the end of the same actuating shaft, a task which could be extraordinarily difficult depending on the internal diameter of the cane tip. Through mathematical modeling an appropriate diametral interference between a solid cylindrical shaft and a rubber can tip was determined for each cane tip used in experimental trials. One stepped shaft was developed for use with each type of rubber cane tip utilized in experimental trials with a different internal diameter ($3/4$", $7/8$", and 1"). The end of the actuating shaft is indicated below in Figure 16.

Figure 16: Existing actuating shaft connection

While initially designed for affixing other test equipment onto the actuating shaft, such as head forms and foot forms, the stepped shaft was designed to attach via a similar methodology. After the correct diametral interference was calculated, a more appropriate interfacing diameter was determined, and calculations were performed to verify that the maximum force experienced does not exceed the calculated critical load. The completed final design is included in Appendix F.

6.1.1 Material Testing

As previously mentioned, the rubber cane tips utilized in experimental trials have differing internal diameters. Two typical rubber cane tips utilized in experiments are indicated below in Figure 17.

Figure 17: Rubber cane tip (Left: Type A, Right: Type B)

Type A has an internal diameter of $3/4$", while type B has an internal diameter of $7/8$". Almost all mechanical properties of the rubber cane tips can be determined through accurate measuring and modeling, however the material composition is not directly specified in any available documentation. Research indicated a significant variance in published data and myriad varieties of potential rubber compositions. As a result, the Young's modulus of the rubber cane tips had to be determined experimentally with the material testing apparatus located in CB-1016. The material testing apparatus, in addition to a test in progress on the lower portion of type B, is indicated below in Figure 18.

Figure 18: Rubber cane tip material testing

The data from the material testing apparatus was utilized in conjunction with measured mechanical properties to create the stress-strain plots indicated in Appendix C. Forward and reverse strokes were found to develop slightly differing stress-strain plots, and as such were plotted separately. The linear portion of each dataset, on each stress-strain curve, was determined and averaged to determined a single representative Young's modulus for both types of cane tips. Our testing indicated an average Young's modulus of $4.786\ MPa$, or approximately $0.694\ ksi$.

6.1.2 Calculations

Using techniques previously introduced in Mechanical Engineering Design I (EMEC-4130), the diametral interference (δ) between two different materials was calculated with the following governing equation:

$$p = \frac{\delta}{D\left[\frac{1}{E_o}\left(\frac{d_o^2 + D^2}{d_o^2 - D^2} + v_o\right) + \frac{1}{E_i}\left(\frac{D^2 + d_i^2}{D^2 - d_i^2} - v_i\right)\right]}$$

The determined diametral interference was accommodated into the design diameter of the interfacing section between the stepped shaft and the rubber cane tip, henceforth referred to as the critical diameter, before being checked for failure due to buckling using the J.B. Johnson formula indicated below:

$$F_{crit} = A_c \sigma_{yp} \left(1 - \frac{B}{4\rho^2}\right)$$

Where:

$$B = \frac{\sigma_{yp} l}{n\pi^2 E_i}$$

Verifying:

$$n = \frac{F_{crit}}{F_{allow}} > 1$$

Consider that the buckling mode of failure typically occurs before axial compression stresses induce yielding in a compression member. In this scenario, the critical load for yielding in the stepped shaft would be calculated as follows:

$$F_{crit} = \sigma_{yp} \cdot \frac{\pi d_{crit}^2}{4}$$

Or:

$$F_{crit} = A_c \sigma_{yp}$$

Thus, the calculated critical load due to yielding is a less conservative variation of the J.B. Johnson formula indicated above. In addition, no load fluctuations are present in our mathematical model, and as such there would be no variation in yielding due to static loading or fatigue loading. As a result, only the critical load due to buckling, and its associated factor of safety, is utilized in calculation iterations.

Preliminary design iterations are shown below to illustrate the design process for a stepped shaft to be utilized with a $3/4$-inch internal diameter cane tip. A similar design procedure was utilized to develop a stepped shaft for a $7/8$-inch and 1-inch internal diameter cane tip.

6.1.2.1 Design Iteration 1

Assumptions:

- Main governing criteria is the amount of force required to replace/remove the rubber cane tip from the stepped shaft (while kneeling).
- Rubber mechanical properties are equivalent to rubber used in a typical 60 A Belt (Natural rubber)
- Shaft mechanical properties are constant, and are equivalent to Stainless Steel 316.
- Cane tip to be secured without the use of specialized tools.

Rubber cane tip mechanical properties:

- Inner Diameter: $D = 0.75\ in$ (measured)
- Inner Length: $l = 1.625\ in$ (measured)
- Wall Thickness: $t = 0.25\ in$ (measured – from thinnest walled tip used in experiments)
- Poisson's Ratio[15]: $v_o = 0.50$
- Young's modulus[4]: $0.0015\ GPa \leq E_o \leq 0.0025\ GPa$
    - Using max value: $E_o = 0.0025\ GPa$
    - In US standard units: $E_o = 0.3626\ ksi$

Actuating shaft mechanical properties:

- Poisson's Ratio[3]: $v_i = 0.25$
- Young's modulus [3]: $E_i = 28000\ ksi$

Other mechanical properties:

- Coefficient of friction (dry contact) between Rubber and Stainless Steel[8]: $\mu = 0.64$
- Recommended force for replacing a component from equipment (kneeling)[11]: $F = 42\ lbf$

The minimum axial force required to press the hub on the shaft can be represented as[14]:

$$F = \mu A_s p = \mu(\pi D l)p$$

Then the contact pressure between the two parts can be calculated as:

$$p = \frac{(42\ lb)}{(0.64)\pi(0.75\ in)(1.625\ in)} = 17.1398\ psi$$

The contact pressure between the two assembled parts can also be represented as[14]:

$$p = \frac{\delta}{D\left[\frac{1}{E_o}\left(\frac{d_o^2 + D^2}{d_o^2 - D^2} + v_o\right) + \frac{1}{E_i}\left(\frac{D^2 + d_i^2}{D^2 - d_i^2} - v_i\right)\right]}$$

Consider:

$$d_o = D + 2t = 1.25\ in$$

$$d_i = 0$$

Then the diametral interference can be calculated as:

$$\delta = (17.1398\ psi)(0.75\ in)\left[\frac{1}{(0.3626 \cdot 10^3\ psi)}\left(\frac{(1.25\ in)^2\ + (0.75\ in)^2}{(1.25\ in)^2 - (0.75\ in)^2} + (0.50)\right)\right.$$

$$\left. + \frac{1}{(28000 \cdot 10^3)}\left(\frac{(0.75\ in)^2\ + (0)^2}{(0.75\ in)^2 - (0)^2} - 0.25\right)\right]$$

$$\delta = 0.0931\ in$$

Stepped shaft mechanical properties:

- Shaft diameter: $d = 0.75\ in$
- Length of narrow section: $l = 3.75\ in$
- Tensile strength, Yield[3]: $\sigma_{yp} = 41.1\ ksi$

Other mechanical properties:

- End-fixity coefficient: $n = 1$ (assumed)
- Design critical load: $F_{allow} = 45\ kN$ (provided by Dr. Liu)
    - (In US standard units: $F_{allow} = 10.1164\ kip$)

In order to determine which instability consideration should be utilized, the constant B must first be determined[12]:

$$B = \frac{\sigma_{yp} l}{n\pi^2 E_i}$$

Then:

$$B = \frac{(41.1 \cdot 10^3)(3.75)}{(1)\pi^2(28000 \cdot 10^3)} = 0.0005577$$

Checking[12]:

$$\frac{B}{\rho^2} = \frac{0.0005577}{\left(\frac{0.75}{4}\right)^2} = 0.015864 < 2$$

$$\therefore J.B. Johnson\ justified$$

Consider the J.B. Johnson buckling formula[12]:

$$F_{crit} = A_c \sigma_{yp} \left(1 - \frac{B}{4\rho^2}\right)$$

For a circular cross section[12]:

$$\rho = \frac{d}{4}$$

Then:

$$F_{crit} = \frac{\pi}{4}(0.75)^2(41.1 \cdot 10^3)\left(1 - \frac{0.0005577}{4\left(\frac{0.75}{4}\right)^2}\right) = 18.0854\ kip$$

Verifying:

$$n = \frac{18.0854\ kip}{10.1164\ kip} \approx 1.8 > 1$$

∴ Design OK, proceed to next iteration

6.1.2.2 Design Iteration 2

Modifications:

- Main governing criteria is still the amount of force required to replace/remove the rubber cane tip from the stepped shaft (due to customer input), but now utilizes the entire body (no longer from kneeling position). This is a more realistic approximation of the force that is utilized.
- Young's modulus for butyl rubber is utilized in place of natural rubber, and determined from the average of two stress-strain plots of material performance. Likely a better approximation for rubber cane tip properties.
- Bucking no longer assumed to occur in the narrow section of the stepped shaft. The critical load in the solid and hollow portion of the stepped shaft are determined independently.
- The length of the contact surface with rubber cane tip has been increased ($1.625\ in \rightarrow 2\ in$) to guarantee a secure fit.

Assumptions:

- Main governing criteria is the amount of force required to replace/remove the rubber cane tip from the stepped shaft (standing, whole body involved).
- Rubber mechanical properties are equivalent to butyl rubber.
- Shaft mechanical properties are constant, and are equivalent to Stainless Steel 316.
- Cane tip to be secured without the use of specialized tools.

Rubber cane tip mechanical properties:

- Inner Diameter: $D = 0.75\ in$ (measured)
- Inner Length: $l = 1.625\ in$ (measured)

- Wall Thickness: $t = 0.25\ in$ (measured – from thinnest walled tip used in experiments)

- Poisson's Ratio[1]: $v_o = 0.50$ (assumed to be the same as natural rubber – no data available)

Young's modulus is the average slope of the elastic region of the following stress strain curves[16]:

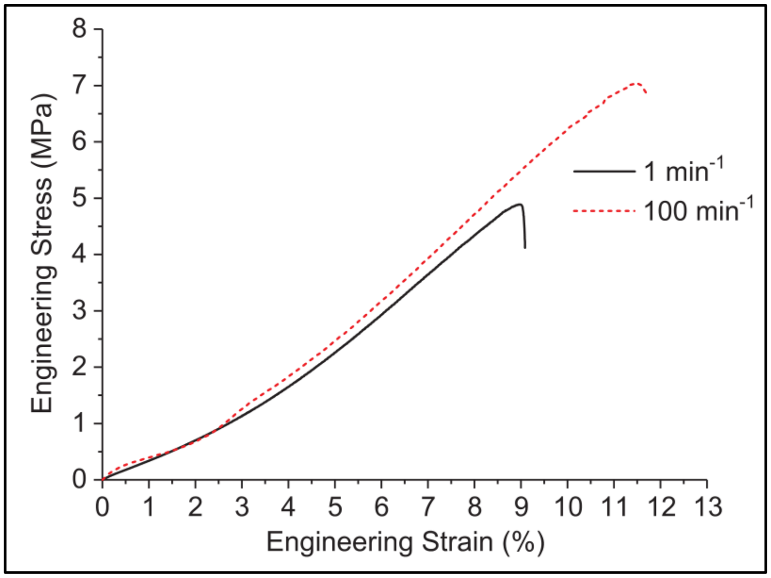


Figure 19: Stress-strain curve for butyl rubber

  - For 1 min^{-1}: $E_1 = 0.8\ MPa$
  - For 10 min^{-1}: $E_{10} = 0.6353\ MPa$
  - The average Young's modulus is: $E_o = \frac{E_1 + E_{10}}{2} = 0.7176\ MPa$

- Young's modulus: $E_o = 0.7176\ MPa$
  - In US standard units: $E_o = 0.1041\ ksi$

Actuating shaft mechanical properties:

- Poisson's Ratio[3]: $v_i = 0.25$

- Young's modulus [3]: $E_i = 28000\ ksi$

Other mechanical properties:

- Coefficient of friction (dry contact) between Rubber and Stainless Steel[8]: $\mu = 0.64$

- Recommended force for replacing a component from equipment (kneeling)[11]: $F = 50\ lbf$

Calculations:

The minimum axial force required to press the hub on the shaft can be represented as[14]:

$$F = \mu A_s p = \mu(\pi D l)p$$

Then the contact pressure between the two parts can be calculated as:

$$p = \frac{(50\ lb)}{(0.64)\pi(0.75\ in)(1.625\ in)} = 20.4045\ psi$$

The contact pressure between the two assembled parts can also be represented as[14]:

$$p = \frac{\delta}{D\left[\frac{1}{E_o}\left(\frac{d_o^2 + D^2}{d_o^2 - D^2} + v_o\right) + \frac{1}{E_i}\left(\frac{D^2 + d_i^2}{D^2 - d_i^2} - v_i\right)\right]}$$

Consider:

$d_o = D + 2t = 1.25\ in$

$d_i = 0$

Then the diametral interference can be calculated as:

$$\delta = (20.4045\ psi)(0.75\ in)\left[\frac{1}{(0.1041 \cdot 10^3\ psi)}\left(\frac{(1.25\ in)^2\ + (0.75\ in)^2}{(1.25\ in)^2 - (0.75\ in)^2} + (0.50)\right)\right.$$

$$\left. + \frac{1}{(28000 \cdot 10^3)}\left(\frac{(0.75\ in)^2\ + (0)^2}{(0.75\ in)^2 - (0)^2} - 0.25\right)\right]$$

$$\delta = 0.3859\ in$$

Stepped shaft mechanical properties (narrow section, contact with rubber cane tip):

- Outer diameter: $d = 0.75\ in$
- Length of narrow section: $l = 2\ in$
- Tensile strength, Yield[3]: $\sigma_{yp} = 41.1\ ksi$

Other mechanical properties:

- End-fixity coefficient: $n = 1$ (assumed)
- Design critical load: $F_{allow} = 45\ kN$ (provided by Dr. Liu)
    - (In US standard units: $F_{allow} = 10.1164\ kip$)

In order to determine which instability consideration should be utilized, the constant B must first be determined[12]:

$$B = \frac{\sigma_{yp} l}{n\pi^2 E_i}$$

Then:

$$B = \frac{(41.1 \cdot 10^3)(2)}{(1)\pi^2(28000 \cdot 10^3)} = 0.0002975$$

For a circular cross section[12]:

$$\rho = \frac{d}{4}$$

Checking[12]:

$$\frac{B}{\rho^2} = \frac{0.0002975}{\left(\frac{0.75}{4}\right)^2} = 0.0085 < 2$$

$$\therefore J.B.Johnson\ justified$$

Consider the J.B. Johnson buckling formula[12]:

$$F_{crit} = A_c \sigma_{yp}\left(1 - \frac{B}{4\rho^2}\right)$$

Then:

$$F_{crit,1} = \frac{\pi}{4}(0.75)^2(41.1 \cdot 10^3)\left(1 - \frac{0.0002975}{4\left(\frac{0.75}{4}\right)^2}\right) = 18.119\ kip$$

Stepped shaft mechanical properties (hollow section, contact with actuating shaft):

- Outer diameter: $D = 1.4\ in$
- Inner diameter: $d = 0.9450\ in$
- Length of narrow section: $l = 1.75\ in$
- Yield strength[3]: $\sigma_{yp} = 41.1\ ksi$

Other mechanical properties:

- End-fixity coefficient: $n = 1$ (assumed)
- Design critical load: $F_{allow} = 45\ kN$ (provided by Dr. Liu)
    - In US standard units: $F_{allow} = 10.1164\ kip$

In order to determine which instability consideration should be utilized, the constant B must first be determined[12]:

$$B = \frac{\sigma_{yp} l}{n\pi^2 E_i}$$

Then:

$$B = \frac{(41.1 \cdot 10^3)(1.75)}{(1)\pi^2(28000 \cdot 10^3)} = 0.0002603$$

For a hollow circular cross section[12]:

$$\rho = \frac{\sqrt{D^2 + d^2}}{4}$$

Checking[12]:

$$\frac{B}{\rho^2} = \frac{0.0002975}{\left(\frac{\sqrt{(1.4)^2 + (0.9450)^2}}{4}\right)^2} = 0.0015 < 2$$

$$\therefore J.B.Johnson\ justified$$

Consider the J.B. Johnson buckling formula[12]:

$$F_{crit} = A_c \sigma_{yp} \left(1 - \frac{B}{4\rho^2}\right)$$

Then:

$$F_{crit,2} = \frac{\pi}{4}(1.4^2 - 0.9450^2)(41.1 \cdot 10^3)\left(1 - \frac{0.0002603}{\left(\frac{\sqrt{(1.4)^2 + (0.9450)^2}}{4}\right)^2}\right) = 34.429\ kip$$

Checking:

$$F_{crit,1} < F_{crit,2}$$

Narrow section buckles first, check critical load of narrow section against design critical load for future design iterations.

$$n = \frac{18.0854\ \ kip}{10.1164\ kip} \approx 1.8 > 1$$

∴ Design OK, proceed to next iteration

6.1.2.3 Design Iteration 3

Modifications:

- Young's modulus for rubber cane tip to be calculated experimentally via the impact tester in the material testing lab (CB-1016).
- Diametral interference fit calculations are now done through the use of a MATLAB script.
- Buckling calculations are now done through the use of a MATLAB script, narrow section buckling diameter updated to include the calculated diametral interference (δ).

Assumptions:

- Main governing criteria is the amount of force required to replace/remove the rubber cane tip from the stepped shaft (standing, whole body involved).
- Young's modulus of rubber cane tip is an experimentally obtained result.
- Shaft mechanical properties are constant, and are equivalent to Stainless Steel 316.
- Cane tip to be secured without the use of specialized tools.

Stepped shaft mechanical properties (narrow section, contact with rubber cane tip):

- Outer diameter: $d = 0.75\ in$
- Length of section: $l = 2\ in$
- Tensile strength, yield[3]: $\sigma_{yp} = 41.1\ ksi$
- Young's modulus [3]: $E_i = 28000\ ksi$

Stepped shaft mechanical properties (hollow section, contact with actuating shaft):

- Outer diameter: $D = 1.4\ in$
- Inner diameter: $d = 0.9450\ in$
- Length of section: $l = 1.75\ in$

- Yield strength[3]: $\sigma_{yp} = 41.1\ ksi$
- Young's modulus [3]: $E_i = 28000\ ksi$

Other mechanical properties

- End-fixity coefficient: $n = 1$ (assumed)
- Design critical load: $F_{allow} = 45\ kN$ (provided by Dr. Liu)
    - In US standard units: $F_{allow} = 10.1164\ kip$

Rubber cane tip mechanical properties:

- Inner Diameter: $D = 0.75\ in$ (measured)
- Inner Length: $l = 1.625\ in$ (measured)
- Wall Thickness: $t = 0.25\ in$ (measured – from thinnest walled tip used in experiments)
- Poisson's Ratio[1]: $v_o = 0.50$ (assumed to be the same – no data)

Using the material tester located in the engineering labs the mechanical properties of the rubber cane tips were determined experimentally. All experimental data is presented in Appendix C, and a final average Young's modulus of $4.786\ MPa$ ($0.694\ ksi$) was determined.

Using the MATLAB file IntFitCheck.m (Appendix L), the required diametral interference was calculated as:

$$\delta = 0.0579\ in$$

With some rounding, the diameter of the narrow portion of the stepped shaft becomes:

$$D = 0.8079\ in$$

Using BuckCheck.m (Appendix M), the critical loads are determined to be:

$$F_{crit,narrow} = 21.030\ kip$$

$$F_{crit,hollow} = 34.429\ kip$$

$\therefore$ narrow section will fail first

Verifying:

$$n = \frac{21.030 \ kip}{10.1164 \ kip} \approx 2.1 > 1$$

$\therefore$ Design OK, proceed to next iteration

6.1.2.4 Design Iteration 4

Process modifications:

- Young's modulus and yield strength of the stepped shaft updated to reflect material available for order (AISI 1018 CD Steel).
- End-fixity coefficient (n) updated to 2 to reflect fixed-free end conditions. This is a more accurate representation of operating conditions.
- Only the narrow section of the stepped shaft is under compression during machine operation. Thus, the buckling will never occur in the hollow section of the stepped shaft, and associated results form the MATLAB program for the hollow section of the stepped shaft can be neglected.

Assumptions:

- Main governing criteria is the amount of force required to replace/remove the rubber cane tip from the stepped shaft (standing, whole body involved).
- Young's modulus of rubber cane tip is an experimentally obtained result.
- Shaft mechanical properties are constant, and are equivalent to Stainless Steel 316.
- Cane tip to be secured without the use of specialized tools.

Stepped shaft mechanical properties (narrow section, contact with rubber cane tip):

- Outer diameter: $d = 0.75 \ in$
- Length of section: $l = 2 \ in$

- Yield strength[2]: $\sigma_{yp} = 50.0\ ksi$
- Young's modulus [2]: $E_i = 29000\ ksi$

Other mechanical properties:

- End-fixity coefficient: $n = 2$ (fixed-free)
- Design critical load: $F_{allow} = 45\ kN$ (provided by Dr. Liu)
    - In US standard units: $F_{allow} = 10.1164\ kip$

Rubber cane tip mechanical properties:

- Inner Diameter: $D = 0.75\ in$ (measured)
- Inner Length: $l = 1.625\ in$ (measured)
- Wall Thickness: $t = 0.25\ in$ (measured – from thinnest walled tip used in experiments)
- Poisson's Ratio[15]: $v_o = 0.50$ (assumed to be the same – no data)

Using the MATLAB file IntFitCheck.m (Appendix L), the required diametral interference was calculated as:

$$\delta = 0.0579\ in$$

With some rounding, the diameter of the narrow portion of the stepped shaft becomes:

$$D = 0.8079\ in$$

Using BuckCheck.m (Appendix M), the critical load is determined to be:

$$F_{crit,narrow} = 25.603\ kip$$

Verifying:

$$n = \frac{25.603\ kip}{10.1164\ kip} \approx 2.5 > 1$$

∴ Design OK, proceed to modeling

6.1.3 Finite Element Analysis

The validity of the calculated parameters were then verified through the use of FEA (finite element analysis) in Autodesk Inventor. In order to simulate operating conditions each stepped shaft was modeled with a compressive axial load applied to the narrow end of the stepped shaft at the interior most recessed face of the hollow portion. Full details of each analysis can be found in Appendix C.

The results of the FEA on a stepped shaft for a cane tip with an internal diameter of 3/4-inch is indicated below in Figure 20.

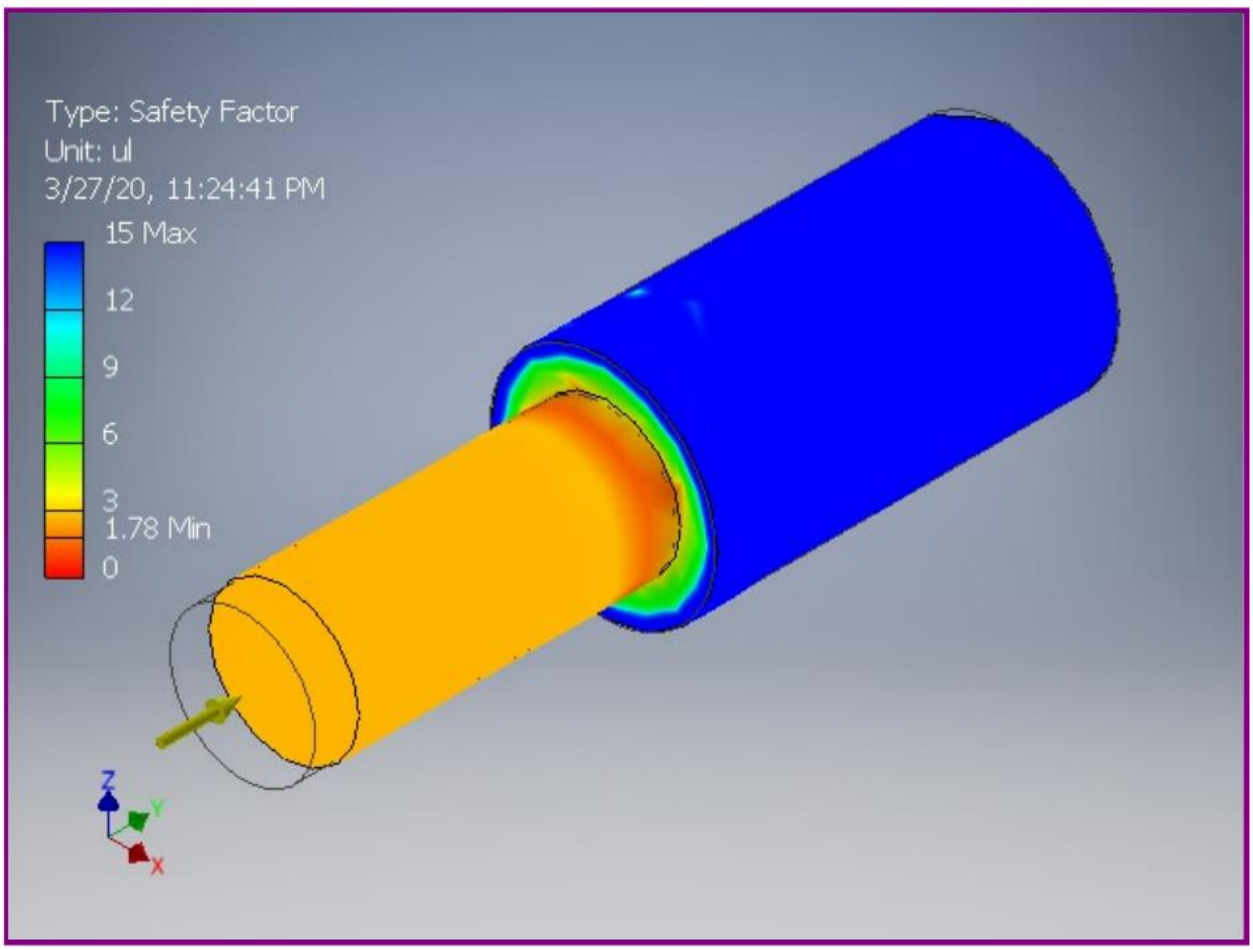


Figure 20: Finite element analysis on 3/4-inch stepped shaft

Simulations determined a minimum factor of safety of 1.78 at the base of the narrow shaft, within safe operating conditions.

The results of the FEA on a stepped shaft for a cane tip with an internal diameter of 7/8-inch is indicated below in Figure 21.

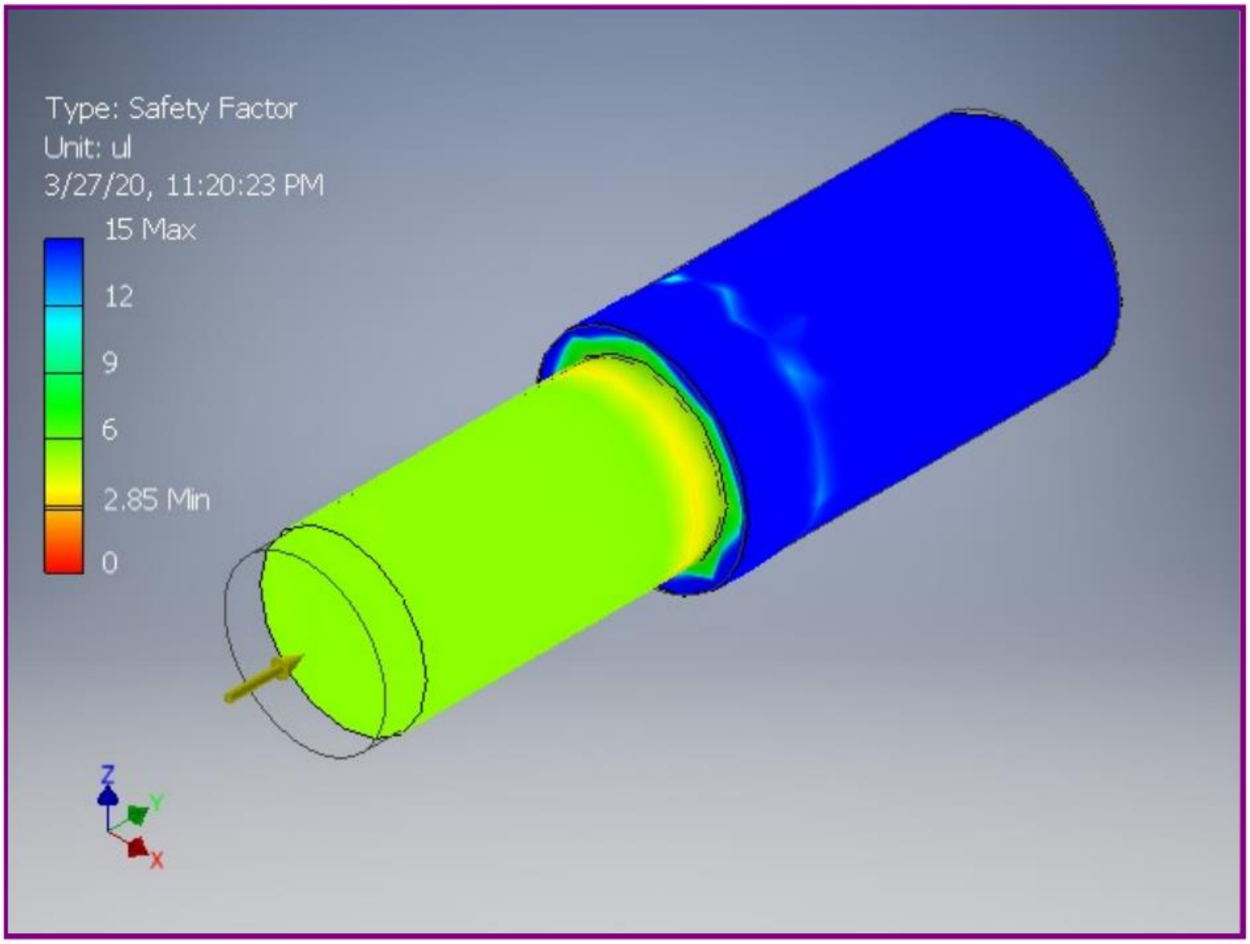


Figure 21: Finite element analysis on 7/8-inch stepped shaft

Simulations determined a minimum factor of safety of 2.85 at the base of the narrow shaft, which is also within safe operating conditions.

Finally, the results of the FEA on a stepped shaft for a cane tip with an internal diameter of 1-inch is indicated below in Figure 22.

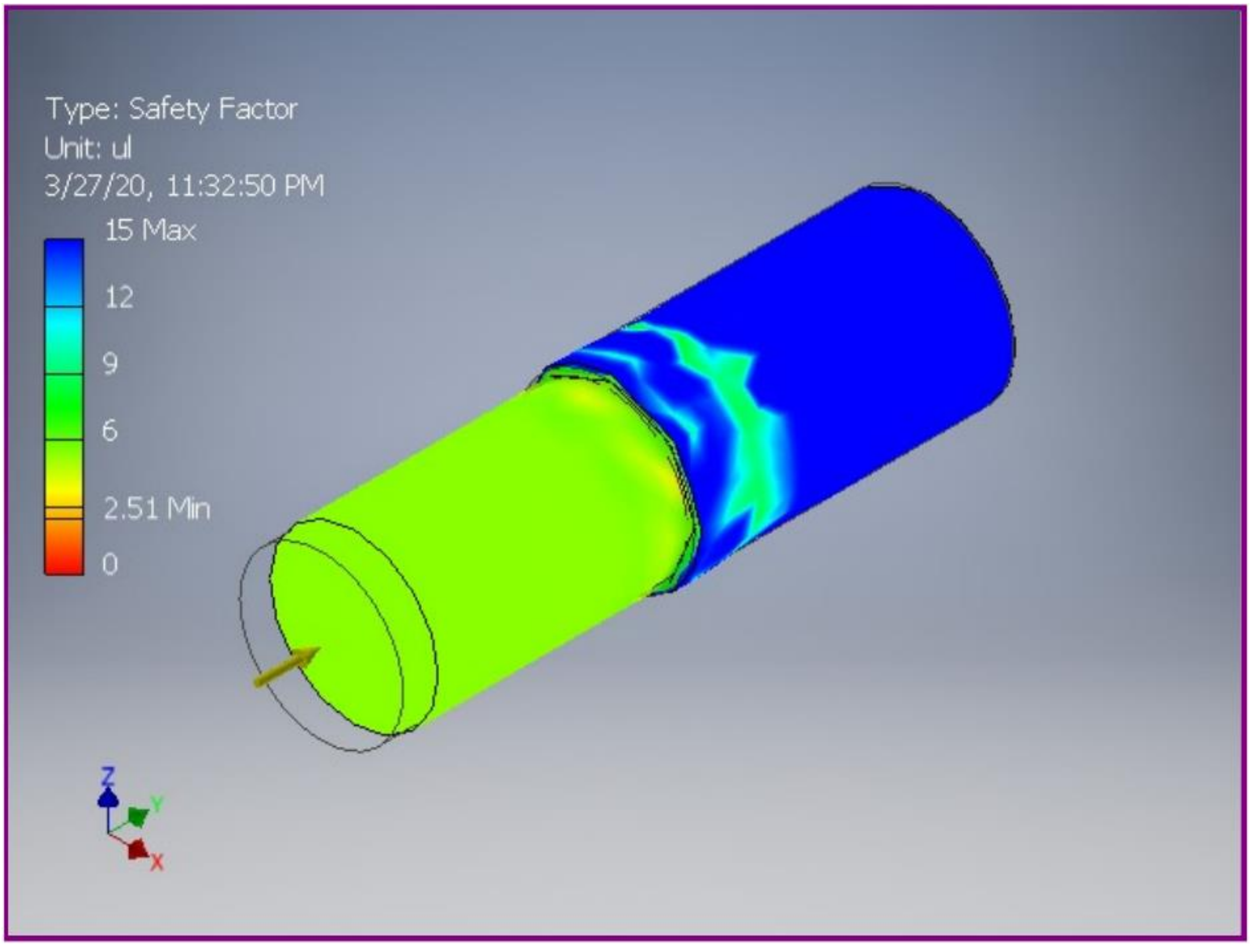


Figure 22: Finite element analysis on 1-inch stepped shaft

Simulations determined a minimum factor of safety of 2.51 at the base of the narrow shaft, which is also within safe operating conditions.

6.1.4 3D-Printed Prototype

The interfacing dimensions between the stepped shaft and the actuating shaft are critical to ensure proper component function. The bolts utilized to fasten the stepped shaft to the actuating shaft are not rated to support any of the impact load, and purely intended to be utilized as a means of attachment. In order to accuracy of this measurement, a test collar was 3D printed and dry fit to the actuating shaft. Several design iterations are indicated below in Figure 23.

Figure 23: 3D printed test collar generations

The first iteration had an internal diameter that was too narrow to be easily removed from the actuating shaft after assembly. As a result, the test collar had to be destroyed in order to be to be removed. Adjustments were made to the internal diameter of the stepped shaft, expanding the interfacing diameter by $2\ mm$. The second iteration was developed with an inferior version of a 3D printer on a particularly humid day, and as a result the layers of the print were prone to separation from one another. Clear packing tape was utilized to hold the layers together so that the test collar could be utilized to approximate any further clearance requirements. It was determined that approximately $1\ mm$ should be added back to the internal interfacing diameter. The third iteration fit accurately, and required minor adjustments of $0.1\ mm$ to the hole locations, shifting them closer to the end of the stepped shaft, and away from the portion that interfaces with the actuating shaft.

The dimensions of the final printed prototype, which were utilized in the final design of all stepped shafts, is indicated in Appendix E.

6.2 Support Bracket

The new design for the force plate base support bracket has removed the slot that was in the original one shown in Figure 24.

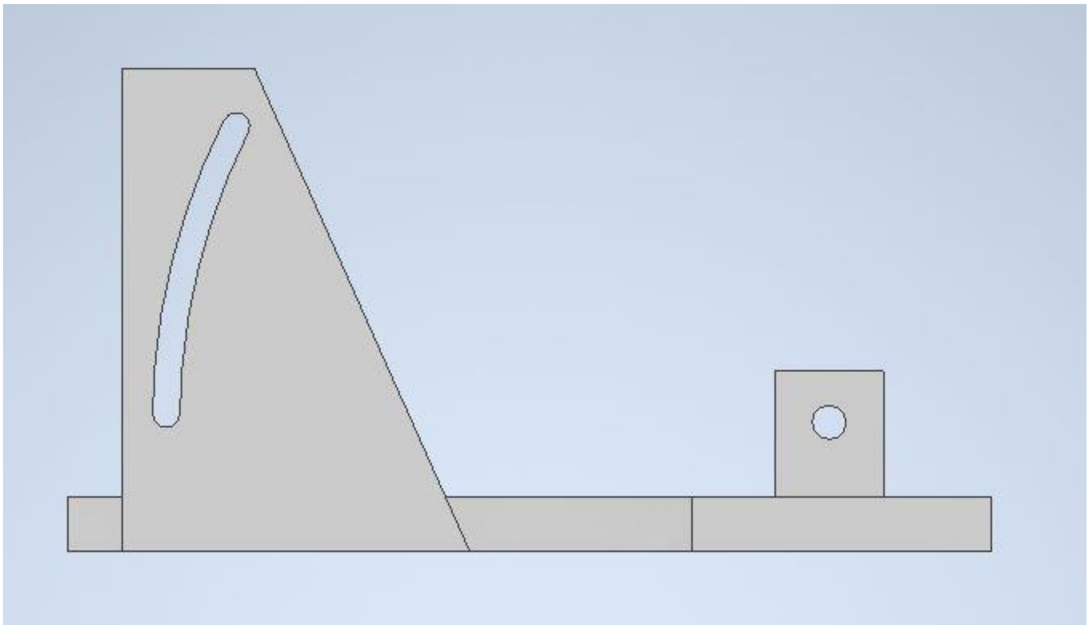


Figure 24: Original Support Bracket

The new and improved design features 3 locking positions: 85 degrees, 90 degrees, and 95 degrees, and is featured in Figure 25 below.

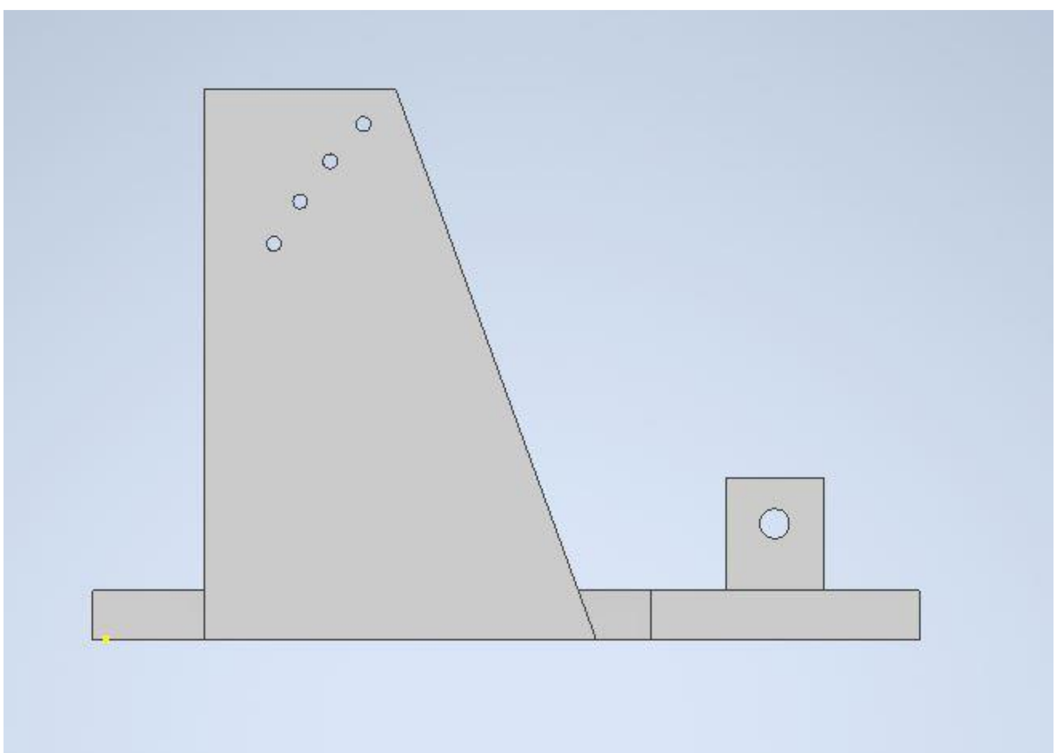


Figure 25: Final Design of Support Bracket

The design also removed the support bracket angle adjustment rod at the back of the bracket that was damaged and ineffective as shown in Figure 26.

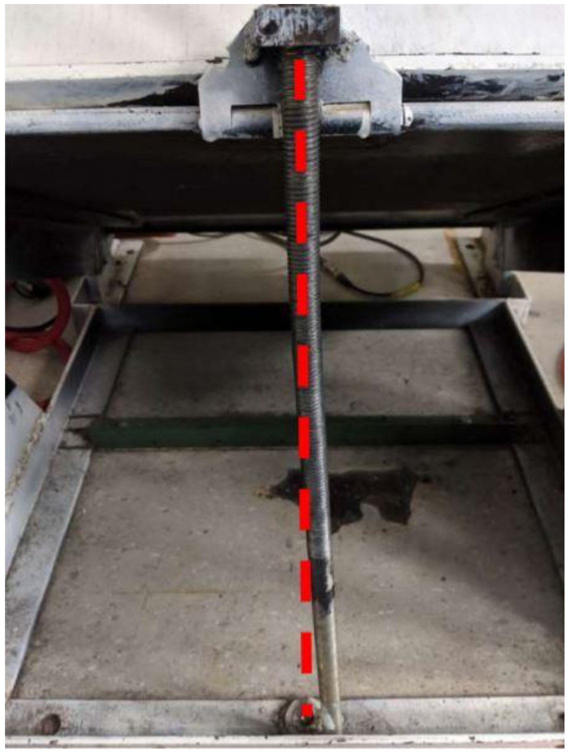


Figure 26: Support bracket angle adjustment rod

Instead of the angle adjustment rod, the new design will be completely supported by bolts. Calculations were performed in order to determine the appropriate bolt diameter in order to successfully meet the force requirements of the machine. Static as well as fatigue analysis was performed. The new support bracket needs to be welded on to the existing base, therefore weld calculations were performed as well to ensure the design is satisfactory. The fourth and lowest hole in the bracket is a safety mechanism that will have a permanent bolt, so that while adjusting the position, the force plate will be stopped from falling to the ground. The final design was only possible to attain after many iterations. The first iteration involved a completely new base for the bracket. This design would have been more expensive and much more difficult to manufacture. The next design only required the brackets to be replaced, but only had three holes, which is where the safety concern was brought to the group's attention. The final design solved all of the problems by adding the fourth hole, and adjusting the position of the support brackets with respect to the rest of the base.

6.2.1 Calculations

All equations utilized in calculations are from Shigley's Mechanical Engineering Design.[15]

Performing static analysis of the bolts:

x-z Plane:

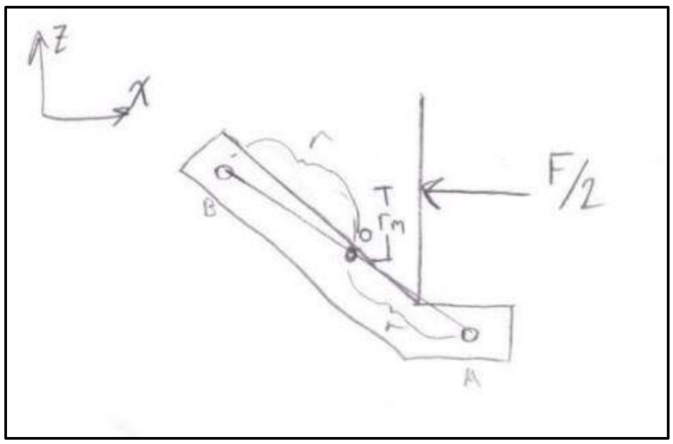


Figure 27: x-z plane for bolt analysis

Primary Shear Load per Bolt:

Given the Maximum Force of 45,000 N:

$$F'_A = F'_B = \frac{\frac{F}{2}}{2} = \frac{F}{4} = \frac{45000}{4} = 11.25\ kN$$

Secondary Shear Load per Bolt:

$$F"_A = F"_B = \frac{Mr}{2r^2} = \frac{M}{2r}$$

$$M = \frac{Fr_m}{2} = \frac{45000 * 0.1924812}{2} = 4330.827\ kN * m$$

$$F" = 7.249\ kN$$

Using Parallelograms and drawing forces to scale:

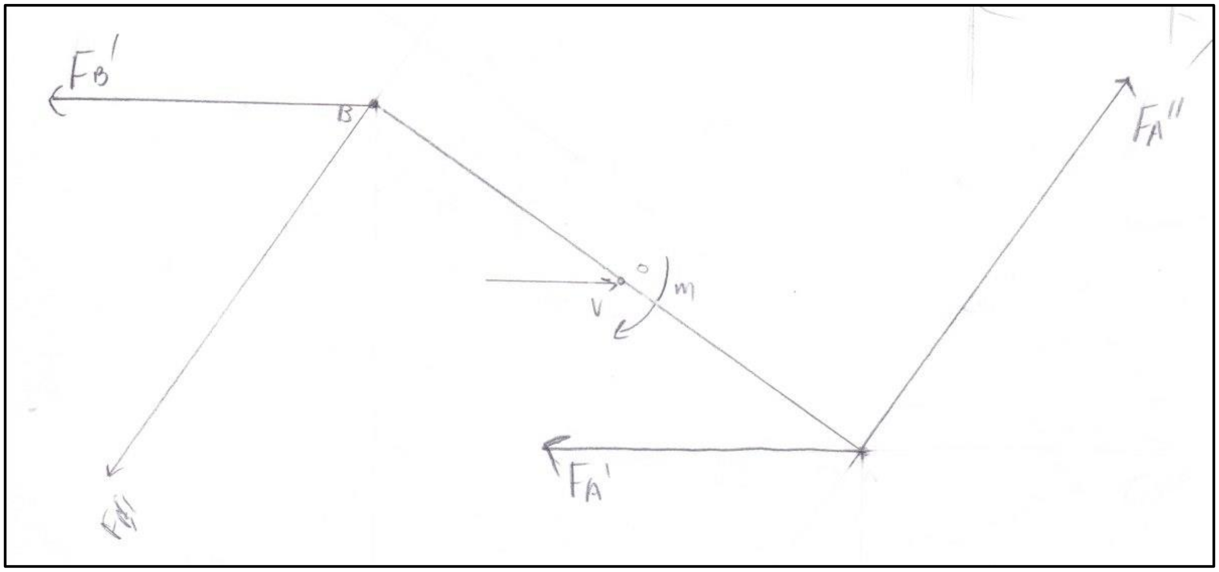


Figure 28: Bolt forces free-body diagram

Thus:

$$F_A = 9.475\ kN \text{ and } F_B = 16.385\ kN$$

$\therefore$ Bolt B is critical

x-y Plane:

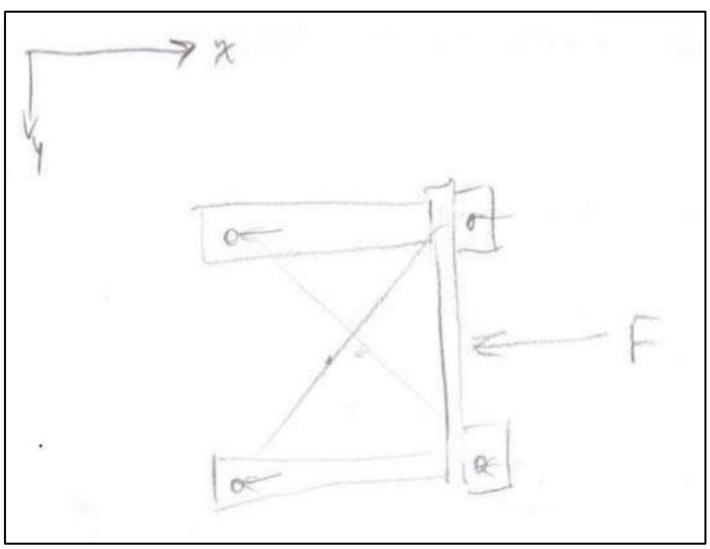


Figure 29: x-y plane for bolt analysis

$$M = 0, \qquad F" = 0, \qquad F' = \frac{F}{4}$$

$$F = F' = 11.25\ kN$$

$\therefore\ F_{xz} = 16.365\ kN$ and $F_{xy} = 11.25\ kN$y

Where 1 $kN = 0.224809\ lbf$

$\therefore\ F_{xz} = 3683.495\ lbf$ and $F_{xy} = 2529.101\ lbf$

Transverse Shear Stress:

$$A = \frac{\pi d\text{^}2}{4}$$

$$A = \frac{\pi(0.625)^2}{4} = 0.30679\ in^2$$

$$\tau_{xz} = \frac{4 * Fxz}{3 * A}$$

$$\tau_{xz} = \frac{4 * 3683.495}{3 * 0.30679} = 16008.755\ psi$$

$$\tau_{xy} = \frac{4 * Fxy}{3 * A}$$

$$\tau_{xy} = \frac{4 * 2529.101}{3 * 0.30679} = 10991.669\ psi$$

$$\tau = \sqrt{{\tau_{xz}}^2 + {\tau_{xy}}^2}$$

$$\tau = 19418.986\ psi$$

Ultimate Tensile Stress of Grade 5 Bolts[5]: $\sigma_{ut} = 120000\ psi$

Ultimate Shear Stress of bolts[13]: $\tau_{ut} = 0.6 * \sigma_{ut}$

$$\tau_{ult,shear} = 0.6 * 120000\ psi = 72000\ psi$$

$$Static\ Factor\ of\ \ Safety\ = \frac{\tau_{ult,shear}}{\tau} = \frac{72000}{19418.96}$$

$$Static\ Factor\ of\ \ Safety\ = 3.7$$

$\therefore$ Acceptable

Fatigue Analysis for Repeated Stress:

$$\tau_m = \tau_a = \frac{\tau}{2} = 9709.5\ psi$$

Endurance Limit:

$$Se = Se' * k_a * k_b * k_c * k_d * k_e * k_f$$

Estimated Endurance Limit:

$$Se' = \ 0.5 * S_{ut} = 0.5 * 120000$$

$$Se' = \ 60000\ psi$$

Modifying Factors:

Surface Modification Factor k_a

$$k_a = a * {S_{ut}}^b$$

Where $S_{ut} = 120\ kpsi$

From Table 6-2 (Shigley's 10th ed.):

$$a = 2.70, \qquad b = \ -0.265$$

$$k_a = 2.70 * 120^{-0.265} = 0.75924$$

Size Factor k_b:

$$d_e = 0.370d = 0.370 * 0.625$$

$$d_e = 0.23125''$$

$$k_b = 0.879{d_e}^{-0.107} = 1.0281$$

Loading Factor k_c:

$$k_c = 1$$

Temperature Factor k_d:

$$k_d = 1$$

Reliability Factor k_e (99% Reliability):

$$k_e = 0.814$$

Miscellaneous Effects Factor k_f:

$$k_f = 1$$

Modified Endurance Limit:

$$S_e = 60000 * 0.75924 * 1.0281 * 1 * 1 * 0.814 * 1$$

$$S_e = 38123.127\, psi$$

Assuming Stress concentration factor:

$$k_t = 1$$

Modified Goodman Criterion:

$$\frac{\tau_a}{S_e} + \frac{\tau_m}{S_{ut}} = \frac{1}{n}$$

$$\frac{9709.5}{38123.127} + \frac{9709.5}{120000} = \frac{1}{n}$$

Fatigue Factor of Safety:

$$n = 2.97$$

$$\therefore \text{Acceptable}$$

Performing weld calculations:

Primary Shear Stress:

$$\tau' = F/A$$

$$A = total\ throat\ area\ of\ weld$$

$$A = 1.414hd$$

$$h = 0.375$$

$$d = 16.1''$$

$$A = 1.414*0.375*16.1 = 8.537025$$

$$F = 45/2\ kN = 10.116/2\ kip$$

$$\tau' = \frac{10.116/2}{8.537025} = 0.5924\ kpsi$$

Secondary Shear Stress:

$$\tau" = Mr/I$$

$M = \frac{10.116}{2} * 18.7$ *18.7" is distance to middle of force plate

$$I = 0.707hIu$$

$$Iu = \frac{d^3}{6}$$

$$\therefore \tau" = \frac{\frac{10.116}{2} * 18.7 * \frac{16.1}{2}}{172.113} = 4.423\ kpsi$$

Shear Magnitude:

$$\tau = \sqrt{\tau'^2 + \tau"^2}$$

$$\tau = \sqrt{0.5924^2 + 4.423^2} = 4.4624\ kpsi$$

$$S_{sy} = 0.577 * S_y$$

$$n = \frac{S_{sy}}{\tau} = \frac{0.577 * 61.5}{4.4624} = 7.925$$

n = 7.9

∴ Acceptable

$$\tau_{rev} = \frac{\tau_a}{\left(1 - \frac{\tau_m}{Sut}\right)} = 10564\ psi$$

a = 253981

$$b = -0.31607$$

$$N = \left(\frac{\tau_{rev}}{a}\right)^{1/b}$$

$$N = 23394.65$$

Therefore, the bolts last 23394 cycles operating at the maximum force ever experienced by the force plate accelerometers.

6.2.2 Modeling

A MATLAB script named BoltShearStress.m (located in Appendix N) was written in order to determine the appropriate diameter for the bolt fasteners. The script performs the calculations shown in the previous section for a three different bolt sizes: ½”, 5/8”, and ¾”. The result from the code was used to determine that the 5/8” bolt is best suited for the design. The other modeling involved in the support bracket design was determining the location of the holes. The bracket was modeled using inventor, and the location of the holes was determined by measuring the x and y coordinates from where the force plate base connects to the support base. The bracket itself had to be moved closer to the force plate base in order to accommodate for the 90-degree lock position. All of the dimensions and positions of the components of the support bracket were located graphically with Autodesk Inventor. MATLAB and Inventor were both essential in the design calculations needed to ensure the design could withstand the forces exerted by the machine. Final drawings of the design can be found in the Appendix. The drawings in the appendix also show the position that the new plate needs to be welded at in order for the bolts to line up properly.

6.3 Adjustment Knuckle

The adjustment knuckle is the part of the repetitive impact testing apparatus that connects the impact rod to the pneumatic system. The knuckle uses two parts that are bolted together to create a press-fit that prevents the impact rod from sliding. The impact rod being held in place is a stainless-steel rod that is $30\ mm$ in diameter.

6.3.1 Knuckle Design

Several design iterations were developed until the desired factor of safety was achieved.

6.3.1.1 Old Knuckle Design

The old design seen in Figure 30 connects the cylindrical part of the knuckle to the base using a bolt that holds the cylinder in a slot.

Figure 30: Old knuckle design

The old design allowed the knuckle to be placed at different angles which allowed the object being tested to be impacted at different locations. This design is flawed because of the slots that allowed for the angle adjustment. The metal between the slots were too thin for the repetitive impacts that occurred during operation and some of them sheared off. This led to the knuckle not fitting together properly which caused excessive vibration in the rod.

6.3.1.2 First Design Iteration

The first Design iteration of the knuckle is seen in Figure 31.

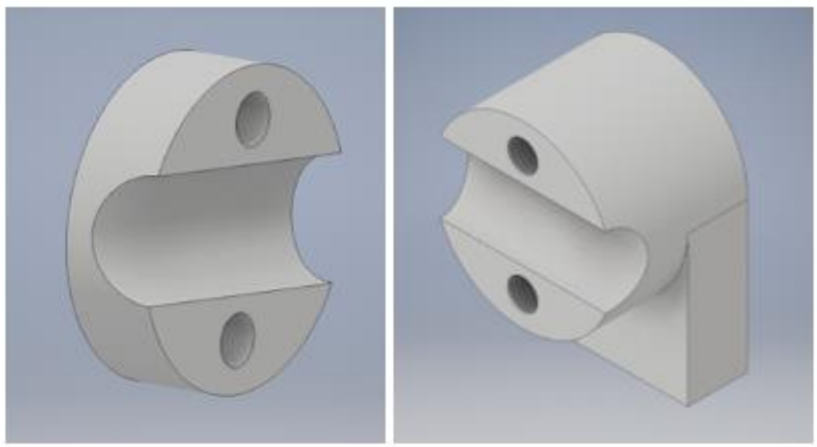


Figure 31: First knuckle design iteration

This Design involved removing the angle adjustment of the knuckle and make it a solid piece. This design fixes the vibration issue coming from the old broken design. The loss of adjustment in this design is made up for in the design of the new impact rod.

6.3.1.3 Final Design

The Final Design, indicated below in Figure 32, takes the first design iteration and adds some improvements.

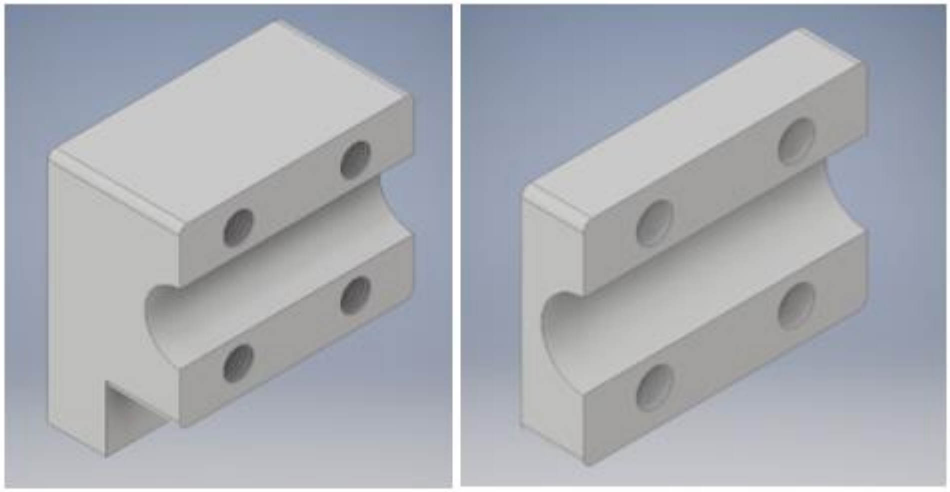


Figure 32: Final knuckle design

The first design being one piece was diffcult to machine so the final design is more squared off for easier machining. The length of the slot that holds the impact rod was extended by a ½ “ which increases the clamping force of the knuckle reducing the torque needed for the bolts to hold the impact rod in place. This extension allowe for two more bolts to be added decreases the force acting on each bolt and prevents bolt failure. The last change that was made in the final design was the offset from the base of the knuckle to the impact ro was reduced by a ½ “. This change was made to allow the part to be made from the same stock as the impact rod and also reduces the torque acting on the bolts holding the knuckle down. This design is made up of AISI 1018 Steel and uses 4-½” bolts to hold the rod in place.

6.3.2 Calculations

Assumptions:

- Press fit calculations assume an outer ring with an outer diameter of 7/8” which is the smallest diameter from the center of the shaft to the outside of the knuckle.
- The radial interference is set to be 0.00015”.

- Mechanical Properties of knuckle and shaft are constant.

Material Properties:

- AISI 1018 Steel (Knuckle)
  - modulus of elasticity $E = 205\ GPa$
  - Ultimate tensile Strength $S_{ut} = 440\ MPa$
  - Yield strength $S_y = 370\ MPa$
  - Poisson's ratio $v = 0.29$
- Stainless Steel 316 (Impact Rod):
  - modulus of elasticity $E = 139\ GPa$
  - Ultimate tensile Strength $S_{ut} = 579\ MPa$
  - Yield strength $S_y = 290\ MPa$
  - Poisson's ratio $v = 0.25$

6.3.2.1 Press Fit Calculations

The contact pressure from the press fit shaft can be found using:

$$p = \frac{\delta}{D\left[\frac{1}{E_o}\left(\frac{d_o^2 + D^2}{d_o^2 - D^2} + v_o\right) + \frac{1}{E_i}\left(\frac{D^2 + d_i^2}{D^2 - d_i^2} - v_i\right)\right]}$$

Using this contact pressure, the force required to hold the impact in place can be found using:

$$F = 2 * R * L * p$$

The holding force of the knuckle, which is the force that the knuckle will prevent from sliding is found from:

$$F_{Hold} = f * \pi * 2 * R * L * p$$

The coefficient of friction used is $f = 0.74$, based on mild steel to mild steel contact.[8]

6.3.2.2 Outer Cover Stress

The tangential and radial stresses are found using the following equations:

$$(\sigma_t)_{max} = p\frac{r_o^2 + R^2}{r_o^2 - R^2}$$

$$(\sigma_r)_{imp} = -p$$

The shear stress developed at the interference surface is also needed and found by using:

$$\tau = \frac{imp}{2\pi RL}$$

Using these stresses, the maximum shear stress was found through stress transformation of a 2D state of stress:

$$\tau_{max} = \left([\sigma_t - \frac{\sigma_t + (\sigma_r)_{imp}}{2}]^2 + \tau^2\right)^{0.5}$$

Next, the static factor of safety is calculated using maximum shear stress theory:

$$n_{static} = \frac{0.5 * S_y}{\tau_{max}}$$

6.3.2.3 Fatigue Stress Analysis

The fatigue stress analysis uses ASME elliptic criteria for fatigue failure. The mean and alternating stresses were found using the equations

$$\sigma_a' = \left((\sigma_r)_a{}^2 - [(\sigma_r)_a * (\sigma_t)_a] + (\sigma_t)_a{}^2 + 3\tau_a^2 \right)^{0.5}$$

$$\sigma_m' = \left((\sigma_r)_m{}^2 - [(\sigma_r)_m * (\sigma_t)_m] + (\sigma_t)_m{}^2 + 3\tau_m^2 \right)^{0.5}$$

Theses equations used the mean and alternating stresses for the tangential, radial and shear stresses for the knuckle. The endurance strength then had to be found using the ultimate strength of the knuckle:

$$S_e' = 0.5 * S_{ut}$$

This endurance strength is multiplied by the following modification factors to find the modified endurance strength:

$$S_e = k_a * k_b * k_c * k_e * S_e'$$

These modification factors are:

$$k_a = 4.51 * (S_{ut} * 10^{-6})^{-0.265} \text{ (Surface Factor)}$$

$$k_b = 1.24 * (2 * R * 1000)^{-0.107} \text{ (Size Factor)}$$

$$k_c = 1 \text{ (Loading Factor)}$$

$$k_d = 1 \text{ (Temperature Factor)}$$

$$k_e = 0.897 \text{ (Reliability Factor)}$$

The fatigue factor of safety for infinite life was then found using:

$$n = \left(\frac{1}{\left[k_f * \frac{\sigma'_a}{S_e}\right]^2 + \left[k_f * \frac{\sigma'_m}{S_y}\right]^2} \right)^{0.5}$$

Where a stress concentration of 1 is utilized.

6.3.2.4 Bolting Calculations

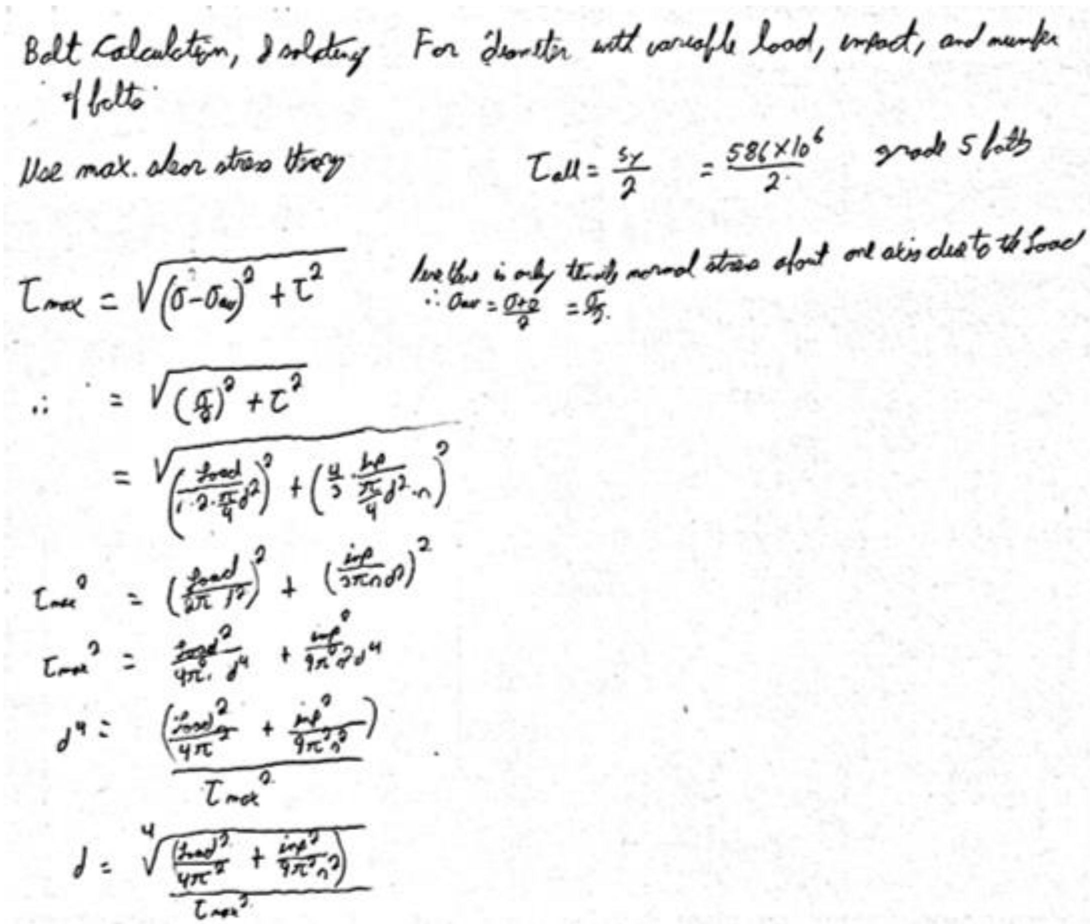
The impact rod is held in the knuckle using 4 bolts, so the load carried by each bolt is

$$Load = F/4$$

Due to the combined shear and axial loaded nature of the bolts, the diameter required to support the load is found using

$$d_{required} = \left[\frac{\left(\frac{Load^2}{4\pi^2} + \frac{imp^2}{9\pi^2 * 16} \right)}{\frac{586}{2} * 10^6} \right]^{\frac{1}{4}}$$

Which was derived from the steps found in Figure 33 below:



Bolt Calculation, Isolating For diameter with variable load, impact, and number of bolts

Use max. shear stress theory $\quad \tau_{all} = \frac{S_y}{2} = \frac{586 \times 10^6}{2}$ grade 5 bolts

$$\tau_{max} = \sqrt{(\sigma - \sigma_{av})^2 + \tau^2}$$

here there is only the normal stress about one axis due to the load $\therefore \sigma_{av} = \frac{\sigma + 0}{2} = \frac{\sigma}{2}$

$$\therefore \quad = \sqrt{\left(\frac{\sigma}{2}\right)^2 + \tau^2}$$

$$= \sqrt{\left(\frac{Load}{1 \cdot 2 \cdot \frac{\pi}{4} d^2}\right)^2 + \left(\frac{4}{3} \frac{imp}{\frac{\pi}{4} d^2 \cdot n}\right)^2}$$

$$\tau_{max}^2 = \left(\frac{Load}{2\pi d^2}\right)^2 + \left(\frac{imp}{3\pi n d^2}\right)^2$$

$$\tau_{max}^2 = \frac{Load^2}{4\pi^2 d^4} + \frac{imp^2}{9\pi^2 n^2 d^4}$$

$$d^4 = \frac{\left(\frac{Load^2}{4\pi^2} + \frac{imp^2}{9\pi^2 n^2}\right)}{\tau_{max}^2}$$

$$d = \sqrt[4]{\frac{\left(\frac{Load^2}{4\pi^2} + \frac{imp^2}{9\pi^2 n^2}\right)}{\tau_{max}^2}}$$

Figure 33: Bolt required diameter derivation and explanation

From this equation ½” bolts were chosen for the design. The torque required from these bolts are the calculated using a friction factor of 0.2 for unlubricated grade 5 bolts. This is done using

$$B_{torq} = fric * d_{bolt} * \frac{25.4}{1000} * Load$$

6.3.3 Modeling

The analysis of the knuckle shown by the equations in the previous section was done using a MATLAB code. The code gave the following values:

Required Bolt torque is: 15.983 Ft-Lb

Static Factor of Safety n= 7.674

ASME Elliptic Fatigue Factor of Safety is $n = 8.444$

Thus, locking Force is 1.763 times the required amount.

6.4 Impact Rod

Once the PCC was completed, it had been decided that we would move further with the preliminary design that Dale had came up with as the basis. This preliminary design had constituted of spherical ball joint style end that would be encased in an outer cover which would then connect to an outgoing shaft. In the conceptualization stage, this design held significant promise. It would have been able to allow for impact testing to take place at multiple locations on the object being tested, therefore meeting an important design objective. While this design had promise and positive aspects to it, it also had negative aspects that simply could not be ignored.

Ultimately, a cylindrical end was chosen over the spherical end for multiple reasons. Relating back to the PCC, the cylindrical end was chosen over the spherical end for the following reasons. As seen in the PCC, the durability criterion was of the second highest importance, with a score of 14.5, for this reason the cylindrical end excelled. While maintaining the same position locking ability, the cylindrical end had a larger area with which it could disperse the impact force through a bearing stress, than that of its spherical counterpart. The third most important criterion was the reliability in its capability to perform as required, with a score of 12. The cylindrical end was once again superior here due to its friction fit only having to resist the torque about a single axis to maintain its position. The spherical end's friction fit would have needed to resist the torque about all three axes, causing the possibility of a larger moment vector to form, thereby decreasing the reliability of its friction fit.

The fourth most important factor of the PCC was a tie between simplicity and cost, both of which had a score of 5. The cylindrical end was a better option here as well, as the manufacturing will be significantly less complicated due to the reduction of manufacturing challenges. The least important part of our PCC was ease of use, which had a score 3.5. This is the one area where the cylindrical end is not as good as the spherical end, since adjustments can only be made about one axis without having to adjust the knuckle as well. The spherical end would have been able to have adjustments about all three axes without having to adjust the knuckle. For these reasons, in relation to the PCC, the cylindrical end was ultimately chosen to be our final design for the impact rod.

Moving forwards with the cylindrical end, there were a few different conceptualized ideas, shown in Appendix I and Appendix J until arriving at the final design shown in Appendix K. As can be seen by the first two revisions of the cylinder cover, dimensions were not included, and a slot for angle adjustment also was not included. These drawings were mainly for a physical representation, and to allow for a better understanding. Both revisions had shortcomings that caused them not to be utilized for the final design. As can clearly be seen in the first revision, there is no place for a shaft leaving the cylinder to go, so this was quickly moved away from. Revision 2 held significant promise initially. Ultimately, its shortcomings proved to be too great to utilize. These shortcomings included its lack of symmetry in the bolt's locations, increased manufacturing difficulty due to the covers rounded features and completely enclosed cylinder. Due to these shortcomings, the boxed cover of the final design in Appendix K was utilized. This design reduced the complexity required for manufacturing, and provided more symmetrical for distribution, helping to reduce stresses, and increase the holding ability of the cover. The cylindrical shaft end, and the outgoing shaft that work with this final design can also be found in Appendix K.

6.4.1 Calculations

The impact rod cover and the incoming cylindrical shaft, which can be found in Appendix K, were modeled as a cylindrical press fit. While there will be slight variation from this, it was seen to be an appropriate approximation. The cover, and cylindrical shaft were both designed with the use of AISI 1018, causing them to have the same properties. Thus, interference pressure became:

$$p = \frac{E\delta}{2R}\left[1 - \frac{R^2}{r_o^2}\right]$$

After having found the interference pressure, the torque holding ability of the cylindrical cover was able to be found from the following:

$$T = \frac{fp\pi(2R)^2 * (L - \frac{d_{inshaft}}{4})}{2}$$

Where $d_{inshaft}$ is the diameter of the incoming shaft, it is divided by 4 as there is a 90° slot in the cover to allow for angle adjustment.

Maximum tangential and radial stresses become the following:

$$(\sigma_t)_{max} = \frac{E\delta}{2R}\left[1 + \frac{R^2}{r_o^2}\right]$$

$$(\sigma_r)_{max} = -\frac{E\delta}{2R}\left[1 - \frac{R^2}{r_o^2}\right] = -p$$

Given that the cover experienced impact, bearing stress was added to the radial stress as well, becoming:

$$(\sigma_r)_{imp} = -\frac{E\delta}{2R}\left[1 - \frac{R^2}{r_o^2}\right] - \frac{imp}{2RL}$$

Where imp is equal to the impact force, and L is equal to the length of the cylinder.

Torsional shear stress is also present within the cylinder at the interference radius; thus the stress can be represented as:

$$\tau = \frac{TR}{J} = \frac{T_{req} * R}{\frac{\pi}{2} * R^4}$$

Where T_{req} is equal to the torque required to hold the cylinder in place. Next, the maximum shear stress was found through stress transformation of a 2D state of stress:

$$\tau_{max} = \left(\left[\sigma_t - \frac{\sigma_t + (\sigma_r)_{imp}}{2}\right]^2 + \tau^2\right)^{0.5}$$

To be conservative, the maximum shear stress theory was used to determine the static factor of safety, with S_y being the yield strength of the material thus:

$$n_{static} = \frac{0.5 * S_y}{\tau_{max}}$$

Once static analysis proved to be acceptable, fatigue analysis was required due to repetitive impacting nature of the machine. ASME elliptic criteria was used for fatigue failure, as it governs by yielding which still makes it very safe, but also does not go to the conservative extremes of the Soderberg criteria. Mean and alternating stresses were obtained for the tangential, radial, and torsional shear stresses of the cover, which are denoted by an 'm' and an 'a' respectively in the following:

$$\sigma_a' = \left((\sigma_r)_a{}^2 - [(\sigma_r)_a * (\sigma_t)_a] + (\sigma_t)_a{}^2 + 3\tau_a^2\right)^{0.5}$$

$$\sigma_m' = \left((\sigma_r)_m{}^2 - [(\sigma_r)_m * (\sigma_t)_m] + (\sigma_t)_m{}^2 + 3\tau_m^2\right)^{0.5}$$

Then, the endurance strength had to be found, and given that AISI 1018 steel has an ultimate strength less than $1400\ MPa$, this became:

$$S_e' = 0.5 * S_{ut}$$

After obtaining the endurance strength the modified endurance strength had to be obtained from the following:

$$S_e = k_a * k_b * k_c * k_e * S_e'$$

$$k_a = 4.51 * (S_{ut} * 10^{-6})^{-0.265} \text{ (Surface Factor)}$$

$$k_b = 1.24 * (2 * R * 1000)^{-0.107} \text{ (Size Factor)}$$

$$k_c = 1 \text{ (Loading Factor)}$$

$$k_d = 1 \text{ (Temperature Factor)}$$

$$k_e = 0.897 \text{ (Reliability Factor)}$$

$$k_f = 1.2 \text{ (Stress Concentration Factor)}$$

Finally, fatigue factor of safety for infinite life was able to be obtained from the following:

$$n = \left(\frac{1}{\left[k_f * \frac{\sigma_a'}{S_e}\right]^2 + \left[k_f * \frac{\sigma_m'}{S_y}\right]^2} \right)^{0.5}$$

Lastly, the force required by the bolts to hold the cylinder together had to be taken into consideration to ensure that the proper bolts were selected. Recalling the pressure from of the interference fit, the force required to hold cover in place becomes:

$$F = p * A = 2 * R * L * p$$

Determining the load required by each bolt based off of 4 bolts is given as:

$$Load = F/4$$

Given a yield strength of 586 MPa for grade 5 bolts, the diameter required to ensure the bolts would not yield was given by:

$$d_{req} = \left(Load * \frac{4}{\pi * 586 * 10^6} \right)^{0.5}$$

Based off of the values obtained, 7/16" UNC bolts were chosen to be used. When grade 5 bolts are used to determine the required bolt torque as shown in the formula shown below, their unlubricated 'fric' value is given as 0.2. Consider:

$$B_{torq} = fric * \frac{7}{16} * \frac{25.4}{1000} * Load$$

To convert this value to pound feet from the newton meters above, the value is multiplied by 0.73756. The above listed equations were all used in an iterative MATLAB code that was ran on a while loop from a bolt torque to 0 to 50 ft-lbs. The recommended torque specification of the bolts is 50 ft-lbs, at which point gives the following values:

Required Bolt torque is: $50.000\ ft - lb$

Static Factor of Safety $n = 1.778653$

ASME Elliptic Fatigue Factor of Safety is $n = 1.275443$

Locking Torque is 1.705375 times the required amount

Given the iterative nature of the code that was used, graphs shown in Figure 34 below were able to be made, relating bolt torque to the fatigue factor of safety and torque holding ability. The full MATLAB code used for this can be found in Appendix P.

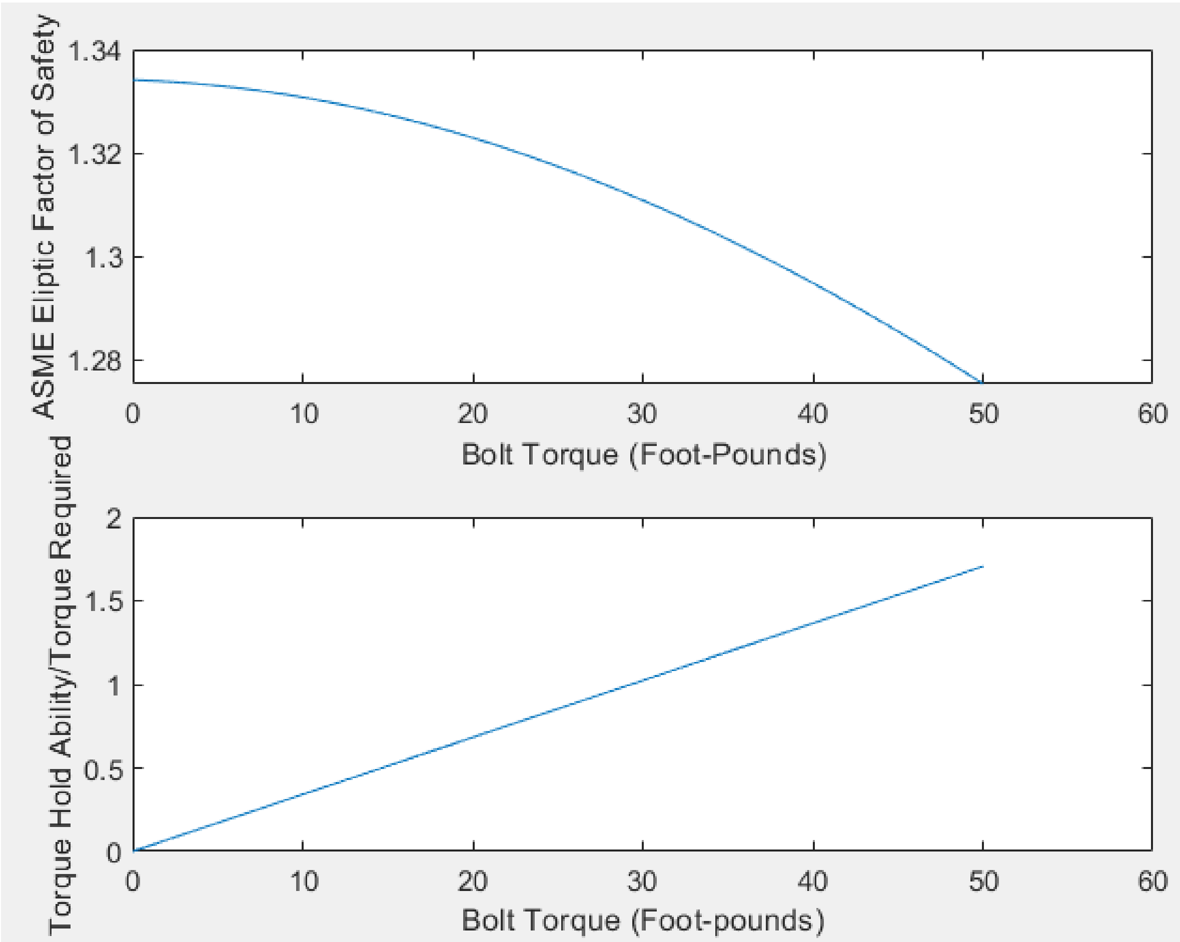


Figure 34: Bolt Torque Vs. Fatigue Factor of Safety and Torque Holding Ability

6.4.2 Modeling

At the mid semester design presentation, a clear and concise method of displaying the design ideas to Dr. Zerpa was extremely important. While drawings and 3D CAD models greatly assist with this, having a real object to manipulate allows for a significantly improved experience. For this presentation, a 1 : 2 scale model of the cylindrical cover and cylindrical shaft end were 3D printed. The 3D print can be seen below in Figure 35.

Figure 35: 3D printed cylindrical cover and cylindrical shaft end

7.0 Operation, Maintenance, Service and Disposal

In this section, each major component of the design is discussed in regard to operation, maintenance, service, and its disposal where applicable. Each component has significant changes in the operation, and each part needs to be used together in order for the machine to operate. Therefore, the user of the machine must know clearly how to use each part of the machine individually in order to get most functionality out of the design. Any short term or long-term maintenance requirements will also be discussed. Each part is designed to last, so the disposal of the parts will not be discussed.

7.1 Cane Tip Stepped Shaft

A stepped shaft was designed for each variety of cane tip utilized in experimental procedures with a different internal diameter. Thus, cane tips of type A would utilize the 3/4-inch stepped shaft, and cane tips of type B would utilize the 7/8-inch stepped shaft. The 1-inch stepped shaft is reserved for future experimental results, since the next most common internal cane diameter is approximately 1-inch.

In order to use a cane tip stepped shaft in experimental procedures, it is attached to the actuating cylinder using the same methodology as existing test apparatuses. The rubber cane tip is then secured onto the narrow end of the rubber cane tip by pressing in onto the stepped shaft.

7.2 Support Bracket

The support bracket is operated differently than the old design. The old design utilized a long, threaded bar to adjust the angle of the force plate in unnecessarily small increments; and it did not allow the force plate to sit vertically. In order to change the position of the new support bracket, the user must remove the bolts from the current position, move the force plate manually to the desired position, and reattach the bolts. The machine is most often used in one position for extended periods of time, so changing the position is a fairly rare event. Due to the rarity of actually changing the angle of the force plate, as well as the added degrees of freedom for maximum range of motion implemented by the other aspects of the design, this more rigid method was chosen. As far as the bolts go, they are designed for infinite life with the maximum force considered, so there should be little to no maintenance involved with them. In the worst-case scenario, the bolts would simply need to be replaced should they experience signs of damage. Little to no maintenance is required for this aspect of the design.

7.3 Adjustment Knuckle

The adjustment knuckle is a fairly basic part when it comes to its uses. It is the part that connects the impact rod to the pneumatic system of the impact machine. The old design allowed the impact rod to be placed at different angles which assisted with testing different parts of an object. The new design is two pieces, the base and the part that bolts down on the impact rod. The rod being press fit in the knuckle and not fixed allows the rod to be placed at different lengths if needed.

7.4 Impact Rod

The impact rod was designed to significantly improve the functionality of the RITA as a whole. The impact rod is to be set to the user's desired angle, at which point the 4 bolts found on the outer cover are to be tightened to ensure the rod stays at its desired location. It is recommended that the bolts be tightened to 50 ft-lbs. Ideally this should be done in roughly 10 ft-lb increments, in a "criss-cross" pattern to ensure that the press fit is as evenly distributed as possible. If the user has a fair amount of experience working hands on "twisting wrenches" they should be able to torque the bolts by their own judgement to near $50\ ft\text{-}lbs$ without requiring a torque wrench. As previously indicated in Figure 34 above, there is a substantial amount of room for error within the $50\ ft\text{-}lbs$ recommended torque that will still be able to lock the impact rod in place while still maintaining an infinite life through fatigue analysis. Essentially what this is saying is that the $50\ ft\text{-}lbs$ is a good judgement recommendation, there is a lot of wiggle room.

In terms of maintenance in an to increase thread life on the bolts the threads should have a small amount of grease occasionally applied to them. Also, if after substantial use the impact rod is not longer able to hold itself in place, the cylinder shaft end that the outer cover mates with could be knurled in an effort to increase the service life of the materials. Given the infinite life that the impact rod was designed for this should not be an issue. However, the possibility exists with all designs that there may have been an oversight, or something not considered. This maintenance aspect should not need to be used but is there if needed.

8.0 Costs and Bill of Materials

Shown below in Table 7 is a finalized bill of materials for the materials required to make the design components of the RITA. The pricing of the fasteners is an estimate based on previous experiences with Intercity Industrial, true pricing of these item is subject to vary slightly. The support plates, 1018 round bar sections, cylinder cover raw material, and the cylinder have all already been order. These were ordered as fabrication and testing was expected to happen before the COVID-19 pandemic made this no longer possible. The required materials with the exception of the fasteners, foam padding, and the miscellaneous items have already been ordered. As can be seen, there has been no cost associated with labour, as all of the manufacturing work is expected to be done at no cost in-house.

Table 7: Bill of materials

Component	Description	Base Qty.	Cost per Unit	Total
Support Plates	24"x24"x0.5" steel plate	2	$154.75	$309.50
Fasteners	Bolts, Washers	2	$3.50	$7.00
1018 Round Bar	Mild Steel Round Bar 1.25" dia. 60" Length	1	$88.00	$88.00
1018 Round Bar	Mild Steel Round Bar 1.5" dia. 60" Length	1	$128.00	$128.00
Cyl. Cover Raw Material	2" x 4" x 3.25" Plate	4	$59.00	$236.00
Cylinder	2.5" Dia. 12" Length AISI 1018	1	$70.00	$70.00
Fasteners	Machine bolts (M5-0.8 x 5 mm)	2	$0.80	$1.60
Fasteners	Washers (M5)	2	$0.25	$0.50
Foam Padding	10" x 60" x 1/2" Rubber for Padding	2	$30.00	$60.00
Fasteners	1/2" UNC To Attach Knuckle Together	4	$2.50	$10.00
Fasteners	To Attach Impact Rod Halves (Socket Head 7/16 UNC)	4	$5.00	$20.00
Miscellaneous	Drill Bits, End Mill, Welding Wire/Rod ETC..	1	$150.00	$150.00
			TOTAL COST:	**$1,080.60**

9.0 Validation and Compliance

All of the major components were designed with having an acceptable factor of safety in mind. What defines an acceptable factor of safety? That depends on the importance of the part and how crucial it is that the part doesn't fail. For example, the cane tip stepped shaft was designed with a minimum factor of safety of 1, as the part is easily replaceable by fabricating a new one. The other three major design components were all analysed under fatigue loading. The criteria for these designs was infinite fatigue life. Therefore, all designs complied with the criteria deemed fit for each part of the design. The support bracket design ended up with a life of 23394 cycles operating at the maximum force, which means it will actually last a lot longer. Even when the bolts do fail, they can be easily replaced. Any and all assumptions made are stated in the calculations or beforehand and are accounted for in the calculations. Some guidelines were used such Government of Canada's recommended force output used to determine the appropriate force to attach the rubber cane tips to the cane tip stepped shaft. Thus, all designs cooperated with all of the codes and standards and comply with the minimum required factors of safety.

10.0 Conclusions and Recommendations

The design objectives and constraints discussed above are satisfied through the major design adjustments previously indicated. Moving forward with the next iteration of the design, several operations are required to complete component manufacturing. While these manufacturing steps are staking place, several additional minor adjustments can be implemented into the testing apparatus.

10.1 Order of Operations

Due to the coronavirus (COVID-19) pandemic, all lab facilities have been closed until further notice. As a result, several component manufacturing steps had to be tabled for completion at a later date. This section indicates a sequential list of steps that are required to complete the fabrication of major design components, to be performed at a later date. Should lab facilities reopen in the next few months, all group members are able to assist in the completion of outstanding machine components.

10.1.1 Cane Tip Stepped Shaft

Lengths of AISI 1018 CD steel have been ordered from New West Metals, and are currently stored on-campus in the Centennial Building labs. In order to complete fabrication of the stepped shafts a few steps must be performed using the lab facilities on-site at Lakehead University. Because of the similarly between each of the three stepped shafts, several steps can be repeated for all shaft variations. The steps required for component completion are below:

1. Using a bandsaw, cut three sections of AISI 1018 CD steel to a minimum of $114.3\ mm$ (4.5-inch) lengths.
2. Using a lathe with a three-jaw chuck, centre the tooling for facing of AISI 1018 CD steel.
3. Perform facing on both ends of all 3 AISI 1018 CD steel segment until the overall length is reduced to $107.8\ mm$ ($\pm 0.010\ mm$).
4. Repeat Step 3 for the other AISI 1018 CD steel segments.
5. For the $3/4 -$ inch stepped shaft, perform a $50.8\ mm$ $(\pm 0.010\ mm)$ long continuous feed cut until a critical diameter of $20.52\ mm$ ($\pm 0.010\ mm$) is achieved.
6. For the $7/8 -$ inch stepped shaft, perform a $50.8\ mm$ ($\pm 0.01\ mm$) long continuous feed cut until a critical diameter of $24.46\ mm$ ($\pm 0.010\ mm$) is achieved.
7. For the $1 -$ inch stepped shaft, perform a $50.8\ mm$ $(\pm 0.010\ mm)$ long continuous feed cut until a critical diameter of $29.18\ mm$ ($\pm 0.010\ mm$) is achieved.
8. Using a lathe with a three-jaw chuck, centre the tooling for face drilling operations.

9. Progressively face drill the wide end of a stepped shaft to a depth of $46\ mm\ (\pm 0.010\ mm)$, until an internal diameter of $25\ mm$ $(\pm 0.010\ mm)$ is achieved.
10. Repeat Step 9 for the other stepped shafts.
11. Use a boring bar to achieve a square cut depth of $47\ mm\ (\pm 0.010\ mm)$, and an internal diameter of $24\ mm\ (\pm 0.010\ mm)$.
12. Repeat Step 11 for the other stepped shafts.
13. Using a milling machine with a four-jaw chuck, bore a $5.5\ in$ diameter hole $12.61\ mm\ (\pm 0.010\ mm)$ from the annular end of the stepped shaft.
14. Repeat Step 13 at a distance of $38.01\ mm\ (\pm 0.010\ mm)$ from the annular end of the stepped shaft.
15. Repeat Steps 13 and Step 14 for the other stepped shafts.
16. Viewing a stepped shaft axially from the annular end, consider the face of a clock, and allow the holes which have just been bored to be considered the $12 -$o’clock position.
17. Using a milling machine with a four-jaw chuck, bore a $5.5\ in$ diameter hole $24.46\ mm\ (\pm 0.010\ mm)$ from the annular end of the stepped shaft at the $3 -$o’clock position.
18. Repeat Step 17 for the other stepped shafts.

10.1.2 Support Bracket

The two support brackets have been ordered from Coastal Steel, and are currently stored on-campus in the Centennial Building labs. They are made of 44W mild steel, and the required holes have already been located and cut into each support bracket. The only remaining task is to attach each bracket to the existing force plate base. The remaining steps are outlined below:

1. Grind off existing brackets with an angle grinder.
2. Weld new support brackets in place of old ones, in the proper position.

10.1.3 Adjustment Knuckle

All dimensions are given in Appendix H.

1. Use a bandsaw to cut two 3.5” sections from the 2”x4” 1018 mild steel bar.
2. Take one of these sections and bore the hole that will hold the impact rod.
3. Using a mill, take a 1” deep and 1” wide cut off one of the 3.5” sides.
4. Add 0.125 radius fillets to the required edges.
5. Drill the holes for bolting the knuckles with 1/2”-13 threads.
6. Drill the holes that for bolting down the knuckle with 3/8”-16 threads.
7. Take the other section of the 2”x4” stock and mill it down to a 7/8”x3”x3.5” piece.
8. Bore the hole that will hold the impact rod.
9. Add 0.125 radius fillets to the required edges.
10. Drill the 1/2” holes in the required places.
11. On the opposite side as the impact rod hole, drill 3/4” holes 1/2” deep where each of the 1/2” holes are.

10.1.4 Impact Rod

In order to successfully fabricate the impact rod, certain steps need to be taken. The instructions for each part will be listed in point form in the following.

10.1.4.1 Outer Cylinder Cover

1. To start off the 7/16” tight clearance holes will have to be drilled in one of the 4”x 2” x 3.25” sections of steel. The other half will need to have threads made into it as per the drawing, and the recessed holes will also be made.
2. The two sections of steel will then need to be tightly bolted together to ensure there is a firm locking fit.

3. In the center of the 4” x 4” face, a 2.5” diameter hole will need to be drilled through the length of the material.
4. Once this has been completed the bottom face will need to have a 1.25” diameter hole drilled through the center of it, until it reaches the 2.5” center bore.
5. The 1.25” bore will need to then be widened out to obtain the 45° of rotation either way.
6. Then the top face of the cover will have a 1.25” diameter hole drilled ½” deep into its center.
7. The cover will then be unbolted.
8. With an end mill the 1.5” diameter section on the upper part of each half of the cover can be made.
9. The end mill can then be used to create the slot for the key-way in each half of the outer cover.

10.1.4.2 Outgoing Shaft

1. For the outgoing shaft, a 5” long piece of 1.5” diameter AISI 1018 steel is required. The first step is to turn 4.75” of the length down to a 1.25” diameter.
2. The next step is to turn the top 1.85” of the bar down to 0.945”.
3. The next steps is to use an end mill to create the ¼” x ¼” slot for the key-way. This will need to be made to extend ¼” from the outer edge of the 1.5” diameter section.
4. On the same edge as the key-way two ¼” UNC threads extending ¼” deep into the shaft are to be placed 0.482” and 1.222” from their centers’ to the top of the 1.25” diameter section.
5. The shaft is then to be rotated 90° at which point another ¼” UNC thread extending ¼” deep into the material is to be placed 1.007” from its center to the top of the 1.25” diameter section.

10.1.4.3 Cylinder Shaft End

1. A 3.25” long section of 2.5” diameter AISI 1018 steel is required. In the center of the non-flat surface a 1.25” diameter hole will need to be drilled 1.25” deep into the material. This can be done with a drill press if the cylindrical section is well secured.

2. Once this has been completed a 4.75” long section of 1.25” diameter shaft will be required. On one end of the shaft a 1.85” deep 0.945” diameter hole will need to be made with a drill press.
3. Once this has been made two 0.217” holes extending to the 0.945” center bore need to be drilled into the side of the 1.25” diameter section of shaft at a distance of 0.4975” and 1.2805” from the center of the holes to the bottom of the shaft where the 0.945” hole has just been made.
4. The 1.25” diameter shaft will then need to be rotated 90° and another 0.217” diameter hole extending to the existing 0.945” center bore will need to be made at a distance of 1.0415” from the center of the hole to the bottom of the shaft where the 0.945” hole exists.
5. The 4.75” long section of AISI 1018 steel will then need to be inserted into the hole that has just be made in the cylinder. The end being inserted needs to be the end without the hole in it. A socket weld will then need to be made at the outer contact area.

10.2 Additional Recommendations

Some additional machine improvements are noted below. While these aspects of the machine were not critical enough to garner major component redesign, these associated improvements are critical to the continued effective and safe operation of the test apparatus.

10.2.1 Machine Anchoring

Currently, the force plate apparatus is mounted directly to the floor of the testing lab, as shown below in Figure 36. As a result, machine operation can be very noisy for patrons in adjacent rooms.

Figure 36: Force plate anchoring

One of the most efficient ways to reduce operating noise would be to mount the apparatus onto a foam rubber base, similar to the vertical impact testing machine located adjacent to this test apparatus. The foam used would be a sturdy hard foam that would be more efficient at dampening the vibration and noise, but may affect the operating of the machine due to the foam not being as sturdy as the floor. Since this machine addition is easy to integrate, it is recommended that testing procedures are analyzed, and this recommendation is utilized as required.

10.2.2 Angle Adjustment Rod

As discussed previously, the angle adjustment rod was found to be significantly deformed (Figure 37). This indicates that the current method of angle adjustment on the support bracket is not ideal, and was the justification for the alternative support bracket design.

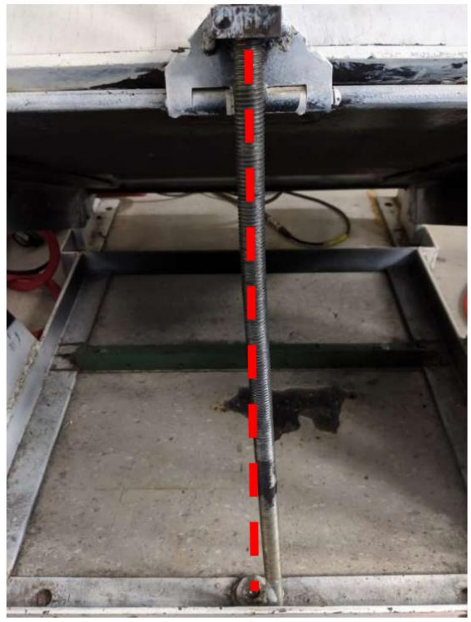


Figure 37: Existing angle adjustment rod

Since the updated design utilized three static testing positions, the angle adjustment rod is no longer required for machine function. As a result, this portion of the machine can be removed entirely.

10.2.3 Force Plate Cover Clasp

The clasps that are currently in place to secure the force plate need replacement. Figure 38 below (left) shows an example of one of the damaged clasps currently on the machine.

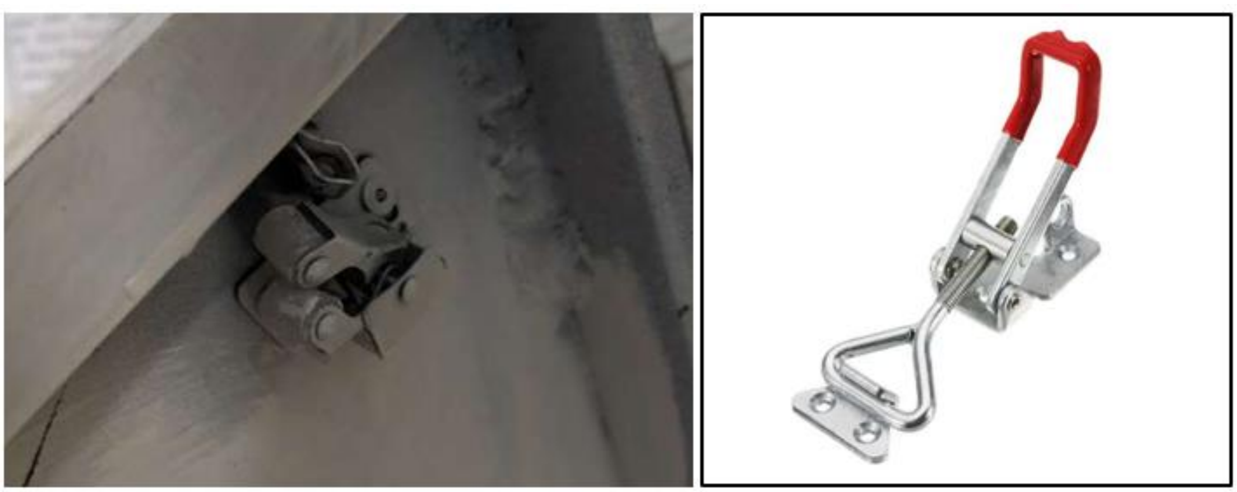


Figure 38: Force plate cover clasp (left) and toggle latch (right)

These clasps will be replaced with a simple toggle latch. Toggle latches are an extremely reliable, economical and effective method to create a strong sealing force. Depending on the chosen holding capacity, these can be found on amazon between $10 to $30 for four of them. Shown above in Figure 38 (right) is an example of a toggle latch.

10.2.4 Acrylic Safety Shield

The acrylic sheet of the safety guard is broken, as seen below in Figure 39.

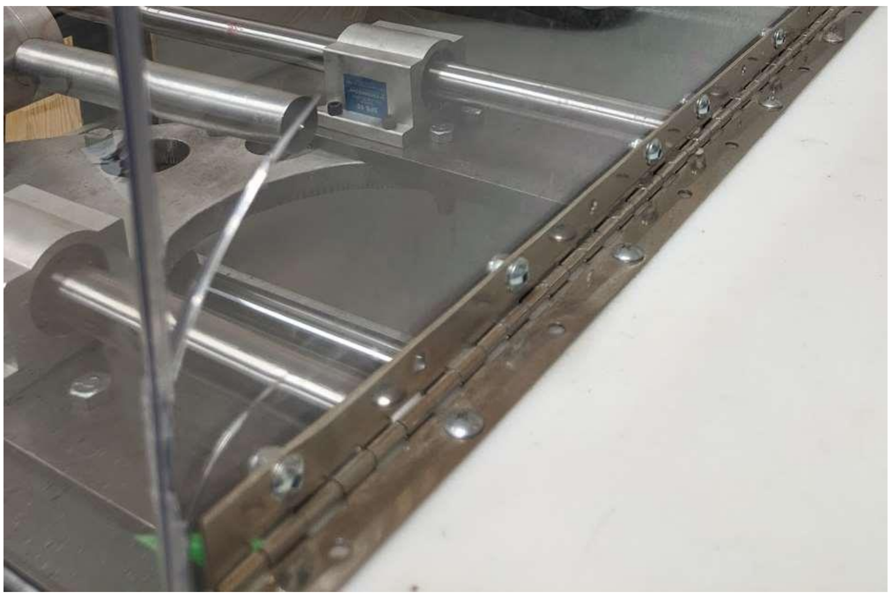


Figure 39: Damaged acrylic

It is recommended that this sheet is replaced with a thicker segment of extruded acrylic. The current sheet is 1/8" thick, a replacement sheet of at least 1/4" is recommended to withstand the stresses encountered at the hinge. An approximate cost for a replacement sheet is $15 + shipping from TAP Plastics. Holes would need to drilled to attach it to the current safety guard. Acrylic cement is also required to secure the replacement acrylic sheet to the adjoining sections.

10.2.5 Pneumatic Devices

Our research has indicated that the current pressure regulator utilized for this equipment (Campbell Hausfeld PA212303AV), indicated below in Figure 40, is not optimized for the pressure fluctuations experienced during equipment operation.

-

Figure 40: Existing air regulator

These pressure fluctuations can largely be accommodated through a correctly sized precision pressure regulator. The following characteristics are recommended for this application:

- 3/8" connection
- NPT thread
- 0,05 bar to 2,5 bar pressure range
- Z Flow direction from right to left)

10.3 Closing Remarks

The design objectives of the repetitive impact testing apparatus have been satisfied through the development of several major and minor designs. The proposed impact rod design introduces a significantly improved range of motion during test procedures. The damaged adjustment knuckle component has been redesigned to better accommodate the conditions encountered during experimental procedures. The updated adjustment knuckle will have an increased interfacing surface area, increasing the stability of the actuating rod during tests and reducing associated oscillations. The rigidity of the force plate will be improved with the use of the new support brackets. The static testing positions, coupled with the maneuverability introduced with the impact rod, allow the force plate to be struck at various angles during testing procedures. Finally, use of the proposed cane tip stepped shafts will allow rubber cane tips to be assembled onto the actuating shaft more easily, by incorporating a more appropriate diametral interference between the contacting surfaces.

All major design adjustments were created with the identified design constraints in mind. All major and minor design components will interface with the existing design as simply as possible – using the same means of attachment as existing components. The exception to this is the support brackets, which need to be welded onto the force plate to function as intended. The pneumatics of the testing apparatus have not changed, with the exception of a suggestion of a precision pressure regular to accommodate pressure losses during machine operation.

The core mechanics of the testing apparatus remain unchanged, while new features have been introduced. Several new testing procedures can be conducted, and the safety of experimental procedures has been improved. The dynamic nature of the proposed testing apparatus makes it irreplaceable in injury biomechanics research, and provides a solid foundation for the next iteration of design.

11.0 References

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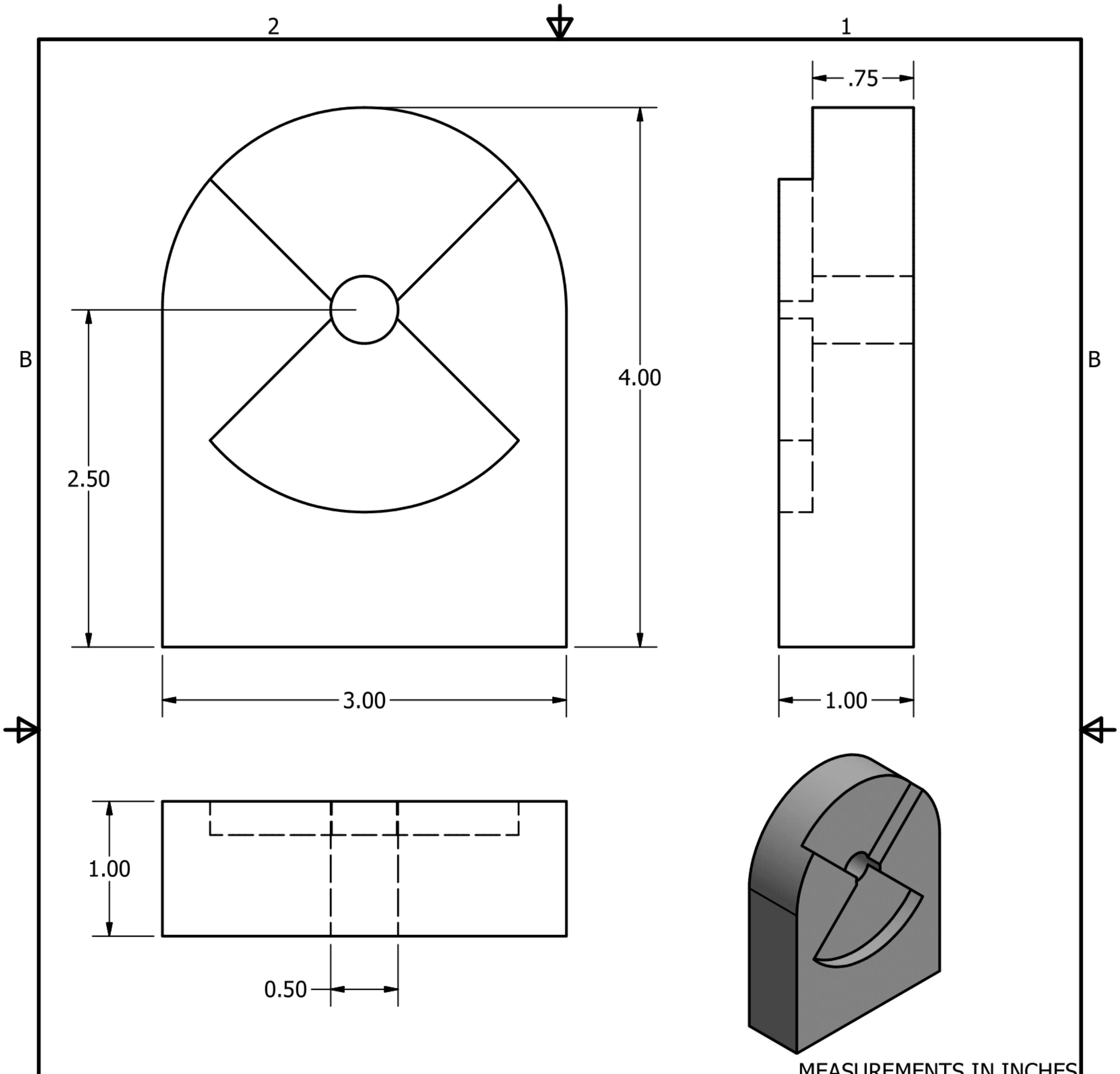
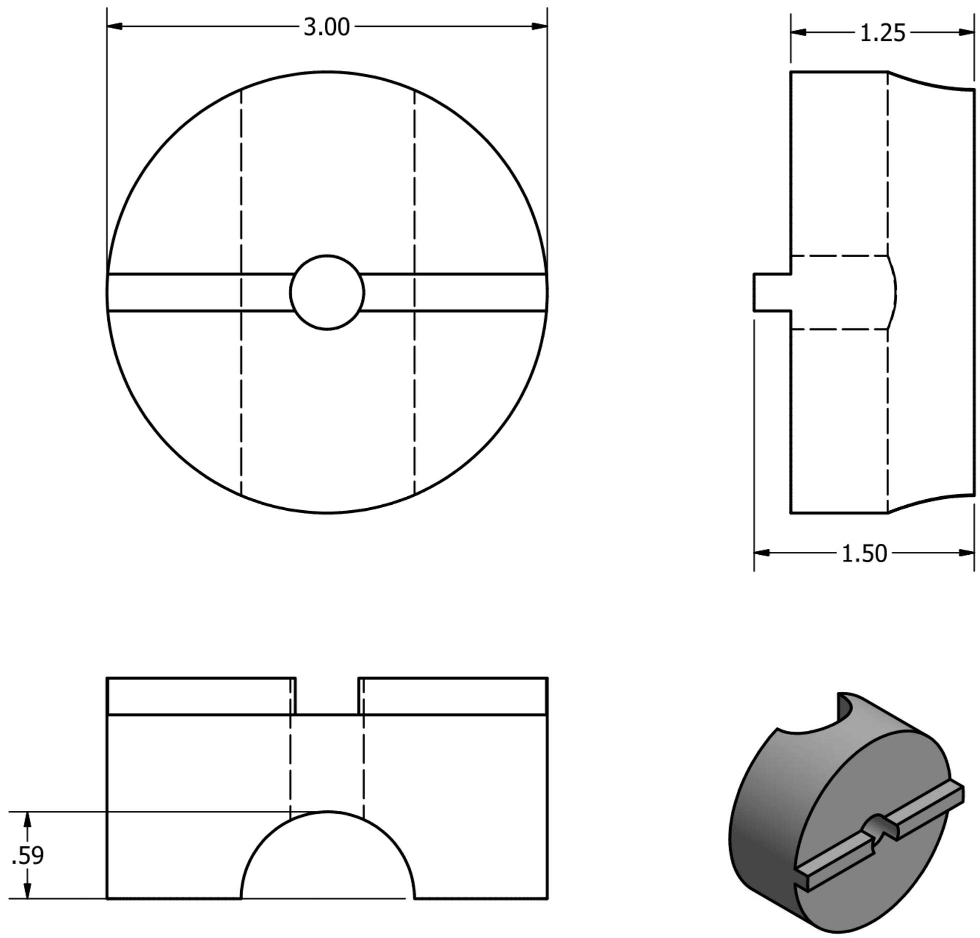
Appendix A: Repetitive Impact Testing Apparatus



DRAWN Adam Genno	10/15/19	Knuckle A
CHECKED		TITLE
QA		Repetitive Impact Testing Apparatus
MFG		
APPROVED		SIZE A / DWG NO RITA Rev0 / REV
		SCALE 1 : 1 / SHEET 1 OF 9

A
A
2
1



MEASUREMENTS IN INCHES

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QA		
MFG		
APPROVED		SIZE A / DWG NO RITA Rev0 / REV
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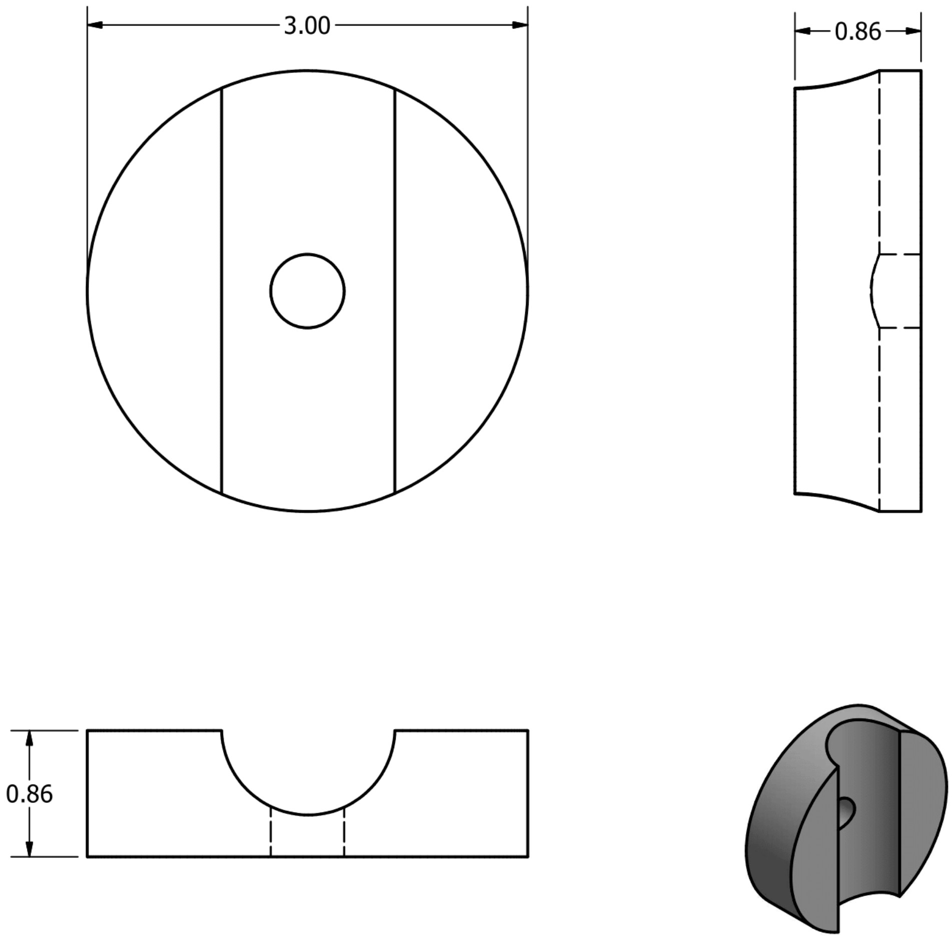

MEASUREMENTS IN INCHES

DRAWN Adam Genno	10/15/19	Knuckle C
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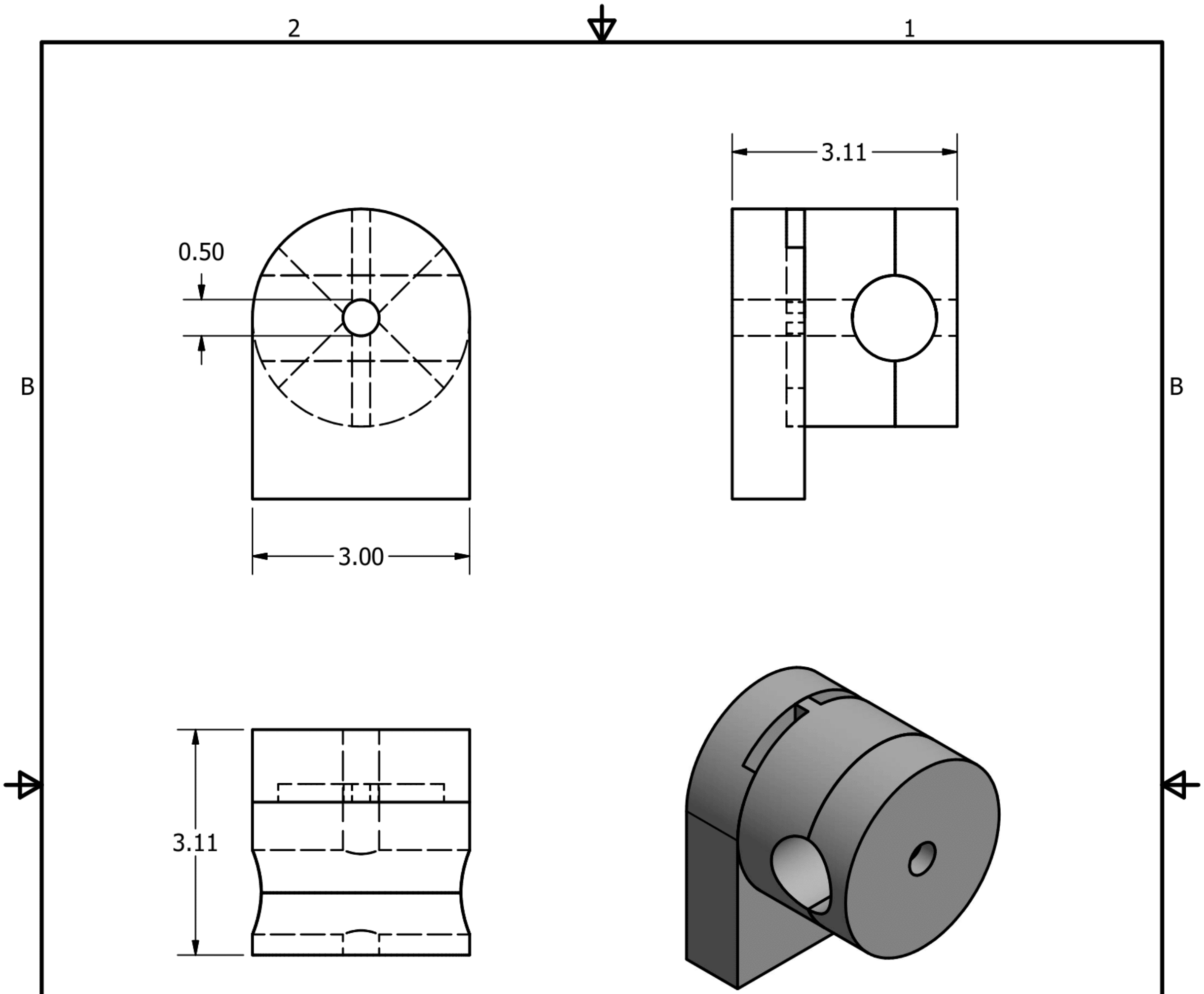
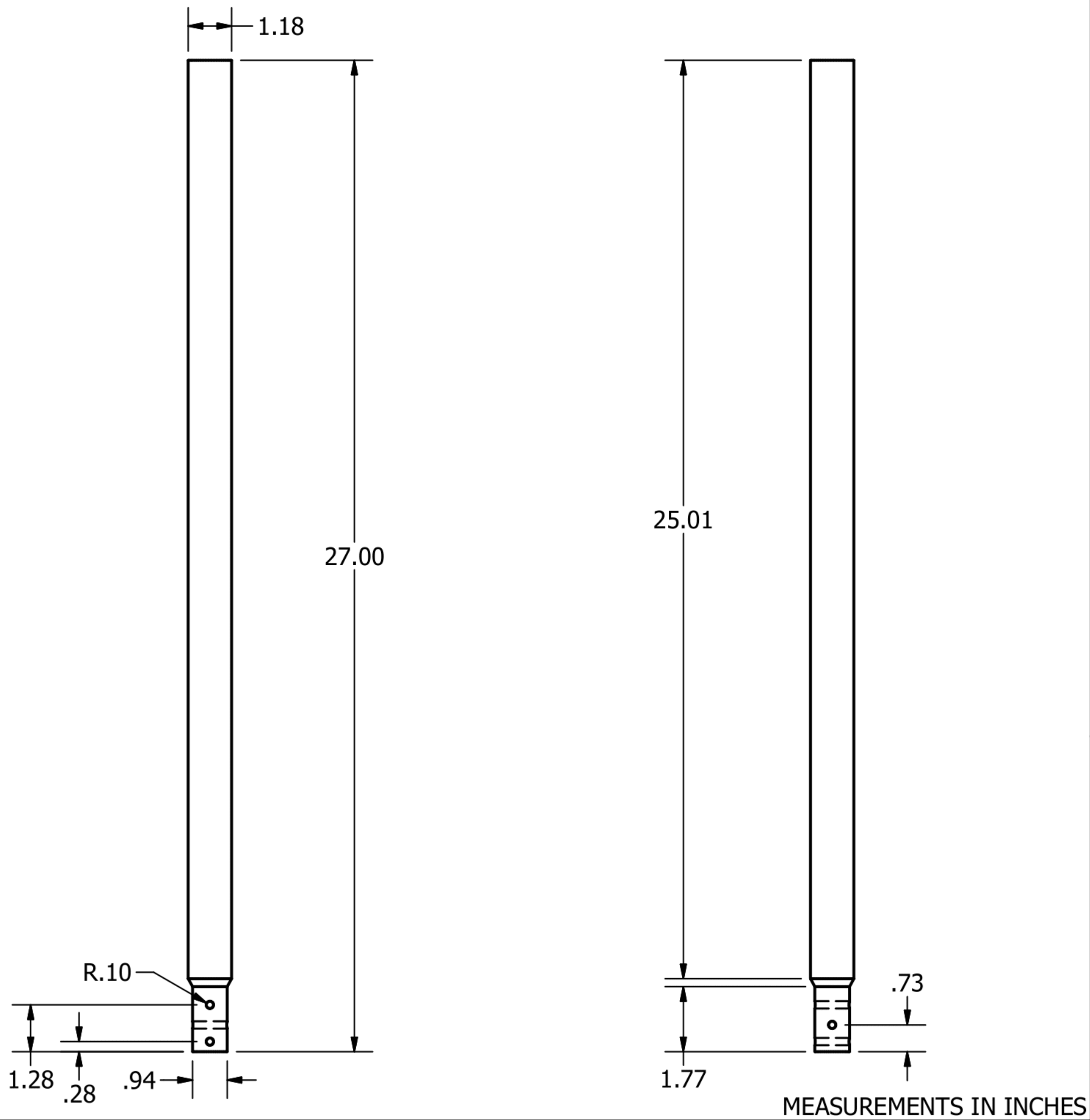
QA		
MFG		
APPROVED		SIZE A — DWG NO RITA Rev0 — REV
		SCALE 1 : 1 — SHEET 3 OF 9



MEASUREMENTS IN INCHES

DRAWN Adam Genno	10/15/19	Assembled Knuckle			
CHECKED		TITLE Repetitive Impact Testing Apparatus			
QA					
MFG					
APPROVED					
		SIZE A		DWG NO RITA Rev0	REV
		SCALE 1 / 2			SHEET 4 OF 9



MEASUREMENTS IN INCHES

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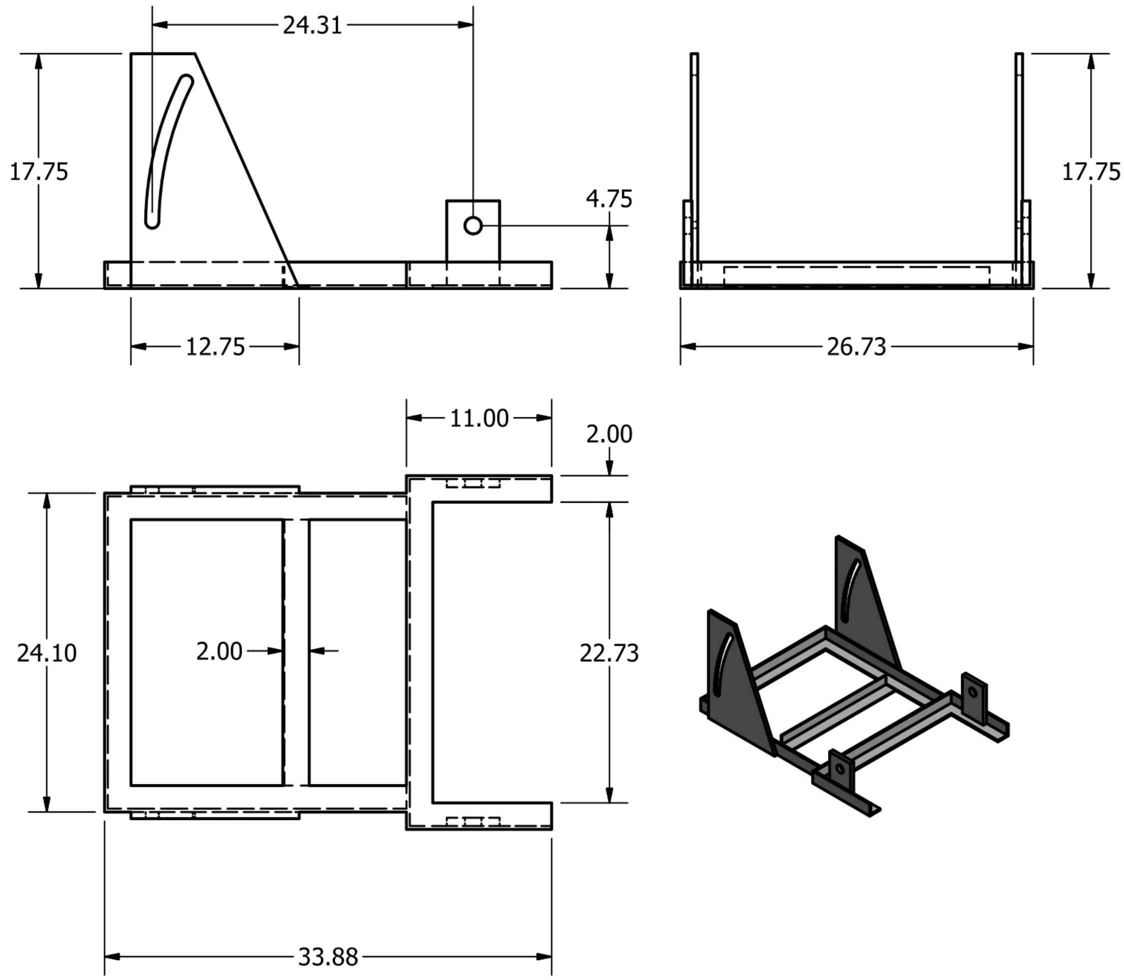
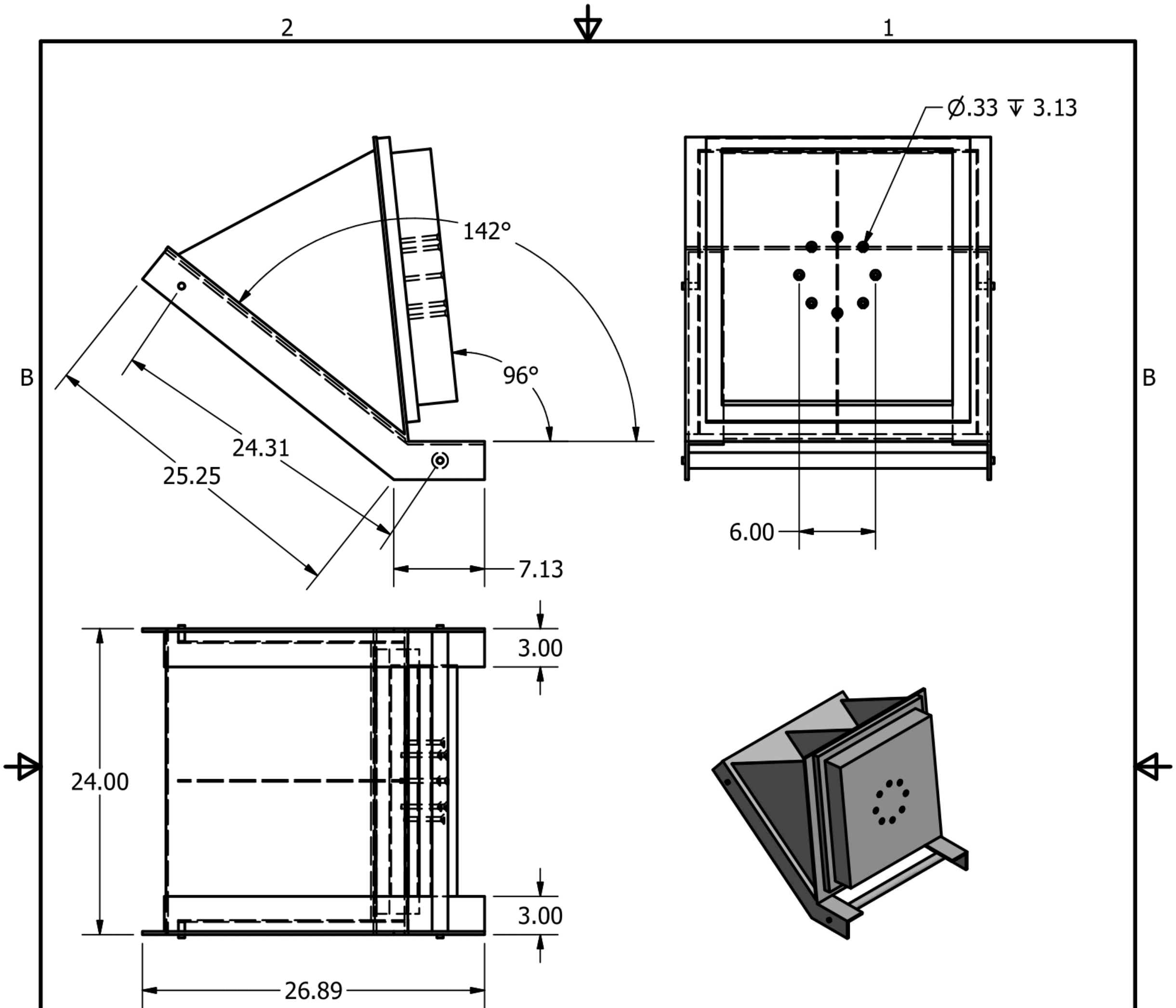
CHECKED		TITLE
QA		Repetitive Impact Testing Apparatus
MFG		
APPROVED		
		SIZE A · DWG NO RITA Rev0 · REV
		SCALE 1:4 · SHEET 5 OF 9



MEASUREMENTS IN INCHES

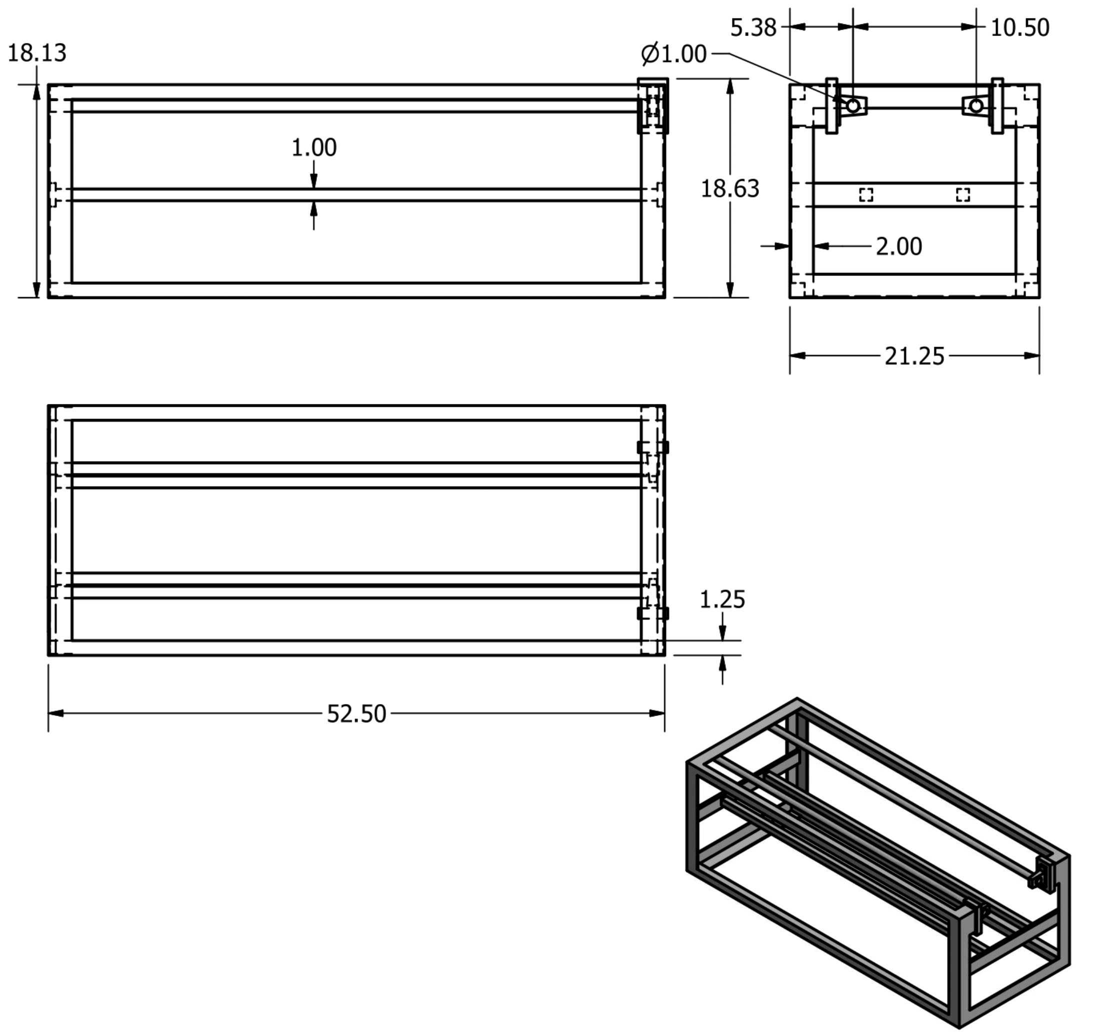
DRAWN Adam Genno	10/15/19	Force Plate Base			
CHECKED		TITLE Repetitive Impact Testing Apparatus			
QA					
MFG					
APPROVED					
		SIZE A		DWG NO RITA Rev0	REV
		SCALE 0.09 : 1		SHEET 6 OF 9	



MEASUREMENTS IN INCHES

DRAWN Adam Genno	10/15/19	Force Plate			
CHECKED		TITLE			
QA		Repetitive Impact Testing Apparatus			
MFG					
APPROVED					
		SIZE A		DWG NO RITA Rev0	REV
		SCALE 0.09 : 1		SHEET 7 OF 9	

A
A
2
1



MEASUREMENTS IN INCHES

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QA		
MFG		
APPROVED		SIZE A · DWG NO RITA Rev0 · REV
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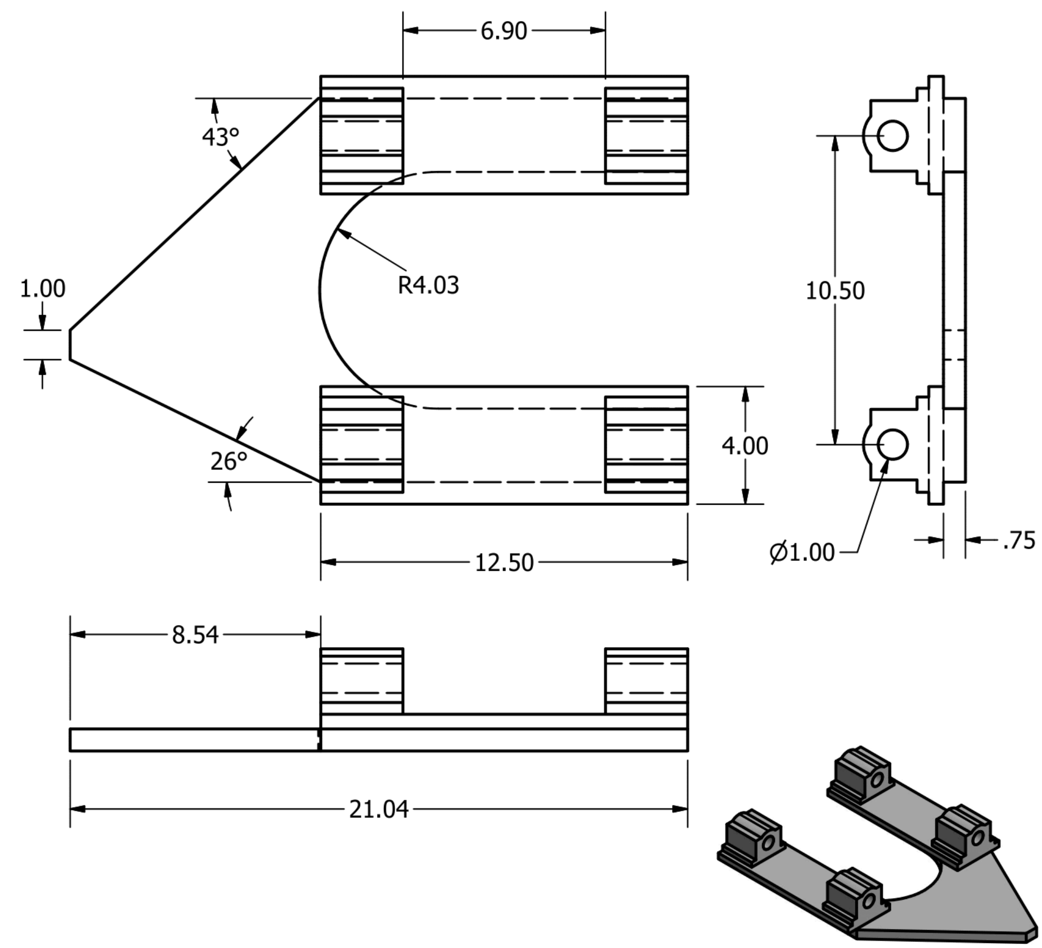

MEASUREMENTS IN INCHES

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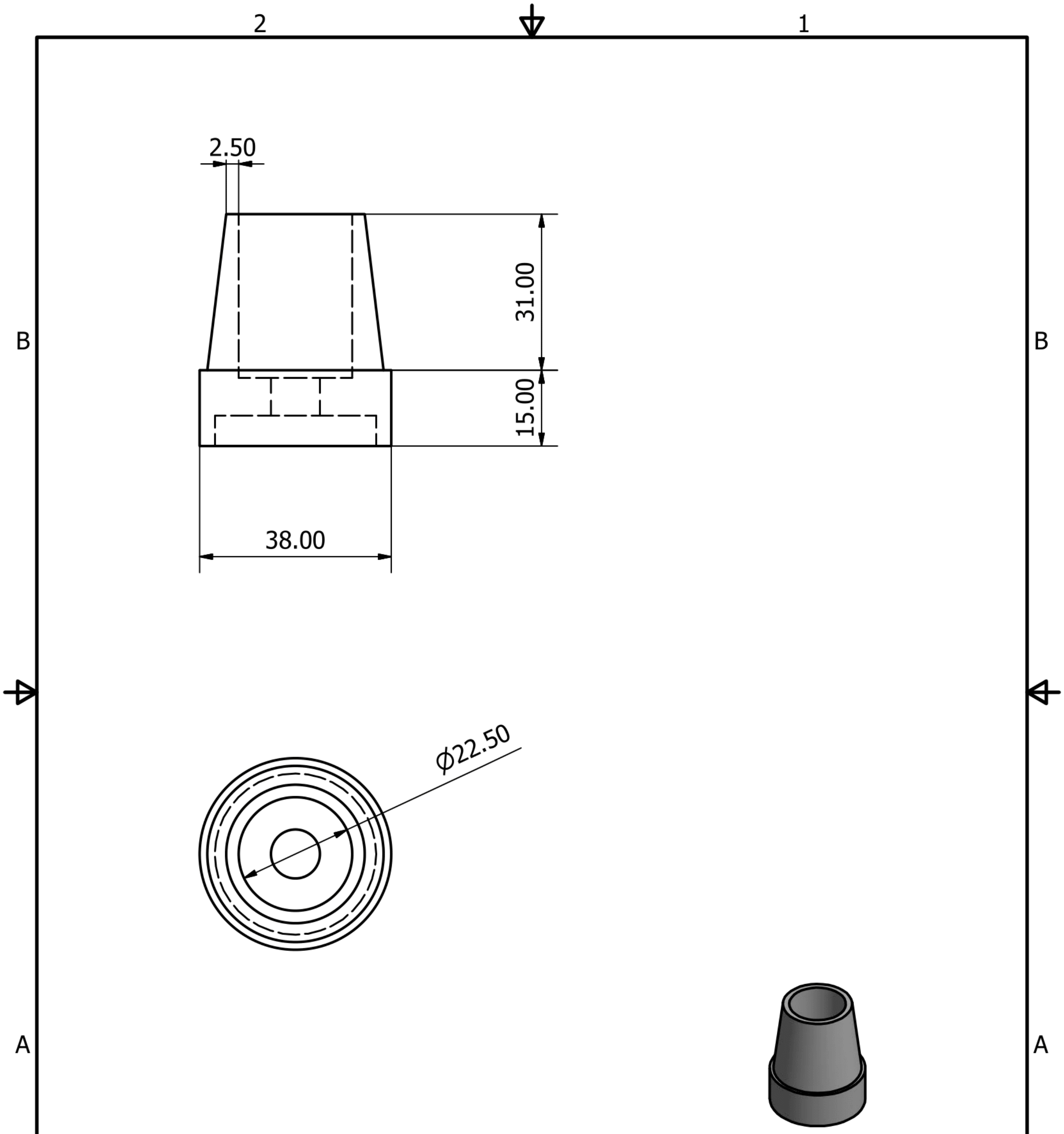
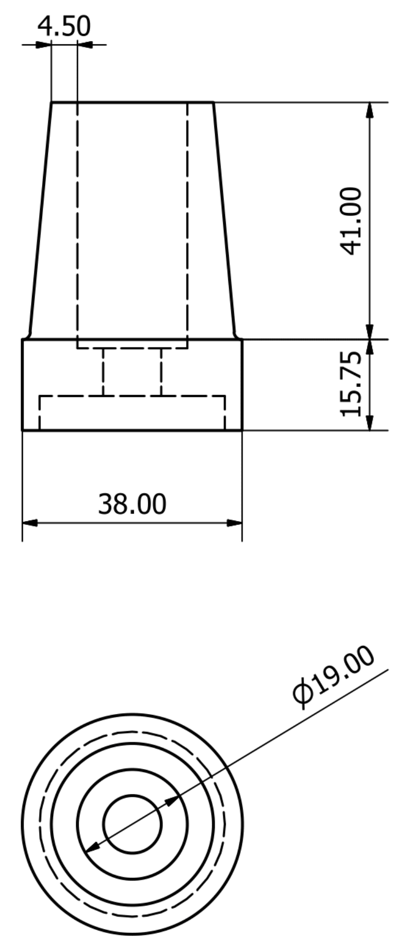
QA		
MFG		
APPROVED		SIZE A DWG NO RITA Rev0 REV
		SCALE 0.2 : 1 SHEET 9 OF 9

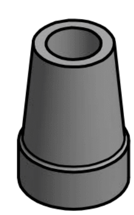
Appendix B: Rubber Cane Tips



DIMENSIONS IN MILLIMETERS

Designed by Adam Genno	Checked by	Approved by	Date	Date 1/23/20	
EMEC-4969		RITA			
		Cane Tip (Type A)		Edition 2	Sheet 1 / 3

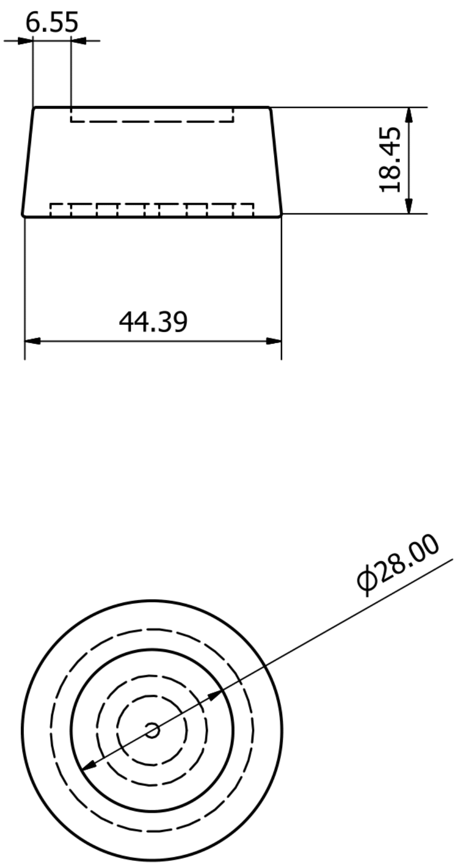


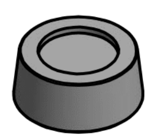


DIMENSIONS IN MILLIMETERS

Designed by	Checked by	Approved by	Date		Date	
Adam Genno					1/23/20	

EMEC-4969	RITA		
	Cane Tip (Type B - Upper)	Edition 2	Sheet 2 / 3





DIMENSIONS IN MILLIMETERS

Designed by	Checked by	Approved by	Date		Date	
Adam Genno					1/23/20	
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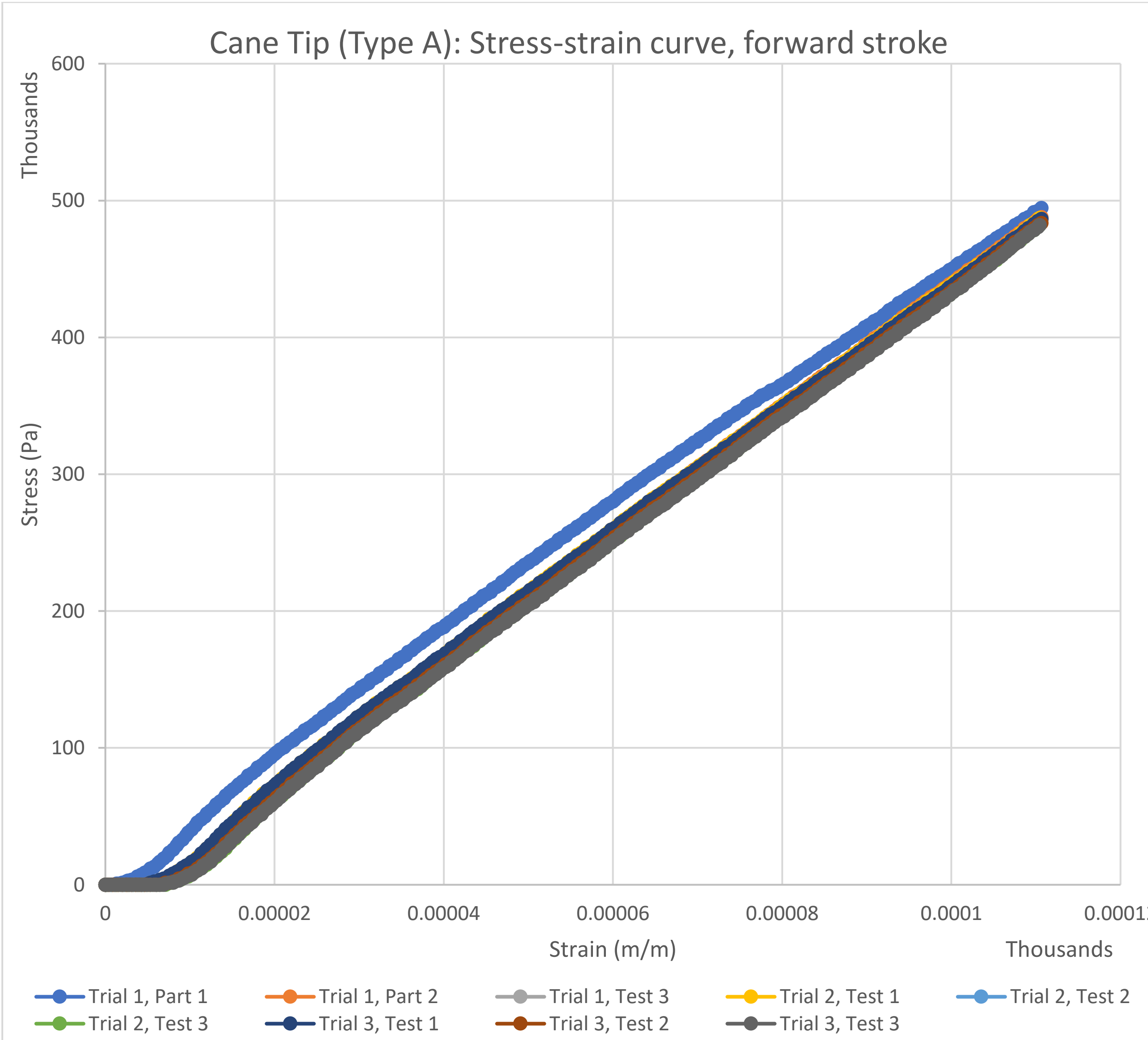
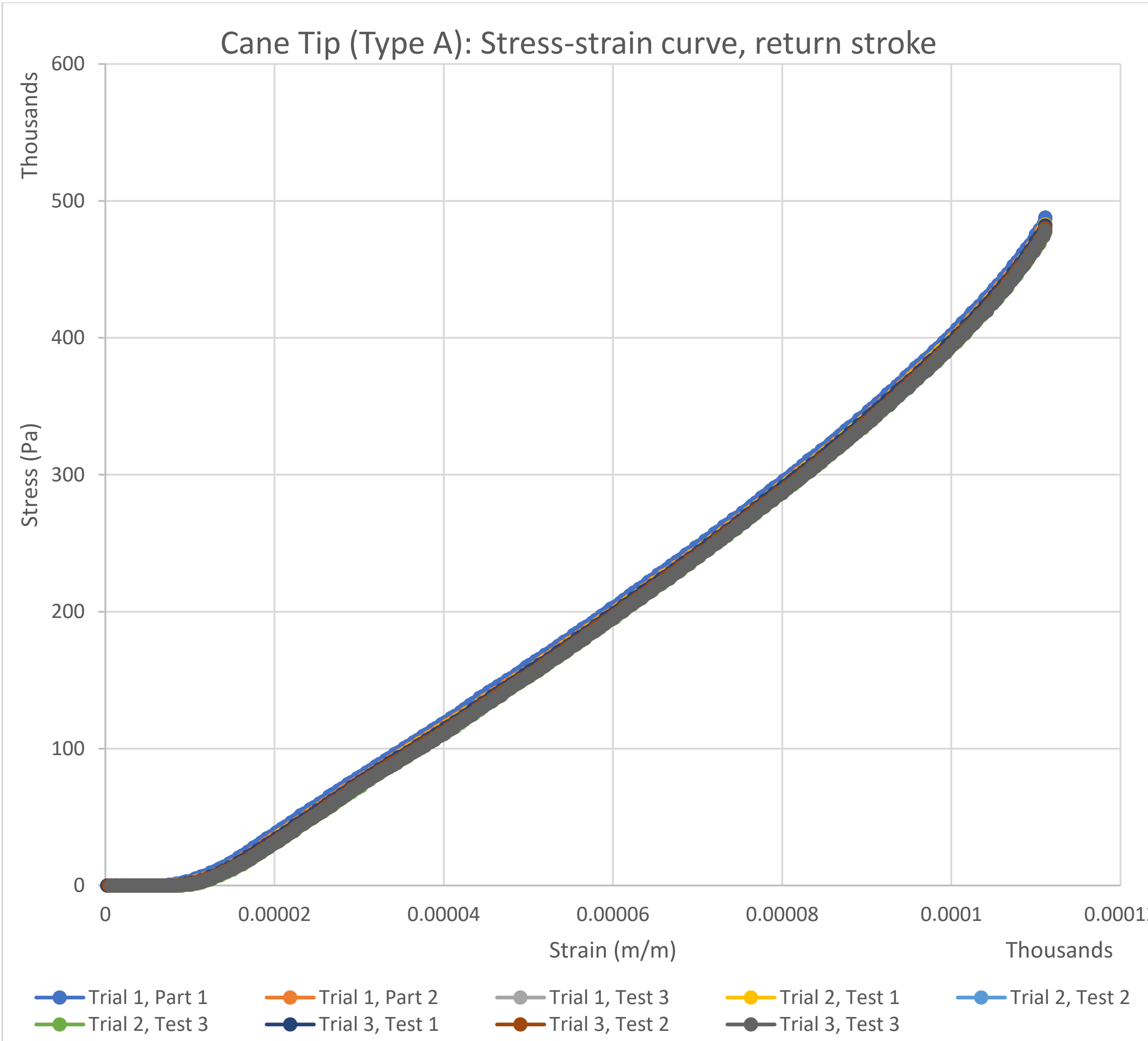
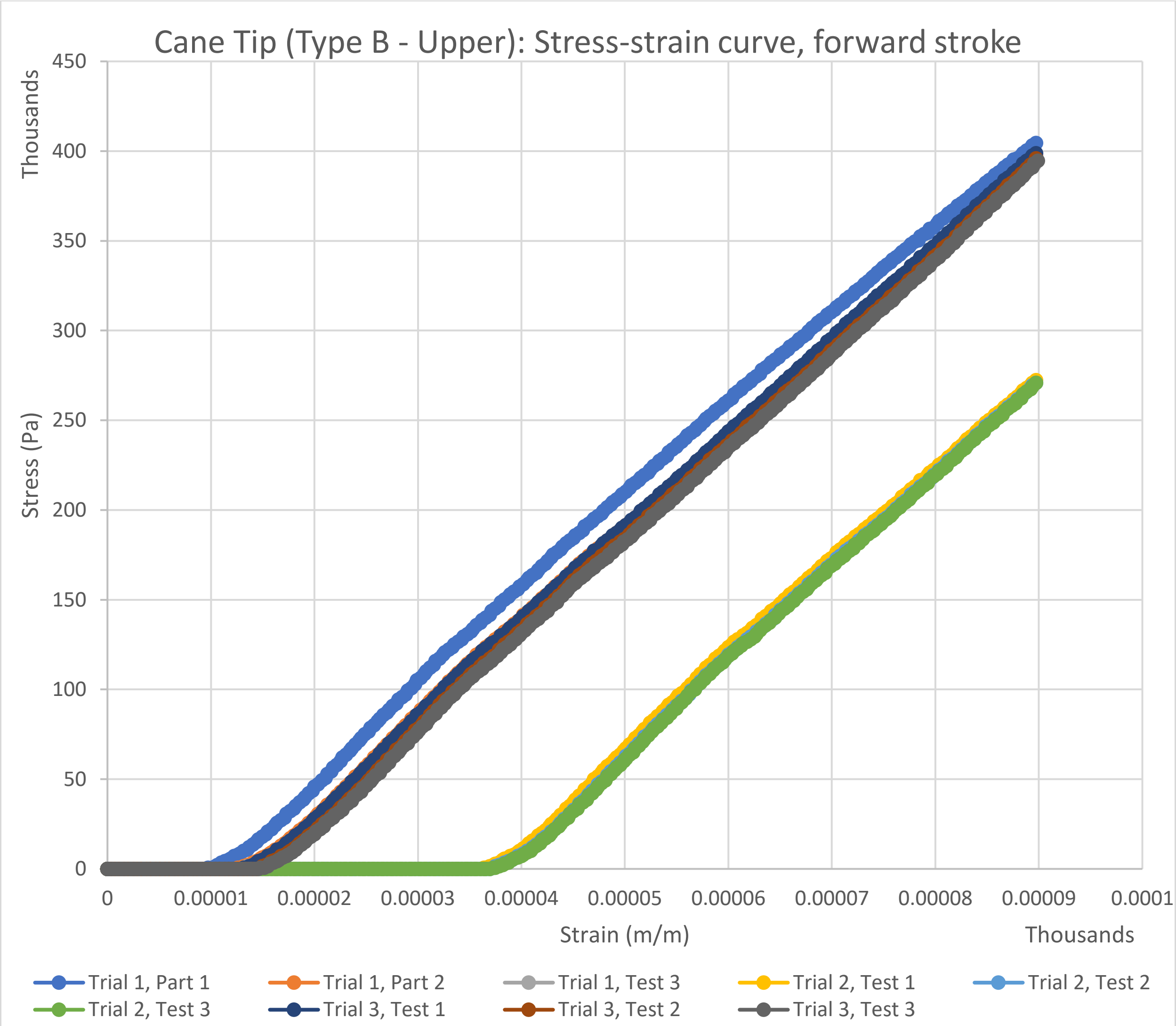
			Cane Tip (Type B - Lower)		Edition 2	Sheet 3 / 3

Appendix C: Rubber Cane Tip Testing

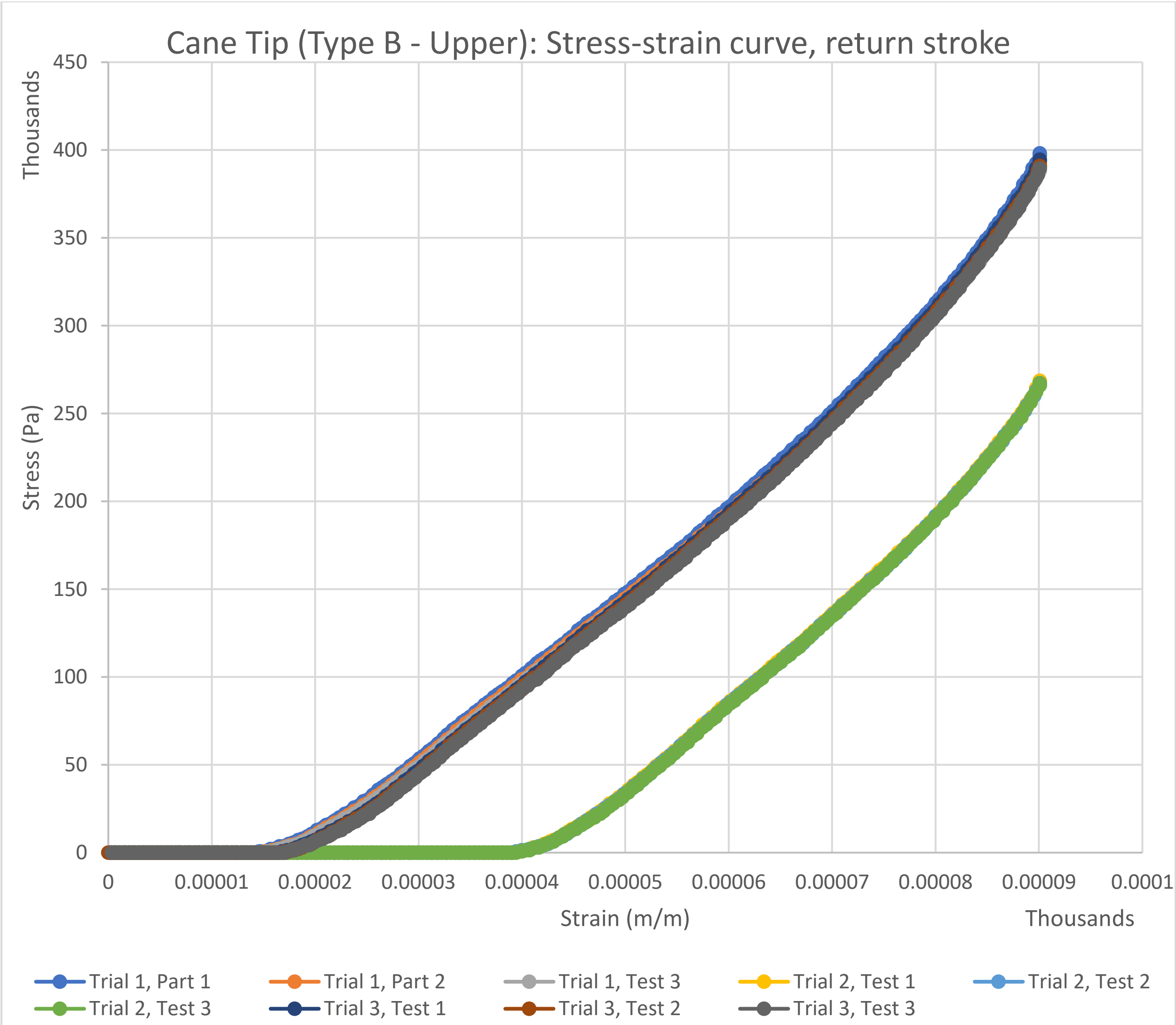




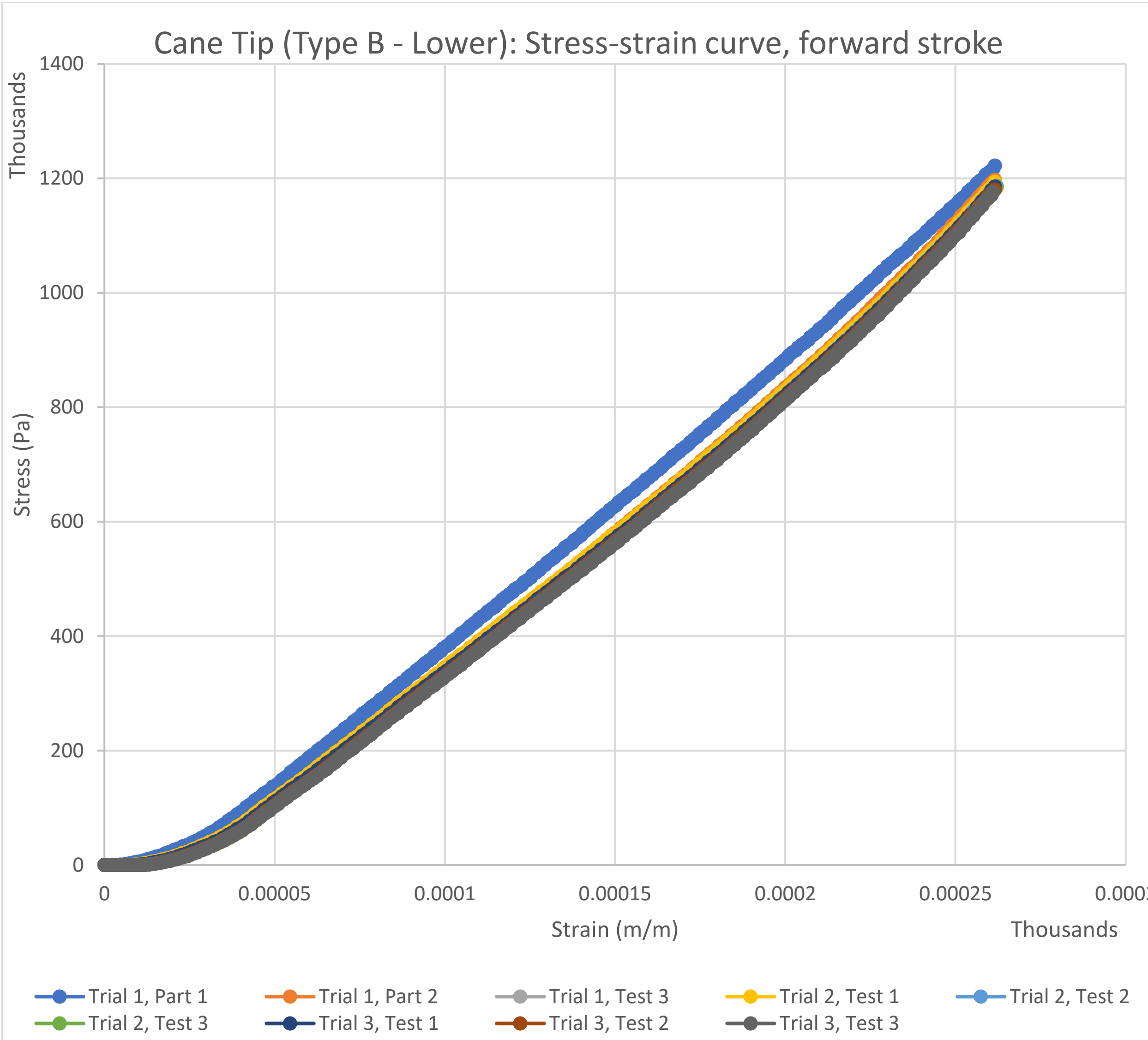
Cane Tip (Type A): Stress-strain curve, return stroke
Thousands
Stress (Pa)
600
500
400
300
200
100
0
0
0.00002
0.00004
0.00006
0.00008
0.0001
0.00012
Strain (m/m)
Thousands
Trial 1, Part 1
Trial 1, Part 2
Trial 1, Test 3
Trial 2, Test 1
Trial 2, Test 2
Trial 2, Test 3
Trial 3, Test 1
Trial 3, Test 2
Trial 3, Test 3



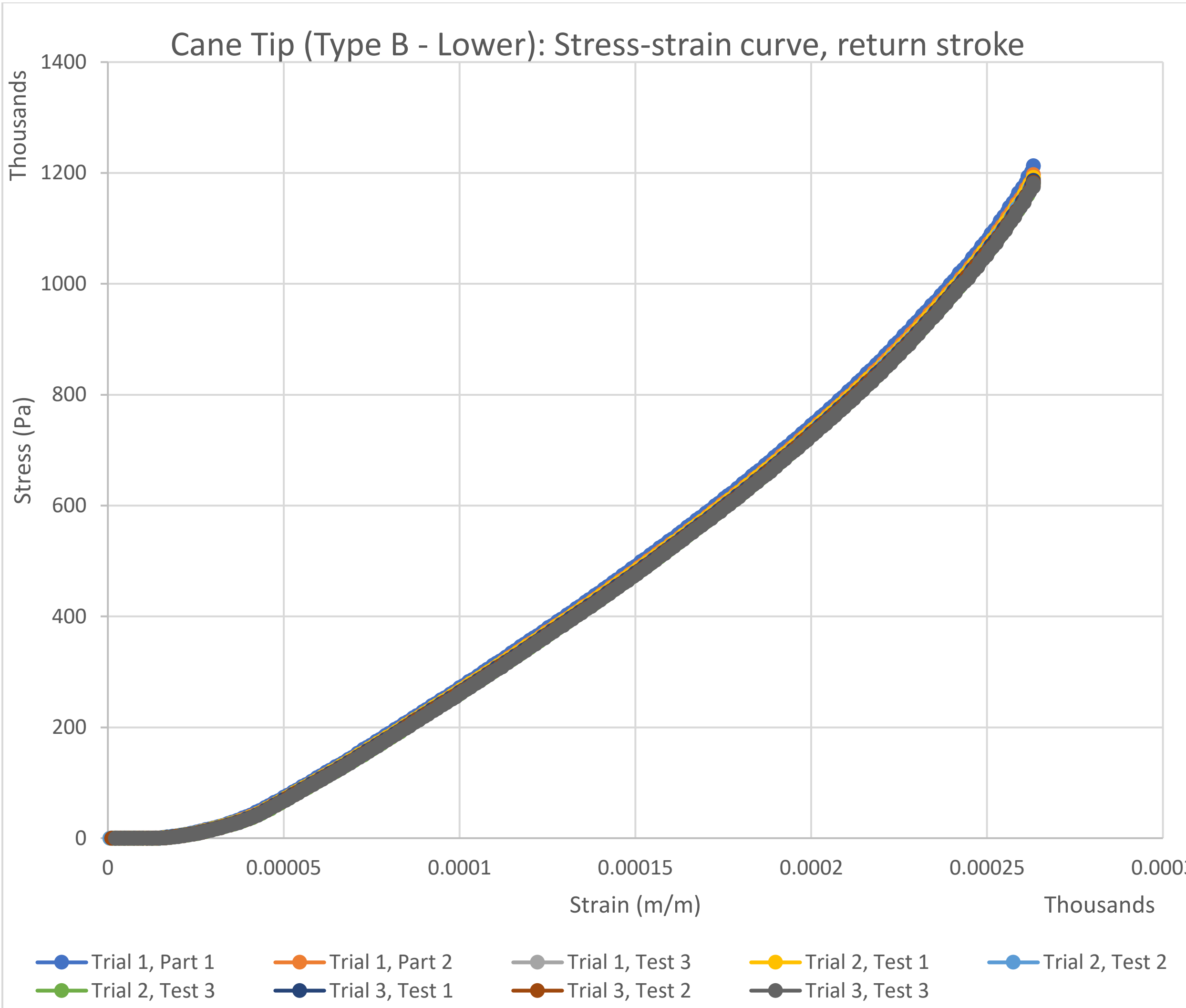
Cane Tip (Type B - Upper): Stress-strain curve, forward stroke
Stress (Pa)
Thousands
450
400
350
300
250
200
150
100
50
0
0
0.00001
0.00002
0.00003
0.00004
0.00005
0.00006
0.00007
0.00008
0.00009
0.0001
Strain (m/m)
Thousands
Trial 1, Part 1
Trial 1, Part 2
Trial 1, Test 3
Trial 2, Test 1
Trial 2, Test 2
Trial 2, Test 3
Trial 3, Test 1
Trial 3, Test 2
Trial 3, Test 3



Cane Tip (Type B - Upper): Stress-strain curve, return stroke
Thousands
Stress (Pa)
450
400
350
300
250
200
150
100
50
0
0
0.00001
0.00002
0.00003
0.00004
0.00005
0.00006
0.00007
0.00008
0.00009
0.0001
Strain (m/m)
Thousands
Trial 1, Part 1
Trial 1, Part 2
Trial 1, Test 3
Trial 2, Test 1
Trial 2, Test 2
Trial 2, Test 3
Trial 3, Test 1
Trial 3, Test 2
Trial 3, Test 3



Cane Tip (Type B - Lower): Stress-strain curve, forward stroke
Thousands
Stress (Pa)
1400
1200
1000
800
600
400
200
0
0
0.00005
0.0001
0.00015
0.0002
0.00025
0.0003
Strain (m/m)
Thousands
Trial 1, Part 1
Trial 1, Part 2
Trial 1, Test 3
Trial 2, Test 1
Trial 2, Test 2
Trial 2, Test 3
Trial 3, Test 1
Trial 3, Test 2
Trial 3, Test 3


Cane Tip (Type B - Lower): Stress-strain curve, return stroke
Thousands
Stress (Pa)
1400
1200
1000
800
600
400
200
0
0
0.00005
0.0001
0.00015
0.0002
0.00025
0.0003
Strain (m/m)
Thousands
Trial 1, Part 1
Trial 1, Part 2
Trial 1, Test 3
Trial 2, Test 1
Trial 2, Test 2
Trial 2, Test 3
Trial 3, Test 1
Trial 3, Test 2
Trial 3, Test 3

Tip size:	**Type A**		Tip size:	**Type A**
Trial #:	**1**		Trial #:	**1**
Stroke:	**Forward**		Stroke:	**Reverse**
X-coord of Linear Elastic Region (Start, in m/m):	0.035556		X-coord of Linear Elastic Region (Start, in m/m):	0.08
Y-coord of Linear Elastic Region (End, in Pa):	167765.6		Y-coord of Linear Elastic Region (End, in Pa):	294505.5
X-coord of Linear Elastic Region (Start, in m/m):	0.090889		X-coord of Linear Elastic Region (Start, in m/m):	0.024222
Y-coord of Linear Elastic Region (End, in Pa):	411721.6		Y-coord of Linear Elastic Region (End, in Pa):	55677.66
Slope (Pa):	**4.409E+06**		Slope (Pa):	**4.282E+06**
Tip size:	**Type A**		Tip size:	**Type A**
Trial #:	**2**		Trial #:	**2**
Stroke:	**Forward**		Stroke:	**Reverse**
X-coord of Linear Elastic Region (Start, in m/m):	0.026889		X-coord of Linear Elastic Region (Start, in m/m):	0.088444
Y-coord of Linear Elastic Region (End, in Pa):	104029.3		Y-coord of Linear Elastic Region (End, in Pa):	332600.7
X-coord of Linear Elastic Region (Start, in m/m):	0.082444		X-coord of Linear Elastic Region (Start, in m/m):	0.032667
Y-coord of Linear Elastic Region (End, in Pa):	361172.2		Y-coord of Linear Elastic Region (End, in Pa):	86446.89
Slope (Pa):	**4.629E+06**		Slope (Pa):	**4.413E+06**
Tip size:	**Type A**		Tip size:	**Type A**
Trial #:	**3**		Trial #:	**3**
Stroke:	**Forward**		Stroke:	**Reverse**
X-coord of Linear Elastic Region (Start, in m/m):	0.018444		X-coord of Linear Elastic Region (Start, in m/m):	0.096889
Y-coord of Linear Elastic Region (End, in Pa):	56410.26		Y-coord of Linear Elastic Region (End, in Pa):	378022
X-coord of Linear Elastic Region (Start, in m/m):	0.074222		X-coord of Linear Elastic Region (Start, in m/m):	0.041333
Y-coord of Linear Elastic Region (End, in Pa):	319413.9		Y-coord of Linear Elastic Region (End, in Pa):	118681.3
Slope (Pa):	4715205		Slope (Pa):	4668132
Average modulus - Forward stroke (Pa):	**4.584E+06**		Average modulus - Forward stroke (Pa):	**4.454E+06**
Average modulus - Type A (Pa):				**4.519E+06**

Tip size:	**Type B1**		Tip size:	**Type B1**
Trial #:	**1**		Trial #:	**1**
Stroke:	**Forward**		Stroke:	**Reverse**
X-coord of Linear Elastic Region (Start, in m/m):	0.028829		X-coord of Linear Elastic Region (Start, in m/m):	0.064685
Y-coord of Linear Elastic Region (End, in Pa):	97212.29		Y-coord of Linear Elastic Region (End, in Pa):	221586.8
X-coord of Linear Elastic Region (Start, in m/m):	0.073874		X-coord of Linear Elastic Region (Start, in m/m):	0.01964
Y-coord of Linear Elastic Region (End, in Pa):	329521.1		Y-coord of Linear Elastic Region (End, in Pa):	10721.94
Slope (Pa):	5157255		Slope (Pa):	4681201
Tip size:	**Type B1**		Tip size:	**Type B1**
Trial #:	**2**		Trial #:	**2**
Stroke:	**Forward**		Stroke:	**Reverse**
X-coord of Linear Elastic Region (Start, in m/m):	0.021982		X-coord of Linear Elastic Region (Start, in m/m):	0.071712
Y-coord of Linear Elastic Region (End, in Pa):	39313.8		Y-coord of Linear Elastic Region (End, in Pa):	256611.9
X-coord of Linear Elastic Region (Start, in m/m):	0.067027		X-coord of Linear Elastic Region (Start, in m/m):	0.026486
Y-coord of Linear Elastic Region (End, in Pa):	276626.2		Y-coord of Linear Elastic Region (End, in Pa):	34310.22
Slope (Pa):	5268335		Slope (Pa):	4915435
Tip size:	**Type B1**		Tip size:	**Type B1**
Trial #:	**3**		Trial #:	**3**
Stroke:	**Forward**		Stroke:	**Reverse**
X-coord of Linear Elastic Region (Start, in m/m):	0.014955		X-coord of Linear Elastic Region (Start, in m/m):	0.078559
Y-coord of Linear Elastic Region (End, in Pa):	5003.574		Y-coord of Linear Elastic Region (End, in Pa):	296640.5
X-coord of Linear Elastic Region (Start, in m/m):	0.06		X-coord of Linear Elastic Region (Start, in m/m):	0.033333
Y-coord of Linear Elastic Region (End, in Pa):	238027.2		Y-coord of Linear Elastic Region (End, in Pa):	65761.26
Slope (Pa):	5173124		Slope (Pa):	5105098
Average modulus - Forward stroke (Pa):	**5.200E+06**		Average modulus - Forward stroke (Pa):	**4.901E+06**
Average modulus - Type B1 (Pa):				**5.050E+06**

Tip size:	**Type B2**	Tip size:	**Type B2**
Trial #:	**1**	Trial #:	**1**
Stroke:	**Forward**	Stroke:	**Reverse**
X-coord of Linear Elastic Region (Start, in m/m):	0.083684	X-coord of Linear Elastic Region (Start, in m/m):	0.189474
Y-coord of Linear Elastic Region (End, in Pa):	301718.2	Y-coord of Linear Elastic Region (End, in Pa):	687628.9
X-coord of Linear Elastic Region (Start, in m/m):	0.215263	X-coord of Linear Elastic Region (Start, in m/m):	0.057895
Y-coord of Linear Elastic Region (End, in Pa):	964261.2	Y-coord of Linear Elastic Region (End, in Pa):	103092.8
Slope (Pa):	5035326	Slope (Pa):	4442474

Tip size:	**Type B2**	Tip size:	**Type B2**
Trial #:	**2**	Trial #:	**2**
Stroke:	**Forward**	Stroke:	**Reverse**
X-coord of Linear Elastic Region (Start, in m/m):	0.063158	X-coord of Linear Elastic Region (Start, in m/m):	0.21
Y-coord of Linear Elastic Region (End, in Pa):	169415.8	Y-coord of Linear Elastic Region (End, in Pa):	793470.8
X-coord of Linear Elastic Region (Start, in m/m):	0.194737	X-coord of Linear Elastic Region (Start, in m/m):	0.078421
Y-coord of Linear Elastic Region (End, in Pa):	806872.9	Y-coord of Linear Elastic Region (End, in Pa):	178350.5
Slope (Pa):	4844674	Slope (Pa):	4674914

Tip size:	**Type B2**	Tip size:	**Type B2**
Trial #:	**3**	Trial #:	**3**
Stroke:	**Forward**	Stroke:	**Reverse**
X-coord of Linear Elastic Region (Start, in m/m):	0.042632	X-coord of Linear Elastic Region (Start, in m/m):	0.230526
Y-coord of Linear Elastic Region (End, in Pa):	73539.52	Y-coord of Linear Elastic Region (End, in Pa):	917869.4
X-coord of Linear Elastic Region (Start, in m/m):	0.174211	X-coord of Linear Elastic Region (Start, in m/m):	0.098947
Y-coord of Linear Elastic Region (End, in Pa):	691408.9	Y-coord of Linear Elastic Region (End, in Pa):	258419.2
Slope (Pa):	4695808	Slope (Pa):	5011821

Average modulus - Forward stroke (Pa):	**4.859E+06**	Average modulus - Forward stroke (Pa):	**4.710E+06**

Average modulus - Type B1 (Pa):	**4.784E+06**

Collected Results	
Cane tip type:	Type A
Minimum area @top (m^2):	0.000295
Max area @base (m^2):	0.00107
Average cross-sectional area (m^2):	0.0006825
Average length (m):	0.045
Cane tip type:	Type B - U
Minimum area @top (m^2):	0.000329
Max area @base (m^2):	0.00107
Average cross-sectional area (m^2):	0.0006995
Average length (m):	0.0555
Cane tip type:	Type B - L
Minimum area @top (m^2):	0.00132
Max area @base (m^2):	0.00159
Average cross-sectional area (m^2):	0.001455
Average length (m):	0.019
Overall average modulus (Pa):	**4.78451E+06**
Overall average modulus (psi):	**6.93935E+02**

Appendix D: Finite Element Analysis (Autodesk Inventor)

Stress Analysis - Three Quarter

Analyzed File:	Cane Tip Stepped Shaft - Three-Quarter Inch.ipt
Autodesk Inventor Version:	2018 (Build 220112000, 112)
Creation Date:	3/27/20, 11:24 PM
Study Author:	Adam Genno

⊟ Project Info (iProperties)

⊟ Physical

Material	Steel AISI 1018 106 HR

Note: Physical values could be different from Physical values used by FEA reported below.

⊟ Static Analysis:1

General objective and settings:

Design Objective	Single Point
Study Type	Static Analysis
Last Modification Date	3/27/20, 8:02 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

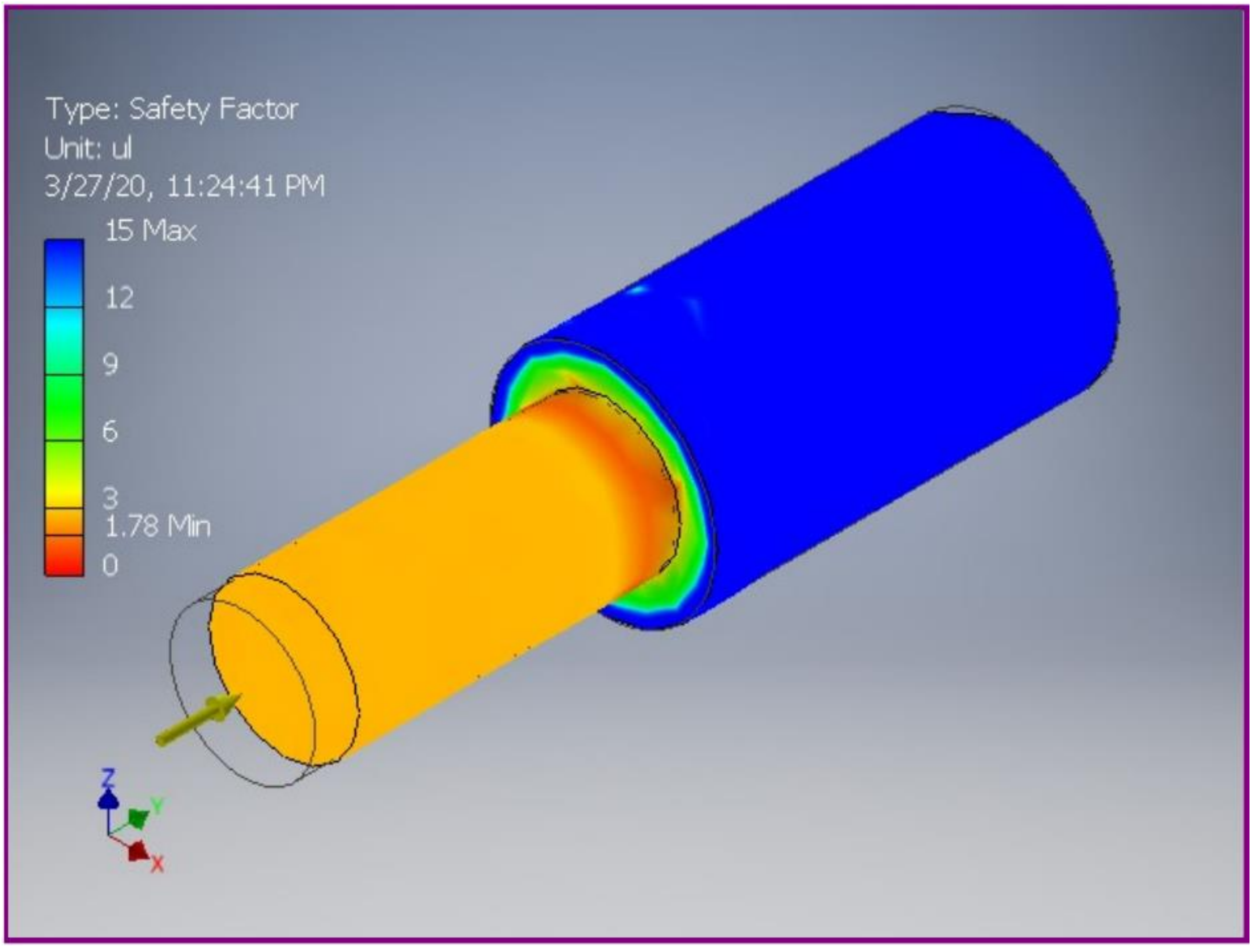
⊟ Results

⊟ Result Summary

Name	Minimum	Maximum
Volume	40230.2 mm^3	
Mass	0.316612 kg	
Safety Factor	1.77731 ul	15 ul

⊟ Figures

⊟ Safety Factor


Type: Safety Factor
Unit: ul
3/27/20, 11:24:41 PM
15 Max
12
9
6
3
1.78 Min
0
Z
Y
X

Stress Analysis - Seven Eighths

Analyzed File:	Cane Tip Stepped Shaft - Seven-Eighths.ipt
Autodesk Inventor Version:	2018 (Build 220112000, 112)
Creation Date:	3/27/20, 11:20 PM
Study Author:	Adam Genno

⊟ Project Info (iProperties)

⊟ Physical

Material	Steel AISI 1018 106 HR

Note: Physical values could be different from Physical values used by FEA reported below.

⊟ Static Analysis:1

General objective and settings:

Design Objective	Single Point
Study Type	Static Analysis
Last Modification Date	3/27/20, 11:14 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

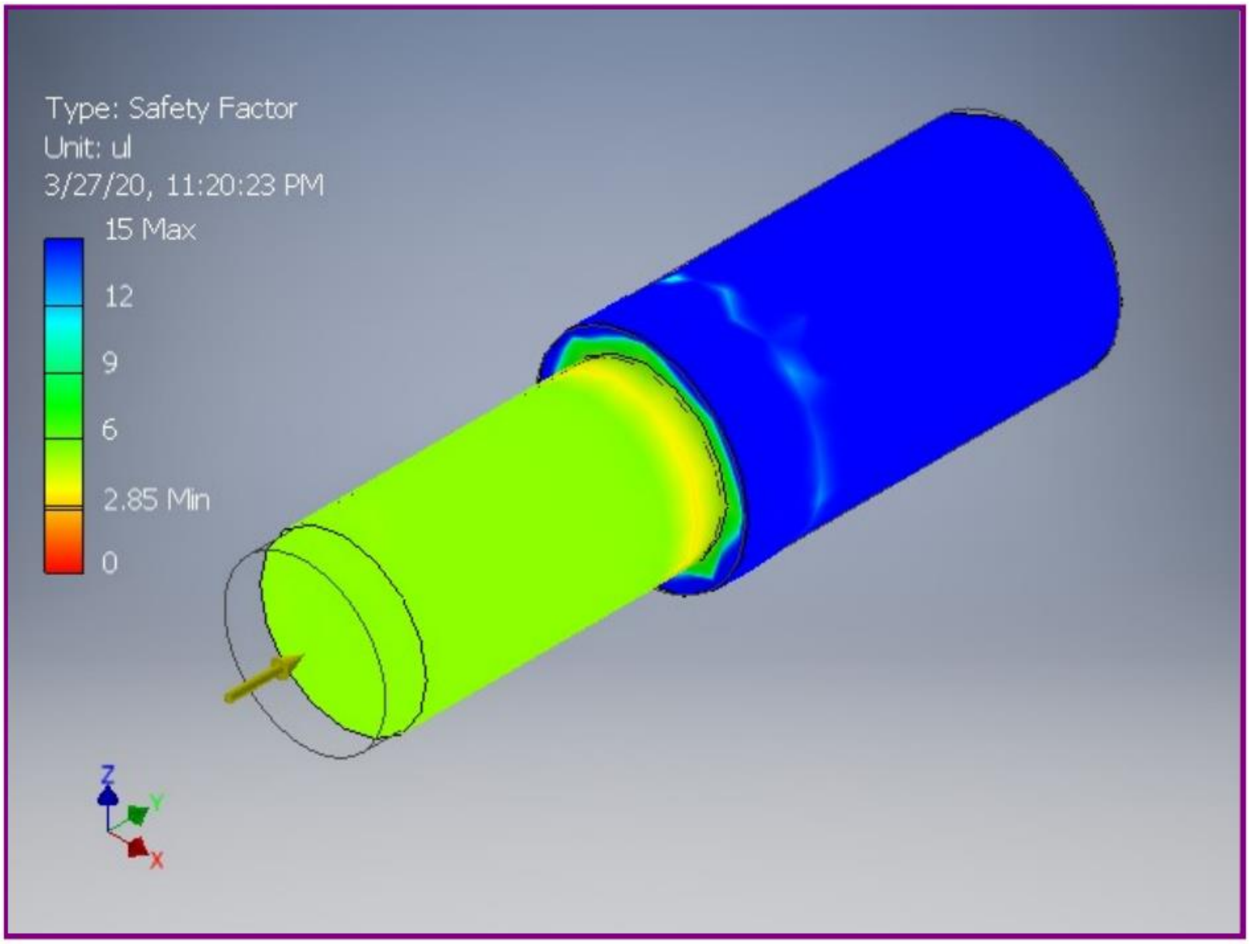
⊟ Results

⊟ Result Summary

Name	Minimum	Maximum
Volume	47302.4 mm^3	
Mass	0.37227 kg	
Safety Factor	2.84794 ul	15 ul

⊟ Figures

⊟ Safety Factor


Type: Safety Factor
Unit: ul
3/27/20, 11:20:23 PM
15 Max
12
9
6
2.85 Min
0
Z
Y
X

Stress Analysis - One

Analyzed File:	Cane Tip Stepped Shaft - One Inch.ipt
Autodesk Inventor Version:	2018 (Build 220112000, 112)
Creation Date:	3/27/20, 11:32 PM
Study Author:	Adam Genno

⊟ Project Info (iProperties)

⊟ Physical

Material	Steel AISI 1018 106 HR

Note: Physical values could be different from Physical values used by FEA reported below.

⊟ Static Analysis:1

General objective and settings:

Design Objective	Single Point
Study Type	Static Analysis
Last Modification Date	3/27/20, 11:31 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

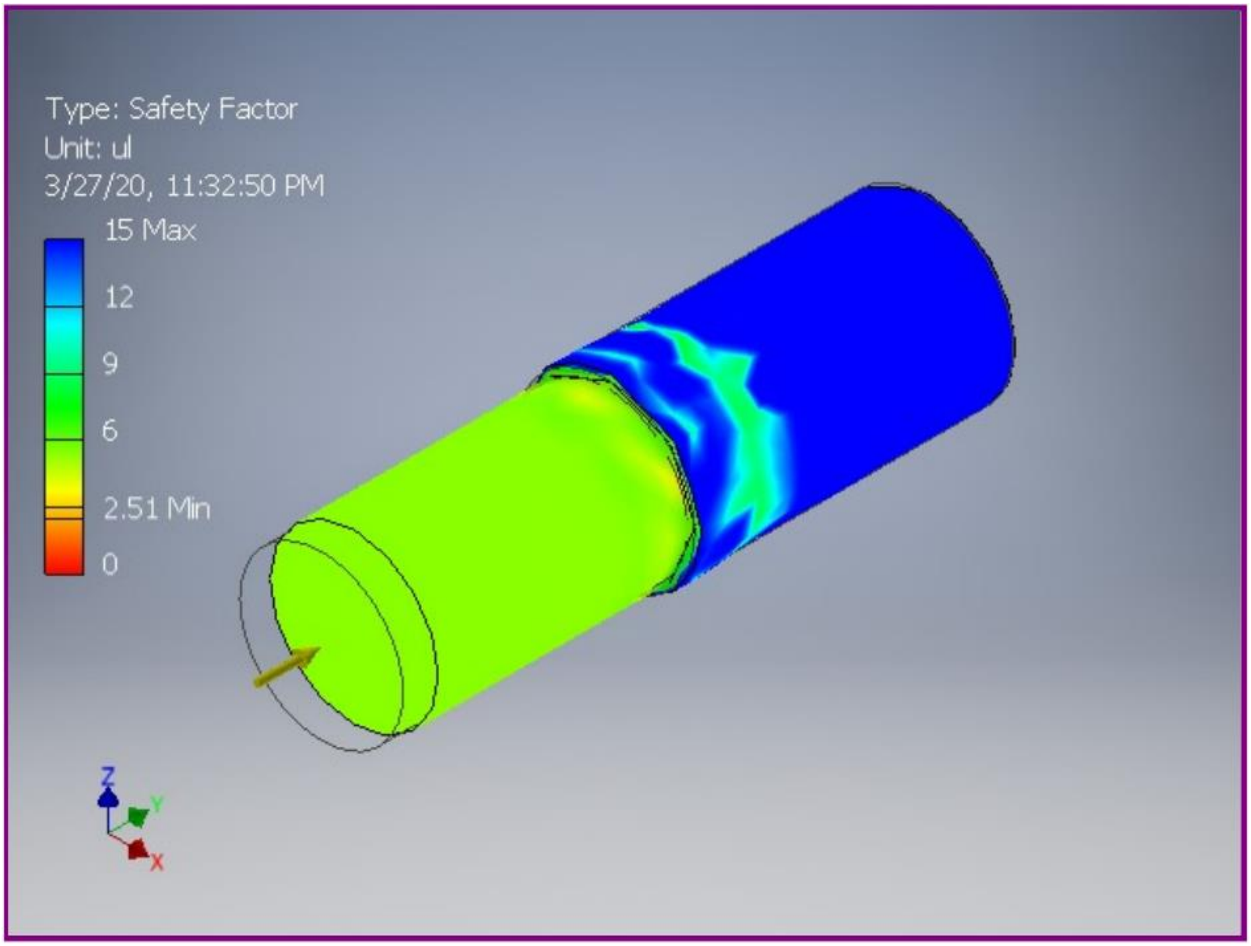
⊟ Results

⊟ Result Summary

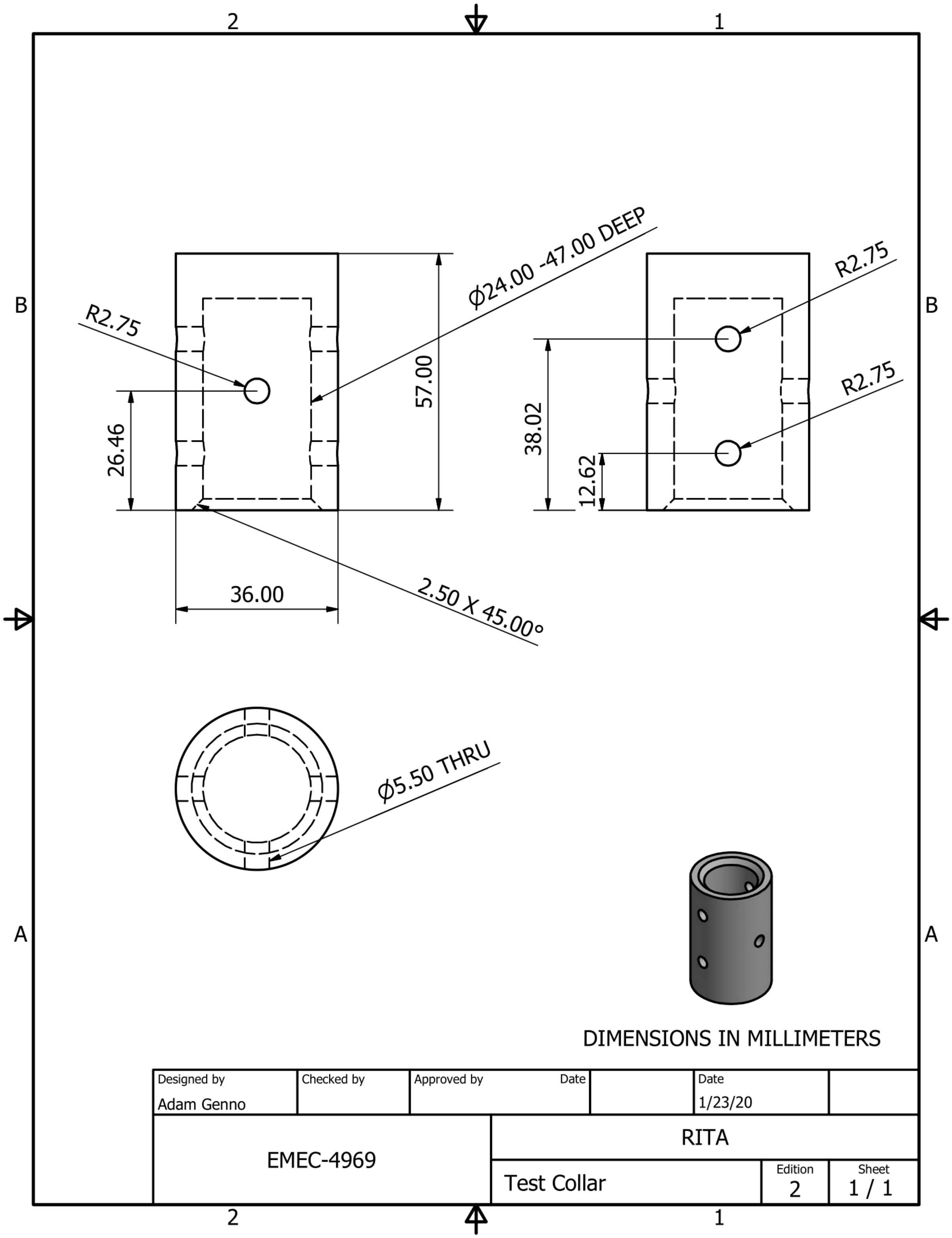
Name	Minimum	Maximum
Volume	57392.1 mm^3	
Mass	0.451676 kg	
Safety Factor	2.51064 ul	15 ul

⊟ Figures

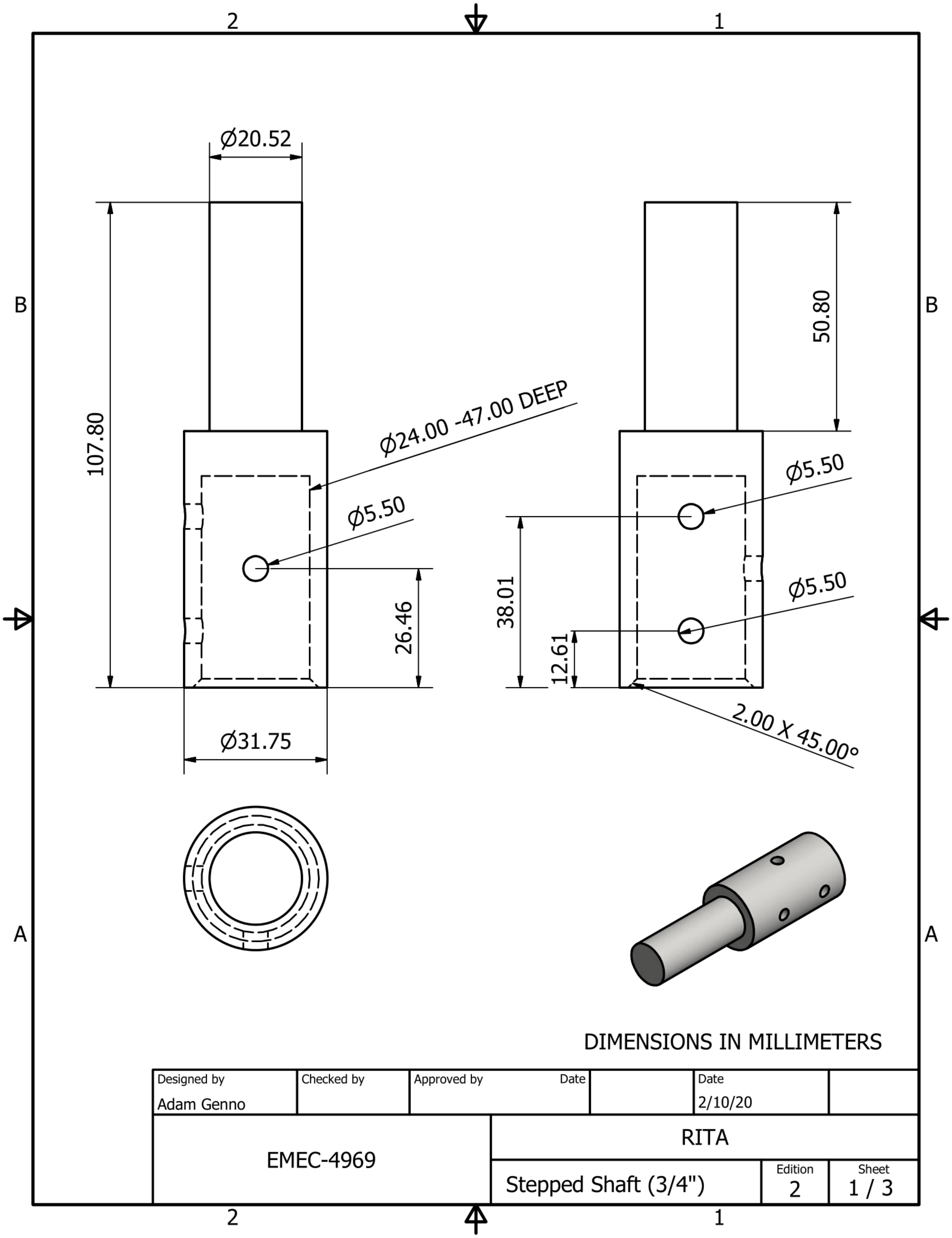
⊟ Safety Factor

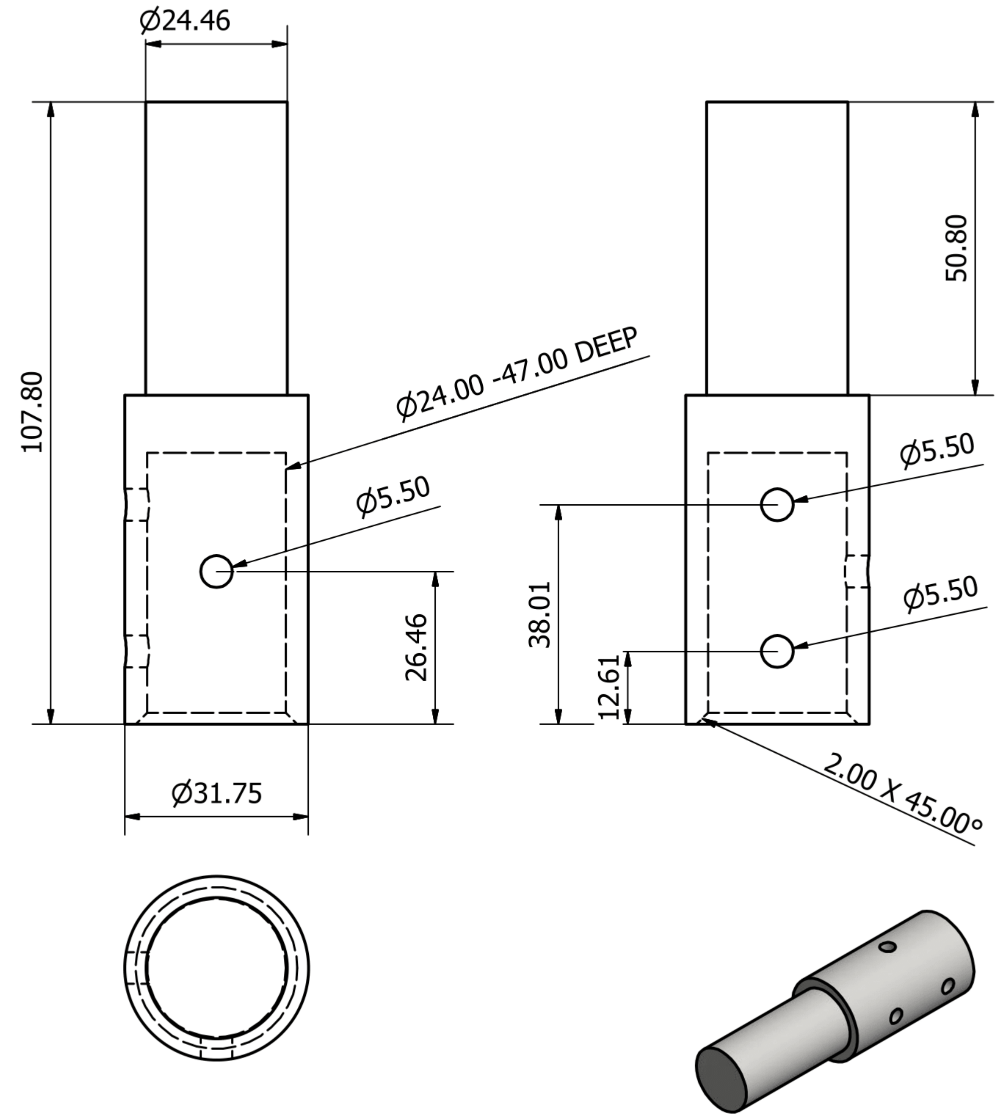

Type: Safety Factor
Unit: ul
3/27/20, 11:32:50 PM
15 Max
12
9
6
2.51 Min
0
Z
Y
X

Appendix E: Test Collar



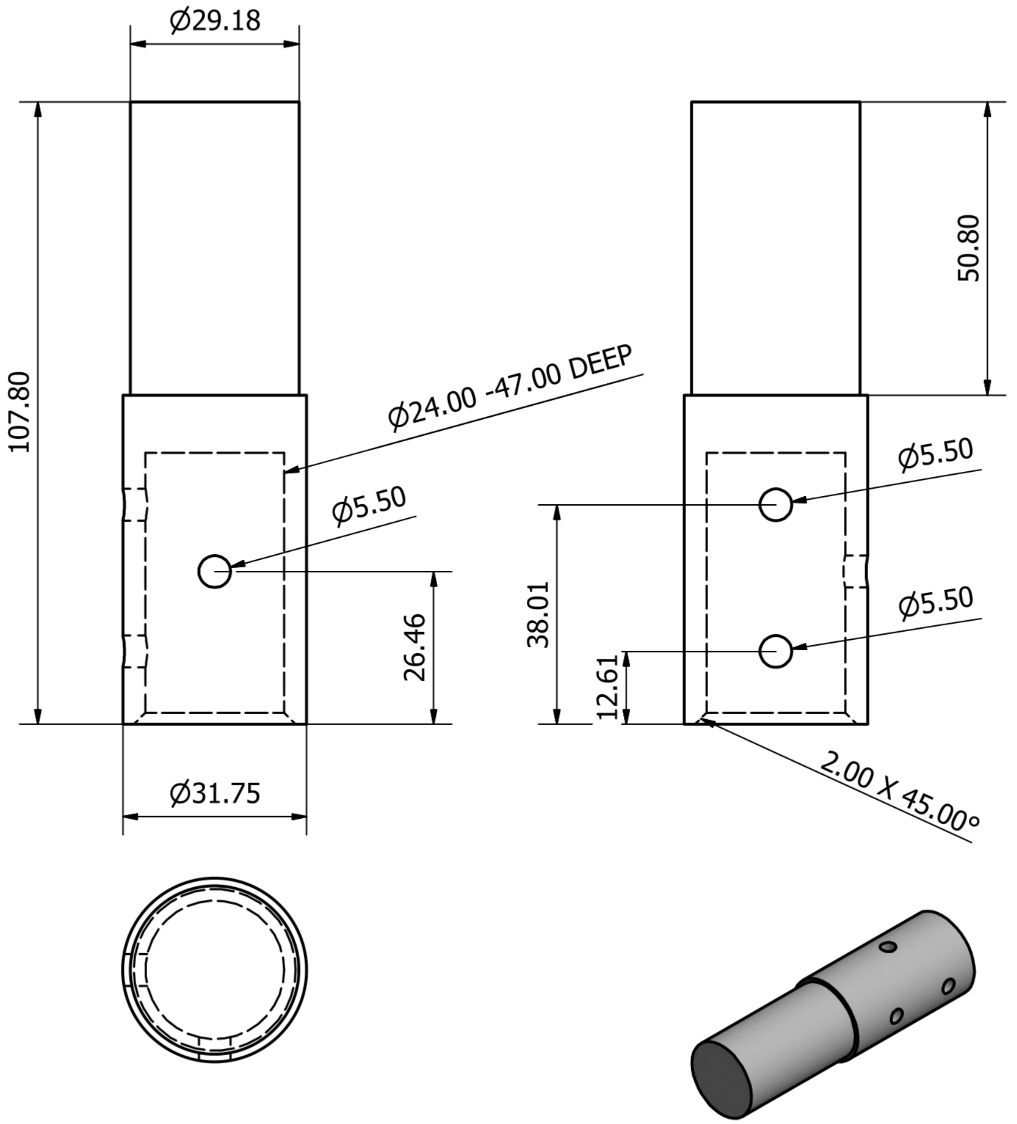
Appendix F: Cane Tip Stepped Shafts





DIMENSIONS IN MILLIMETERS

Designed by	Checked by	Approved by	Date	Date	
Adam Genno				2/10/20	
EMEC-4969	RITA				
	Stepped Shaft (7/8")			Edition 2	Sheet 2 / 3

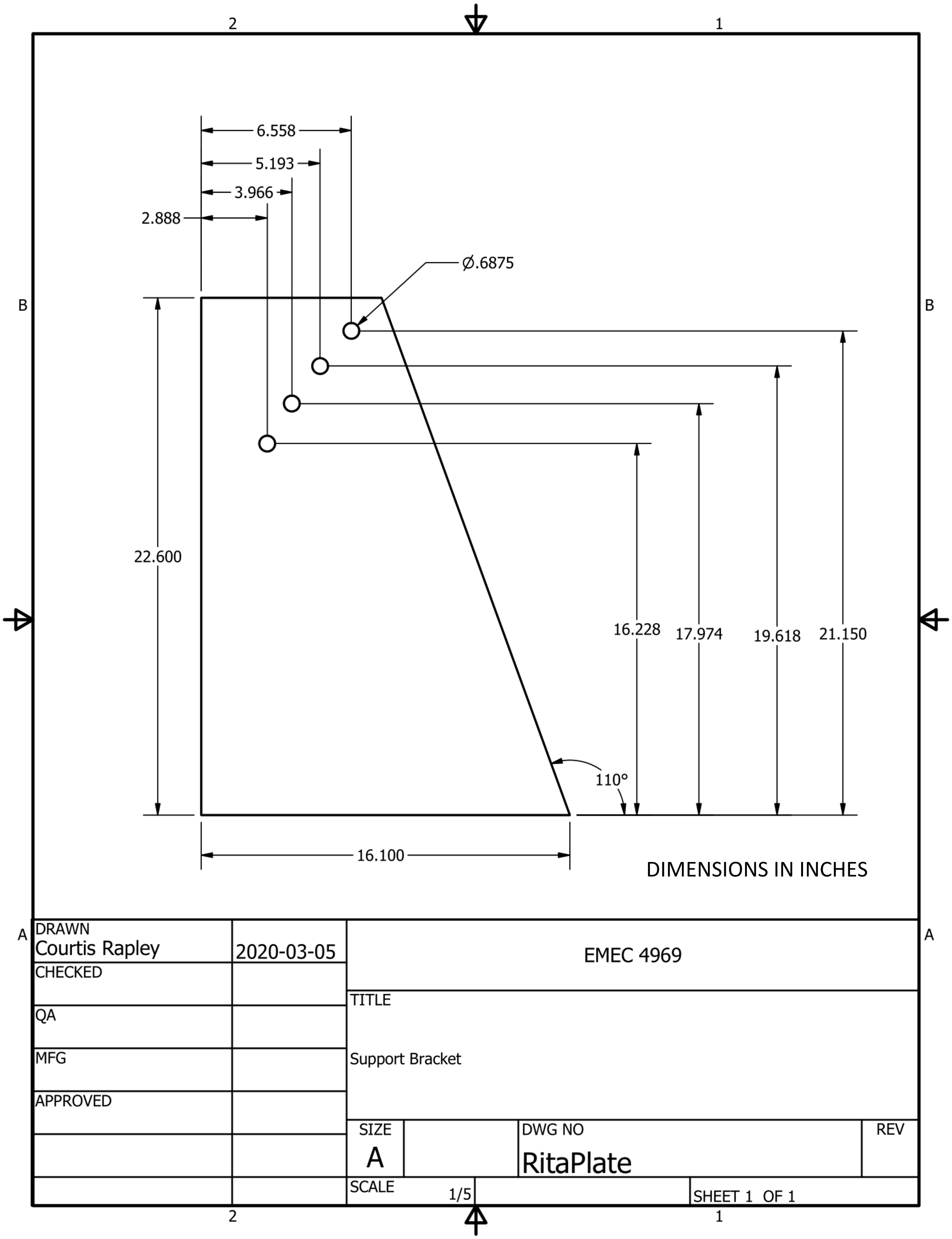


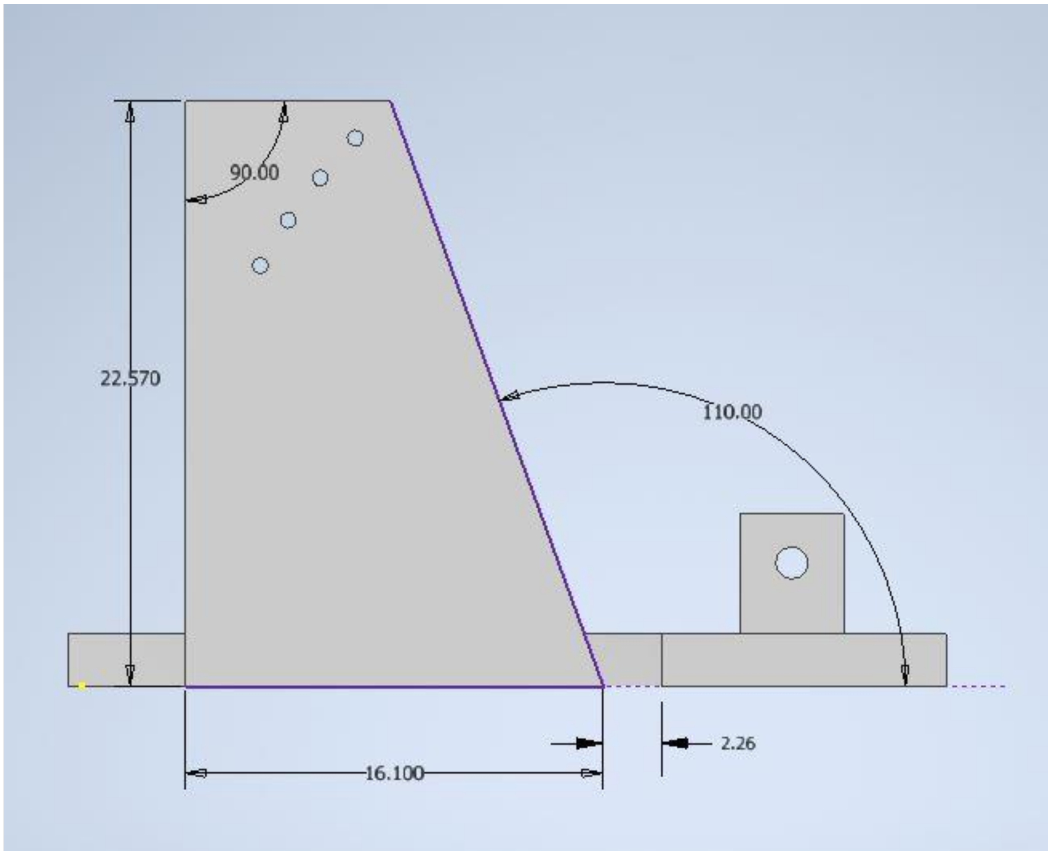
DIMENSIONS IN MILLIMETERS

Designed by	Checked by	Approved by	Date	Date
Adam Genno				2/10/20

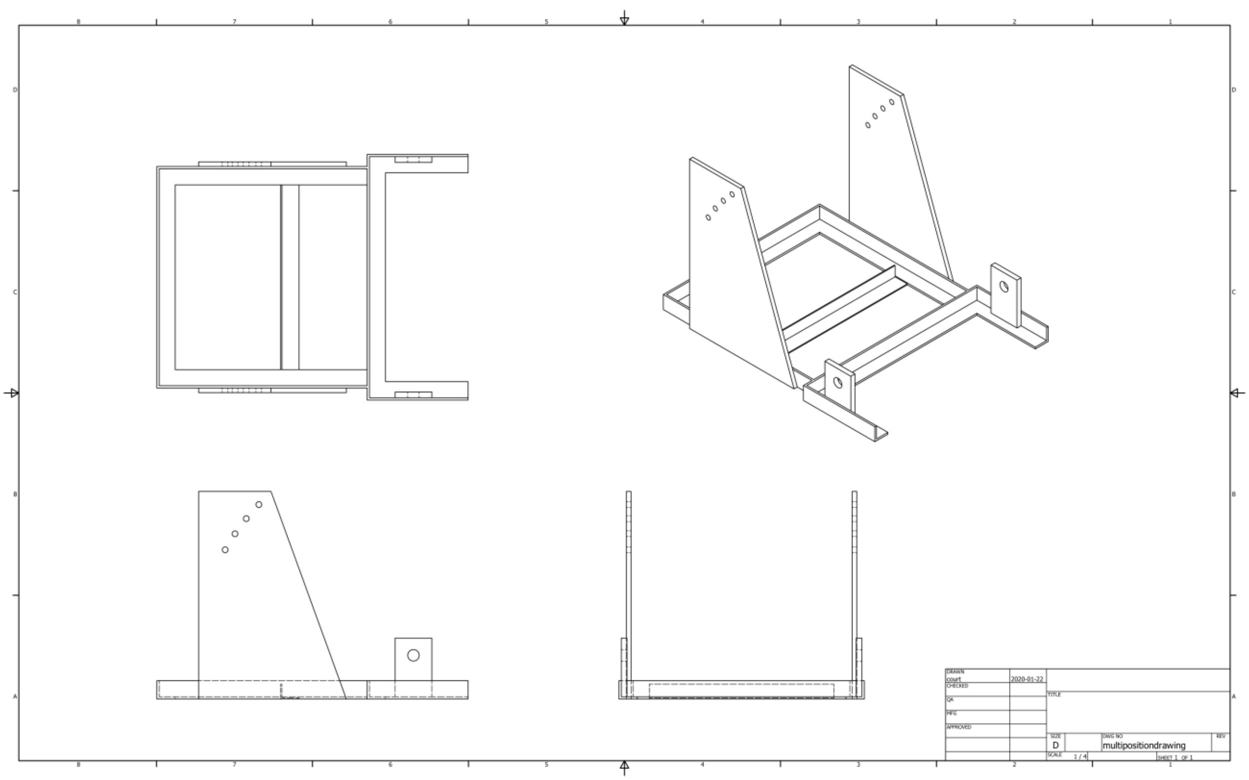
EMEC-4969	RITA		
	Stepped Shaft (1")	Edition 2	Sheet 3 / 3

Appendix G: Support Bracket



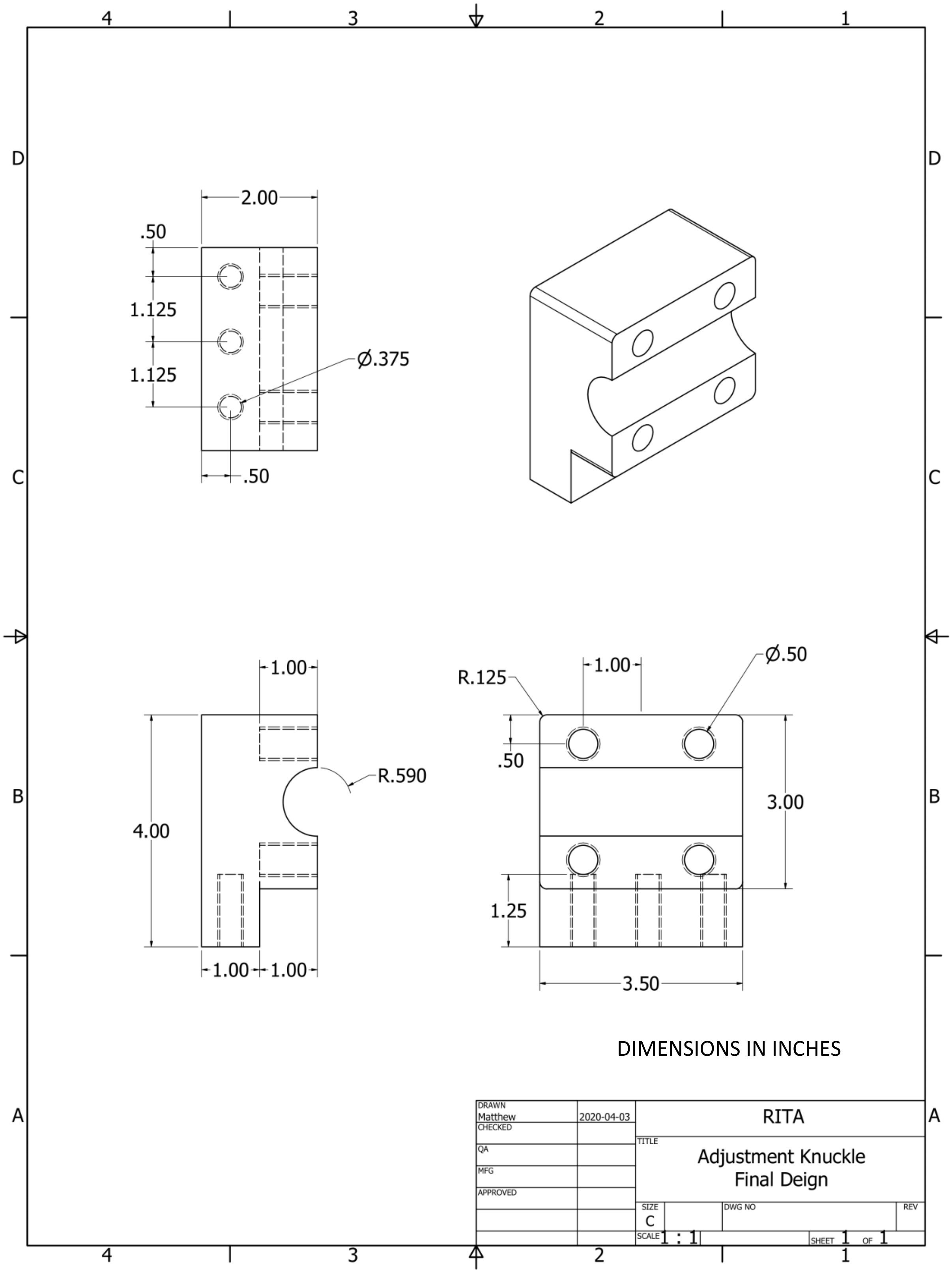


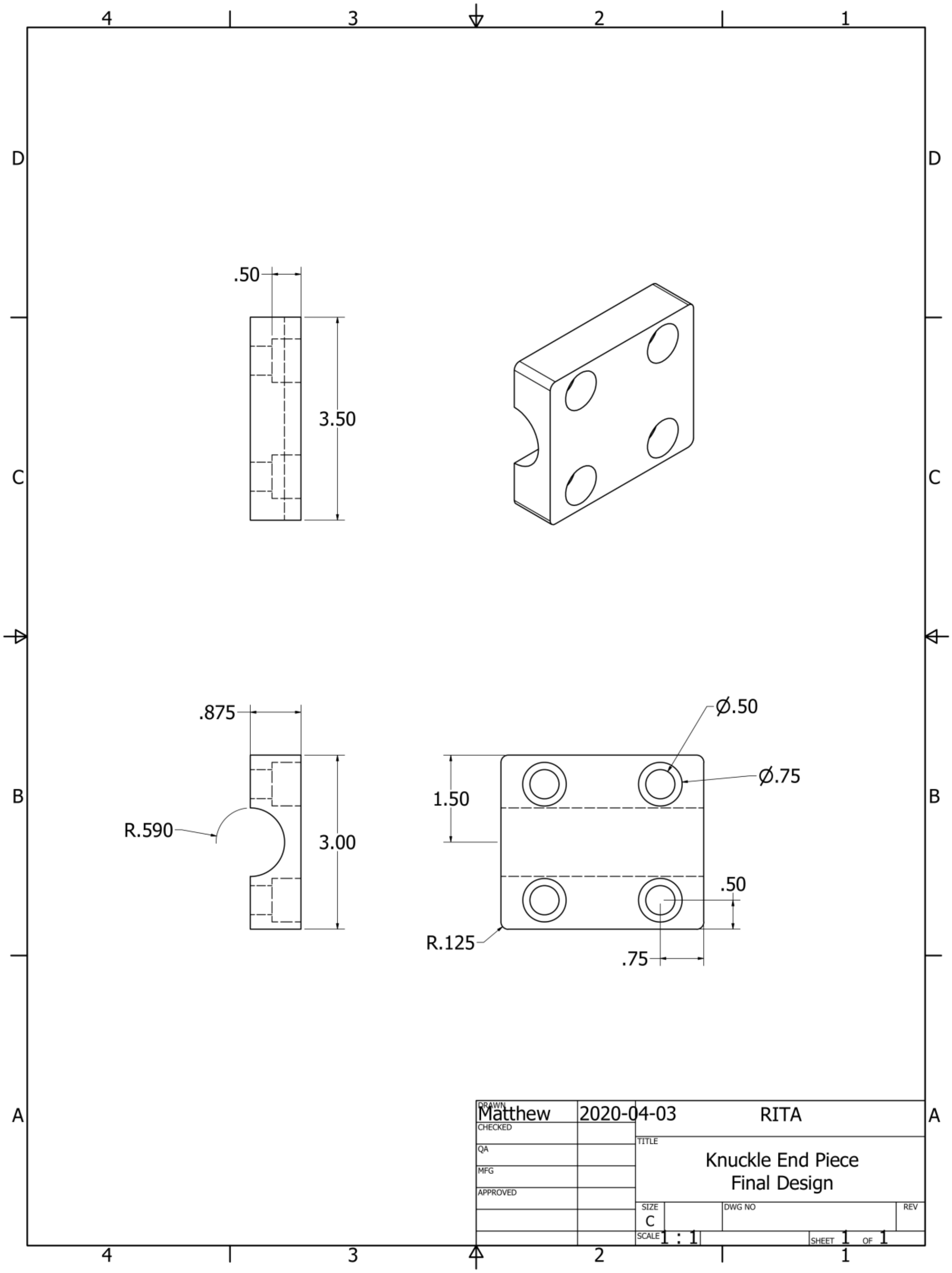
90.00
22.570
110.00
16.100
2.26



DRAWN
court
2020-01-22
CHECKED
TITLE
QA
MFG
APPROVED
SIZE
D
DWG NO
multipositiondrawing
REV
SCALE
1 / 4
SHEET 1 OF 1

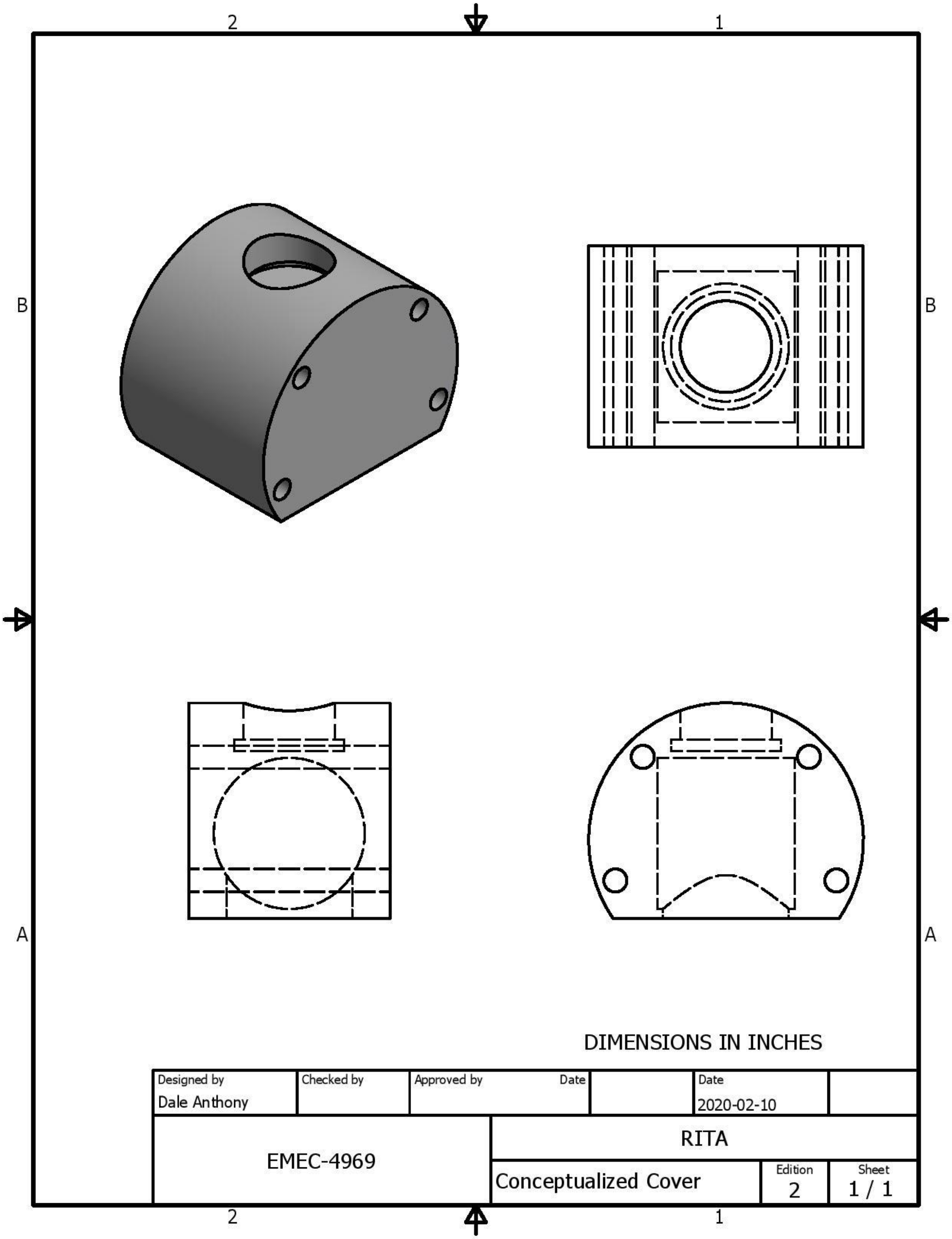
Appendix H: Adjustment Knuckle



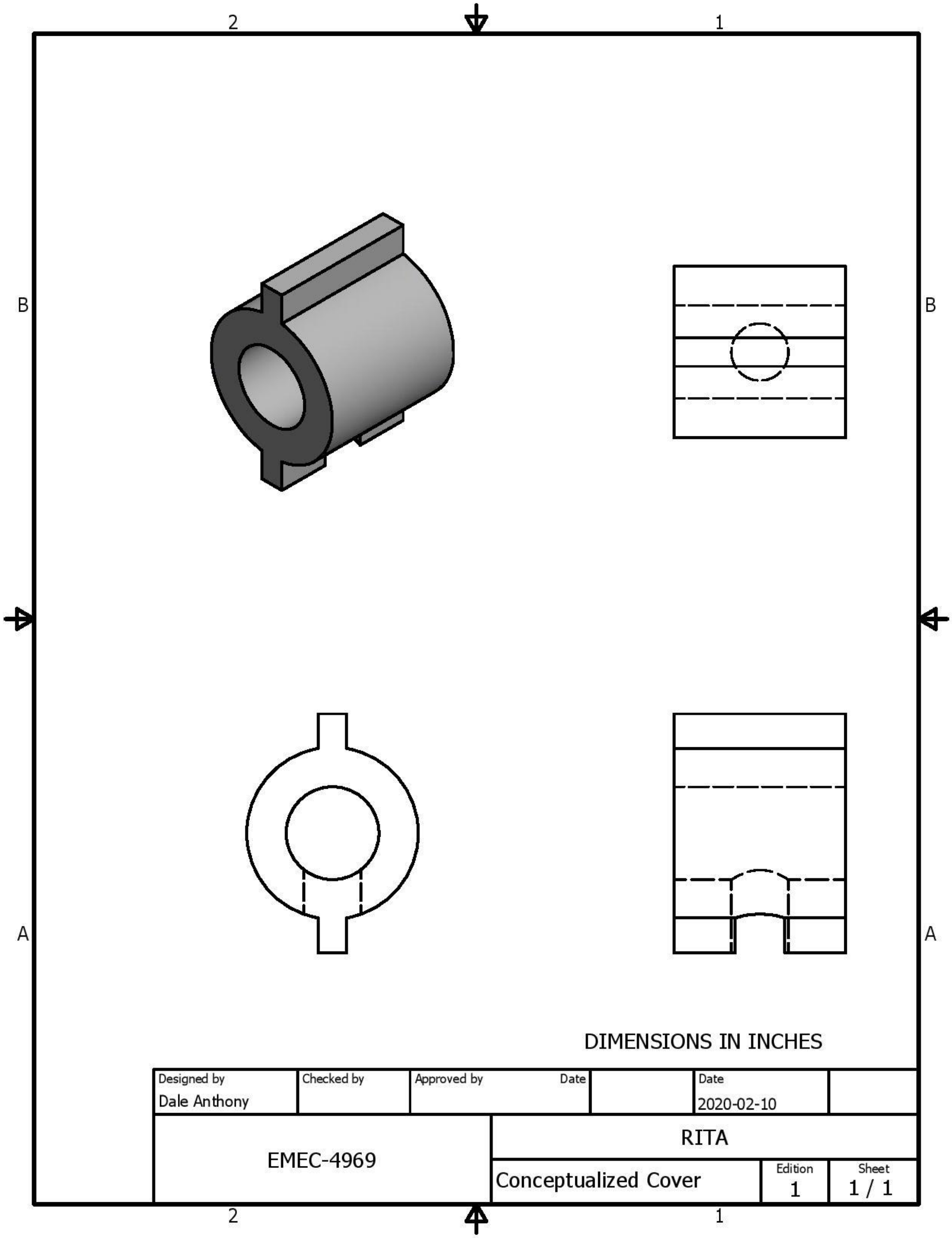


.50
3.50
.875
R.590
3.00
Ø.50
Ø.75
1.50
.50
R.125
.75
DRAWN Matthew 2020-04-03
CHECKED
QA
MFG
APPROVED
RITA
TITLE
Knuckle End Piece
Final Design
SIZE C
DWG NO
REV
SCALE 1 : 1
SHEET 1 OF 1

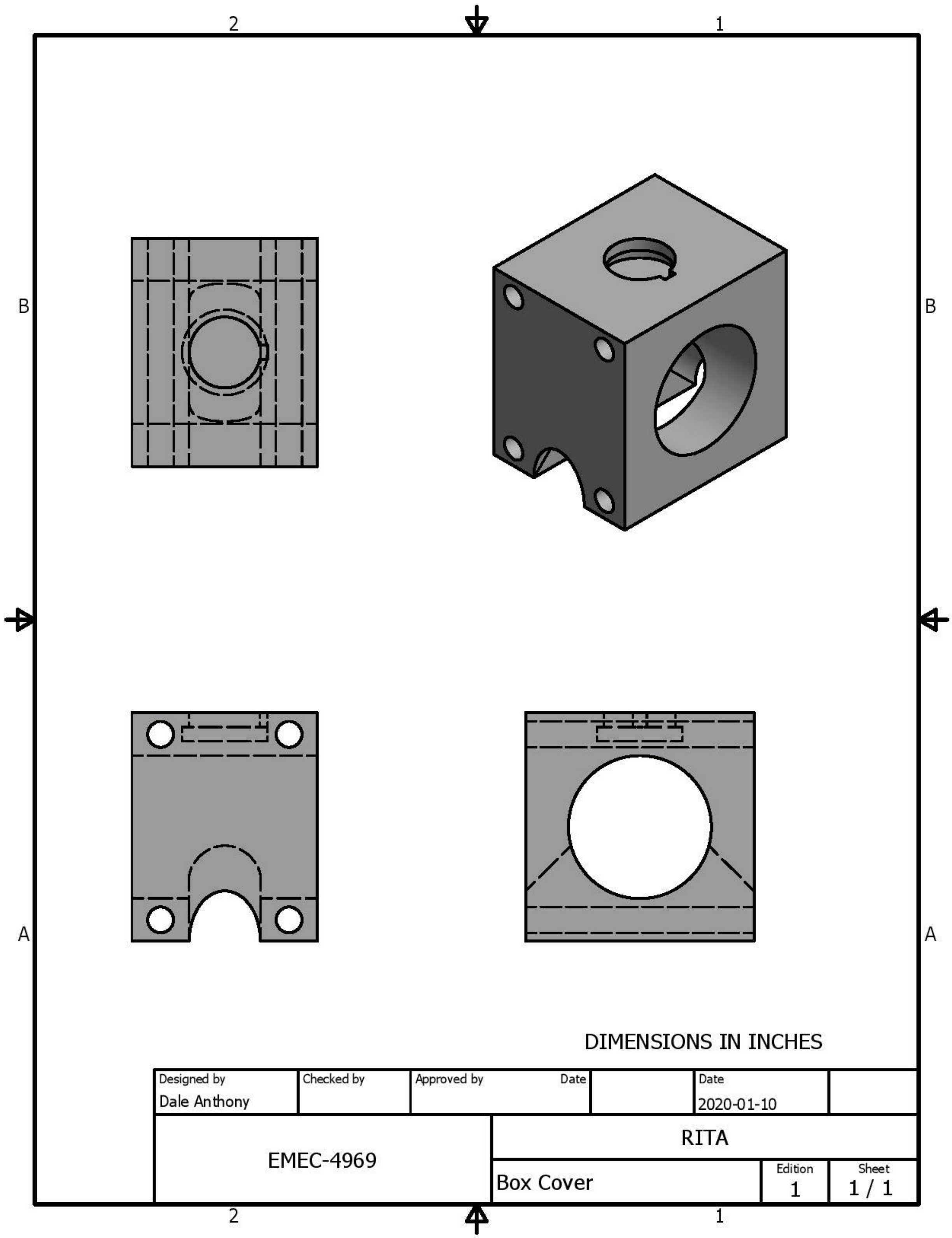
Appendix I: Impact Rod (Revision 1)

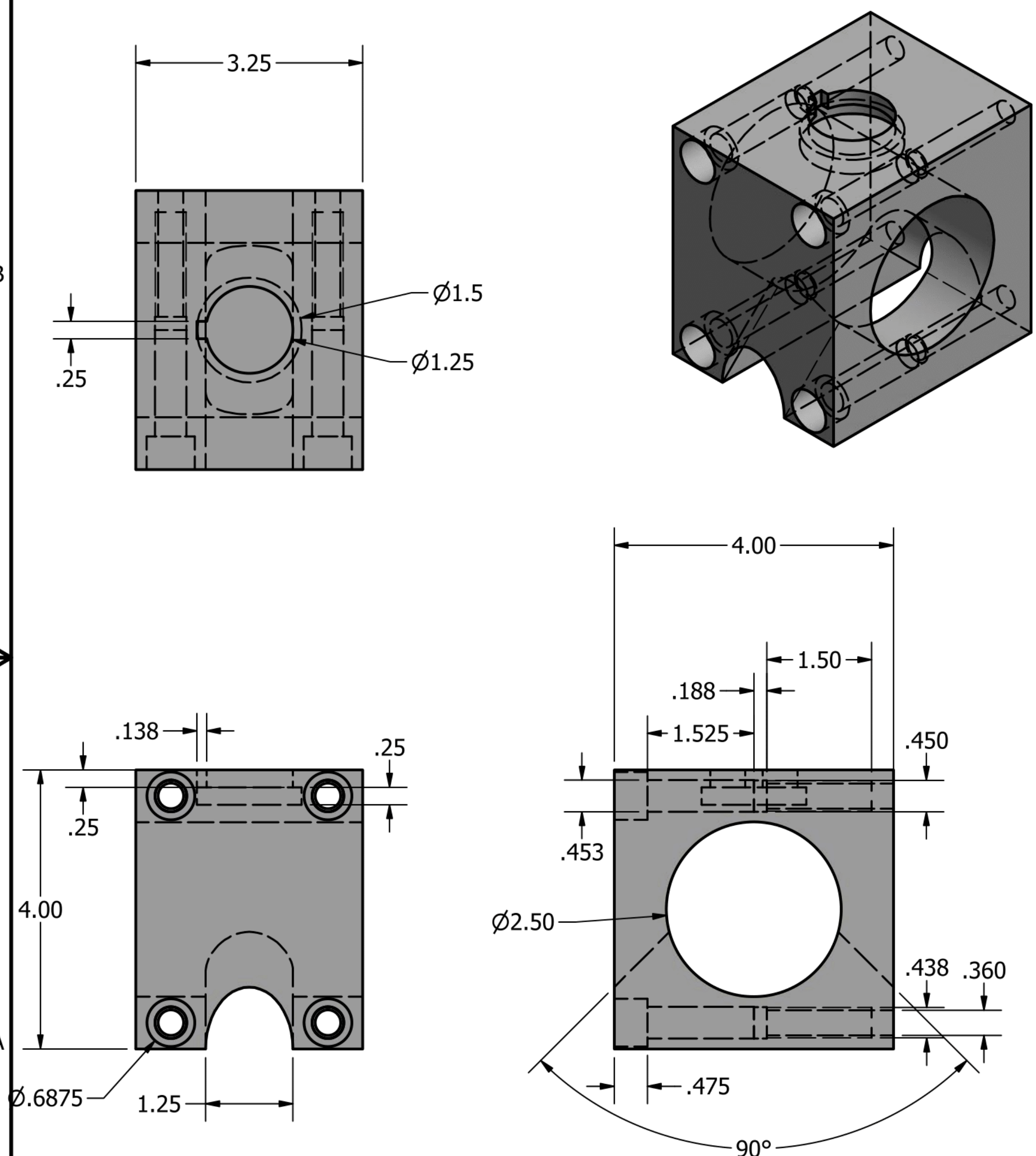


Appendix J: Impact Rod (Revision 2)



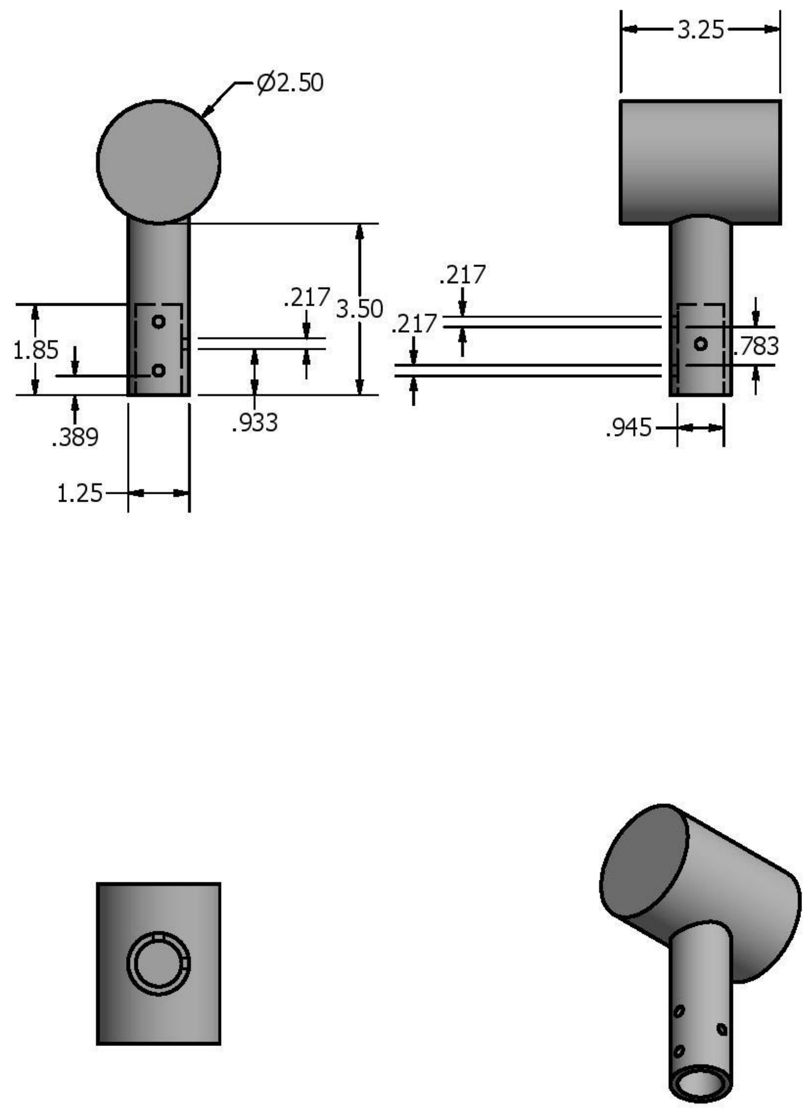
Appendix K: Impact Rod (Final)





DIMENSIONS IN INCHES

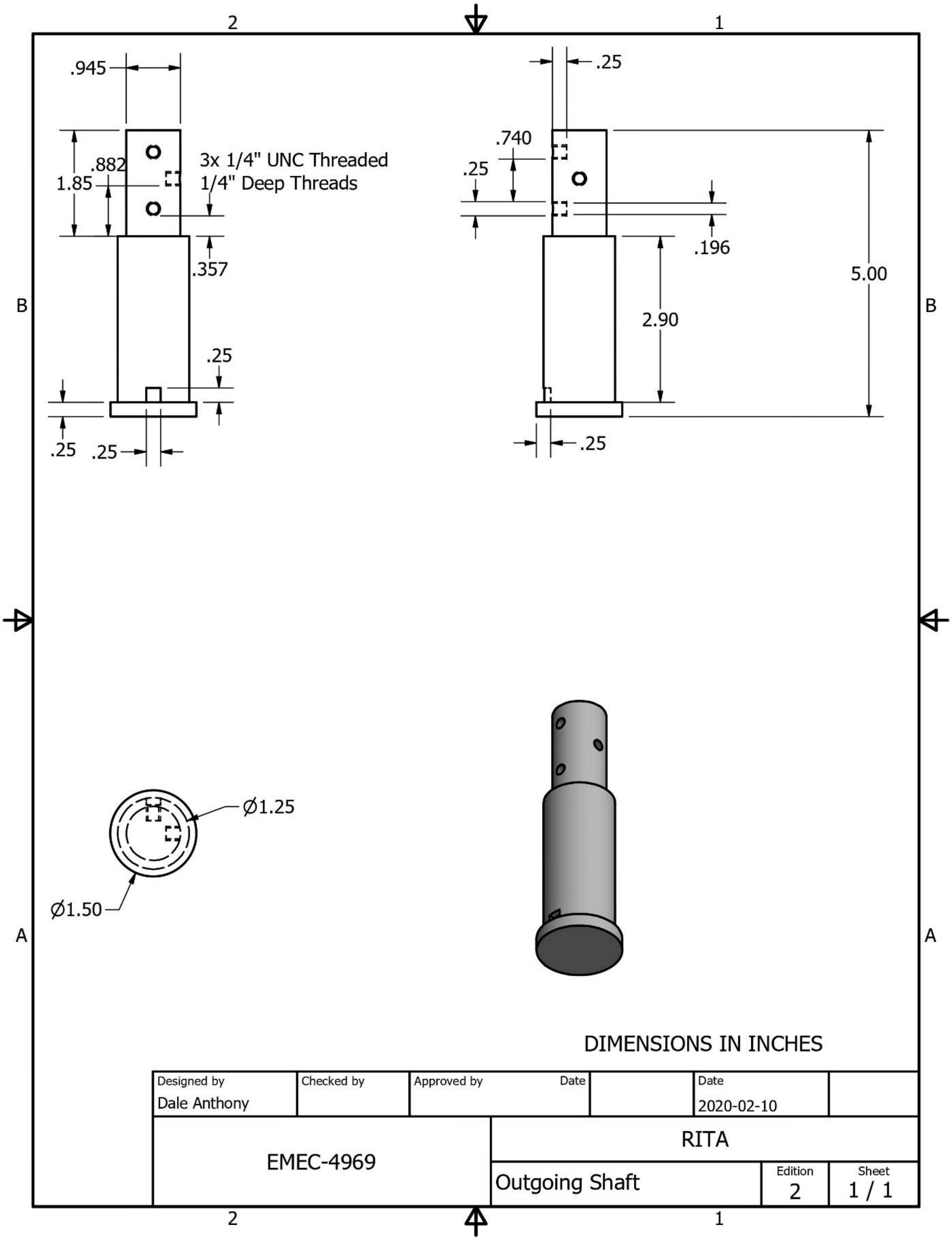
Designed by	Checked by	Approved by	Date	Date	
Dale Anthony				2020-02-10	
EMEC-4969	RITA				
	Cylinder Cover			Edition 3	Sheet 1 / 1



DIMENSIONS IN INCHES

Designed by	Checked by	Approved by	Date	Date	
Dale Anthony				2020-02-10	

EMEC-4969	RITA		
	Cylinder Shaft End	Edition 3	Sheet 1 / 1



.945
1.85
.882
3x 1/4" UNC Threaded
1/4" Deep Threads
.357
.25
.25
.25
.25
.740
.25
.196
5.00
2.90
.25
Ø1.25
Ø1.50
DIMENSIONS IN INCHES
Designed by
Dale Anthony
Checked by
Approved by
Date
Date
2020-02-10
EMEC-4969
RITA
Outgoing Shaft
Edition
2
Sheet
1 / 1

Appendix L: Interference Fit Test Program

```
% ###################################################################### %
%                              EMEC-4969-YA                              %
%                             Degree Project                             %
%                    Repetitive Impact Testing Apparatus                 %
% ###################################################################### %
%                      Interference Fit Test Program                     %
% ###################################################################### %

%initialize
clc
clear

%skip initial trial results (skip = 1, don't skip = 0)
skip = 0;

%%% MATERIAL PROPERTIES %%%
l = 1.625; %interior length of cane contact surface (walls only), in
D = 0.75; %interference diameter, in
t = 0.25; %wall thickness of cane tip, in
d_i = 0; %inner diameter of rod, in
d_o = D+(2*t); %outer diameter of cane tip, in
E_o = 0.3626*(10^3); %modulus of cane tip, psi
v_o = 0.50; %poissons ratio of cane tip
E_i = 29700*(10^3); %modulus of rod, psi UPDATED: 2/10/20
v_i = 0.29; %poissons ratio of rod
mu = 0.64; %dynamic friction coefficient between rod and cane tip
A_s = pi * D * l; %contact surface area between rod and cane tip, in^2

if (skip == 0)
fprintf('### Natural Rubber ###\n')
%%% USING RECOMMENDED FORCE %%%
F = 42; %force used to apply cane tip to shaft, lb
p = F/(mu*A_s); %calculate interference pressure, psi
% Calculate diametral interference, in
delta = p*(D*((1/E_o)*((((d_o^2)+(D^2))/((d_o^2)-
(D^2)))+v_o)+(1/E_i)*((((D^2)+(d_i^2))/((D^2)-(d_i^2)))-v_i)));
fprintf('Using F = %0.f lbf \n', F)
fprintf('Using D = %0.3f in \n', D)
fprintf('Diametral interference is %f in \n', delta)
fprintf('Outer Diameter should be: %f in \n', D+delta)
%%% USING HIGHER THAN RECOMMENDED FORCE %%%
F = 50; %force used to apply cane tip to shaft, lb
p = F/(mu*A_s);%calculate interference pressure, psi
% Calculate diametral interference, in
delta = p*(D*((1/E_o)*((((d_o^2)+(D^2))/((d_o^2)-
(D^2)))+v_o)+(1/E_i)*((((D^2)+(d_i^2))/((D^2)-(d_i^2)))-v_i)));
fprintf('Using F = %0.f lbf \n', F)
fprintf('Using D = %0.3f in \n', D)
fprintf('Diametral interference is %f in \n', delta)
fprintf('Outer Diameter should be: %f in \n', D+delta)
%%% USING MAXIMUM ACCELERATION %%%
F = 15443.5; %force used to apply cane tip to shaft, lb
p = F/(mu*A_s); %calculate interference pressure, psi
```

```
% Calculate diametral interference, in
delta = p*(D*((1/E_o)*((((d_o^2)+(D^2))/((d_o^2)-
(D^2)))+v_o)+(1/E_i)*((((D^2)+(d_i^2))/((D^2)-(d_i^2)))-v_i)));
fprintf('Using F = %0.f lbf \n', F)
fprintf('Using D = %0.3f in \n', D)
fprintf('Diametral interference is %f in \n', delta)
fprintf('Outer Diameter should be: %f in \n', D+delta)
end

if (skip == 0)
fprintf('\n')
fprintf('### Butyl Rubber ###\n')
%%% USING UPDATED MATERIAL PROPERTIES (11/1/2019) %%%
E_o = 0.1041*(10^3); %modulus of cane tip, psi
F = 50; %force used to apply cane tip to shaft, lb
p = F/(mu*A_s); %calculate interference pressure, psi
% Calculate diametral interference, in
delta = p*(D*((1/E_o)*((((d_o^2)+(D^2))/((d_o^2)-
(D^2)))+v_o)+(1/E_i)*((((D^2)+(d_i^2))/((D^2)-(d_i^2)))-v_i)));
fprintf('Using F = %0.f lbf \n', F)
fprintf('Using D = %0.3f in \n', D)
fprintf('Diametral interference is %f in \n', delta)
fprintf('Outer Diameter should be: %f in \n', D+delta)
end

fprintf('\n')
fprintf('### Tested Cane Tip Properties ###\n')
D = 0.5; %for 1/2" cane tip
%%% USING UPDATED MATERIAL PROPERTIES (1/21/2020) %%%
E_o = 693.935; %modulus of cane tip, psi
F = 50; %force used to apply cane tip to shaft, lb
p = F/(mu*A_s); %calculate interference pressure, psi
% Calculate diametral interference, in
delta = p*(D*((1/E_o)*((((d_o^2)+(D^2))/((d_o^2)-
(D^2)))+v_o)+(1/E_i)*((((D^2)+(d_i^2))/((D^2)-(d_i^2)))-v_i)));
fprintf('Using F = %0.f lbf \n', F)
fprintf('Using D = %0.3f in \n', D)
fprintf('Diametral interference is %f in \n', delta)
fprintf('Outer Diameter should be: %f in \n', D+delta)
D = 0.625; %for 5/8" cane tip
E_o = 693.935; %modulus of cane tip, psi
F = 50; %force used to apply cane tip to shaft, lb
p = F/(mu*A_s); %calculate interference pressure, psi
% Calculate diametral interference, in
delta = p*(D*((1/E_o)*((((d_o^2)+(D^2))/((d_o^2)-
(D^2)))+v_o)+(1/E_i)*((((D^2)+(d_i^2))/((D^2)-(d_i^2)))-v_i)));
fprintf('Using F = %0.f lbf \n', F)
fprintf('Using D = %0.3f in \n', D)
fprintf('Diametral interference is %f in \n', delta)
fprintf('Outer Diameter should be: %f in \n', D+delta)
D = 0.75; %for 3/4" cane tip
%%% USING UPDATED MATERIAL PROPERTIES (1/21/2020) %%%
E_o = 693.935; %modulus of cane tip, psi
F = 50; %force used to apply cane tip to shaft, lb
p = F/(mu*A_s); %calculate interference pressure, psi
% Calculate diametral interference, in
```

```
delta = p*(D*((1/E_o)*((((d_o^2)+(D^2))/((d_o^2)-
(D^2)))+v_o)+(1/E_i)*((((D^2)+(d_i^2))/((D^2)-(d_i^2)))-v_i)));
fprintf('Using F = %0.f lbf \n', F)
fprintf('Using D = %0.3f in \n', D)
fprintf('Diametral interference is %f in \n', delta)
fprintf('Outer Diameter should be: %f in \n', D+delta)
D = 0.875; %for 7/8" cane tip
E_o = 693.935; %modulus of cane tip, psi
F = 50; %force used to apply cane tip to shaft, lb
p = F/(mu*A_s); %calculate interference pressure, psi
% Calculate diametral interference, in
delta = p*(D*((1/E_o)*((((d_o^2)+(D^2))/((d_o^2)-
(D^2)))+v_o)+(1/E_i)*((((D^2)+(d_i^2))/((D^2)-(d_i^2)))-v_i)));
fprintf('Using F = %0.f lbf \n', F)
fprintf('Using D = %0.3f in \n', D)
fprintf('Diametral interference is %f in \n', delta)
fprintf('Outer Diameter should be: %f in \n', D+delta)
D = 1; %for 1" cane tip
E_o = 693.935; %modulus of cane tip, psi
F = 50; %force used to apply cane tip to shaft, lb
p = F/(mu*A_s); %calculate interference pressure, psi
% Calculate diametral interference, in
delta = p*(D*((1/E_o)*((((d_o^2)+(D^2))/((d_o^2)-
(D^2)))+v_o)+(1/E_i)*((((D^2)+(d_i^2))/((D^2)-(d_i^2)))-v_i)));
fprintf('Using F = %0.f lbf \n', F)
fprintf('Using D = %0.3f in \n', D)
fprintf('Diametral interference is %f in \n', delta)
fprintf('Outer Diameter should be: %f in \n', D+delta)
```

Program output:

```
### Natural Rubber ###
Using F = 42 lbf
Using D = 0.750 in
Diametral interference is 0.093061 in
Outer Diameter should be: 0.843061 in
Using F = 50 lbf
Using D = 0.750 in
Diametral interference is 0.110787 in
Outer Diameter should be: 0.860787 in
Using F = 15444 lbf
Using D = 0.750 in
Diametral interference is 34.218854 in
Outer Diameter should be: 34.968854 in

### Butyl Rubber ###
Using F = 50 lbf
Using D = 0.750 in
Diametral interference is 0.385892 in
Outer Diameter should be: 1.135892 in

### Tested Cane Tip Properties ###
Using F = 50 lbf
```

```
Using D = 0.500 in
Diametral interference is 0.027654 in
Outer Diameter should be: 0.527654 in
Using F = 50 lbf
Using D = 0.625 in
Diametral interference is 0.039818 in
Outer Diameter should be: 0.664818 in
Using F = 50 lbf
Using D = 0.750 in
Diametral interference is 0.057890 in
Outer Diameter should be: 0.807890 in
Using F = 50 lbf
Using D = 0.875 in
Diametral interference is 0.088032 in
Outer Diameter should be: 0.963032 in
Using F = 50 lbf
Using D = 1.000 in
Diametral interference is 0.148654 in
Outer Diameter should be: 1.148654 in
>>
```

Appendix M: Shaft Buckling Test Program

```
% ####################################################################### %
%                               EMEC-4969-YA                              %
%                              Degree Project                             %
%                    Repetitive Impact Testing Apparatus                  %
% ####################################################################### %
%                          Shaft Buckling Test Program                    %
% ####################################################################### %

%initialize
clc
clear

%%% MATERIAL PROPERTIES %%%
sigma_yp = 41.1*(10^3); %yield stress, psi
n = 1; %end coefficient condition
E_i = 29700*(10^3); %modulus of elasticity of rod, psi UPDATED: 2/10/20
max_load = 10116.4; %lbf

fprintf('### Narrow section (1/2" Cane Tip) ### \n')
%%% NARROW SECTION %%%
l = 2; %length of buckling section, in
D = 0.527654; %outer diameter, in
B = ((sigma_yp)*l)/(n*(pi^2)*E_i); %Check is J.B. Johnson is justified
rho = (D/4);
if ((B/(rho^2)) < 2)
    fprintf("Johnson Justified \n")
    F = (pi/4)*(D^2)*sigma_yp*(1-(B/(4*(rho^2))));
    fprintf("For narrow section: D = %0.3f in \n", D)
    fprintf("Critical load is: %0.3f lbf \n", F)
    if (F < max_load)
        fprintf("### WARNING: Critical load exceeds design load, D = %0.3f in
NOT SUITABLE ### \n", D)
    end
    fprintf("\n")
else
    fprintf("Euler formula must be used")
    fprintf("\n")
end

fprintf('### Narrow section (5/8" Cane Tip) ### \n')
%%% NARROW SECTION %%%
l = 2; %length of buckling section, in
D = 0.664818; %outer diameter, in
B = ((sigma_yp)*l)/(n*(pi^2)*E_i); %Check is J.B. Johnson is justified
rho = (D/4);
if ((B/(rho^2)) < 2)
    fprintf("Johnson Justified \n")
    F = (pi/4)*(D^2)*sigma_yp*(1-(B/(4*(rho^2))));
    fprintf("For narrow section: D = %0.3f in \n", D)
    fprintf("Critical load is: %0.3f lbf \n", F)
    if (F < max_load)
        fprintf("### WARNING: Critical load exceeds design load, D = %0.3f in
NOT SUITABLE ### \n", D)
```

```
    end
    fprintf("\n")
else
    fprintf("Euler formula must be used")
    fprintf("\n")
end

fprintf('### Narrow section (3/4" Cane Tip) ### \n')
%%% NARROW SECTION %%%
l = 2; %length of buckling section, in
D = 0.807890; %outer diameter, in
B = ((sigma_yp)*l)/(n*(pi^2)*E_i); %Check is J.B. Johnson is justified
rho = (D/4);
if ((B/(rho^2)) < 2)
    fprintf("Johnson Justified \n")
    F = (pi/4)*(D^2)*sigma_yp*(1-(B/(4*(rho^2))));
    fprintf("For narrow section: D = %0.3f in \n", D)
    fprintf("Critical load is: %0.3f lbf \n", F)
    fprintf("\n")
else
    fprintf("Euler formula must be used")
    fprintf("\n")
end
if (F < max_load)
    fprintf("### Critical load exceed design load, D = %0.3f in NOT SUITABLE
### \n", D)
end

fprintf('### Narrow section (7/8" Cane Tip) ### \n')
%%% NARROW SECTION %%%
l = 2; %length of buckling section, in
D = 0.963032; %outer diameter, in
B = ((sigma_yp)*l)/(n*(pi^2)*E_i); %Check is J.B. Johnson is justified
rho = (D/4);
if ((B/(rho^2)) < 2)
    fprintf("Johnson Justified \n")
    F = (pi/4)*(D^2)*sigma_yp*(1-(B/(4*(rho^2))));
    fprintf("For narrow section: D = %0.3f in \n", D)
    fprintf("Critical load is: %0.3f lbf \n", F)
    fprintf("\n")
else
    fprintf("Euler formula must be used")
    fprintf("\n")
end
if (F < max_load)
    fprintf("### Critical load exceed design load, D = %0.3f in NOT SUITABLE
### \n", D)
end

fprintf('### Narrow section (1" Cane Tip) ### \n')
%%% NARROW SECTION %%%
l = 2; %length of buckling section, in
D = 1.148654; %outer diameter, in
B = ((sigma_yp)*l)/(n*(pi^2)*E_i); %Check is J.B. Johnson is justified
rho = (D/4);
if ((B/(rho^2)) < 2)
```

```
    fprintf("Johnson Justified \n")
    F = (pi/4)*(D^2)*sigma_yp*(1-(B/(4*(rho^2))));
    fprintf("For narrow section: D = %0.3f in \n", D)
    fprintf("Critical load is: %0.3f lbf \n", F)
    if (F < max_load)
        fprintf("### WARNING: Critical load exceeds design load, D = %0.3f in
NOT SUITABLE ### \n", D)
    end
    fprintf("\n")
else
    fprintf("Euler formula must be used")
    fprintf("\n")
end

fprintf('### Hollow section ### \n')
%%% HOLLOW SECTION %%%
l = 1.75; %length of buckling section, in
D = 1.25; %outer diameter, in
d = 0.945; % inner diameter, in
B = ((sigma_yp)*l)/(n*(pi^2)*E_i); %Check is J.B. Johnson is justified
rho = (sqrt((D^2)+(d^2)))/4;
if ((B/(rho^2)) < 2)
    fprintf("Johnson Justified \n")
    F = (((pi/4)*(D^2))-((pi/4)*(d^2)))*sigma_yp*(1-(B/(4*(rho^2))));
    fprintf("Critical load is: %0.3f lbf \n", F)
    if (F < max_load)
        fprintf("### WARNING: Critical load exceeds design load, D = %0.3f in
NOT SUITABLE ### \n", D)
    end
    fprintf("\n")
else
    fprintf("Euler formula must be used")
    fprintf("\n")
end
```

Program Output:

```
### Narrow section (1/2" Cane Tip) ###
Johnson Justified
For narrow section: D = 0.528 in
Critical load is: 8951.111 lbf
### WARNING: Critical load exceeds design load, D = 0.528 in NOT
SUITABLE ###

### Narrow section (5/8" Cane Tip) ###
Johnson Justified
For narrow section: D = 0.665 in
Critical load is: 14230.942 lbf

### Narrow section (3/4" Cane Tip) ###
Johnson Justified
For narrow section: D = 0.808 in
```

```
Critical load is: 21032.416 lbf

### Narrow section (7/8" Cane Tip) ###
Johnson Justified
For narrow section: D = 0.963 in
Critical load is: 29901.127 lbf

### Narrow section (1" Cane Tip) ###
Johnson Justified
For narrow section: D = 1.149 in
Critical load is: 42554.039 lbf

### Hollow section ###
Johnson Justified
Critical load is: 21601.924 lbf
```

Appendix N: Bolt Shear Stress Test Program

```
Fxz = 3683.49455; %lbf, from 16.365 kN
Fxy = 2529.1006; %lbf, from 11.25 kN
A1 = 0.19635; %in
A2 = 0.30679; %in
A3 = 0.44179; %in
St_min = 120000; %psi

%transverse shear stresses in xz plane for
tao_xz1 = (4/3)*(Fxz/A1);
tao_xz2 = (4/3)*Fxz/A2
tao_xz3 = (4/3)*Fxz/A3;

tao_xy1 = (4/3)*Fxy/A1;
tao_xy2 = (4/3)*Fxy/A2
tao_xy3 = (4/3)*Fxy/A3;

tao_1 = sqrt(tao_xz1^2 + tao_xy1^2)
tao_2 = sqrt(tao_xz2^2 + tao_xy2^2)
tao_3 = sqrt(tao_xz3^2 + tao_xy3^2)

St_ult1 = St_min;
St_ult2 = St_min;
St_ult3 = St_min;

St_ult_fatigue = St_min %psi

%for bolts, ultimate shear stress is 0.6 times unltimate tensile stress.
Ss_ult1 = St_ult1*0.6
Ss_ult2 = St_ult2*0.6
Ss_ult3 = St_ult3*0.6

fs_1 = Ss_ult1/tao_1
fs_2 = Ss_ult2/tao_2
fs_3 = Ss_ult3/tao_3

%Fatigue Analysis for Repeated Stress
tao_m = tao_2/2
tao_a = tao_2/2;

%Estimated Endurance Limit
Se_ = 0.5*St_ult_fatigue

%Modifying Factors
%Ka
a = 2.70;
b = -0.265;
ka = a*((St_ult_fatigue*10^-3)^b)

%Kb
de = 0.37*(5/8)
```

```
kb = 0.879*de^(-0.107)

%Kc
kc = 1

%Kd
kd = 1

%Ke from table 6-5 for 99% reliability
ke = 0.814

%Kf
kf = 1

%Modified endurance limit
Se = Se_*ka*kb*kc*kd*ke*kf

tao_rev = tao_a/(1-(tao_m/St_ult2))
f = 0.82 %figure 6-18
a2 = ((f*St_ult2)^2)/Se
b = -[log(f*St_ult2/Se)]/3
N = (tao_rev/a2)^(1/b)
```

Program output:

```
tao_xz2 = 1.6009e+04
tao_xy2 = 1.0992e+04

tao_1 = 3.0341e+04
tao_2 = 1.9419e+04
tao_3 = 1.3485e+04

St_ult_fatigue = 120000
Ss_ult1 = 72000
Ss_ult2 = 72000
Ss_ult3 = 72000

fs_1 = 2.3730
fs_2 = 3.7077
fs_3 = 5.3393

tao_m = 9.7095e+03

Se_ = 60000
ka = 0.7592
de = 0.2313
kb = 1.0281
kc = 1
kd = 1
ke = 0.8140
kf = 1
```

```
Se = 3.8123e+04

tao_rev = 1.0564e+04
f = 0.8200
a2 = 2.5398e+05
b = -0.3161
N = 2.3395e+04

>>
```

Appendix O: Knuckle Stress Analysis

```
%degree project knuckle stress calcs
%AISI 1018 Mild Steel yield stress is 370 MPa
clc
clear

imp=45000; %impact force
delta=.00015*0.0254; % radial interference m
R=30/2/1000; %interference radius m
L=3.5*0.0254; %length of cylinder in meters
E_o=205*10^9; %modulus of elasticity Pa 205 GPa outer knuckle
E_i=139*10^9; %modulus of eleasticity Pa 139 shaft
Sut_o=440*10^6; % ultimate tensile strength outer knuckle
Sut_i=579*10^6; % ultimate tensile strength shaft
Sy_o=370*10^6; %yield strength Pa outer knuckle
Sy_i=290*10^6; %yield strenght Pa shaft
r_o=7/8*0.0254; %outer cylinder radius m knuckle
r_i=0; %inner radius
v_o=0.29; %poisson's ratio outer knuckle
v_i=0.25; %poisson's ratio shaft
p=delta/(R*(1/E_o*((r_o^2+R^2)/(r_o^2-R^2)+v_o)+1/E_i*(1-v_i)));%contact
pressure different material
f=0.74; %coeffiction of friction reference
https://web.mit.edu/8.13/8.13c/references-fall/aip/aip-handbook-section2d.pdf
F=2*R*L*p; %Force required to hold the cylinder in place
F_hold=f*pi*2*R*L*p;

%Outer cover stresses
sigma_t=p*((r_o^2+R^2)/(r_o^2-R^2)); %tangentail stress
sigma_rimp=-p; %radial stress with impact
tao=imp/(pi*2*R*L); %shear stress developed at interference location
normav=(sigma_t+sigma_rimp)/2;%average normal stress
tao_max=((sigma_t-normav)^2+tao^2)^0.5; %maximum shear stress in the body
nstatic=Sy_o/2/tao_max; %static factor of safety by max shear stress theory

%fatigue stress analysis
sigma_r=-p; %radial stress without impact
sigma_rm=(sigma_rimp+sigma_r)/2; % mean radial stress
sigma_ra=(sigma_rimp-sigma_r)/2; %radial stress amplitude
sigma_tm=sigma_t; %mean tangential stress
sigma_ta=0; %tangential stress amplitude
tao_m=(tao+0)/2; %torsional stress mean
tao_a=(tao-0)/2; %torsional stress amplitude
sigma_aprime=(sigma_ra^2-sigma_ra*sigma_ta+sigma_ta^2+3*tao_a^2)^0.5; %Mises
henky's alternating stress
sigma_mprime=(sigma_rm^2-sigma_rm*sigma_tm+sigma_tm^2+3*tao_m^2)^0.5; %Mises
henky's mean stress
ka=4.51*(Sut_o*10^-6)^-.265;%surface factor
kb=1.24*(2*R*1000)^-0.107;%size factor
kc=1; %Loading factor (combined loading)
kd=1; %temp factor at room temp
ke=0.897; %reliability factor .897 is 90%
kf=1; %more precise to be determined 1/n is max allowed with 90% reliability
and infinite life
S_eprime=0.5*Sut_o; %Sut<1400 Mpa eq. (6-8) from the design textbook
```

```
S_e=ka*kb*kc*kd*ke*S_eprime; %endurance strength

n=(1/((kf*sigma_aprime/S_e)^2+(kf*sigma_mprime/Sy_o)^2))^0.5; %factor of
safety for infinite life with ASME Eliptic

%bolting calculations
n_b=4; %number of bolts on knuckle
Load=F/n_b; %Load on each bolt
b_dia=8/16; %bolt diameter in inches
O=Load/(pi/4*(b_dia*.0254)^2); %normal stress in the bolts
tao_b=imp/n_b*4/3/(pi/4*(b_dia*.0254)^2); %transverse sheat
tao_bmax=(O^2+tao_b^2)^0.5;
d_req=((Load^2/(4*pi^2)+(imp^2/(9*pi^2*n_b^2)))/(586/2*10^6)^2)^0.25;
%diameter in meters with grade 5 bolt as per
https://www.boltdepot.com/fastener-information/materials-and-grades/bolt-
grade-chart.aspx
d_reqinches=d_req*1000/25.4; %bolt diameter required in inches with grade 5
bolt
fric=0.2; %common grade 5 bolt friction
B_torq=(fric*b_dia*25.4/1000*Load); %Newton Meters torque required on bolts
B_torqftlb=B_torq*0.73756;

fprintf("Required Bolt torque is: %0.3f Ft-Lb \n", B_torqftlb) %0.3f gives 3
points after decimal %f bring black in,
%Ft-Lb prints Ft-Lb after non purple, \n gives new line after the print,
black part is inserted directly after the :
fprintf("Static Factor of Safety n= %0.3f \n", nstatic)
fprintf("ASME Elliptic Fatigue Factor of Safety is n= %0.3f \n", n)
fprintf("Locking Force is %0.3f times the required amount \n", F_hold/imp)
```

Program output:

```
Required Bolt torque is: 15.983 Ft-Lb

Static Factor of Safety n= 7.674

ASME Elliptic Fatigue Factor of Safety is n= 8.444

Locking Force is 1.763 times the required amount
```

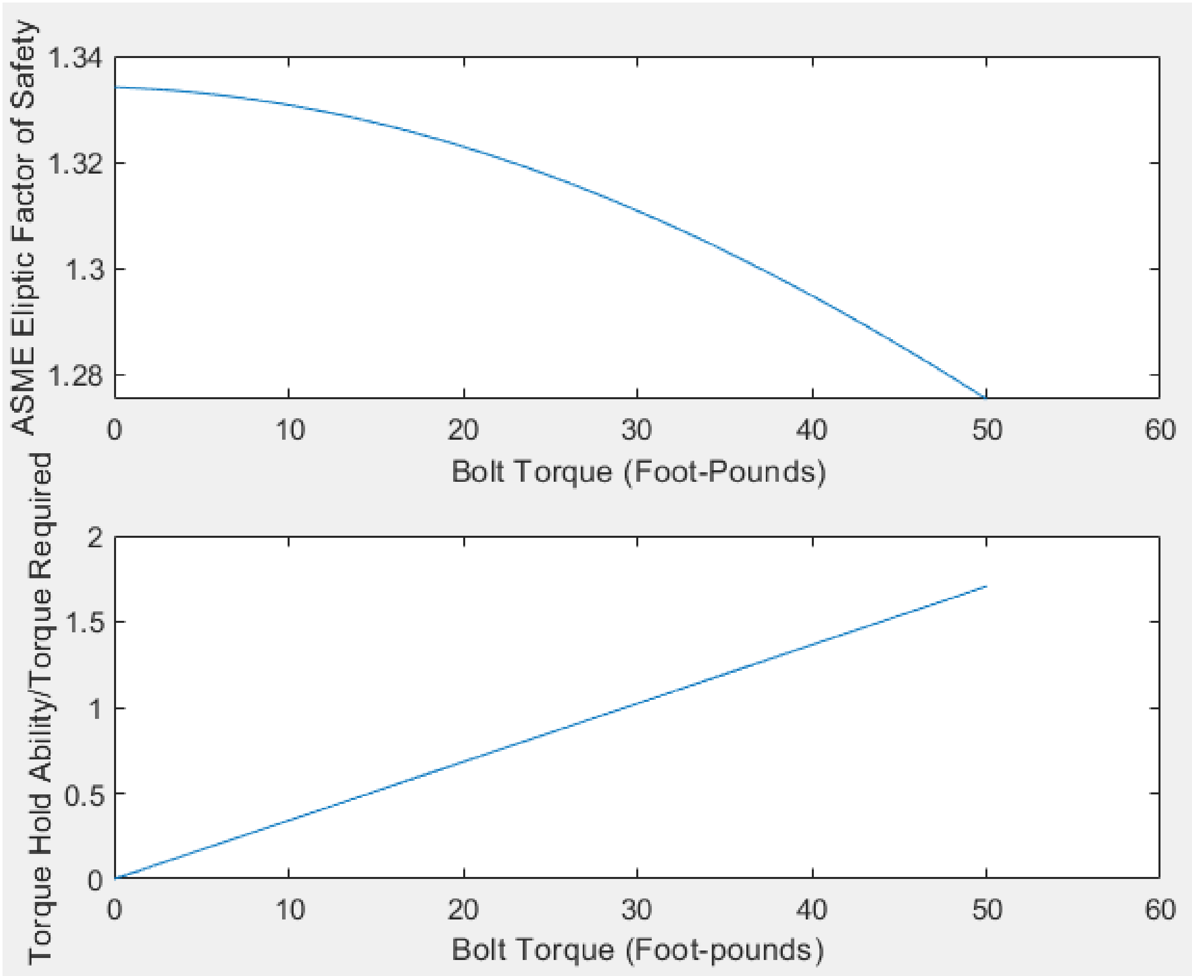
Appendix P: Impact Rod Stress Analysis

```
%degree project stress calcs
clc
clear
iter=0;
B_torqftlb=0;
while (B_torqftlb< 50)
    iter=iter+1;

imp=45000; %impact force
delta=iter*10^(-7)*0.0254; % radial interference m
R=1.25*0.0254; %interference radius m
L=3.25*0.0254; %length of cylinder in meters
E=205*10^9; %modulus of elasticity Pa 205 GPa
Sut=440*10^6; % ultimate tensile strength
Sy=370*10^6; %yield strength Pa
r_o=2*0.0254; %outer cylinder radius m
r_i=0; %inner radius
v=0.29; %poisson's ratio
d_inshaft=1.25*0.0254; %diameter of incoming impact shaft
p=E*delta/(2*R)*(1-(R^2)/(r_o^2));%contact pressure same material r_i=0
f=0.74; %coeffiction of friction reference
https://web.mit.edu/8.13/8.13c/references-fall/aip/aip-handbook-section2d.pdf
T(iter)=f*p*pi*((2*R)^2)*(L-d_inshaft/4)/2;%hold ability torque of cylinder
minshaft=2*(r_o^2+(L/2)^2)^0.5; % distance of two boxed corners from center
(no longer using two boxes back to back -not required)
T_req=imp*0.15*cos(45/180*pi); %2nd part is length beyond center
F=2*R*L*p; %Force required to hold the cylinder in place

%Outer cover stresses
sigma_t=E*delta/2/R*(1+(R^2)/(r_o^2)); %tangentail stress
sigma_rimp=-E*delta/(2*R)*(1-(R^2)/(r_o^2))-imp/(2*R*L); %radial stress with
impact
tao=T_req*R/(pi/2*R^4); %torsional shear stress developed
normav=(sigma_t+sigma_rimp)/2;%average normal stress
tao_max=((sigma_t-normav)^2+tao^2)^0.5; %maximum shear stress in the body
nstatic(iter)=Sy/2/tao_max; %static factor of safety by max shear stress
theory

%fatigue stress analysis
sigma_r=-E*delta/(2*R)*(1-(R^2)/(r_o^2)); %radial stress without impact
sigma_rm=(sigma_rimp+sigma_r)/2; % mean radial stress
sigma_ra=(sigma_rimp-sigma_r)/2; %radial stress amplitude
sigma_tm=sigma_t; %mean tangential stress
sigma_ta=0; %tangential stress amplitude
tao_m=(tao+0)/2; %torsional stress mean
tao_a=(tao-0)/2; %torsional stress amplitude
sigma_aprime=(sigma_ra^2-sigma_ra*sigma_ta+sigma_ta^2+3*tao_a^2)^0.5; %Mises
henky's amplitude stress, week 10 Mechancial Engineering Design 1
sigma_mprime=(sigma_rm^2-sigma_rm*sigma_tm+sigma_tm^2+3*tao_m^2)^0.5; %Mises
henky's mean stress
ka=4.51*(Sut*10^-6)^-.265;%surface factor
kb=1.24*(2*R*1000)^-0.107;%size factor
kc=1; %Loading factor (combined loading)
kd=1; %temp factor at room temp
```

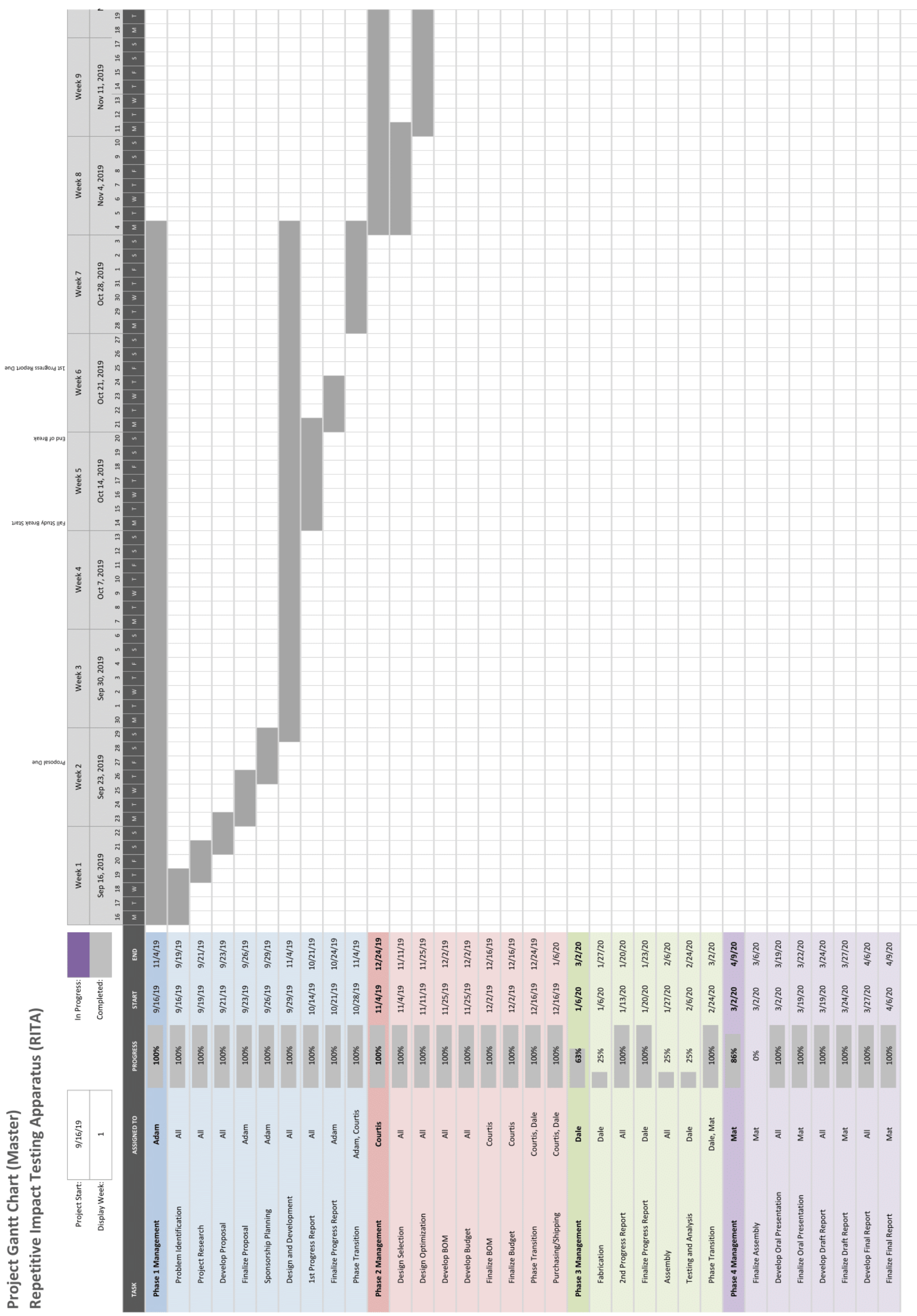
```
ke=0.897; %reliability factor .897 is 90%
kf=1.2; %stress concentration factor
S_eprime=0.5*Sut; %Sut<1400 Mpa eq. (6-8) from the design textbook
S_e=ka*kb*kc*kd*ke*S_eprime; %endurance strength

n(iter)=(1/((kf*sigma_aprime/S_e)^2+(kf*sigma_mprime/Sy)^2))^0.5; %factor of
safety for infinite life with ASME Eliptic

%bolting calculations
Load=F/4; %Load on each bolt
O=Load/(pi/4*(7/16*.0254)^2); %normal stress in the bolts
d_req=(Load*4/pi/(586*10^6))^0.5; %diameter in meters with grade 5 bolt as
per https://www.boltdepot.com/fastener-information/materials-and-grades/bolt-
grade-chart.aspx
d_reqinches=d_req*1000/25.4; %bolt diameter required in inches with grade 5
bolt
fric=0.2; %common grade 5 bolt friction
B_torq=(fric*7/16*25.4/1000*Load); %Newton Meters torque required on bolts
B_torqftlb(iter)=B_torq*0.73756; %Newton meters to foot pounds.
end

subplot (211)
plot (B_torqftlb, n)
xlabel('Bolt Torque (Foot-Pounds)'); ylabel ('ASME Eliptic Factor of
Safety');
subplot (212)
plot (B_torqftlb, T/T_req)
xlabel('Bolt Torque (Foot-pounds)'); ylabel ('Torque Hold Ability/Torque
Required');

fprintf("Required Bolt torque is: %0.3f Ft-Lb \n", B_torqftlb(4658)) %0.3f
gives 3 points after decimal %f bring black in,
%Ft-Lb prints Ft-Lb after non purple, \n gives new line after the print,
black part is inserted directly after the :
fprintf("Static Factor of Safety n= %f \n", nstatic(4658))
fprintf("ASME Elliptic Fatigue Factor of Safety is n= %f \n", n(4658))
fprintf("Locking Torque is %f times the required amount \n", T(4658)/T_req)
```

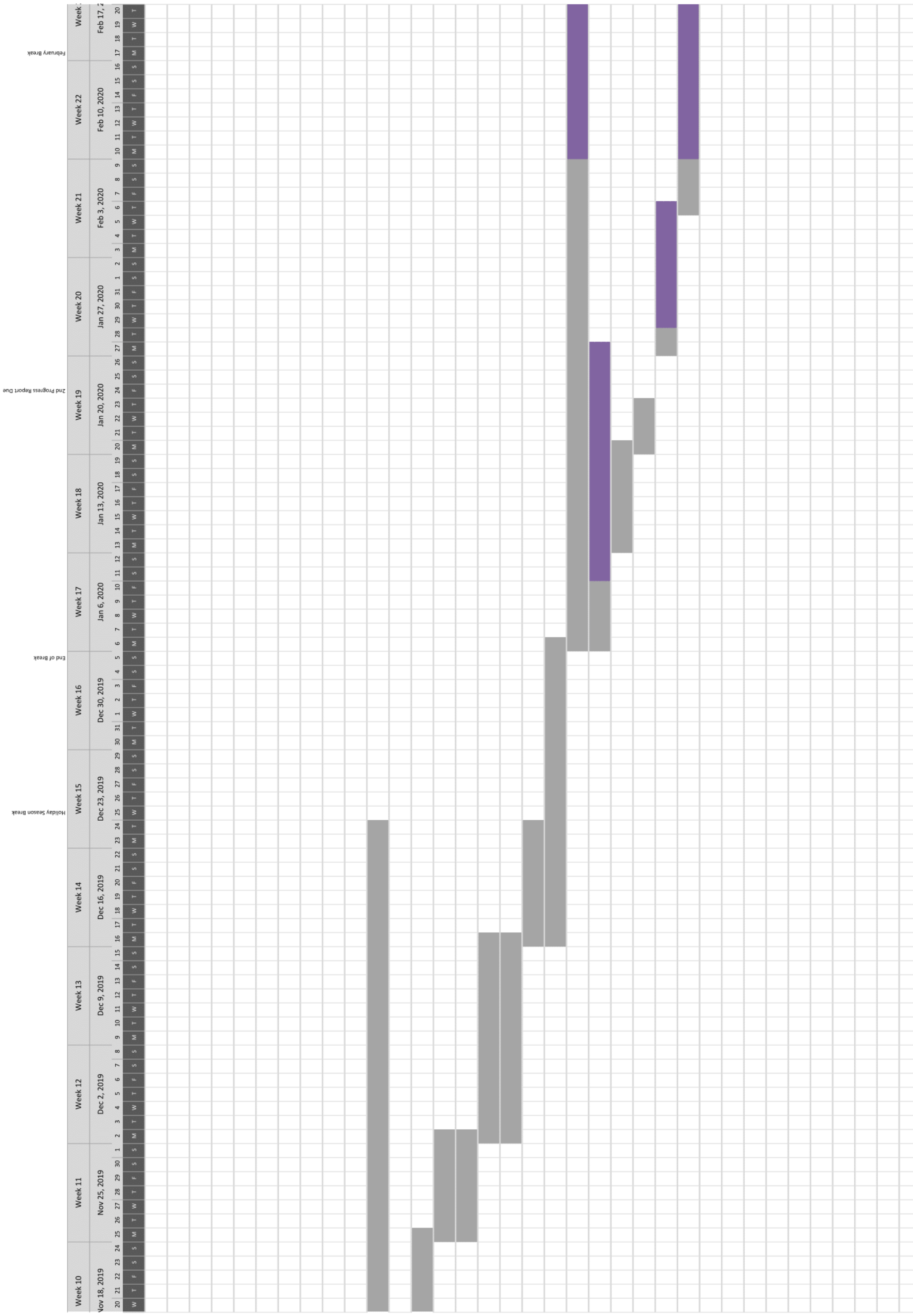
Program Output:

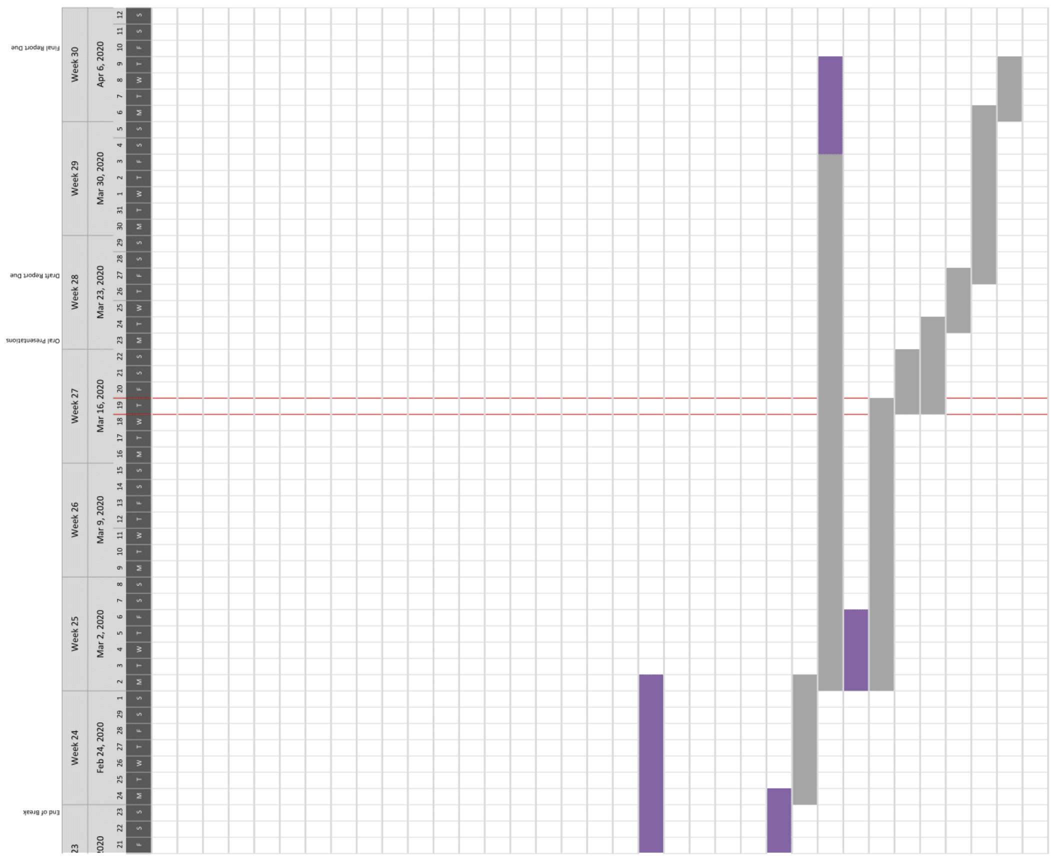
```
Required Bolt torque is: 50.000 Ft-Lb

Static Factor of Safety n= 1.778653

ASME Elliptic Fatigue Factor of Safety is n= 1.275443

Locking Torque is 1.705375 times the required amount
```


ASME Eliptic Factor of Safety
1.34
1.32
1.3
1.28
0
10
20
30
40
50
60
Bolt Torque (Foot-Pounds)
Torque Hold Ability/Torque Required
2
1.5
1
0.5
0
0
10
20
30
40
50
60
Bolt Torque (Foot-pounds)

Appendix Q: Gantt Chart




Week 10
Nov 18, 2019
Week 11
Nov 25, 2019
Week 12
Dec 2, 2019
Week 13
Dec 9, 2019
Week 14
Dec 16, 2019
Week 15
Dec 23, 2019
Week 16
Dec 30, 2019
Week 17
Jan 6, 2020
Week 18
Jan 13, 2020
Week 19
Jan 20, 2020
Week 20
Jan 27, 2020
Week 21
Feb 3, 2020
Week 22
Feb 10, 2020
Holiday Season Break
End of Break
2nd Progress Report Due
February Break


Week 24
Feb 24, 2020
Week 25
Mar 2, 2020
Week 26
Mar 9, 2020
Week 27
Mar 16, 2020
Week 28
Mar 23, 2020
Week 29
Mar 30, 2020
Week 30
Apr 6, 2020
End of Break
Oral Presentations
Draft Report Due
Final Report Due